STEAM

ITS GENERATION AND USE



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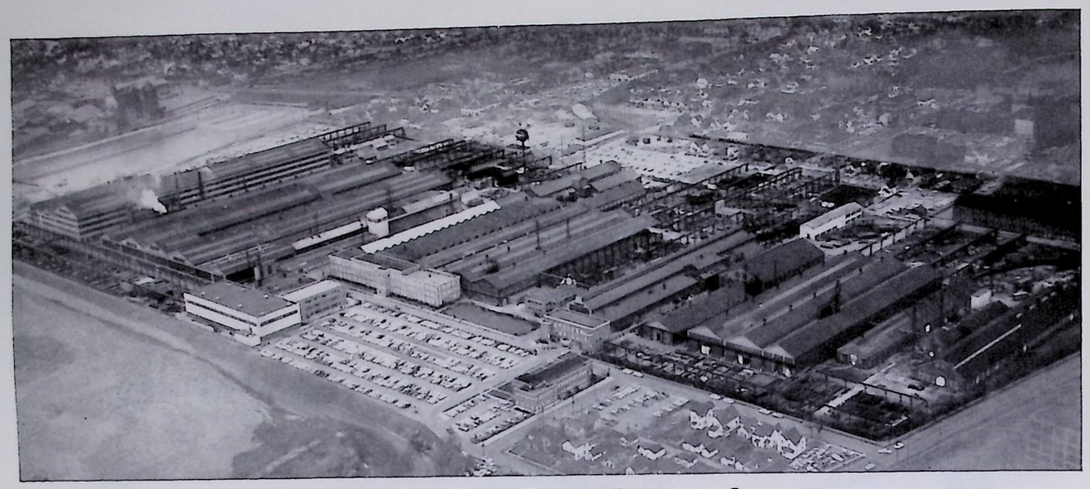


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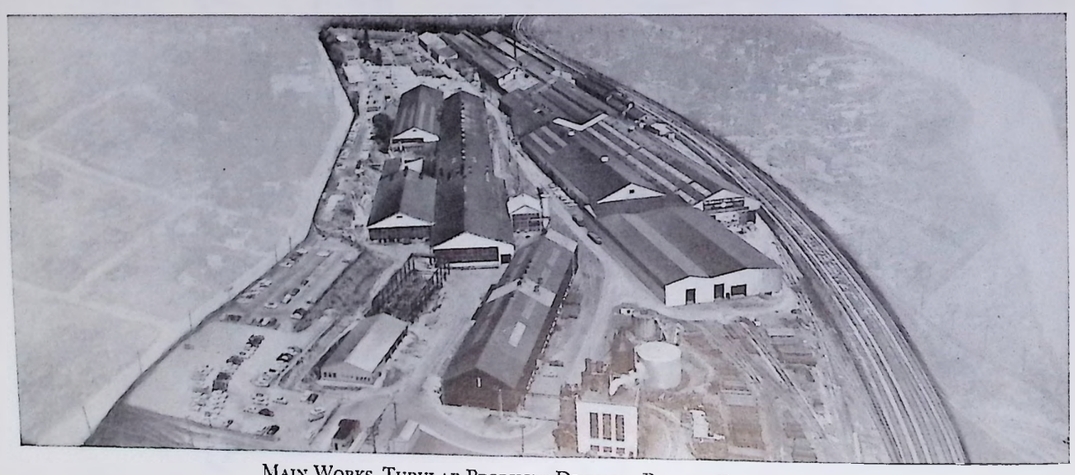
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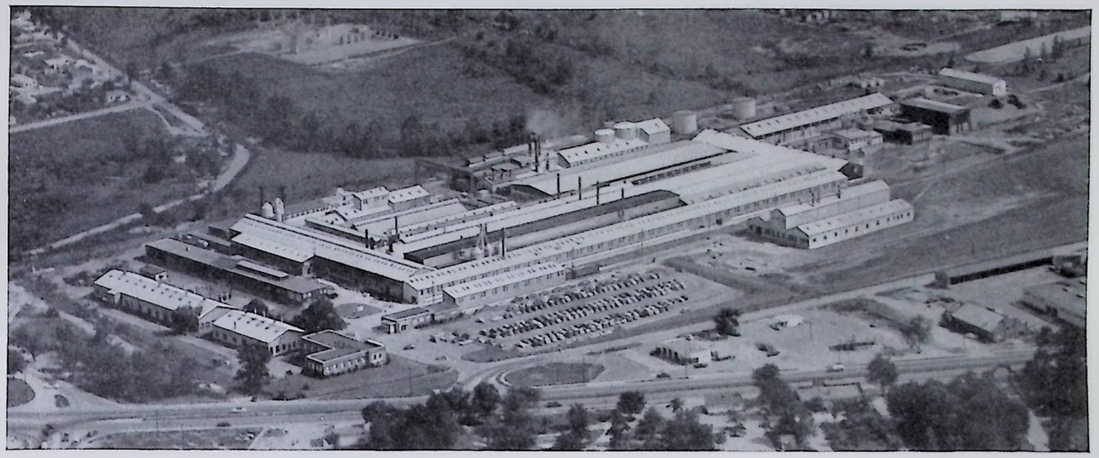


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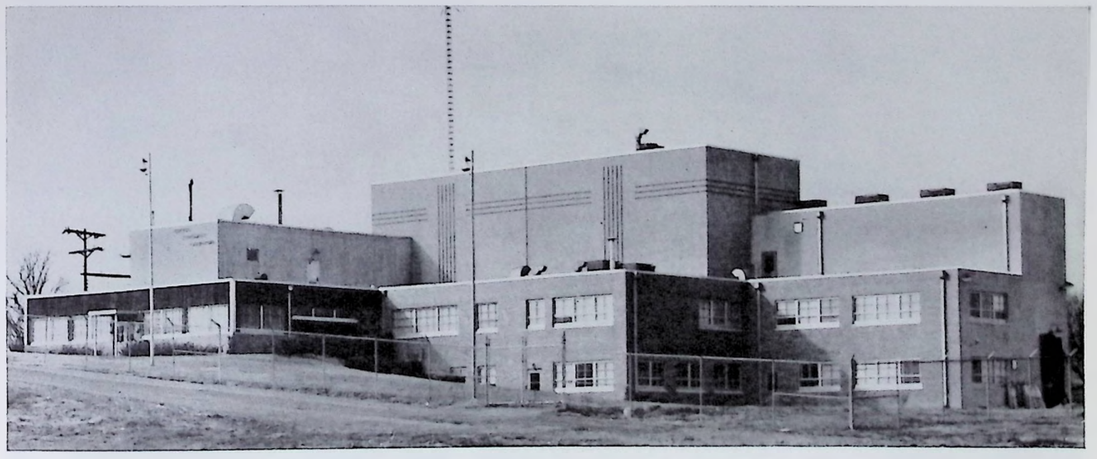


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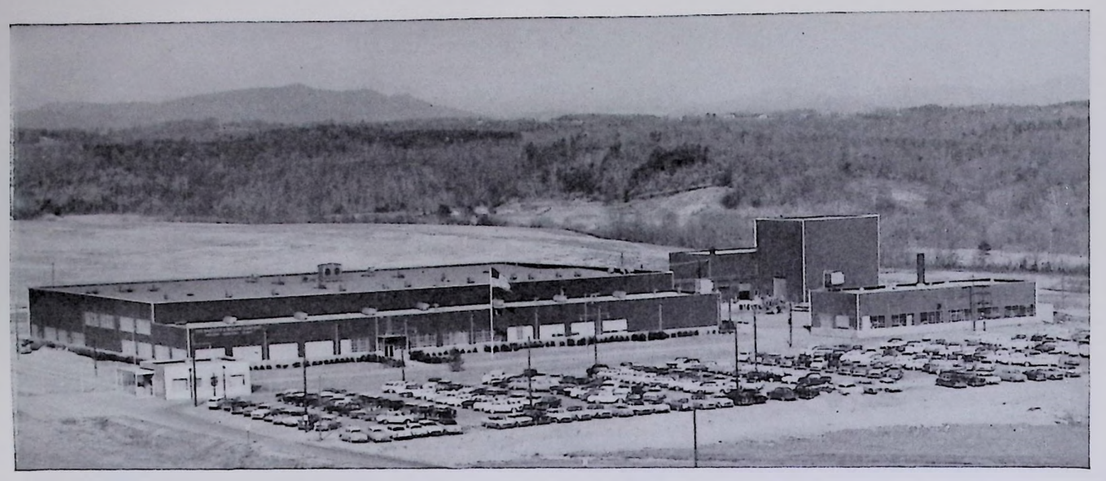
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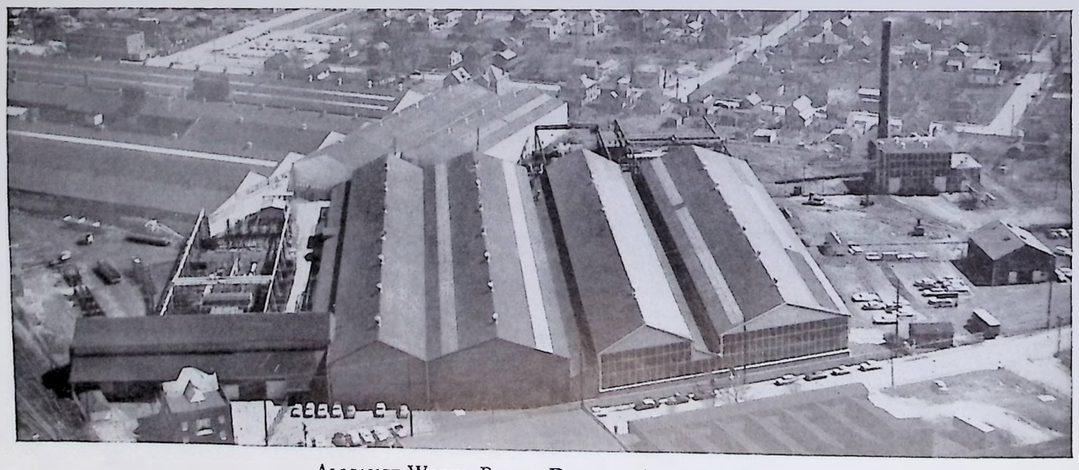
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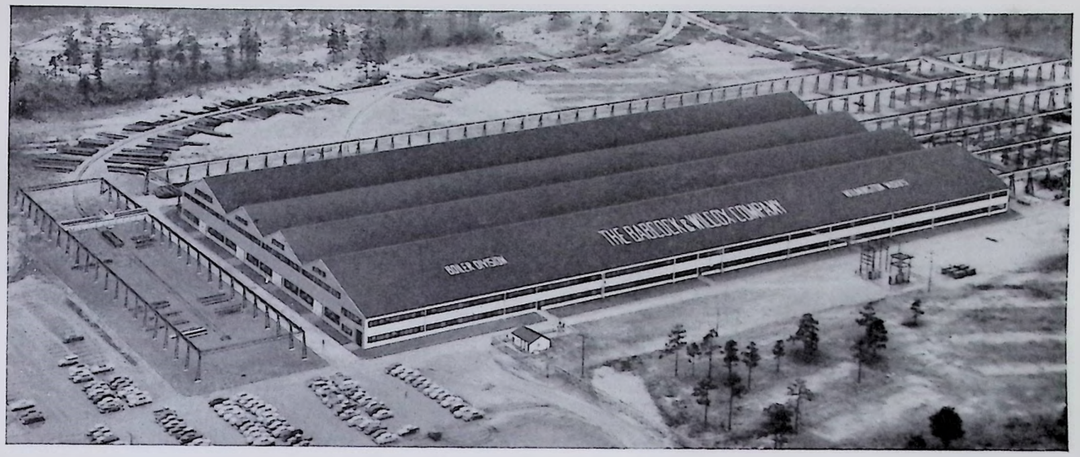


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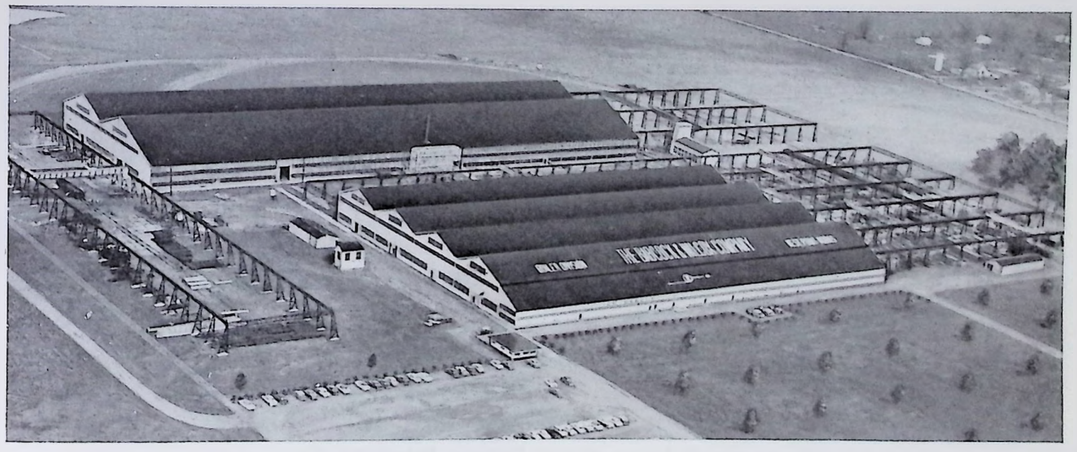


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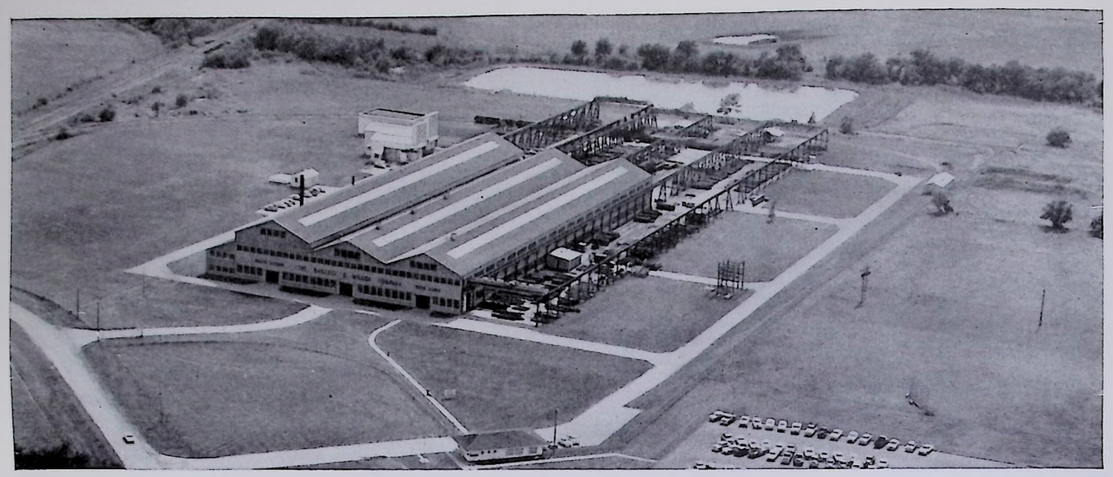
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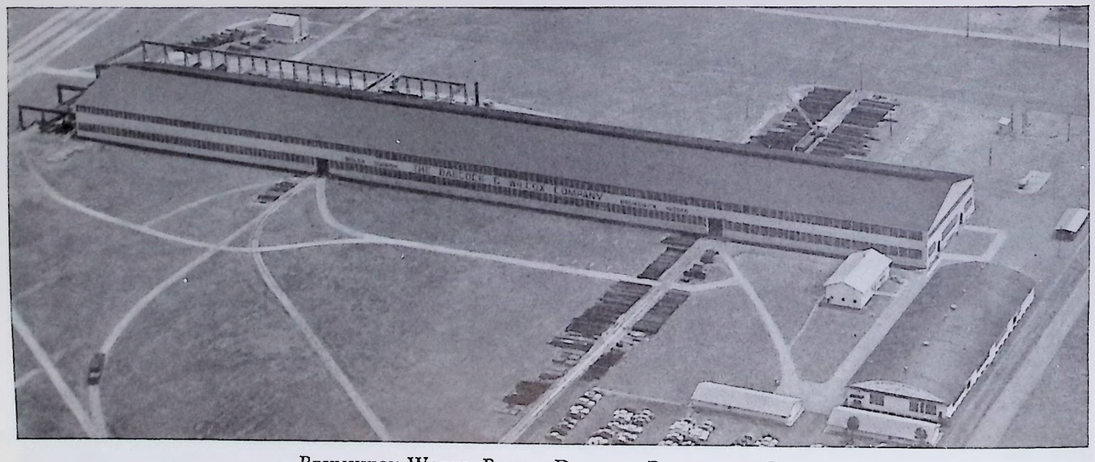
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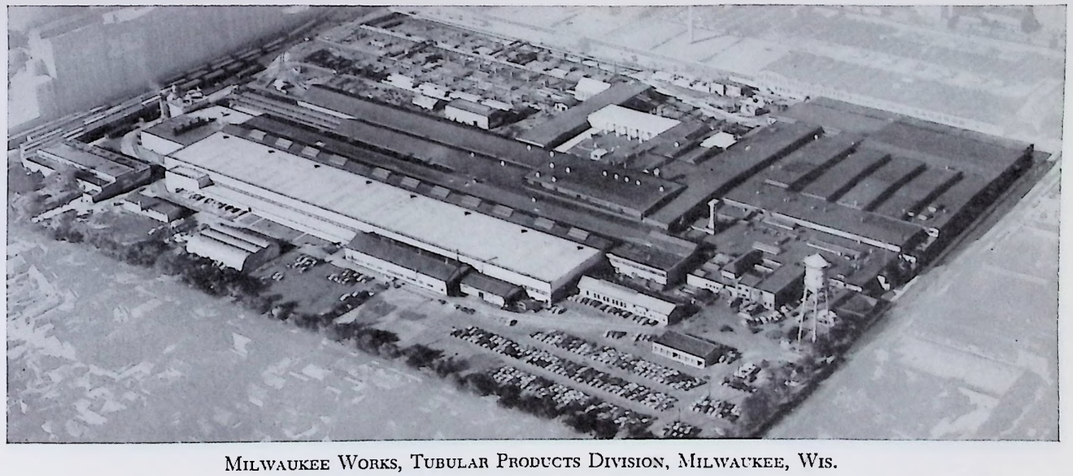


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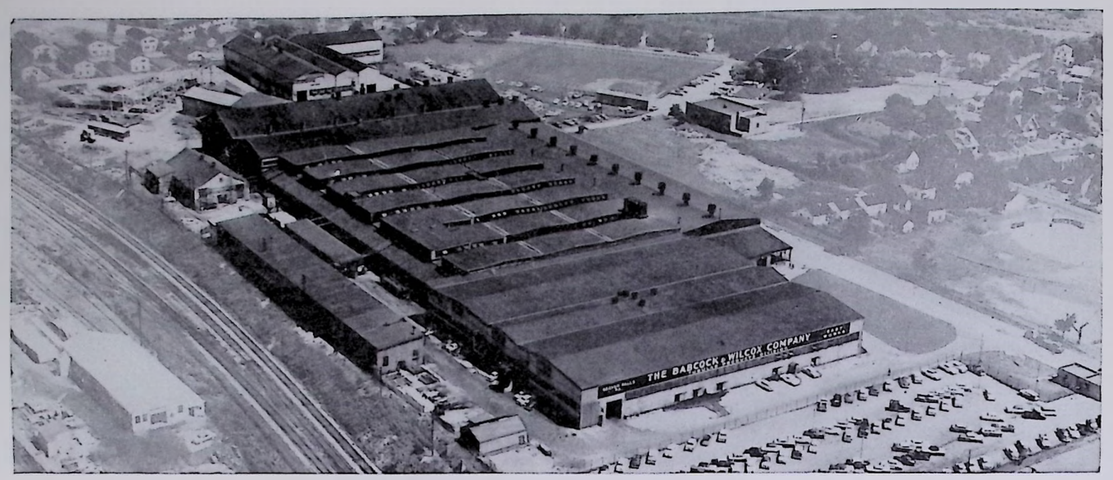
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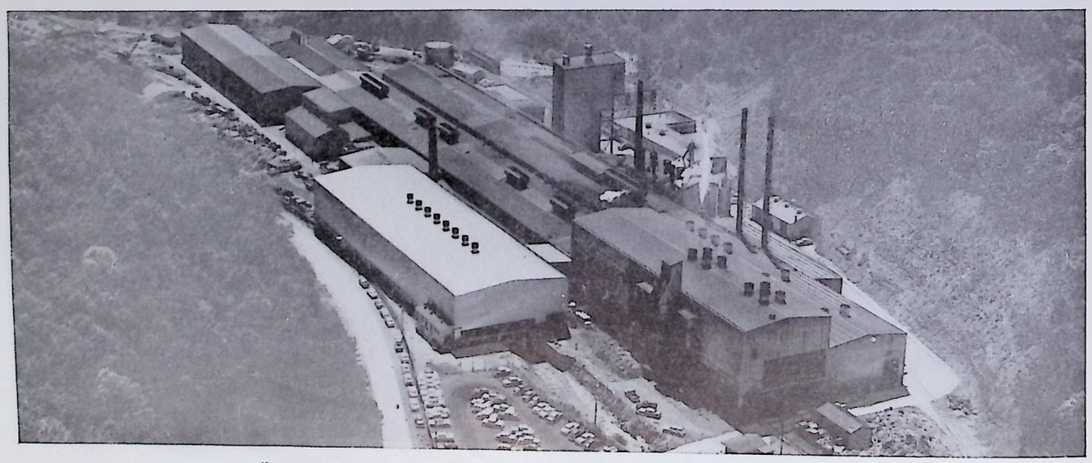
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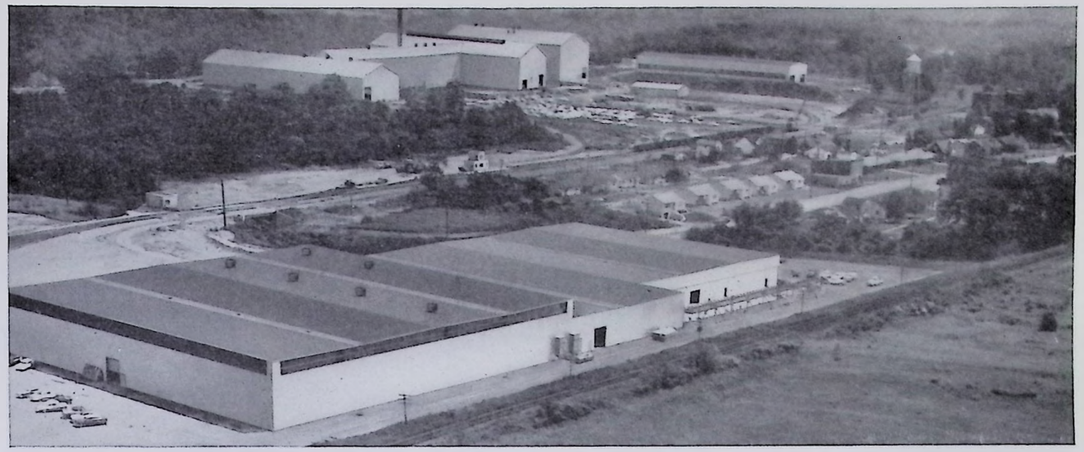


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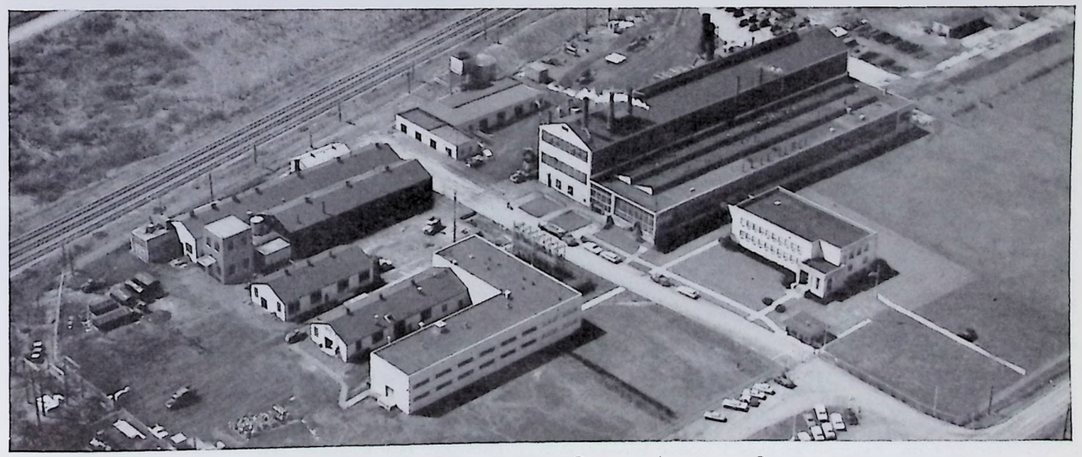


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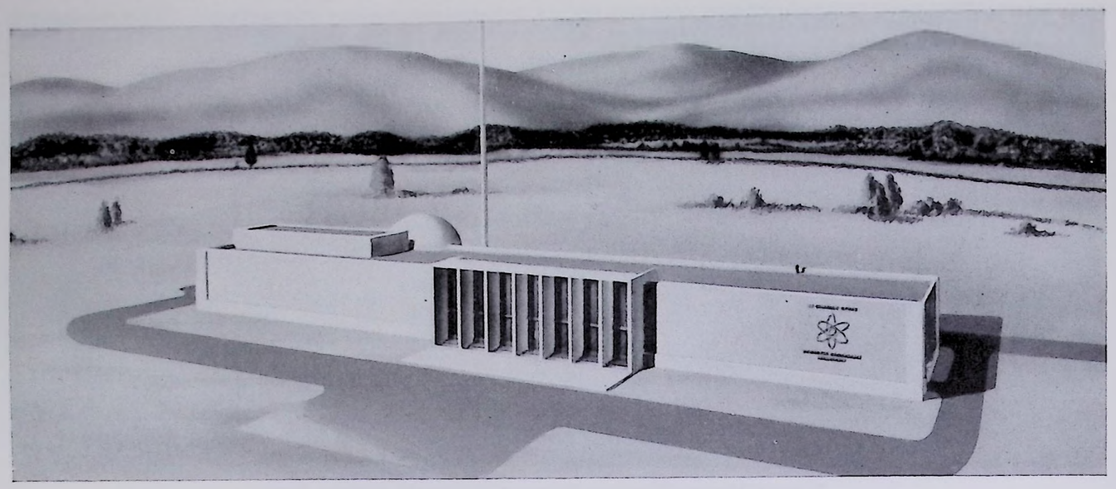


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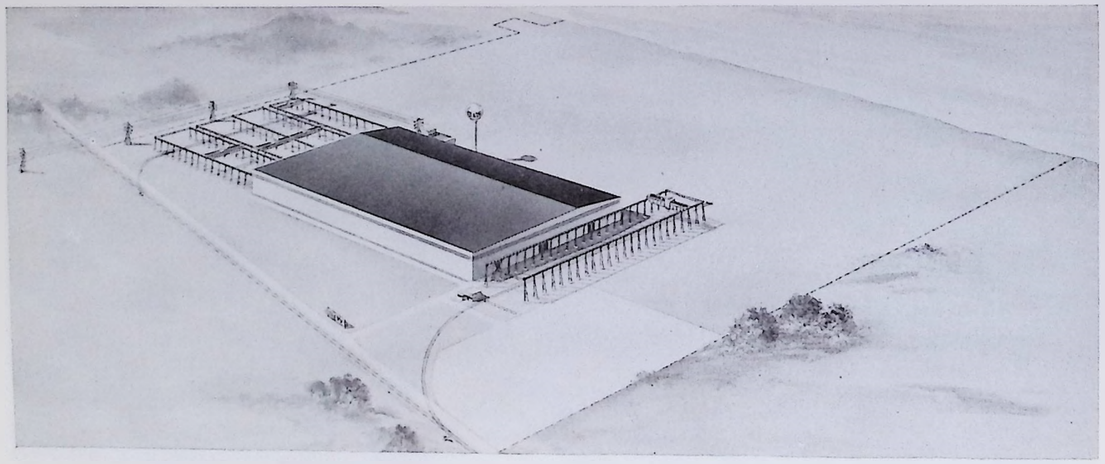


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Foreword

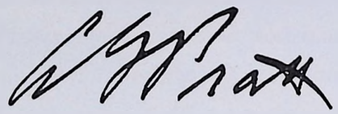
In 1867 George H. Babcock and Stephen Wilcox formed a partnership. The present company succeeded it in 1881.

This 37th edition of STEAM—the first appeared in 1879—marks tremendous advances in the production of steam and in the burning of fuels.

Countless improvements have been devised by the ingenuity of man to give him mastery of his environment. Surely some of the greatest are the steam engine and the developments which followed its introduction.

This intensive age depends upon power for convenience and release from physical drudgery and, indeed, for its adequate continuation. Industrial power is still, and will long be, dependent upon the use of fuel for the generation of steam.

The Company gratefully acknowledges the support, cooperation, and loyalty of its patrons and employees, which have made possible its contributions in the field of steam generation. It expresses the hope that this volume may help to stimulate further advances in the art.



CHAIRMAN OF THE BOARD

1

INTRODUCTION TO STEAM

Records indicate that certain properties of steam were recognized and utilized prior to the year 150 B.C. At about that time, Hero of Alexandria described in his treatise *Pneumatica* devices of his predecessors and contemporaries, as well as an invention of his own, which used the expansive force of steam to raise water above its natural level. He mentioned that this expansive power could be used to elevate a weight and that rotary motion could be produced by the reactive force of steam on the atmosphere. In the latter application, known as "Hero's Engine," a hollow sphere was supported over a boiler by two trunnions, one of which was hollow and connected the interior of the sphere with the steam space of the boiler. Two open pipes bent at right angles were inserted at opposite poles of the sphere, the terminal legs pointing in opposite directions. Steam generated in the boiler passed through the hollow trunnion to the sphere, from which it was exhausted to the atmosphere through the two bent pipes. The reaction of the steam on the atmosphere caused the sphere to rotate as a primitive reaction turbine.

Hero did not suggest any commercial application of the devices he described. Although they were used for trivial purposes, there is no record of related development until the late sixteenth and early seventeenth centuries.

Early Studies and Applications

Treatises bearing on the generation of steam were published by Matthesius, a German author, in 1571; Besson, a philosopher and mathematician at Orleans, in 1578; Ramelli, in 1588; Battista Della Porta, a Neapolitan mathematician and philosopher, in 1601; De Caus, a French engineer and architect, in 1615; and Branca, an Italian architect, in 1629.

Credit for proposing, if not making, the first useful steam engine is apparently due Edward Somerset, second Marquis of Worcester. In 1663 he published a treatise describing devices for raising water, not only by forcing it from two receivers by direct steam pressure, but also by a reciprocating piston actuating one end of a lever, the other operating a pump. His descriptions are rather obscure and without illustrations. There is no authentic record that any of these devices was actually constructed, but it is claimed by many that he did build and operate a steam-driven piston engine.

In 1680 Dr. Denis Papin, a Frenchman, invented a steam digester for culinary purposes, using "a boiler under heavy pressure." To avoid an explosion, he added a contrivance which is the first safety valve of record.

First Successful Steam Engine

The first commercially successful steam engine was developed by Thomas Savery. In 1699 he exhibited a model steam engine before the Royal Society of England, of which Sir Isaac Newton was at that time president. The engine consisted of two copper receivers alternately connected, by a three-way hand-operated valve, with a boiler and a source of water supply. When the water in one receiver had been driven out by the steam, cold water was poured over its outside surface, creating a partial vacuum through condensation and causing it to fill again, while the water in the other receiver was being forced out. Many such machines were placed in actual use as mine pumps.

A serious drawback in the use of Savery's engine was that the height to which it could lift water was limited by the pressure the boiler and vessels could bear. Before its eventual displacement by Newcomen's engine, Desaguliers improved the Savery engine by applying the Papin safety valve to the boiler and using an internal jet for condensation.

The earliest cylinder-and-piston steam engine was based on Papin's suggestion of 1690 that the condensation of steam should be used to make a vacuum beneath the piston after it had been raised by the expansion of steam. Newcomen's atmospheric engine was a practical application of this principle. Papin's first engine was unworkable because the same vessel served as both boiler and cylinder. Heat was applied to a small quantity of water in the bottom of the vessel. After the piston had been raised by the expansion of the steam generated, the heat source was withdrawn, and work was done by the piston on its

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down stroke under atmospheric pressure. An improved form of this engine was developed by Papin after hearing of Savery's engine. In Papin's engine of 1705, a floating diaphragm, or piston, on top of the water in a displacement chamber, kept the steam and water from direct contact. The water, delivered to a closed tank by the downward movement of the piston under pressure, flowed in a continuous stream against the vanes of a water wheel. After the steam in the displacement chamber had expanded, it was exhausted to the atmosphere through a valve instead of being condensed. The engine was, in fact, a noncondensing single-action steam pump with the steam and pump cylinders in one. A curious feature of this engine was a mass of heated metal placed within the diaphragm to keep the steam dry and thus prevent condensation during expansion. This device might be called the first superheater.

Among the several inventions attributed to Papin was a boiler with an internal firebox, the earliest record of such construction.

Newcomen's Engine for Pumping Water

While Papin neglected his own earlier suggestion of a steam-and-piston engine to develop Savery's ideas, Thomas Newcomen, with his assistant, John Cawley, adapted Papin's suggestion of 1690 in a practical engine. Steam admitted from the boiler to a cylinder raised a piston by its expansion, assisted by a counterweight on the other end of a beam actuated by the piston. The steam valve was then closed and the steam condensed by a jet of cold water. The piston was forced downward by atmospheric pressure and did work on a pump. The condensed water in the cylinder was expelled through an escapement valve by the next entry of steam, which was at a pressure only slightly above atmospheric.

In 1711 this engine was introduced into mines for pumping. Whether its action was originally automatic or dependent upon the hand operation of the valves is unknown. The story commonly believed is that a boy, Humphrey Potter, in 1713, whose duty it was to open and shut such valves of an engine he attended, caused the engine to manipulate these valves automatically by suitable cords and catches attached to the beam. This device was simplified in 1718 by Henry Beighton, who suspended from the beam a rod called the plug tree, which actuated the valves by tappets. By 1725 this engine was in common use in collieries and was changed but little in the next seventy years. Since the pressure in the pumps did not depend upon the steam pressure, Newcomen's engine was a distinct advance compared with Savery's engine. In common with the latter, the losses from the alternate heating and cooling of the steam cylinder were enormous. Though this engine might have been modified to serve many purposes, its use seems to have been limited almost entirely to the pumping of water.

The rivalry between Savery and Papin stimulated attention to the saving of fuel. In 1730 Dr. John Allen noted that nearly half of the heat of the fire was lost because of the short duration of contact between the gases and the boiler heating surfaces. To diminish this loss he used an internal furnace with a smoke flue winding through the water in the form of a worm in a still. To prevent the length of gas passage from throttling the fire, he recommended the use of bellows to force the gases through the flue. This is probably the first suggested use of forced draft. In estimating the quantity of fuel lost up the stack, Allen probably made the first boiler test.

Toward the end of the period of use of Newcomen's atmospheric engine, John Smeaton, who, about 1770, built and installed a number of large engines of this type, greatly improved the design in its mechanical details.

Earliest Water-Tube Boiler

A patent taken out by William Blakey in 1766, covering an improvement in Savery's engine, included a novel form of steam generator (see Fig. 1). This is probably the first step in the development of the water-tube boiler. Several tubes alternately inclined at opposite angles were arranged in the furnace, the adjacent tube ends being connected by small pipes. The first successful user of water-tube boilers, however, was James Rumsey, an American inventor celebrated for his early experiments in steam navigation, and it is he who was truly the originator of the water-tube boiler. In 1788 he patented, in England, several forms of boilers, some of which were of the water-tube type. One had a firebox with flat top and sides, with horizontal tubes across the firebox connecting the water spaces. Another had a cylindrical firebox surrounded by an annular water space, and a coiled tube connected at its two ends with the water space was within the box. This was the first of the "coil boilers." Another form in the same patent was the vertical tubular boiler, practically as made at the present time.

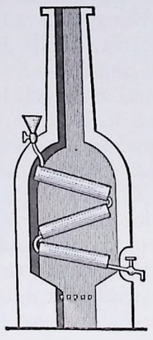


Fig. 1. *First water-tube boiler. Built and patented by William Blakey in 1766*

Great Contributions of James Watt

The improvements in boiler and engine design by Smeaton, Newcomen, and their contemporaries were followed by those of the great engineer, James Watt, an instrument maker of Glasgow. In 1763, while repairing a model of Newcomen's engine, he was impressed by the great waste of steam in the alternate cooling and heating of the engine. His remedy was to maintain the cylinder at the temperature of the entering steam. With this in view he added a vessel, separate from the cylinder, into which the steam should pass from the cylinder and be condensed either by the application of cold water outside or by a jet from within. To preserve a vacuum in his condenser, he added an air pump to remove the water of condensation and the air that leaked in or was brought in with the injection water. Since the cylinder no longer acted as a condenser, it could be maintained at a high temperature by steam-jacketing or by covering it with insulating material. Further, and with the same object in view, he covered the top of the cylinder and introduced steam above the piston to do the work previously accomplished by atmospheric pressure. After several trials with an experimental apparatus based on these ideas, Watt patented his improvements in 1769.

Though Watt's patent describes a noncondensing engine using high steam pressure, his aversion to such practice was strong. Notwithstanding his knowledge of the advantage of greater expansion from higher pressures, he continued to use pressures not more than 7 psi. To overcome such pressures, the boilers were fed through standpipes of sufficient height for the column of water to offset the pressure within the boiler.

In 1782 Watt patented two other features that he had invented as early as 1769. These were the double-acting engine, using steam on both sides of the piston, and the use of steam expansively by shutting off steam from the cylinder when the piston had made but a portion of its stroke, the power for the completion being supplied by the expansive force of the steam already admitted.

He further added a throttle valve for the regulation of steam admission and invented the automatic governor, the steam indicator, a mercury steam gage, and a glass water column.

While little is said in the biographies of Watt about the improvement of steam boilers, all the evidence indicates that Boulton and Watt introduced the first "wagon boiler," so called because of its shape. In 1785 Watt took out a number of patents for variations in furnace construction. Because of the low steam pressures used until the early part of the nineteenth century, little attention was given to the form of the boiler for supplying steam to the engine described above.

About 1800 Richard Trevithick, in England, and Oliver Evans, in America, introduced noncondensing and, for that time, high-pressure steam engines. To the initiative of Evans may be attributed the general use of high-pressure steam in the United States, a feature which for many years distinguished American from European practice. The demands for light weight and for economy of space, which followed the beginning of steam navigation and the invention of the locomotive, required boilers designed and constructed to withstand heavier pressures and forced the adoption of the cylindrical form of boiler, which, as might be expected, varied greatly in detail and principles of operation.

The plain shell-type vessel or one of its many modifications was increasingly used. It incorporated a greater or lesser number of flues, conducted across the water space through which the hot gases passed. However, as the number of steam engines and boilers in use continued to multiply because of the intense industrial activity of the middle nineteenth century, the increase in the number and severity of shell-boiler explosions served to emphasize the inherent safety advantages of the water-tube boiler.

Early American Steam Boilers

The first water-tube boiler, made of a combination of small tubes connected at one end to a reservoir (see Fig. 2), was the invention of an American, John Stevens, who secured a U.S. patent on the design in 1803. This boiler was actually used to generate steam for the engine of a steamboat operating on the Hudson River, but like all the "porcupine" boilers, of which type it was the first, it did not have the elements required for continued success.

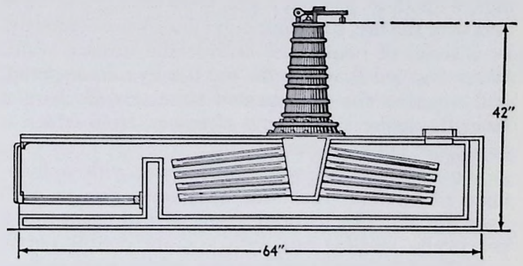


Fig. 2. *Water-tube boiler of small tubes connected at one end to a reservoir. John Stevens, 1803*

Another form of water-tube boiler was patented in 1805 by John Cox Stevens, a son of John Stevens. This boiler consisted of twenty vertical tubes, 1¾-in. internal diameter and 40½ in. long, arranged in a circle, the outside diameter of which was approximately 12 in., connecting a water chamber at the bottom with a steam chamber at the top. Both steam and water

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chambers were annular spaces of small cross sections and contained approximately 33 cu in. The illustration, Fig. 3, shows the cap of the steam chamber secured by bolts. One of these boilers was for a long time at the Stevens Institute of Technology at Hoboken, N. J., and is now in the Smithsonian Institution in Washington, D. C.

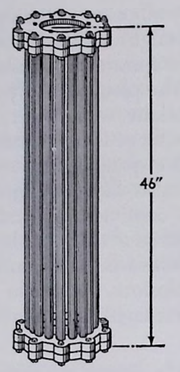


Fig. 3. *Water-tube boiler with tubes connecting water chamber below and steam chamber above. John Cox Stevens, 1805*

At about the same time, Arthur Woolf built a boiler of large horizontal tubes extending across the furnace and connected at the ends to a longitudinal drum above. The first purely sectional water-tube boiler was built by Julius Griffith in 1821. In this boiler, horizontal water tubes were connected to vertical side pipes, which were joined to horizontal gathering pipes, and the latter were connected, in turn, to a steam drum.

In 1822 Jacob Perkins constructed a flash boiler for what was then considered high pressure. Cast-iron bars, with 1½-in. annular holes, were arranged in three tiers over the fire, and their outer ends were connected by a series of bent pipes outside the furnace walls. Water was fed slowly to the top tier by a force pump, and steam in the superheated state was discharged from the lower tiers into a chamber, from which it was taken to the engine.

The first sectional water-tube boiler with well-defined circulation was built by Joseph Eve in 1825 (see Fig. 4). The sections, composed of small tubes, practically vertical but with a slight double curve, were fixed in horizontal headers, which were in turn connected to a steam space above and a water space below formed of larger pipes. The steam and the water spaces were connected by outside pipes to secure a circulation of the water up through the sections and down through the external pipes. In the same year, John M'Curdy of New York built a "Duplex Steam Generator" of "tubes of wrought or cast iron or other material" arranged in several horizontal rows, connected together alternately at the front and the rear by return bends. Cylindrical vessels closed at their ends were placed within the tubes below the water line, so that a thin sheet of water would be exposed to the action of the fire.

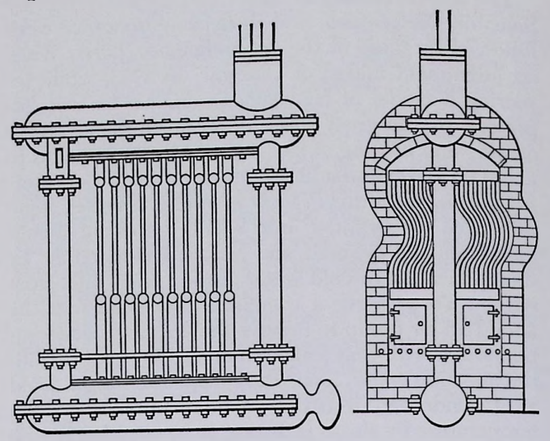


Fig. 4. *First sectional water-tube boiler with well-defined circulation. Joseph Eve, 1825*

In 1826 Goldsworthy Gurney built a number of boilers for use on his steam carriages. Small U-shaped tubes were laid sideways, and their ends were connected with larger horizontal pipes (see Fig. 5). These were interconnected by vertical pipes to permit circulation, and they were also connected to a vertical cylinder which served as a steam-and-water reservoir.

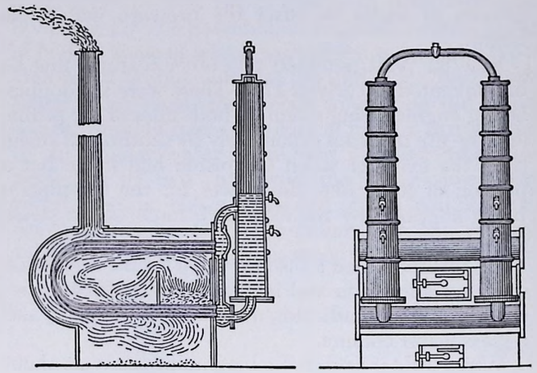


Fig. 5. *Water-tube boiler with small U-shaped tubes connected at ends with larger pipes. Goldsworthy Gurney, 1826*

In 1828 Paul Steenstrup made the first shell boiler with vertical water tubes in the large flues, similar to the boiler known as the "Martin" and suggesting the "Galloway."

The first water-tube boiler with fire tubes within

the water tubes was built in 1830 by Summers and Ogle. Horizontal sections at the top and bottom were connected by a series of vertical water tubes, with fire tubes extending through the water tubes and the horizontal sections. The fire tubes were held in place by nuts, which also served to make the joints.

Early Development of the B&W Boiler

Stephen Wilcox, in 1856, was the first to use inclined water tubes to connect water spaces at the front and rear with a steam space above (see Fig. 6).

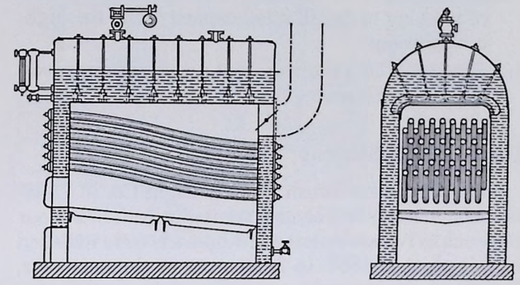


Fig. 6. *Inclined water tubes connecting front and rear water spaces complete circuit with steam space above. Stephen Wilcox, 1856*

The first to make such inclined tubes into a sectional form was Twibill, in 1865 (see Fig. 7). He used wrought-iron tubes connected at the front and rear with standpipes through intermediate connections. These standpipes carried the steam to a horizontal cross drum beneath a dry drum at the top, the entrained water being carried to the rear.

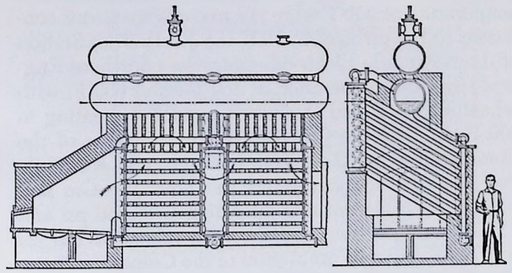


Fig. 7. *First water-tube boiler with inclined tubes in sectional form. Twibill, 1865*

In 1866 George Herman Babcock became associated with Stephen Wilcox, and the first Babcock & Wilcox boiler (see Fig. 8) was patented in 1867. The nest of horizontal cast-iron tubes at the top served in place of a steam-and-water drum. The inclined steam-gathering tubes below, also of cast iron, were placed one above the other in vertical rows, each row and its ends forming a single casting. During the next ten years, development of the boiler continued with great vigor. Drums were soon substituted for the nest of cast-iron tubes originally used for steam-and-water storage, and, as might be expected, steel replaced cast iron for the generating tubes. By 1877 the B&W boiler had been modified as shown in Fig. 9. A little later, the headers were made of wrought steel to meet the demand for boilers of higher pressure. Still later developments are illustrated in Figs. 10 and 11. In these boilers the cast-iron drumheads of Fig. 9 had been changed to dished heads of pressed steel.

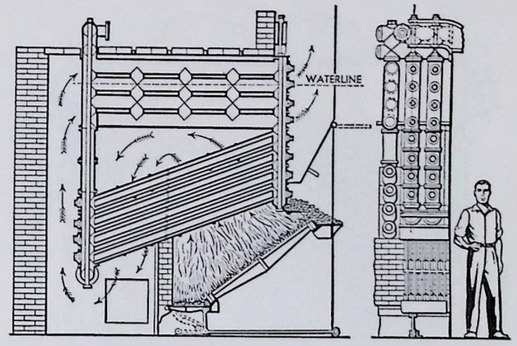


Fig. 8. *First Babcock & Wilcox boiler, patented in 1867*

Requirements of a Perfect Steam Boiler

The following rules, formulated by Stephen Wilcox and originally published in 1875, are of interest since they are reasonably sound today.

1. Proper workmanship and simple construction, using materials which experience has shown to be the best, thus avoiding the necessity of early repairs.
2. A mud drum to receive all impurities deposited from the water and so placed as to be removed from the action of the fire.
3. A steam-and-water capacity sufficient to prevent any fluctuation in steam pressure or water level.
4. A water surface for the disengagement of the steam from the water, of sufficient extent to prevent foaming.
5. A constant and thorough circulation of water throughout the boiler, so as to maintain all parts at the same temperature.
6. The water space divided into sections so arranged that, should any section fail, no general explosion can occur and the destructive effects will be confined to the escape of the contents; large and free

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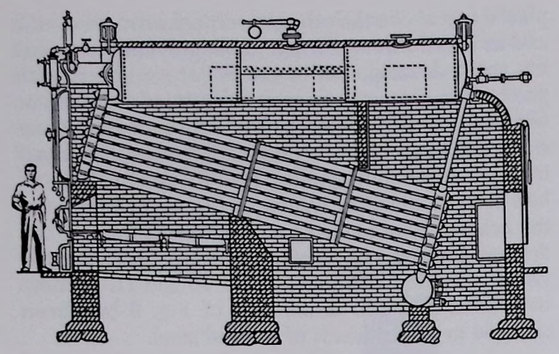


Fig. 9. *Babcock & Wilcox boiler, developed in 1877*

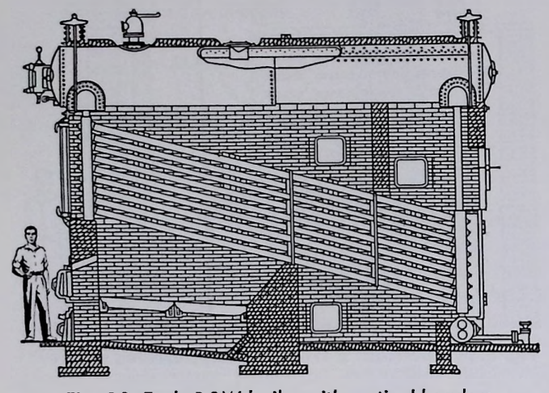


Fig. 10. *Early B&W boiler with vertical headers*

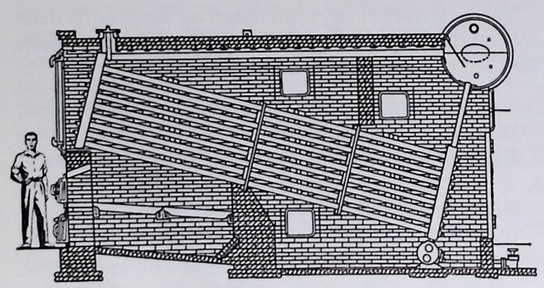


Fig. 11. *Early B&W boiler of cross-drum type*

passages between the different sections to equalize the waterline and pressure in all.

7. A great excess of strength over any legitimate strain, the boiler being so constructed as to be free from strains resulting from unequal expansion and, if possible, to avoid joints exposed to the direct action of the fire.
8. A combustion chamber so arranged that the combustion of the gases started in the furnace may be completed before the gases escape to the chimney. (Editor's note: Today it is essential that combustion be virtually completed in the furnace.)
9. The heating surface as nearly as possible at right angles to currents of heated gases, so as to break up the currents and extract the entire available heat from the gases.
10. All parts readily accessible for cleaning and repairs, a point of the greatest importance as regards safety and economy.
11. Proportioned for the work to be done and capable of working to its full rated capacity with the highest economy.
12. Equipped with the very best gages, safety valves, and other fixtures.

Early Central Stations

The plant of the Brush Electric Light Co. in Philadelphia was the first central station in America. Four Babcock & Wilcox boilers of 73 hp each were installed in this plant in 1881. In September of the next year, boilers of similar make were placed in operation at the Pearl Street Station of the Edison Electric Illuminating Co., New York. The Fisk Street Station of the Commonwealth Edison Co., Chicago, commenced in 1902 and placed in service in October of 1903, was the first public utility plant exclusively using steam turbines for electrical generation. Ninety-six Babcock & Wilcox boilers of 508 hp each were installed in this plant. The turbines operated at 170 psi with steam at 70 F superheat.

From about 1900 to the period of World War I, steam pressures in the larger central stations were gradually increased. By 1914, 250 to 275 psi and a temperature of 560 F were not uncommon steam conditions at the turbine. In 1921 the North Tees Station of the Newcastle Electric Supply Co. (northern England) went into operation at 450 psi and 650 F, with reheat at 500 F and feedwater extraction heating to 300 F. In 1924 the Crawford Avenue Station of the Commonwealth Edison Co., Chicago, and the Philo and Twin Branch stations of the American Gas and Electric Co. all went into operation with 550 psi and 725 F at the turbine throttle, with reheat at 105 psi and 700 F. Miami Fort Station of the Columbia Power Co., Cincinnati, with similar steam conditions, went into operation in 1925. Two turbines each of 50,000 kw and one turbine of 60,000 kw were installed in the Crawford Avenue Station, the largest of these power plants. Steam was supplied to each turbine from four standard boilers and one reheat boiler—a total of 15 boilers built for a working pressure of 650 psi. Side views of the standard and reheat boilers are shown by Figs. 12 and 13, respectively.

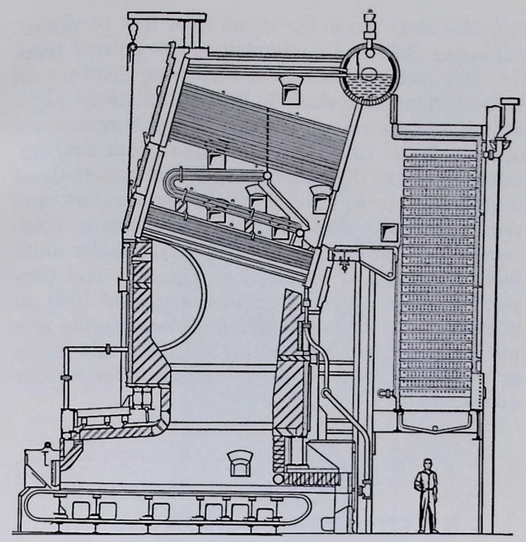


Fig. 12. *B&W boiler for 650 psi working pressure and 750 F steam temperature, 1924*

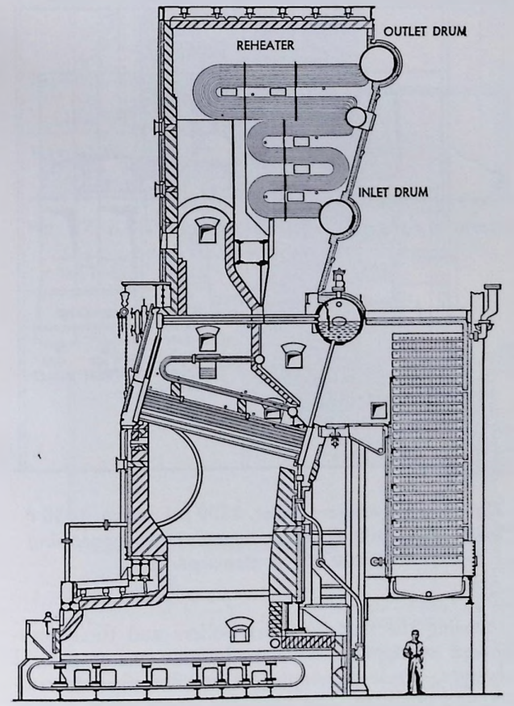


Fig. 13. *B&W boiler with superheat and reheat, for 650 psi working pressure, 750 F initial and 725 F reheat temperatures, 1924*

Bent-Tube Boilers

The leadership of B&W in the production of straight-tube boilers did not preclude recognition of the merits and potentialities of the drum-and-tube type of boiler. Its participation in the perfection of this type dates from 1880, when Allan Stirling initiated his designs. In 1906 the Stirling Consolidated Boiler Company was amalgamated with the other activities of the B&W organization. Since that date there has been a cross-fertilization of accumulated experience which has led to the development of the best ideas in the boiler art. Fig. 14 illustrates one of the early designs of bent-tube boilers which contributed so much to make the name of Stirling world famous.

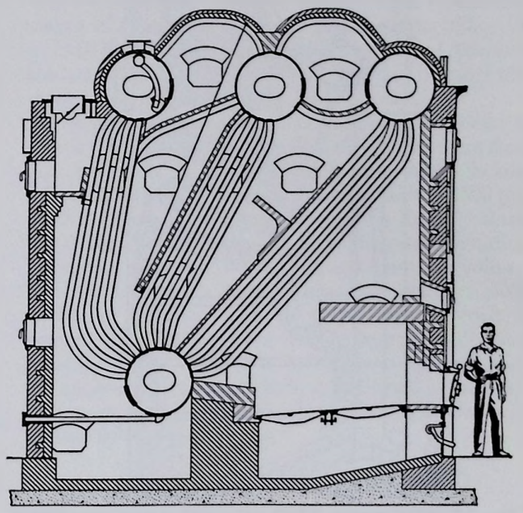


Fig. 14. *Early Stirling boiler arranged for hand firing*

Later Development of B&W Boilers

The really big jump in steam pressure for central stations came in 1924 and 1925 with the design and placing in service of the 1200 psi unit at the Weymouth (later renamed Edgar) Station of the Boston Edison Co. The boiler, built to supply steam for the high-pressure 3150-kw 3600-rpm turbine, is illustrated by Fig. 15. This boiler was designed to supply 143,000 lb steam per hr at 1200 psi and a temperature of 700 F. After passing through the high-pressure turbine, the steam was exhausted at 360 psi to the reheater. Here it was reheated to 700 F and was mixed with the steam from other boilers going to the main turbines, where it developed an additional 12,000 kw. This boiler was known as a topping unit and its turbine as a topping turbine.

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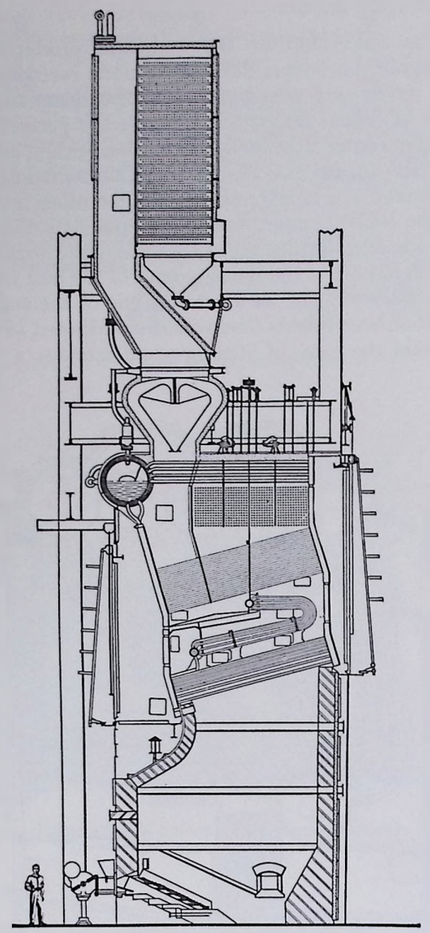


Fig. 15. *B&W boiler, 1200 psi working pressure, 700 F initial and reheat temperatures, and reheat pressure of 360 psi, 1924*

Major changes in the production and utilization of steam occurred between 1925 and the early thirties. The design of larger boilers, operating at higher rates of output and particularly with the use of pulverized coal as fuel (then increasing at an exceedingly rapid pace), necessitated the development of the water-cooled furnace. The proportion of boiler surface hitherto arranged in banks could therefore be reduced, since new steam-generating surface was provided in cooling the furnace. Feedwater heating by steam bled from early stages of the turbine had a similar effect. Above all, the increase in steam temperature required for greater turbine efficiency served still further to reduce boiler-bank surface as such, to be replaced by additional superheater surface since, with higher steam pressures and temperatures, it is necessary that a greater proportion of the total heat input be absorbed in the steam after it is produced. Following 1930 and continuing to the present time, the same factors have had an increasing influence on boiler design, particularly in large installations. Thus, for a steam pressure of 2500 psi, steam temperatures of the order of 1050 F, and reheated steam temperatures of 1050 F, the boiler unit consists essentially of drums, furnace water-wall tubes, superheaters, and reheaters, plus such heat-recovery adjuncts as economizers and air heaters (see Fig. 16). Smaller units, built to operate at lower steam pressures and temperatures—in the general pressure region of 1000 psi with moderate superheat—retain a considerable proportion of their total generating surface in tube banks in addition to the water-cooled surface in the furnace (see Fig. 17).

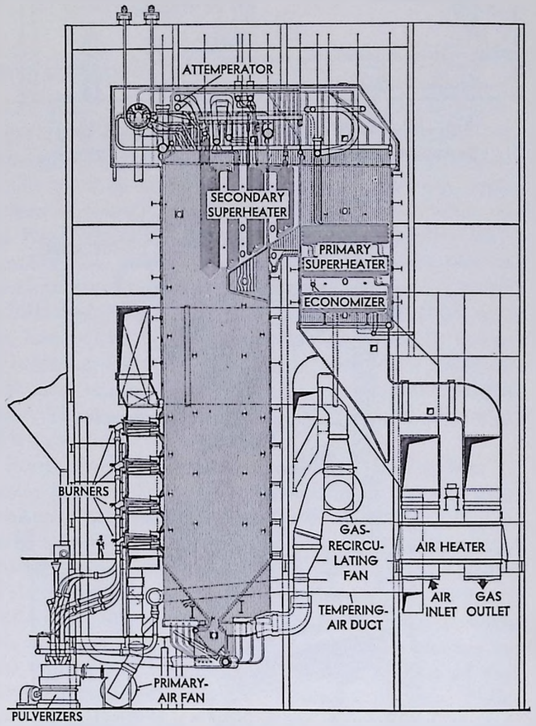


Fig. 16. *B&W Radiant boiler, 2500 psi design, 1050 F initial and 1000 F reheat temperatures, generating 2,400,000 lb steam per hr*

During the past 30 years boilers and their component equipment have undergone many radical changes. These developments are described in subsequent chapters. The need for larger units, higher pressures and temperatures, and the demand for long periods of uninterrupted service have stimulated design of the drum-and-tube type of boiler (see Chapter

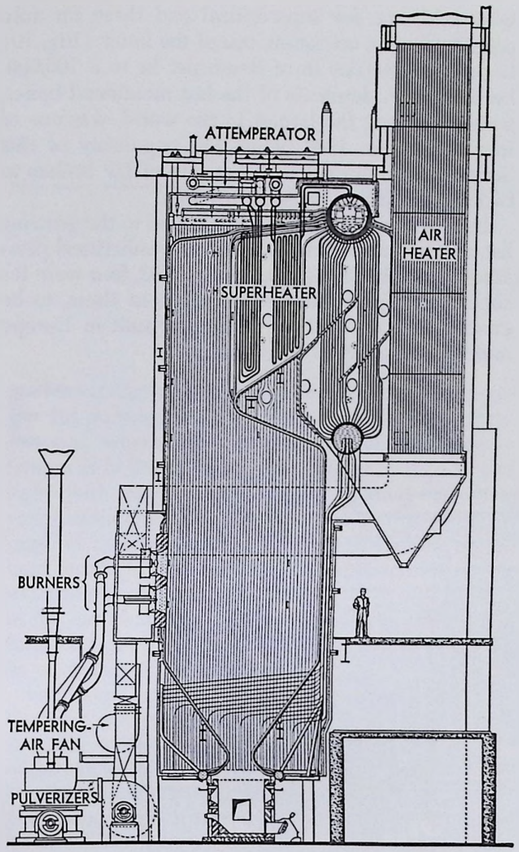


Fig. 17. *B&W Stirling boiler, 925 psi working pressure and 900 F steam temperature*

23). Much work of sound and lasting effect was done during this period in the following categories: the coordination of fuel burning and combustion control with boiler performance to attain greater efficiency; the development of pulverized-coal firing; the measurement of steam and other fluids; and the development of integral-furnace and high-pressure designs.

Steam Production above Critical Pressure

For many years the Company considered the "once-through" principle of steam generation. These studies culminated in the development of the Universal-Pressure boiler for operation at pressures above or below the critical. A contract for the first commercial installation of this type was concluded in 1953. The unit supplies steam at 4500 psi and 1150 F with two stages of reheat, 1050 F and 1000 F, respectively, for a 120,000-kw electric-generating machine. Operation started in 1957, and a new peak in plant thermal efficiency was attained.

The general arrangement of various components is shown diagrammatically by Fig. 18. The working fluid makes one fast uninterrupted trip from the feedwater inlet at 5500 psi, 525 F, to the turbine inlet at 4500 psi, 1150 F. There is no drum. The unit is fired by three 9-ft-diam Cyclone Furnaces, all at the same elevation. There is a relatively small amount of furnace volume and surface and a large convection pass. A wide variety of Eastern Ohio and West Virginia coals is burned. Ash disposal from the Cyclone Furnaces and from the boiler furnace is by the conventional slag-tap method. Gas recirculation, as indicated in the illustration, is used to control ash conditions in the convection tube bank.

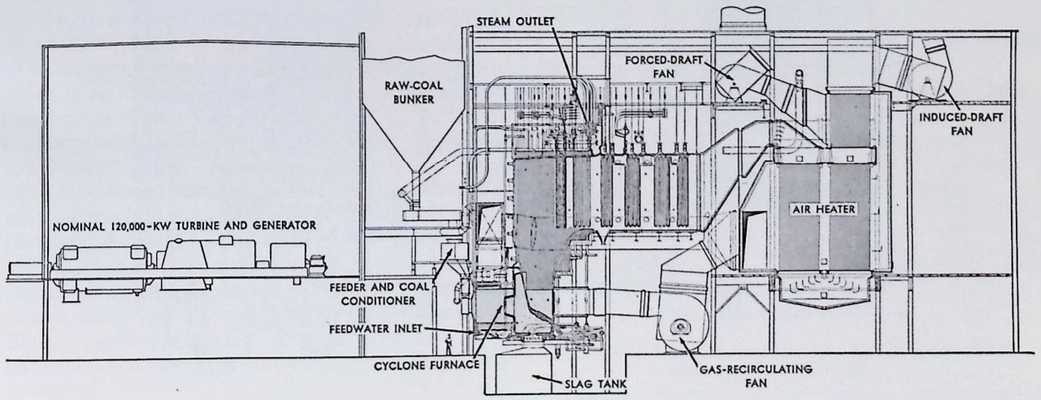


Fig. 18. *Elevation view of a 120,000-kw generating unit served by a B&W Universal-Pressure (UP) boiler operating at 5500-4500 psi, 1150 F, with reheats to 1050 F and 1000 F*

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In 1956 two additional Universal-Pressure (UP) boilers were ordered, each for a capacity of 2,923,000 lb steam per hr, design pressure 4635 psi, initial steam temperature 1050 F, with two stages of reheating to 1050 F at 1265 and 306 psi, respectively. One of these units is Cyclone Furnace fired with slag-tapping, the other pulverized-coal fired with dry-ash removal. Each unit serves a 500,000-kw turbine.

Three more UP boilers were ordered, two in 1957 for oil and gas firing and one in 1958 for Cyclone Furnace firing, all three to operate subcritically. This demonstrates the versatility of the Universal-Pressure boiler with regard to both pressure and type of firing.

The popularity of the Universal-Pressure boiler was again manifest when, in 1959, four boilers of this type were sold, one for supercritical and three for subcritical pressure operation, one of the latter (Fig. 19) to supply 4,900,000 lb of steam per hr to a 700,000-kw turbine. A duplicate of the last mentioned boiler, which is among the largest in the world, was one of four ordered in 1960, when the popularity of this design spread abroad with the sale of UP boilers to two electric power stations in Japan.

In 1961 four UP boilers were added to the growing list, two for supercritical and two for subcritical pressure operation. Of the five sold in 1962, four were for supercritical pressure operation; one of these, to be erected in Italy, will be the largest unit in Europe and will serve a 600,000-kw turbine.

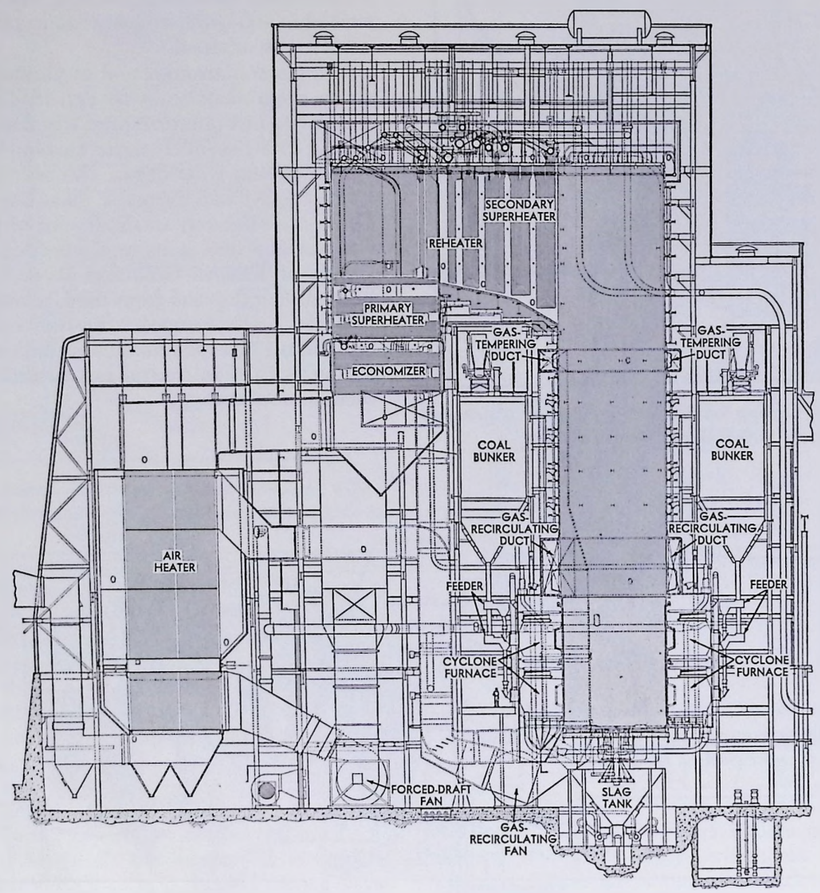


Fig. 19. A 700,000-kw-capacity B&W Universal-Pressure boiler, one of the largest in the world

2

SOURCES OF HEAT ENERGY—COAL

ESSENTIALLY all energy that can be converted to useful purposes is of "atomic origin" by way of the sun, where the transmutation of hydrogen to helium is believed to produce energy in the form of light, with minor supplements of thermal and ultraviolet radiation. The only exception to this is the energy released by nuclear reactions. Most of the earth's heat results from the atomic degradation of a few of its elements in situ. Aside from this natural process in the earth's geological development, practically all future supplies of energy, as far as is now known, will be of atomic origin from the sun.

Estimates indicate that only one two-billionths of the sun's energy reaches the earth's atmosphere and that this is 100,000 times the present annual energy requirement of the world. About half of this energy is reflected back into interstellar space by the atmosphere, and some of the energy absorbed by the earth by day is lost by radiation at night. The net energy, therefore, theoretically recoverable from this source in all forms is estimated to be about 50 times the world requirement.

An estimate of the theoretical heat value at a given time of all vegetation if used directly as fuel, as alcohol, or as synthetic gasoline made from vegetation, appears in Table 1. Production of vegetation requires CO_2, sunlight, and plant nutrients including moisture. There is an abundance of sunlight and moisture and there seems to be an ample supply of CO_2. Estimates indicate that CO_2 constitutes 0.03% of the atmosphere and 0.0012% of the hydrosphere (the aqueous envelope of the earth), or a carbon total of several thousand times as much as the carbon in the coal of the world. However, it would be impossible to use more than a small part of all the vegetation as fuel, since for continued plant growth a large portion of the vegetal matter must be returned to the soil as nutrients. Hence, vegetation cannot be considered as a continuing source of energy in the magnitude indicated in Table 1.

The energy available per year, by the better known means of using the continuous energy of the sun, is given in Table 2. At present, only 19% of the energy requirement of the world is obtained from these sources. Should the world energy requirement double in the next 100 years, it is conceivable that it could be satisfied by improved methods of utilization of these renewable sources, which are about 830 times the present annual requirement, equivalent to 2100 million tons of 13,000 Btu/lb coal.

TABLE 1

Coal Equivalent of World Vegetation*
(Coal at 13,000 Btu/lb)

	A	B
Fuel (direct)	5800	2.76
Alcohol	2940	1.40
Gasoline	2450	1.17

A = Equivalent million tons coal
B = Times yearly world requirement

* Source, *Energy Sources—The Wealth of the World*, Ayres and Scarlott, 1952

The recoverable reserves in the U.S. of coal, oil, natural gas, and shale as of January 1, 1960, reported

TABLE 2

Solar Energy Annually Available to the World from Renewable Sources*
(Equivalent million tons coal @ 13,000 Btu/lb)

	A	B
Solar collectors for power	1,700,000	trace
Waterfalls	35,300	98
Vegetation	5,900	294
Solar collectors for space heating†	882	trace
Wind	196	trace
Totals	1,742,278	392

A = Maximum available
B = Approximate current use

Source of data, *Energy Sources—The Wealth of the World*, Ayres and Scarlott, 1952

* Fissionable elements and fossil fuels are not considered renewable.

† Present energy requirements for space heating.

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TABLE 3

United States Reserves and Production of All Fossil Fuels

(Coal, billion tons; petroleum and shale oil, billion barrels; natural gas, trillion cu ft)

	All Mineral Fuels		Bituminous Coal Equivalent* (13,000 Btu per lb)		
	Reserves† Jan. 1, 1960	Annual Production, 1960	Reserves Billion Tons, 1960	Reserves Per Cent of Total	Annual Production, 1960
Coal:					
Anthracite	7.7	0.018	7.5	0.99	0.018
Bituminous coal and lignite	822.4	0.416	664.2	87.26	0.418
Total, all ranks	830.1	0.434	671.7	88.25	0.436
Crude petroleum	31.7	2.6	7.1	0.93	0.575
Oil from oil shale	324.0	negligible	72.2	9.49	negligible
Natural gas	262.6	13.1	10.1	1.33	0.504
			761.1	100.00	1.515

*Heating values for conversions are: anthracite, 12,700 Btu/lb; bituminous coal and lignite equivalent, 10,500 Btu/lb; petroleum and shale oil, 5,800,000 Btu/bbl; natural gas, 1,000 Btu/cu ft.

† The quantities noted under reserves are: coal, recoverable reserves assumed as ½ of the total reserves as reported by the Bureau of Mines; petroleum, proved reserves as of Dec. 31, 1959; oil from oil shale assumed to be 60% of total estimated reserves in place from deposits averaging not less than 25 gallons of oil a ton; natural gas, proved reserves as of Dec. 31, 1959.

by the U.S. Bureau of Mines, are given in Table 3. In equivalent tons of 13,000 Btu coal, the total is 761.1 billion tons, of which coal represents 88.25%, petroleum 0.93%, oil shale 9.49%, and natural gas 1.33%. On the basis of the 1960 production of 1.5 billion tons of equivalent 13,000 Btu/lb coal, these reserves would last about 700 years.

Information from the United States Atomic Energy Commission indicates that the heat energy available from uranium reserves in the U.S. is equivalent, with breeder reactors, to 35 quintillion (35×10^{19}) Btu exclusive of heat energy from thorium and thermonuclear reactions. In accordance with Table 3, the heat energy from U.S. coal reserves is equivalent to 17.5 quintillion Btu and that from all fossil reserves 20 quintillion Btu. However, were the uranium used in converter instead of breeder reactors, the energy available would be less than that from the coal reserves.

World coal production and consumption figures for 1960, arranged in the order of the ten leading countries in each category, are given in Table 4 as are the world coal reserves, arranged quantitatively. According to these figures the U.S. has 34.6% of the world's coal supply. It is of interest to note that 80% of the world's coal lies within three geographical areas—the United States, Russia, and China—and that China has taken the lead in both the production and the consumption of coal, followed closely by the U.S.S.R.

TABLE 4

World Coal Reserves, Production, and Consumption by Countries in 1960*

(Million Tons)

Total Reserves		%	Production, 1960		Consumption, 1960	
U.S.S.R.	1,763,200	35.5	China†	476.9	China†	475.9
United States‡	1,720,000	34.6	U.S.S.R.	440.7	U.S.S.R.	430.4
China	495,900	10.0	United States	432.4	United States	392.4
Germany	315,200	6.4	Germany	268.4	Germany	262.4
United Kingdom	187,300	3.8	United Kingdom	216.9	United Kingdom	216.8
Canada	98,400	2.0	Poland	118.2	Poland	96.5
Poland	88,000	1.8	Czechoslovakia	67.5	France	75.9
India	66,100	1.3	France	63.2	Japan	68.6
Australia	64,500	1.3	India	58.1	Czechoslovakia	65.4
Union of S. Africa	61,000	1.2	Japan	56.8	India	55.7
Others	104,400	2.1	Others	229.8	Others	299.3
	4,964,000	100.0		2428.9		2439.3

* Source—*United Nations Statistical Paper*, Series J, No. 5, 1962.

† Communist China, including China mainland, North Korea, North Vietnam, and Mongolia.

‡ Includes Alaska.

United States Reserves and Production

Coal reserves in the U.S. have been estimated from drillings, outcroppings, gorges cut by rivers, and explorations for minerals and oil. The distribution of the coal reserves by states, the rank, the amount produced in 1960, and the value per ton, fob mine, are given in Table 5.

The ten states with the highest recoverable coal reserves as of Jan. 1, 1960, and the ten states with the highest coal production in 1960 are listed in Table 6.

In the U.S. the two great coal-producing regions are the Appalachian, including the states of Pennsylvania, West Virginia, Ohio, western Maryland, eastern Kentucky, Virginia, Tennessee, and Alabama, and the Central States region including Illinois, Indiana, western Kentucky, Iowa, Missouri, western Kansas, Oklahoma, and Arkansas.

TABLE 5

United States Coal Reserves by States, 1960 Production, and Value per Ton

(Million Tons)

	Estimated Original Reserves (1)							
State	Bituminous	Sub-bituminous	Lignite	Anthracite (3)	Total	Recoverable Reserves (2) Jan. 1, 1960	Production, 1960	Avg $/ Ton fob Mine, 1960
Alabama	13,754	0	20	0	13,774	6,864	13.0	7.10
Alaska	21,401	71,136	(4)	2,101	94,638	47,306	0.7	8.75
Arkansas	1,816	0	350	456	2,622	1,212	0.4	7.61
Colorado	63,203	18,492	0	90	81,785	40,387	3.6	5.85
Georgia	100	0	0	0	100	38	0.04	5.00
Illinois	137,329	0	0	0	137,329	68,190	46.0	4.00
Indiana	37,293	0	0	0	37,293	17,499	15.5	3.96
Iowa	29,160	0	0	0	29,160	14,223	1.0	3.60
Kansas	20,774	0	(3)	0	20,774	10,374	0.9	4.73
Kentucky	72,318	0	0	0	72,318	33,513	66.8	4.22
Maryland	1,200	0	0	0	1,200	594	0.7	3.74
Michigan	297	0	0	0	297	102	—	—
Missouri	79,362	0	0	0	79,362	39,394	2.9	4.31
Montana	2,363	132,151	87,553	0	222,047	110,853	0.3	3.79
New Mexico	10,948	50,801	0	6	61,755	30,753	0.3	5.93
North Carolina	112	0	0	0	112	55	—	—
North Dakota	0	0	350,910	0	350,910	175,359	2.5	2.29
Ohio	46,488	0	0	0	46,488	21,192	34.0	3.85
Oklahoma	3,673	0	(3)	0	3,673	1,656	1.3	6.79
Oregon	20	180	0	0	200	97	—	—
Pennsylvania								
Anthracite	—	0	0	22,805	22,805	6,189	18.8	7.82
Bituminous	75,093	0	0	—	75,093	29,252	65.4	5.29
South Dakota	0	0	2,033	0	2,033	1,015	0.02	4.08
Tennessee	1,912	0	0	0	1,912	950	5.9	3.57
Texas	8,000	0	7,070	0	15,070	7,440	—	—
Utah	28,222	156	0	0	28,378	13,929	4.9	6.35
Virginia	11,696	0	0	355	12,051	5,244	27.8	4.41
Washington	11,413	52,442	(4)	23	63,878	31,790	0.2	7.54
West Virginia	116,618	0	0	0	116,618	51,940	118.9	5.02
Wyoming	13,235	108,319	(4)	0	121,554	60,375	2.0	3.45
Other States	620	4,065	50	0	4,735	2,360	0.47	—
Total	808,420	437,742	447,966	25,836	1,719,964	830,145	434.33	5.12

(1)—Source, Bureau of Mines.
(2)—Assuming 50% recovery.
(3)—Some thin seam deposits.
(4)—Small reserves of lignite included with subbituminous

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TABLE 6

Recoverable Coal Reserves, Jan. 1, 1960
(Million Tons)

In the Ten Leading States of the U.S.

North Dakota	175,359
Montana	110,853
Illinois	68,190
Wyoming	60,375
West Virginia	51,940
Alaska	47,306
Colorado	40,387
Missouri	39,394
Pennsylvania	35,441
Kentucky	33,513
Total	662,758
Total in U.S.	830,145

Coal Production, 1960
(Million Tons)

In the Ten Leading States of the U.S.

West Virginia	118.9
Pennsylvania	84.2
Kentucky	66.8
Illinois	46.0
Ohio	34.0
Virginia	27.8
Indiana	15.5
Alabama	13.0
Tennessee	5.9
Utah	4.9
Total	417.0
Total in U.S.	434.3

Source—Bureau of Mines

However, two thirds of the coal reserves lie in the Great Plains, the Rocky Mountains, and the West. These coals are subbituminous and lignite and, at some future time, among other uses will probably serve as a source of synthetic oil. Although they are not now widely used because of the distance of the fields from industrial areas, their development, nevertheless, for use as electric utility fuels and as a source of synthetic liquid fuels, is being actively prosecuted.

It is estimated that 32 states have 340,000 square miles underlaid with coal. This represents 11% of the total area of the U.S.

Consumption of Coal by Industries

In 1960, as shown in Table 7, the largest consumption of coal, 45.7%, was by electric utilities. This is a tremendous increase since 1936-40, when electric utilities used only 10.4% of the total coal consumed. In the five-year period 1956-60, electric utility consumption of coal grew from 114 million to 174 million tons. It has been estimated that by 1965 consumption will approach 225 million tons.

Second to electric utilities in 1960 was the production of coke, gas, and steel which consumed 23.2%, followed closely by "other industries" using 22.3%.

The greatest drop in coal consumption occurred in the railroad industry, which used 20.8% of the coal consumed in the period 1936-40 and only 0.6% of the total consumed in 1960. This reflects the effect of the widespread conversion to diesel motive power. Practically all transportation in the U.S. today receives its energy from oil.

The amount of all fuels used in recent years in electric utilities, the per cent of total, and the per cent of steam and hydroelectric power are given in Table 8.

TABLE 7

United States Coal Consumption by Industries
(Bituminous and Lignite, Thousand Tons)

Year, Avg. for Range	Electric Utilities	Coke, Gas, and Steel	Other Industries	Retail Dealers	Railroads, All Uses	Foreign Trade	Total Consumption
1936-1940	41,304	81,805	112,569	75,266	82,519	2901	396,373
1941-1945	69,131	114,492	136,573	111,615	120,049	3167	555,027
1946-1950	83,849	110,484	116,711	90,989	88,678	2474	493,185
1951-1955	114,655	112,139	99,948	61,207	30,509	1728	420,186
1956-1960	160,995	97,231	90,880	35,908	5,827	1141	391,986
1960	173,882	88,393	84,703	30,405	2,101	945	380,429
Percentage Change from (1936-40) to (1956-60)							
1936-1940	+320.9	+ 8.0	−24.8	−59.6	−97.4	−67.5	− 4.0
1941-1945	+151.5	−22.8	−38.0	−72.8	−98.2	−70.2	−31.5
1946-1950	+107.4	−20.0	−27.4	−66.6	−97.6	−61.8	−22.9
1951-1955	+ 51.7	−21.1	−13.1	−50.3	−93.1	−45.3	− 9.5
1956-1960	+ 8.0	− 9.1	− 6.8	−15.3	−63.9	−17.2	− 2.9

Percentage Coal Consumption by Industries

Industry	1960	1936-1940
Electric utilities	45.7	10.4
Coke, gas & steel	23.2	20.6
Other industries	22.3	28.5
Retail sales	8.0	19.0
Railroads	0.6	20.8
Foreign trade	0.2	0.7
Total, %	100.0	100.0

Source—Bureau of Mines

How Coal was Formed

Some of the luxurious vegetation produced by the sun's heat in early geological times has become the coal of today. Great trees and ferns grew in the hot humid climate, and successive layers of fallen vegetation, partly decomposed, piled one upon another and prevented free access of air. At intervals, wind and water covered this mass with sediment, sand, and dirt to form the overburden now found above coal seams. During millions of years, these layers of vegetation were aged by heat and pressured into coal by converting carbohydrates, $C_x(H_2O)_y$, to hydrocarbons, C_xH_y.

The progressive changes were from wood and vegetation to peat, which in turn was changed by heat and pressure to brown coals and lignite, subbituminous coals, and finally to anthracite and graphite.

The intrinsic ash of the coal was largely in the vegetation from the beginning. The extraneous ash came from mud deposits of shale and pyrites that filled the cracks and crevices caused by the movement of the earth's crust.

It has been estimated that about 20 ft of plant matter were required to form a 1-ft layer of bituminous coal. Most of the coal mined in the U.S. comes from seams 3 to 6 ft thick, with the average about 5.4 ft. One seam in Lincoln County, Wyoming, is 118 ft thick. The thickest known seam in the world, 425 ft, is in Manchuria.

Sampling of Coal

The first scientific work in coal sampling was published by E. G. Bailey in the *Journal of Industrial and Engineering Chemistry*, March 1909. The size of the sample that must be taken was based on the ratio of the size to weight of the maximum size lump present in the coal being sampled. The rules for reducing the sample to laboratory size were established in the same manner. The first ASTM Standard for Sampling Coal (D 21), now obsolete, was based largely upon Bailey's work. The method was calculated to give an accuracy within 1% of the ash in the coal in 10,000 samples.

The early standard, requiring a fixed-size sample, came to be regarded as too laborious, and the ASTM Standard D 492 now in use was developed and

TABLE 8

Electric Energy Production in the United States by Source

Year	Fuels Consumed, Bituminous Coal Equivalent° (1000 Ton) Coal	Oil	Gas	Total	Kind of Fuel, Per Cent of Total Coal	Oil	Gas	Steam & Hydro, Per Cent of Each All Fuels	Hydro
'40-'49†	81,158	9,782	15,969	106,909	75.9	9.1	15.0	67.3	32.7
1950	91,871	19,338	26,314	137,523	66.8	14.1	19.1	70.9	29.1
1951	105,768	16,395	31,961	154,124	68.6	10.6	20.8	73.1	26.9
1952	107,071	17,235	38,079	162,385	65.9	10.6	23.5	73.7	26.3
1953	115,897	21,086	43,274	180,257	64.3	11.7	24.0	76.2	23.8
1954	118,385	17,113	48,764	184,262	64.3	9.3	26.4	77.3	22.7
1955	143,759	19,300	48,253	211,312	68.0	9.1	22.9	79.3	20.7
1956	158,279	18,643	51,853	228,775	69.2	8.2	22.6	79.7	20.3
1957	160,769	20,433	55,904	237,106	67.8	8.6	23.6	79.4	20.6
1958	155,724	19,914	57,440	233,078	66.8	8.5	24.7	78.3	21.7
1959	168,427	22,654	68,077	259,158	65.0	8.7	26.3	80.6	19.4
1960	176,634	21,882	72,165	270,681	65.2	8.1	26.7	80.7	19.3
1961	182,140	21,962	76,374	280,476	65.0	7.8	27.2	80.8	19.2

Source of data, Federal Power Commission

° 3.9 bbl of fuel oil and 23,900 cu ft of gas, each taken as equivalent to 1 ton of bituminous coal.

† Average.

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adopted in 1948. In this standard, allowances are made for the probable ash content of the coal, thus permitting the use of smaller samples for the lower-ash coals.

The present standard recognizes two sampling procedures:

1. Commercial-Sampling Procedure
2. Special-Purpose-Sampling Procedure

The "Commercial-Sampling Procedure" applies to the average commercial sampling of coal and is intended for an accuracy such that, if a large number of samples were taken from a single lot of coal, the test results in 95 out of 100 cases would fall within ± 10% of the average ash content of the samples.

The "Special-Purpose-Sampling Procedure" applies to the sampling of coal when special accuracy is required. This procedure should be used to supply samples for the classification of coals and the establishment of design or performance parameters. The special-purpose sample can be one of two sizes: 1) 4 times that of the commercial sample, giving an accuracy of ± 5% of the ash content of the coal sampled, and 2) 9 times that of the commercial sample, giving an accuracy of ± 3.33% of the ash content of the coal sampled.

Standard D 492 also sets procedures for reducing gross samples and for obtaining samples for standard and special moisture determinations. Of particular interest to those concerned with the sampling of coal are two additional publications by the ASTM: *Symposium on Bulk Sampling* (STP 242, 1958) and *Symposium on Coal Sampling* (STP 162, 1955).

It is of prime importance that coal samples be carefully obtained, since any data resulting from subsequent analyses are only as representative as the sample provided.

Coal Analysis

In reporting the components of a coal it is customary to use two different analyses, known as "proximate" and "ultimate." The proximate analysis is defined as the determination of moisture, volatile matter, and ash, and the calculation of fixed carbon by difference. The ultimate analysis from a dried sample is defined as the determination of carbon, hydrogen, sulfur, nitrogen, and ash, and the estimation of oxygen by difference. The scope of each is indicated in the analyses of a West Virginia coal, as noted below, where the ultimate analysis has been converted to the as-received basis. The analysis on the as-received basis includes the total moisture content of the coal that is received at the plant; similarly, the as-fired basis includes the total moisture content of the coal as it enters the boiler furnace or pulverizers. The various values from such analyses are needed to implement the formulas in this chapter and in Chapter 4. Standard laboratory procedures for making these analyses appear in ASTM D 271.

Coal Analyses

On As-Received Basis
(Pittsburgh Seam Coal, West Virginia)

Proximate Analysis		Ultimate Analysis	
Component	Weight, %	Component	Weight, %
Moisture	2.5	Moisture	2.5
Volatile matter	37.6	Carbon	75.0
Fixed carbon	52.9	Hydrogen	5.0
Ash	7.0	Sulfur	2.3
Total	100.0	Nitrogen	1.5
		Oxygen	6.7
		Ash	7.0
Heating value, Btu/lb	13,000	Total	100.0

Classification by Rank

Coals are classified in an effort to provide the data for predicting their probable performance under given operating conditions. Coal performance, particularly methods and principles of burning, is closely associated with specific types of equipment, and reference should be made to Chapters 4, 16, 17, 18, and 28. The nature of impurities in coal is discussed in Chapter 15, and reference to it should also be made.

One classification of coal is by rank or age. Volatile matter, fixed carbon, bed moisture, and oxygen are all indicative of rank, but no one item completely defines it. In the ASTM classification, the basic criteria are the fixed carbon and the calorific values calculated on a mineral-matter-free basis.

In establishing the rank of coals it is necessary to use criteria which show an appreciable and systematic variation with age. For the older coals, a good criterion is the dry, mineral-matter-free fixed carbon or volatile. But this criterion is not suitable for designating the rank of the younger coals. A dependable means of classifying the latter is the moist, mineral-matter-free Btu, which varies little for the older coals but appreciably and systematically for the younger coals.

By the ASTM standard (see Table 9) for classification according to age, older coals are classified by "dry, mineral-matter-free fixed carbon," and younger coals are classified by "moist, mineral-matter-free Btu." The term "moist" refers to bed moisture only, and analyses of constituents must be of bed samples collected as prescribed by the ASTM Standard D 388. Agglomerating and weathering (or slacking) indices are used to differentiate between certain adjacent groups and are defined in this standard. The Parr formulas, given below, are used for calculating these values from the proximate analysis and calorific value (high heating value).

Table 9

Classification of Coals by Rank,[a] ASTM D 388

FC = Fixed Carbon; VM = Volatile Matter; Btu = British Thermal Units

Class	Group	Limits of Fixed Carbon or Btu Mineral-Matter-Free Basis	Requisite Physical Properties
I Anthracitic	1. Meta-anthracite	Dry FC, 98% or more. (Dry VM 2% or less)	
	2. Anthracite	Dry FC, 92% or more and less than 98%. (Dry VM, 8% or less and more than 2%)	
	3. Semianthracite	Dry FC, 86% or more and less than 92%. (Dry VM, 14% or less and more than 8%)	Nonagglomerating[b]
II Bituminous[c]	1. Low-volatile bituminous	Dry FC, 78% or more and less than 86%. (Dry VM, 22% or less and more than 14%)	
	2. Medium-volatile bituminous	Dry FC, 69% or more and less than 78%. (Dry VM, 31% or less and more than 22%)	
	3. High-volatile A bituminous	Dry FC, less than 69%. (Dry VM, more than 31%); and moist[d] Btu, 14,000 or more	
	4. High-volatile B bituminous	Moist[d] Btu, 13,000 or more and less than 14,000[e]	
	5. High-volatile C bituminous	Moist Btu, 11,000 or more and less than 13,000[e]	Either agglomerating or nonweathering[f]
III Subbituminous	1. Subbituminous A	Moist Btu, 11,000 or more and less than 13,000[e]	Both weathering and nonagglomerating
	2. Subbituminous B	Moist Btu, 9,500 or more and less than 11,000[e]	
	3. Subbituminous C	Moist Btu, 8,300 or more and less than 9,500[e]	
IV Lignitic	1. Lignite	Moist Btu, less than 8,300	Consolidated
	2. Brown coal	Moist Btu, less than 8,300	Unconsolidated

[a]Does not include a few coals of unusual physical and chemical properties which come within the limits of fixed carbon or Btu of the high-volatile bituminous and subbituminous ranks. [b]If agglomerating, classify in low-volatile group of the bituminous class. [c]There may be noncaking varieties in each group of the bituminous class. [d]Moist Btu refers to coal containing only its natural bed moisture. [e]Coals having 69 per cent or more fixed carbon on the dry, mineral-matter-free basis shall be classified according to fixed carbon regardless of Btu. [f]There are three varieties in the high-volatile C bituminous coal group, 1) agglomerating and nonweathering, 2) agglomerating and weathering, and 3) nonagglomerating and nonweathering.

$$\text{Dry, Mm-free FC} = \frac{\text{FC} - 0.15\text{S}}{100 - (\text{M} + 1.08\text{A} + 0.55\text{S})} \times 100, \text{ per cent} \quad (1)$$

$$\text{Dry, Mm-free VM} = 100 - \text{Dry, Mm-free FC, per cent} \quad (2)$$

$$\text{Moist, Mm-free Btu} = \frac{\text{Btu} - 50\text{S}}{100 - (1.08\text{A} + 0.55\text{S})} \times 100, \text{ per lb} \quad (3)$$

The approximate formulas used to calculate the values given in Table 10 are:

$$\text{Dry, Mm-free FC} = \frac{\text{FC}}{100 - (\text{M} + 1.1\text{A} + 0.1\text{S})} \times 100, \text{ per cent} \quad (4)$$

$$\text{Dry, Mm-free VM} = 100 - \text{Dry, Mm-free FC, per cent} \quad (5)$$

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$$\text{Moist, Mm-free Btu} = \frac{\text{Btu}}{100 - (1.1A + 0.1S)} \times 100, \text{ per lb} \qquad (6)$$

where:

Mm = mineral matter
Btu = heating value per lb
FC = fixed carbon, %
VM = volatile matter, %
M = bed moisture, %
A = ash, %
S = sulfur, %
All for coal as received (moist basis).

In Table 10, seventeen selected U.S. coals are arranged in the order of the ASTM classification, with the values for the two criteria for the older and younger coals in italics. The bases for the two ASTM classification criteria are shown in Fig. 1 (from U.S. Bureau of Mines, R.I. 3296) for over 300 typical coals of the United States. The groups of Table 9 are indicated in this figure.

Validity of Other Criteria

It may be of interest to consider the validity of other criteria in the classification of coal by rank. Referring to the ultimate analysis, dry, ash-free basis, Table 11:

For the younger coals, the volatile (VM) either "as received" (Table 10) or "dry, ash-free" (Table 11) does not line up the coals in anything like the same order as the ASTM method.

Carbon (C), however, does line up the 17 coals in the same order as the ASTM method except for a slight discontinuity between coals 8 and 9, both in the high-volatile A group. As a criterion, the carbon content would be quite simple and could undoubtedly be used as a suitable basis for classification.

Hydrogen (H_2) is of no use in classification.

Oxygen (O_2) generally decreases with the age of coals but is not consistent, and higher values are indicated for oxidized coal.

Nitrogen (N_2) varies little for all ranks.

TABLE 10

Seventeen Selected U.S. Coals Arranged in Order of ASTM Classification

Coal Samples Rank Classification					Coal As Received — Proximate Analysis, %						Mm Free Dry	Mm Free Moist
No.	Class	Group	State	County	M	VM	FC	A	S	Btu	FC	Btu
1	I	1	Pa.	Schuylkill	2.0	1.8	86.2	10.0	0.79	13,070	99.2	14,720
2	I	2	Pa.	Lackawanna	2.0	6.3	79.7	12.0	0.60	13,000	*94.0*	15,000
3	I	3	Va.	Montgomery	3.0	10.5	66.5	20.0	0.61	11,800	*88.7*	15,140
4	II	1	W.Va.	McDowell	3.0	16.3	75.7	5.0	0.73	14,420	*82.8*	15,250
5	II	1	Pa.	Cambria	4.0	17.0	69.0	10.0	1.63	13,430	*81.1*	15,130
6	II	2	Pa.	Somerset	3.0	20.4	66.6	10.0	1.65	13,520	*77.4*	15,230
7	II	2	Pa.	Indiana	3.0	23.1	63.9	10.0	2.17	13,600	*74.5*	15,340
8	II	3	Pa.	Westmoreland	3.0	30.3	55.7	11.0	1.80	13,130	*65.4*	15,000
9	II	3	Ky.	Letcher; Pike	3.0	34.4	56.6	6.0	0.72	13,800	*62.5*	14,800
10	II	3	Ohio	Jefferson	6.0	34.8	49.2	10.0	2.44	12,450	*59.2*	14,030
11	II	4	Ill.	Saline; Perry	10.0	31.7	48.3	10.0	1.60	11,610	61.1	*13,100*
12	II	4	Utah	Carbon; Emery	8.0	36.6	43.4	12.0	0.56	11,480	54.9	*13,230*
13	II	5	Ill.	Sangamon	14.0	34.3	39.7	12.0	4.07	10,470	54.6	*12,100*
14	III	1	Iowa	Polk	13.9	36.9	35.2	14.0	6.15	10,244	50.0	*12,200*
15	III	2	Col.	Weld; Boulder; Sheridan	24.0	30.2	40.8	5.0	0.36	9,200	57.7	*9,750*
16	III	3	Wyo.	Campbell	24.0	30.0	36.0	10.0	0.33	8,450	55.1	*9,500*
17	IV	1	N.D.	McLean; Morton	40.0	27.6	23.4	9.0	1.42	6,330	46.8	*7,040*

NOTES: For definition of Rank Classification according to ASTM requirements, see Table 9.

Data on Coal As Received

M = bed moisture, %; VM = volatile matter, %
FC = fixed carbon, %; A = ash, %; S = sulfur, %
Btu = Btu per lb

Mm-free, dry FC = Dry, mineral-matter-free fixed carbon in %, calculated by formula:

$$\frac{FC \times 100}{100 - (M + 1.1A + 0.1S)}$$

Mm-free, moist Btu = Moist, mineral-matter-free Btu per lb, calculated by formula:

$$\frac{\text{Btu} \times 100}{100 - (1.1A + 0.1S)}$$

Values in italics for Mm-free, dry FC and Mm-free, moist Btu are used in ASTM classification by rank.

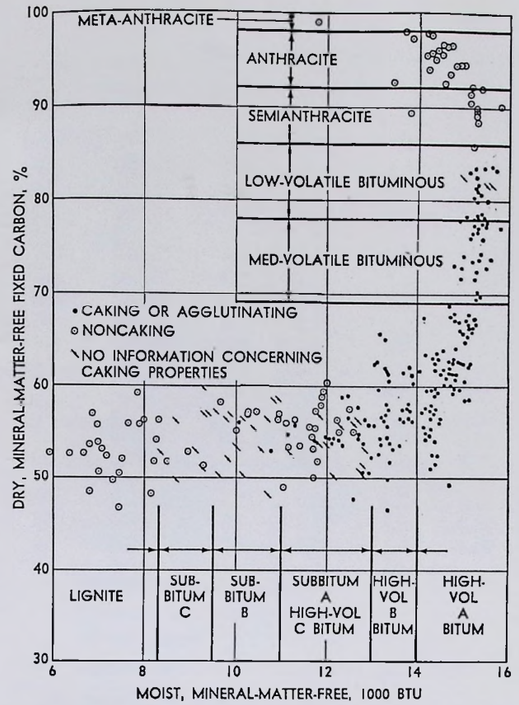


Fig. 1. *Distribution plot for over 300 coals of the United States, illustrating ASTM classification by rank as defined in Table 9*

International Classification of Coal by Rank

There are classifications of coal by rank (or type) which are currently in limited use on the European continent. These are the International Classification of Hard Coals by Type and the International Classification of Brown Coals. The systems were developed by a Classification Working Party that was established in 1949 by the Coal Committee of the Economic Commission for Europe.

The classification system of hard coals, which also includes a simplified statistical grouping combining coals having the same general characteristics, appears in Table 12 (see Appendix, page 2-A2 ff). The term "hard coal" is based on European terminology and is defined as coal with a calorific value of more than 10,260 Btu/lb (5700 Kg-cal/Kg) on the moist, ash-free basis. The term "type" is equivalent to rank in American coal-classification terminology and the term "class" approximates the ASTM rank.

The nine classes of coal, based on dry ash-free volatile-matter content and moist ash-free calorific value, are divided into groups according to their caking properties. Either the free-swelling test or the Roga test may be used to determine caking properties, which are a measure of the behavior of coal when heated rapidly.

The coal groups are further subdivided into subgroups according to their coking properties. Either the Audibert-Arnu test or the Gray-King coking test may be used to determine coking properties, which are a measure of the behavior of coal when heated slowly.

A three-figure code number is used to express the classification of a coal. The first figure indicates the class of the coal, the second figure the group, and the third figure the subgroup. For example, a 635 coal would be a class six coal with a free-swelling index greater than 4 and expansion greater than 140.

This rank classification is about the same as ASTM classification except for determination of calorific-value correction. The International system uses the ash-free basis, whereas the ASTM system uses the mineral-matter-free basis. American coals fit in the International classification as shown by the open-faced blocks in Table 12 (see Appendix 2-A2 ff), based on available data from U.S. Bureau of Mines R.I. 5435. A general comparison of the systems with several American coals is shown in Table 13.

The International Classification of Brown Coals is shown in Table 14. The term "brown coal" refers to coal containing less than 10,260 Btu/lb on the moist, ash-free basis.

The six classes of coal, based on the ash-free equilibrium-moisture content, are divided into groups according to their tar yield on a dry, ash-free basis. This grouping indicates the value of these low-rank coals as a fuel and as a raw material for chemical purposes. In general, brown coals with a high tar content are too valuable to the chemical industry as a raw material for them to be used as fuel.

Other Criteria

Other criteria for the classification of coal by rank have been proposed by various authorities.

A method of classifying coal, by computing the heating value of the residual coal with moisture, ash, and sulfur removed, has been used by Lord. This is called the "H" value.

$$\text{"H"} = \frac{\text{Btu} - 4050S}{100 - (M + A + S)} \times 100 \qquad (7)$$

While the "H" values as listed in Table 11 are of interest, they do not arrange the 17 coals in the same order as the ASTM.

Another method of classification has been reported by Perch and Russell based on the following ratio:

$$\text{Ratio} = \frac{\text{Moist, Mm-free Btu}}{\text{Dry, Mm-free VM}} \qquad (8)$$

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TABLE 11

Study of the Suitability of Other Criteria in the Classification of Coals

Coals Table 10 No.	Btu Lord's "H" Value	Perch & Russell Ratio	Btu per lb VM_{pc}	Proximate Analysis VM	Proximate Analysis FC	Ultimate Analysis Dry, Ash Free C	H_2	O_2	N_2	Btu	Grindability
1	15,000	7250	—	2.0	98.0	93.9	2.1	2.3	0.3	14,850	37
2	15,170	2020	28,400	7.3	92.7	93.5	2.6	2.3	0.9	15,100	26
3	15,450	1110	24,700	13.6	86.4	90.7	4.2	3.3	1.0	15,320	83
4	15,800	855	22,000	17.7	82.3	90.4	4.8	2.7	1.3	15,670	100
5	15,870	757	21,750	19.8	80.2	89.4	4.8	2.4	1.5	15,600	112
6	15,750	643	20,000	23.5	76.5	88.6	4.8	3.1	1.6	15,540	105
7	15,950	572	19,850	26.5	73.5	87.6	5.2	3.3	1.4	15,630	95
8	15,580	423	17,350	35.4	64.6	85.0	5.4	5.8	1.7	15,260	88
9	15,290	389	16,650	37.8	62.2	85.2	5.4	7.0	1.6	15,160	56
10	15,180	335	15,800	41.5	58.5	82.2	5.5	7.7	1.7	14,820	61
11	14,750	328	15,140	39.6	60.4	80.6	5.4	10.3	1.7	14,510	66
12	14,470	287	14,550	45.8	54.2	80.3	5.7	11.7	1.6	14,350	45
13	14,850	257	14,550	46.3	53.7	77.5	5.4	10.2	1.4	14,160	60
14	15,150	232	14,640	51.2	48.7	75.8	7.7	26.0	1.2	14,210	62
15	13,000	228	11,000	42.5	57.5	75.0	5.1	17.9	1.5	12,960	58
16	12,890	206	11,100	45.3	54.7	74.1	5.1	18.7	1.3	12,800	55
17	12,740	128	10,900	54.0	46.0	72.4	4.7	18.6	1.5	12,410	—

NOTES:

$$\text{Btu "H" value} = \frac{\text{Btu} - 4050S}{100 - (M + A + S)} \times 100$$

where Btu, M, A, and S are as given in Table 10 for the coal as received.

$$\text{Perch \& Russell Ratio} = \frac{\text{Moist, Mm-free Btu}}{\text{Dry, Mm-free VM}}$$

where Moist, Mm-free Btu is as given in Table 10. In this formula also, VM on a dry, ash-free basis may be used without appreciable error.

$$\text{Btu/lb } VM_{pc} = \frac{\text{Btu}_{pc} - \left\{\frac{14{,}460}{100} \times FC_{pc}\right\}}{VM_{pc}} \times 100$$

where the subscript pc means on a "pure coal" basis. (See text for conversion of Btu, FC, and VM of coal as received to "pure coal" basis.)

Grindability is determined by the ASTM method (see Page 17-13).

The values of this ratio for the 17 coals listed in Table 11 arrange these coals in exactly the same order as the ASTM method. It is said that this "ratio" is closely related to the coking properties of coals used in coke-oven practice.

Heating Value of the Volatile

Since the quality of the volatile matter in coal is an index of the extent of the conversion of the original carbohydrates to hydrocarbons, studies have been made suggesting that the heating value of the volatile matter will serve as an accurate criterion for classification by rank. This criterion, developed on a "pure coal" basis, is listed in Table 11 under Btu/lb volatile matter (VM), and its derivation is explained below.

Composition of the fixed carbon in all types of coal is substantially all carbon. The variable constituents of coals can, therefore, be considered as concentrated in the volatile matter. One index of the quality of the volatile matter, its heating value, is perhaps the most important property as far as combustion is concerned, and it bears a direct relation to the properties of the pure coals (dry, mineral-matter-free). The volatile matter in the coals of lower rank, where the conversion of carbohydrates to hydrocarbons has not progressed far, is relatively high in water and CO_2 and, consequently, low in heating value. In coals of higher rank, the volatile matter is relatively high in hydrocarbons, such as methane (CH_4), and, consequently, relatively high in heating value.

TABLE 13

System Comparison of Selected American Coals

State	County	Seam	Classification ASTM	Classification International
W.Va.	McDowell	Pocahontas #3	Lvb	333
Pa.	Clearfield	Lower Kittaning	Mvb	435
W.Va.	Monongalia	Pittsburgh	Hvab	635
Ill.	Williamson	No. 6	Hvbb	734
Ill.	Knox	No. 6	Hvcb	821

Lvb = low-volatile bituminous
Mvb = medium-volatile bituminous
Hvab = high-volatile A bituminous
Hvbb = high-volatile B bituminous
Hvcb = high-volatile C bituminous

TABLE 14

International Classification of Brown Coals
(Gross Calorific Value Below 10,260 Btu/lb)*

Group No.	Group Parameter Tar Yield (dry, ash-free), %	Code Number					
40	>25	1040	1140	1240	1340	1440	1540
30	>20-25	1030	1130	1230	1330	1430	1530
20	>15-20	1020	1120	1220	1320	1420	1520
10	>10-15	1010	1110	1210	1310	1410	1510
00	10 & less	1000	1100	1200	1300	1400	1500

	Class Number	10	11	12	13	14	15
Class Parameter	Total Moisture, % (Ash Free)	20 and less	>20 –30	>30 –40	>40 –50	>50 –60	>60 –70

NOTES: The total-moisture content refers to freshly mined coal. For internal purposes, coals with a gross calorific value over 10,260 Btu/lb (moist, ash-free basis), considered in the country of origin as brown coals but classified as hard coals for international purposes, may be classified under this system, to ascertain, in particular, their suitability for processing.

When the total-moisture content is over 30%, the gross calorific value is always below 10,260 Btu/lb.

*Moist, ash-free basis (86 F/96% relative humidity).

To establish reasonably accurate heating values for the volatile matter, the analyses and heating value of the coal must be converted to the mineral-matter-free basis. The only difference in the conversion used for this method and the conversion used in the ASTM Standard D 388 is that half the sulfur is assumed to be pyritic. The assumption that only half the sulfur is pyritic and the remainder is organic is in closer agreement with the average for a large number of U.S. coals. The formulas for converting the analyses and coal heating values to the "pure coal" basis and for calculating the heating value of the volatile matter are given below:

$$VM_{pc} = \frac{VM - (0.08A + 0.2S)}{100 - (1.08A + 0.2625S + M)} \times 100, \% \quad (9)$$

$$FC_{pc} = \frac{FC - 0.0625S}{100 - (1.08A + 0.2625S + M)} \times 100, \% \quad (10)$$

$$\text{Btu}_{pc} = \frac{\text{Btu} - 26.2S}{100 - (1.08A + 0.2625S + M)} \times 100, \text{ per lb} \quad (11)$$

$$\text{Btu/lb } VM_{pc} = \frac{\text{Btu}_{pc} - \left\{\frac{14{,}460}{100} \times FC_{pc}\right\}}{VM_{pc}} \times 100 \quad (12)$$

where:

VM, FC, A, S, M, and Btu are the same as noted above for the Parr formulas. Subscript pc denotes "pure coal" basis.

The Btu per lb volatile matter, as calculated from the above formulas, is given in Table 11 for 16 of the 17 coals listed covering the entire range of rank. It is not given for the first coal listed, since this formulation may not be entirely suitable for extremely low-volatile fuels. For only two coals, Nos. 14 and 15, Class III, Subbituminous A and B, Groups 1 and 2, is the order substantially different from the ASTM arrangement, Table 10. The range of values of Btu per lb volatile matter, from 10,900 to 28,400, is large and should serve as a useful classification.

The relation of the heating value of the volatile matter and the heating value of the pure coal is shown in Fig. 2 for a large number of coals. It is evident that a fair line could be drawn, without serious error, to indicate the path of this relationship.

A listing of ASTM standards for testing coal will be found in the Appendix, page 2-A1.

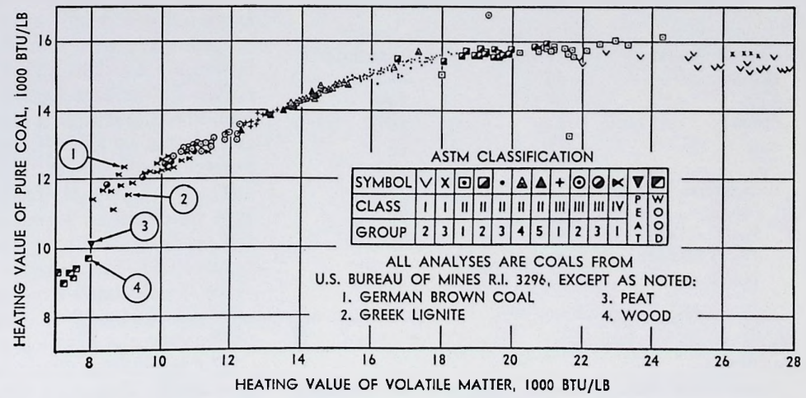


Fig. 2. *To illustrate a suggested coal classification using the relationship of the respective heating values of "pure coal" and the volatile matter*

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Classification by Grade

In the ASTM Standard D 389 for the classification of coal by grade, the ash, softening temperature, sulfur, and Btu are represented by symbols. This means of classification is not generally used in industry, but it is still retained as a standard method.

Commercial Sizes of Coal

Bituminous

Sizes of bituminous coal are not well standardized, but the following sizings are common:

Run of Mine. Run of mine is shipped without screening. It is used for both domestic heating and steam production.

Run of Mine (8 in.). This is run of mine with oversize lumps broken up.

Lump (5 in.). This size will not go through a 5-in. round hole. It is used for hand firing and domestic purposes.

Egg (5 in. × 2 in.). This size goes through 5-in. and is retained on 2-in. round-hole screens. It is used for hand firing, gas producers, and domestic firing.

Nut (2 in. × 1¼ in.). This size is used for small industrial stokers, gas producers, and hand firing.

Stoker Coal (1¼ in. × ¾ in.). This size is used largely for small industrial stokers and domestic firing.

Slack (¾ in. and under). Used for pulverizers, Cyclone Furnaces, and industrial stokers.

Several other sizes of bituminous coal are prepared by different producers, especially for domestic-stoker requirements.

Anthracite

Definite sizes of anthracite are standardized as given in Table 15.

TABLE 15

Commercial Sizes of Anthracite
(Graded on Round-Hole Screens)

Trade Name	Diameter of Holes, Inches: Through	Retained On
Broken	4⅜	3¼ to 3
Egg	3¼ to 3	2 7/16
Stove	2 7/16	1⅞
Nut	1⅞	13/16
Pea	13/16	9/16
Buckwheat		
No. 1	9/16	5/16
No. 2 (Rice)	5/16	3/16
No. 3 (Barley)	3/16	3/32
No. 4	3/32	3/64

The broken, egg, stove, nut, and pea sizes are largely used for hand-fired domestic units and gas producers.

Buckwheat Nos. 1 and 2 are used in domestic stokers and hand-fired steam boilers.

Buckwheat Nos. 3 and 4 are used for traveling-grate stokers, and No. 4 is also burned in pulverized form.

Moisture Determination

Moisture in coal is essentially water, quantitatively determined by definite prescribed methods which may vary according to the nature of the material. It is preferable to determine the moisture in coal in two steps: 1) prescribed air drying to equilibrium at 10-15 C above room temperature, and 2) prescribed oven drying for one hr at 104-110 C, after pulverizing (see also ASTM Standard D 271).

The air-dried component of the total-moisture value should be reported separately, because this useful information is required in the design and selection of equipment. For efficient pulverizing of anthracite and bituminous coal, it is the surface moisture that must be evaporated.

In 1956 a tentative method of test for equilibrium moisture of coal, at 96 to 97% relative humidity and 30 C, was accepted by the ASTM (see ASTM Standard D 1412-56T) for use pending adoption as standard.

This method affords a means of estimating the bed moisture of either coal that is wet and shows visible surface moisture or coal that may have lost some moisture. It may be used for estimating the surface, or extraneous, moisture of wet coal, such moisture being the difference between total moisture, as determined by ASTM Standard D 271, and equilibrium moisture. Also, the equilibrium moisture of a sample of coal is considered to be equal to the bed moisture for classification of coal by rank (see ASTM Standard D 388).

Calorific Value of Coal

The heating value of coal is determined accurately in an oxygen bomb submerged in cooling water. Several acceptable makes or types of bomb calorimeters are listed in ASTM Standard D 271. See also ASME Power Test Codes, part 9, *Heat of Combustion.* The coal sample is compressed into a hard pellet and ignited by a hot wire. The heating value is determined from the rise in the water temperature.

The gross (higher) calorific value is defined as the heat produced by a unit quantity of the fuel, with the products in the form of ash, liquid water, and gaseous CO_2, SO_2, and N_2. The net (lower) calorific value is calculated from the gross calorific value as the heat produced by a unit quantity of fuel when all water in the products remains as vapor.

The calculation (given in ASTM Standard D 407) is made by deducting 1030 Btu per pound of water

derived from a unit quantity of fuel, including both the water originally present as moisture and that formed by combustion.

In America the gross calorific value is commonly used in heat-balance calculations, while in Europe the net value is generally used.

Grindability of a Coal

Grindability is a term used to measure the ease of pulverizing a coal in comparison with a standard coal chosen as 100 (unity) grindability. A description of the method of testing to determine grindability of a coal in accordance with ASTM Standard D 409 and an illustration of the machine used in making the test (Fig. 12) appear in Chapter 17. The grindability of 16 of the 17 representative coals in Table 10 is given in the last column of Table 11.

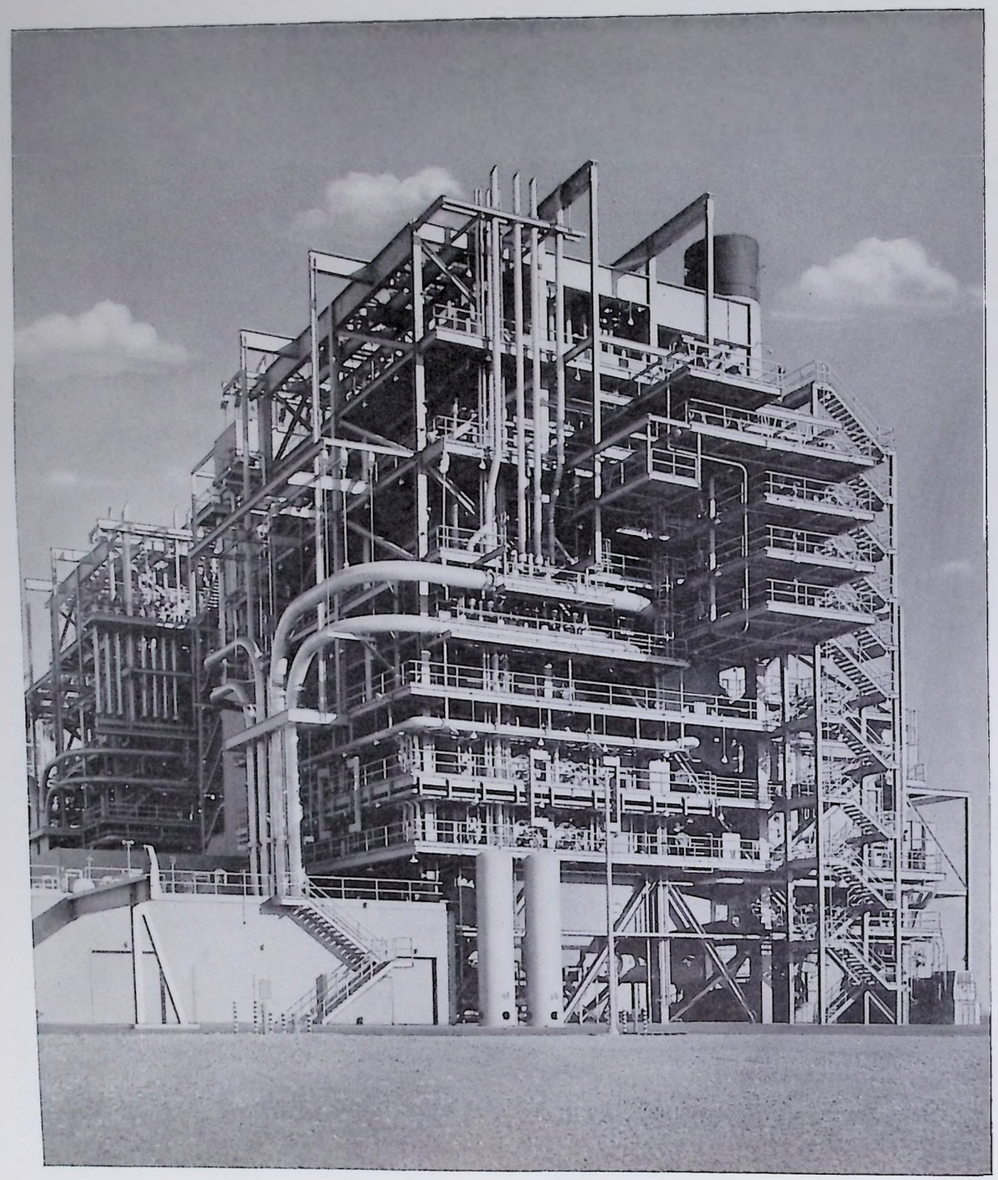
Free-Swelling Index

The ASTM Standard Method D 720 is used for obtaining information regarding the free-swelling property of a coal. Since it is a measure of the behavior of coal when heated rapidly, it may be used as an indication of the coking characteristic of the coal when burned as a fuel.

Coal Ash

The nature, composition, and properties of coal ash are discussed in Chapter 18.

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THIS CENTRAL STATION IN CALIFORNIA USES OIL AND GAS AS FUELS TO GENERATE 6,400,000 POUNDS OF STEAM PER HOUR

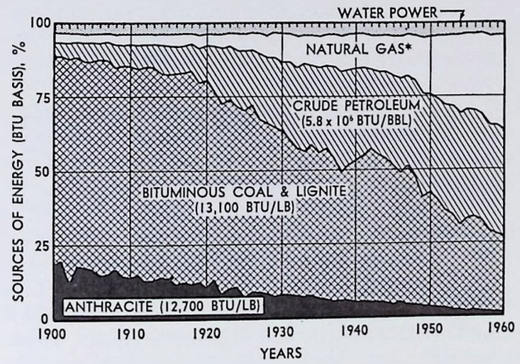
3

SOURCES OF HEAT ENERGY—OTHER THAN COAL*

ASIDE from coal, the principal sources of heat energy are petroleum, natural gas, and oil shale, which, like coal, come from fossil formations and are nonreplenishable. Vegetation, primarily wood, which is replenishable, is also a minor source (approximately 5%) of heat energy. The use of petroleum and natural gas as sources of heat energy is a comparatively recent development. Of the cumulative amount of petroleum used in the world, 40% has been produced during the past ten years.

While coal in large quantities is widely distributed throughout the world, the reserves of petroleum and natural gas are much more limited, and extensive investigations are already underway to improve practical methods of producing oil from coal and oil shale. The commercial production of liquid fuels from coal has been in successful operation in some parts of the world for several years.

During the period 1899 through 1959, petroleum and natural gas have been used in the U.S. to supply a constantly increasing part of the heat-energy requirements, with a corresponding reduction in the use of coal, as indicated in Fig. 1. The annual world production of petroleum increased approximately 100% during the period 1949 to 1959, while consumption in the U.S. during the same period increased 58%.



*TOTAL PRODUCTION X 1075 BTU MINUS REPRESSURING VENT AND WASTE GAS X 1035.

Fig. 1. *Per cent of total Btu produced each year (1900-1960) from various sources of heat energy in the United States (Water power converted to Btu equivalent)*

Availability of Fuels—Other than Coal

Petroleum. Availability of petroleum is based on the estimated proven reserves in areas where it has actually been found by exploration and drilling. In 1959 the proven reserves of the world were estimated to be about 293 billion barrels. At the rate of consumption in 1959, this quantity would be sufficient for 15 yr. The petroleum fields containing these reserves are located in four great basins: the Eastern Mediterranean; the Caribbean, United States, and Canada; the Far Eastern; and the North Polar.

The location of oil reserves, refining areas, and pipelines in the U.S. is shown in Fig. 2. The proven petroleum reserves in the U.S. are presently about 38 billion bbl. The estimate of unproven petroleum reserves in the U.S. is 50 billion bbl, excluding offshore reserves that may considerably increase this amount. During recent years the proven reserves have ranged from about 6 to 13 times the annual requirements, and up to the present the rate of development of new reserves has kept pace with the rapidly increasing demand. World production of petroleum in 1959 was of the order of 7.1 billion bbl/yr, of which the U.S. produced 2.6 billion bbl, or about 37%. Since the proven reserves of the U.S. are about 13% of the world total, it appears possible that imports will be needed in the future to supply a substantial portion of the U.S. requirements. Throughout the years to the end

* For analyses and properties, see Appendix 3; for nuclear fuel, see Chapter 27.

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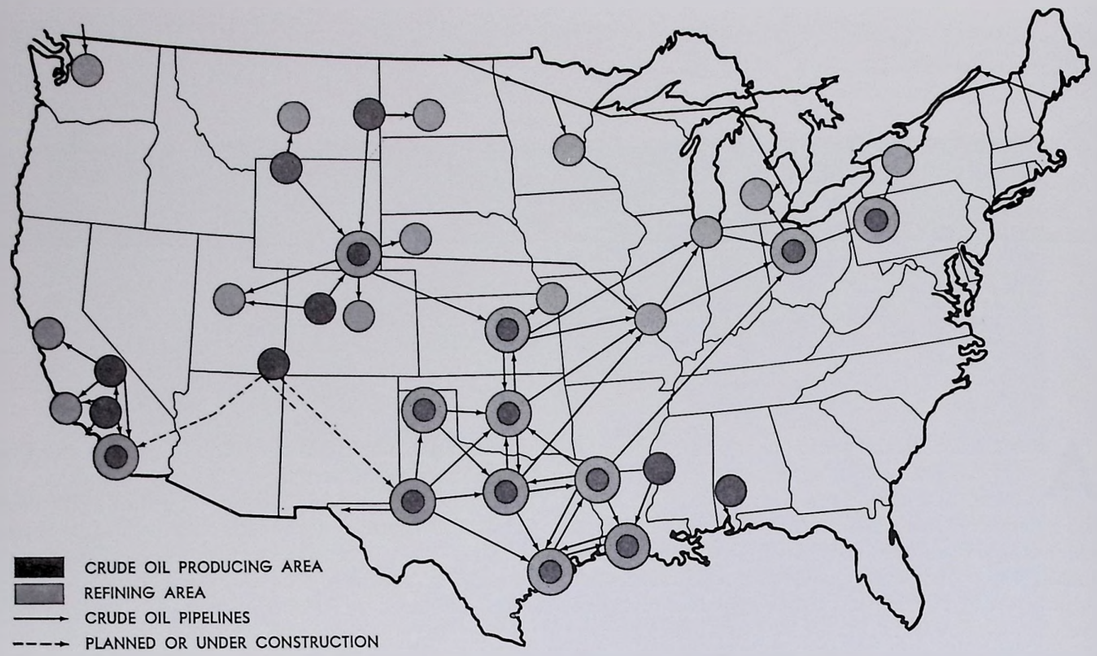


Fig. 2. *Oil fields, refining areas, and pipelines in the United States*

of 1959, the total production of petroleum in the U.S. was approximately 62.9 billion bbl. Assuming that the current total petroleum resources in proven and unproven reserves is of the order of 88 billion bbl, approximately ⅖ of the total original oil resources of the U.S. has been used.

The proportion of petroleum resources actually recovered depends upon the rate of withdrawal from existing wells, the economics of continued exploration and drilling, and the development or improvement of techniques that will permit economic recovery of more of the oil from the reservoirs. The rate of discovery of new large fields is liable to diminish, and it will be necessary to drill deeper in less favorable areas in the U.S. The number of dry holes drilled is increasing as are the average drilled depths and the costs. The recovery of petroleum from the wells is another economic factor. In 1961 the 591,116 oil wells in the U.S. produced an average of 12.2 bbl/well, day, and about 89% of the wells required pumping to bring the oil to the surface. About ⅔ of the oil wells in the U.S. are marginal or stripper wells that average only about 3.5 bbl/well, day.

In the past few decades considerable progress has been made in the extraction and recovery of oil from known fields. Research is responsible for the development of new recovery techniques. Moreover, during this period improved engineering and production practices were adopted by the oil industry. Formerly, as little as 25% of the oil in a field was recoverable. This percentage has been greatly increased and, equally important, many supposedly depleted fields have been restored to commercial production through improved recovery practices, such as gas repressuring of underground reservoirs. Dry gas is pumped into the reservoir to maintain a reservoir pressure, or "drive," at a level which assures an optimum rate of recovery. This gas-cycling method also produces condensate liquids, such as gasoline, butane, and propane, which are stripped from the gas before it is returned to the reservoir.

After the wells in a reservoir cease to produce oil at an economical rate, it is considered to be in the secondary recovery period, and artificial methods are used to force the remaining oil to the wells for removal. These secondary recovery methods consist of repressuring the reservoirs with water (water flooding), gas, or air. More recently "fire flooding," or in situ combustion, has been used. Briefly, oil is burned underground to heat up the remaining oil so that it may flow more easily and improve recovery. It has been estimated that this process may require 10 to 15% of the oil as fuel, with essentially complete recovery of the remainder.

Natural Gas. Areas containing petroleum can also be expected to yield natural gas, but there are many natural gas wells that yield no petroleum. The locations of natural gas fields and pipelines are shown in Fig. 3. The quantity of natural gas reserves of the

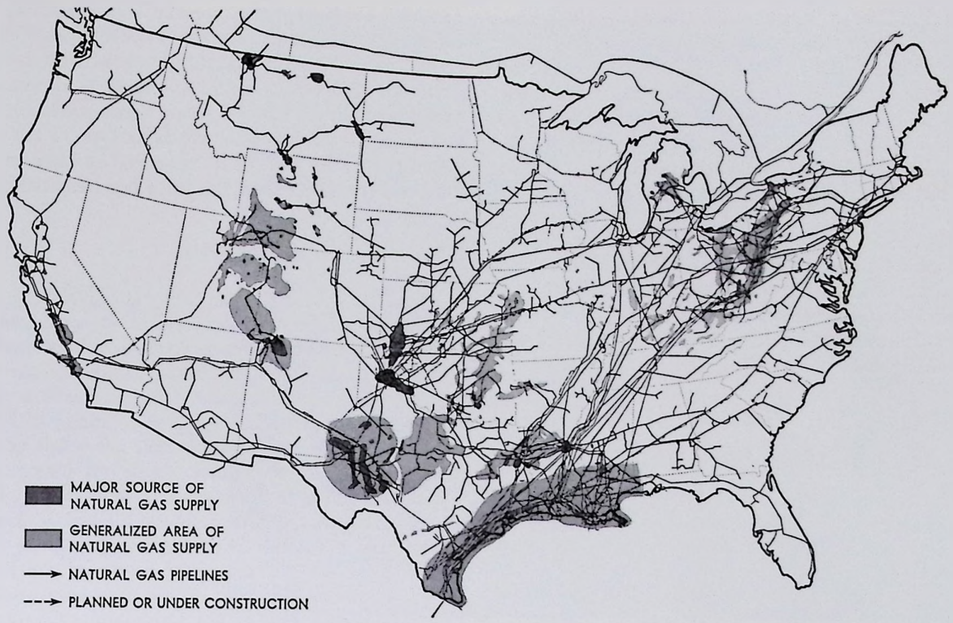


Fig. 3. *Natural gas fields and pipelines in the United States*

world is not known. In the U.S. the proven reserves of natural gas at the end of 1958 were 254 trillion cu ft, the equivalent in heating value of approximately 44 billion bbl of petroleum—somewhat more than the proven petroleum reserves.

Throughout the years to the end of 1958, the marketed production of natural gas in the U.S. was of the order of 115 trillion cu ft. A large quantity has also been flared or used in the production of carbon black. It is estimated that in 1944, 30% of the net production of natural gas was so used. The marketed production of natural gas in the U.S. during 1958 was of the order of 11.5 trillion cu ft. By the end of 1958, assuming reserves of 254 trillion cu ft, which do not include undiscovered resources, about ⅓ of the total marketed and proven reserves of natural gas in the U.S. had been consumed. At the 1958 rate of consumption, the present proven reserves will last approximately 23 years (this is referred to as the "life index"). The life index may be misleading, inasmuch as it does not include future discoveries that have kept the life index relatively constant, nor does it include possible greatly increased rates of imports from Canada. The life index should be considered as a measure of our working inventory of natural gas.

Oil Shale. Oil shale deposits are widely distributed throughout the world, the largest reserves being in the U.S., where they are exceeded only by coal as the most abundant fossil fuel. This potential resource is greater than the known petroleum reserves and assures adequate liquid fuels, gases, and petrochemicals for many years. The principal oil shale reserves of the world are given by country in Table 1.

TABLE 1

Oil Shale Reserves of the World

Country	Potential Oil in Shale, million barrels
Australia	399
Brazil	102,350
Estonia	6,500
France	410
Germany	1,919
Israel	25
Manchuria	1,863
New Zealand	213
Scotland	222
Sicily	23,500
Spain	80
Sweden	4,290
Union of South Africa	6
U.S.S.R.	7,000
United States:	
Green River Formation	1,132,000
Other	18,660

In the 17th century, medicinal oils were obtained from oil shale in England. Commercial production of shale oil was started in France in 1838, Scotland

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in 1850, the United States and Australia in 1860, and in Estonia, Spain, and Manchuria in the 1920's. The largest installation in the world in Southern Manchuria produces 9 to 12 million barrels of shale oil per year. Spain, Brazil, Manchuria, West Germany, Estonia, and the U.S.S.R. are the countries where commercial activity for production of shale oil and its products is of significance.

Wood. In 1923, forest lands of the world were estimated at 7.5 billion acres. This is 23.6% of the land area of the world exclusive of the polar regions. Of the forest area, 5.5 billion acres were considered productive. Forest areas of the world by continent are given in Table 2.

TABLE 2

Forest Areas of the World

Continent	Forest Area, million acres	Forest to Total Area, %
Asia	2096	21.6
South America	2093	44.0
North America	1444	26.8
Africa	797	10.7
Europe	774	31.1
Australia & Oceania	283	15.1
Total	7487	23.6

Until the latter part of the nineteenth century, when it was replaced by coal, wood supplied the major part of the heat-energy requirement. Wood is no longer a major factor as a source of heat energy, partly because of the depletion of the forests and the increasing demands for its use as lumber and in the production of paper, plywood, rayon, and other products. Today the burning of wood and bark for steam generation is largely confined to locations where it is available as a by-product or waste, as in the lumber, furniture, plywood, and pulp industries. For instance, in producing lumber from logs, half the wood may be discarded as sawdust, bark, shavings, slabs, and ends, all of which can be used as fuel. Also waste-wood liquors from the pulp industry are burned for the recovery of heat and process chemicals. Various wood wastes are used extensively in other parts of the world, and to a lesser extent in the U.S., to produce alcohol as fuel. New methods of barking logs and producing pulp, however, are continually reducing the quantity of wood available as fuel. More profitable end uses, such as bark for the production of roofing materials, are also reducing the availability of wood for fuel.

Other Vegetation. Numerous other types of vegetation, generally in the form of waste or by-products, are used for heat energy. In the U.S., however, they are an almost negligible factor in the total heat-energy requirements. The availability of these wastes or by-products as a source of heat energy is diminishing because, as in the case of wood, more profitable uses are being found for many of them. One of the more common of such wastes is "bagasse," the fibrous portion of sugar cane after the juice has been extracted.

Preparation for Use as Fuel

Petroleum. At the present time petroleum is the most important source of hydrocarbons. In crude oil, hydrogen and carbon are combined in various ways and proportions to form an almost limitless number of compounds, from methane, CH_4, which is a gas at room temperature, through a wide range of liquids to heavy waxes and asphalts that are solids at room temperature. As recovered, crude oil may contain sufficient water to form an emulsion. Most of this water is removed at the well by electrical or chemical means before delivery to the refinery.

Although small amounts of crude oil are burned without processing, most of the petroleum used has been refined to some extent. Originally, refining crude oil or petroleum was simply the process of separating the lighter compounds (higher in hydrogen) from the heavier compounds by fractional distillation. Through the development of refining techniques, such as thermal and catalytic cracking (Fig. 4) and hydrogenation of the heavier hydrocarbons to produce lighter ones, petroleum has now come to be considered as a raw-material source of hydrogen and carbon that can be combined as required to meet a variety of specific end uses. Consequently, the characteristics of the portion available as fuel oil depend more and more upon economic and national-security requirements rather than upon the nature of the crude oil from which it comes.

The principal volume products obtained from refining petroleum are: 1) gasoline, 2) Diesel oil, 3) fuel oil in various grades including kerosene, 4) lubricating oil, 5) asphalt, wax, and grease, and 6) synthetic rubber. In addition to the items listed as sources of heat energy, relatively small quantities of refinery gas, acid sludge, and petroleum coke are obtained as by-products and are often consumed as fuels at the refineries or used to enrich gases manufactured from coal.

Natural Gas. Since natural gas almost always accompanies petroleum, it may well be considered as part of the petroleum. To reduce the viscosity of the crude oil and facilitate raising it to the surface and handling, it is desirable to keep the lighter hydrocarbon compounds in solution by subjecting the oil to pressure. When the pressure is released, the light

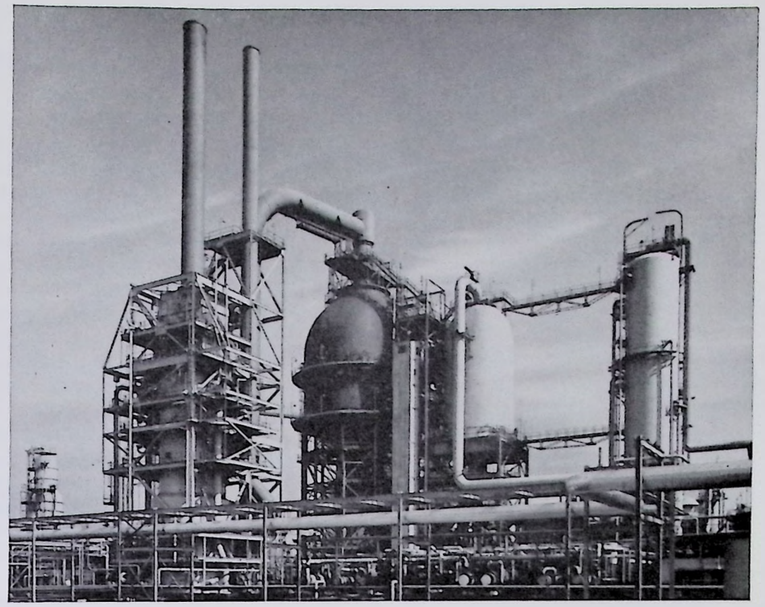


Fig. 4. *Catalytic cracking of petroleum at a Southern refinery*

hydrocarbons, such as methane, CH_4, ethane, C_2H_6, propane, C_3H_8, butane, C_4H_{10}, and others, volatilize as gases or vapors. Propane and butane are often separated from the lighter gases and are widely used as bottle gas. They are distributed and stored under pressure as liquefied gas. When the pressure is released the liquid boils, producing gaseous fuel. The portion heavier than butane may be separated and distributed as natural gasoline.

Much of the natural gas produced is suitable for use without further preparation. However, some natural gas may contain enough sand or gaseous sulfur compounds to be obnoxious or troublesome in use. The sand is usually removed at the source. Natural gases containing excessive amounts of sulfur compounds are known as "sour." One authority defines "sour natural gas" as gas containing more than 1.5 grains of hydrogen sulfide or more than 30 grains total sulfur per 100 cu ft of gas. The sulfur compounds are removed before distribution. Where natural gas is used to replace or supplement manufactured gas, it is sometimes reformed to bring its heating value in line with the manufactured gas. Natural gas may also be mixed directly with manufactured gas to increase the heating value of the distributed product.

Oil Shale. Oil shale, a sedimentary rock containing organic matter, will yield oil on heating. Subsequent refining of the oil is a necessary step in its recovery. Although the recovery of oil from shale has not been done on a commercial basis in the U.S., it has been demonstrated in pilot-plant operations by the Bureau of Mines that a range of acceptable fuel oils may be produced from shale oil by relatively simple refining techniques and that motor and Diesel fuels can be produced by special refining methods. Since the yield of oil may be only 25-30 gal/ton of shale, and even less with some lean shales (Swedish shale has about 5% oil), recovery of oil from this source involves a tremendous problem in the disposal of the waste.

Wood. Reduction to convenient size for handling and burning is the primary preparation required when wood is to be used for the generation of steam. The moisture content may also be reduced to improve combustion and efficiency. Bark and slabs from the lumber and pulp industries are often reduced in size by a machine cutter known as a hog. Bark from water-borne logs, bark removed by hydraulic barking methods, and bark from wood stored in the open may contain as much as 60 to 70% moisture by weight. Where such bark is to be used as a fuel, its moisture content is often reduced.

Transportation and Storage

Petroleum. Because of its many nonfuel applications, its high heating value as a fuel, and its ease in handling and transportation as a liquid, petroleum and its products are widely distributed throughout the world.

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Currently the over-all rate of production in the U.S. is approximately equal to the demand. However, in recent years the production of heavy fuel oils has been between 85 and 88% of the demand, and the balance has been supplied by imports of heavy oils from Venezuela and the Middle East. Lesser quantities have also been received from Mexico, Trinidad, and Canada.

Oil from the Gulf is transported to East Coast refineries by tanker and pipeline. A considerable part of the demand in areas bordering on the Mississippi and its tributaries is supplied from the Gulf by barge shipments. Only a small quantity is shipped from the West to the East Coast. Because of the relatively higher cost of haulage as compared with water or pipeline transportation, comparatively small quantities are shipped overland by rail and truck. The approximate relative costs of shipping petroleum by various means of transportation, although subject to wide variations, are noted in the following tabulation:

Various Transportation Methods	Relative Cost per Ton-Mile
Water	1
Pipeline	9
Railroad	27
Truck	81

To prevent contamination between certain types of crude oils, careful precautions must be taken, such as cleaning the tanks before loading a new shipment. In pipeline transportation this may necessitate the running of parallel lines.

The serious hazard of oil-storage-tank failure is overcome by storage in underground tanks or by building cofferdams around surface tanks with sufficient capacity to hold the entire contents of the tank. On the West Coast, crude oil is commonly stored in huge concrete reservoirs holding several million barrels. In the East, oil is usually stored in steel tanks to eliminate evaporation loss and to immunize against fire hazard from lightning. A standard set of rules for the storage and handling of oils, which serves as the basis for many local ordinances and as an excellent guide to safe practices, has been prepared by the National Fire Protection Association. Loss in storage of heavy fuel oils is negligible. Lighter products, such as gasoline, may volatilize sufficiently in warm weather to cause appreciable loss, unless special storage tanks are provided.

It may be necessary to reduce the viscosity of heavy fuel oil by heating before it can be pumped. Consequently transportation and storage facilities are generally provided with heating equipment. Storage tanks, piping, and heaters used for heavy oils require cleaning at intervals because of fouling or sludge formation. Various commercial compounds may be helpful in reducing sludge.

Natural Gas. Lacking suitable means of transportation, the use of natural gas would necessarily be limited to areas close to the source. The rapid increase in consumption of natural gas throughout the country and in areas remote from the source resulted from the continued extension of a vast system of long-distance pipelines. The distribution of natural gas is subject to some practical limitations because of the energy required for transportation. It is estimated that the energy required to transmit 1 cu ft of gas 1000 miles is about 150 Btu.

In general it is not practical to vary the supply of natural gas to accommodate hourly or daily fluctuations in consumer demand. For economic reasons, long-distance pipelines should operate with a high load factor. The rate of withdrawal from the wells may often be limited for conservation reasons, and the cost of the pipeline to provide the peak rate would be prohibitive. Therefore, to meet fluctuations in demand it is usually necessary to provide storage near at hand or briefly to supplement the supply with manufactured gas.

Storage facilities, either above ground or underground, are being rapidly increased. Above ground the usual methods of storage are: a) in the familiar large water-seal tanks; b) in liquid form in insulated steel tanks; c) in liquid form absorbed in a granular substance, released by passing warm gas over the grains; d) in pipe holders (at or near the surface) of commercial gas-line pipe laid parallel and interconnected; and e) using the trunk transmission line as a reservoir by building up the pressure. Because of low cost, underground storage of gas pumped back into depleted or partially depleted gas and oil wells, is the best method for the large storage required to meet seasonal variations in demand. This method is coming more and more into use, and the closer the storage is to the consumer the better, especially for accommodating hourly fluctuations.

Oil Shale. Because of the low combustible content of oil shale it appears certain that the oil will be extracted from the shale close to the source. Commercial use of shale oil will require establishment of distribution storage systems similar to those for petroleum.

Wood. As logged, wood is transported by rail, truck, or water. It has not been general practice to transport logs far from the source except by water. The means of water transportation are:

1. Driving: Floating down stream with the current.
2. Towing in Booms: An assembly of floating logs, enclosed within a loop of boom timbers fastened end to end, towed by tug.
3. Towing in Rafts: A number of large logs, arranged several logs deep, chained together, and towed by tug; usually in the open waters along the Pacific Coast.

The logs may or may not be barked at the source, depending upon the season, the distance of transportation, and the destination. To reduce deterioration, it is general practice to remove the bark before storing in piles. If floated to destination, holding areas are generally provided where a large supply of rough logs may be floated until required for use.

In many areas where the weather affects transportation, the movement of logs is seasonal. The supply of bark as fuel is therefore seasonal where a year's supply of logs may be received in a few months and barked for storage.

Properties and Utilization

Gasoline and Diesel oils produced from petroleum, although very important sources of heat energy, are not normally used for steam generation.

Fuel Oils. In refining petroleum it is common practice to produce fuel oils complying with the several specifications prepared by the ASTM and adopted as a commercial standard by the U.S. Bureau of Standards. To meet changes in supply and demand, these standards have been revised several times in past years, and further changes may be expected. The current standards are tabulated in the Appendix, page 3-A1. Fuel oils are graded according to gravity and viscosity, the lightest being No. 1 and the heaviest No. 6. For satisfactory pumping and burning, grades 5 and 6 generally require heating. Typical analyses, relative cost, and range of analyses and heating values of the several grades of fuel oils are given in the Appendix, page 3-A2.

The gross heating values of various fuel oils of different gravity are noted in Fig. 5. The gross heating values, after correction for the nonhydrocarbon constituents, are closely related to the gravity of the oil. The corrected gross heating value in Btu/lb is equal to:

$$\frac{\text{Btu} \times [100 - (\text{A} + \text{B} + \text{C})]}{100} + 40.5\text{C}$$

where:

Btu is taken from Fig. 4
A = % by wt of water
B = % by wt of ash
C = % by wt of sulfur

Where the percentages by weight of water and ash are not known, the percentages by volume of water and sediment can be used without appreciable error.

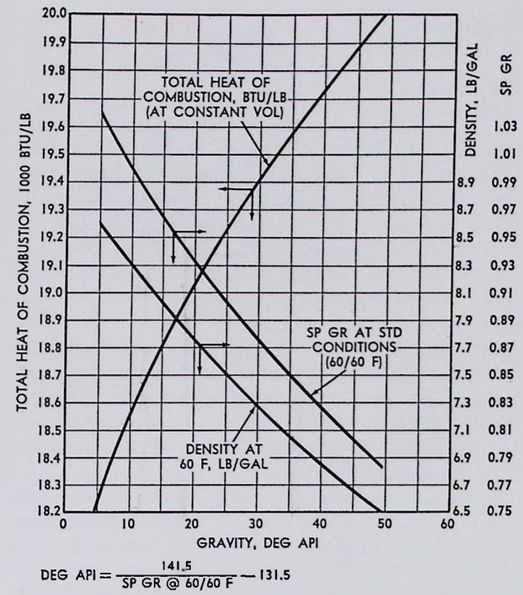


Fig. 5. Heating value, weight (lb per gal), and specific gravity of fuel oil for a range of API gravities

Fuel oils are generally sold on a volume basis with 60 F as the base temperature. To determine what the volume of oil will be at 60 F, a correction factor must be applied to the volume at any other temperature. Correction factors corresponding to temperature and API gravity are given in Fig. 6 (see Fig. 5 for

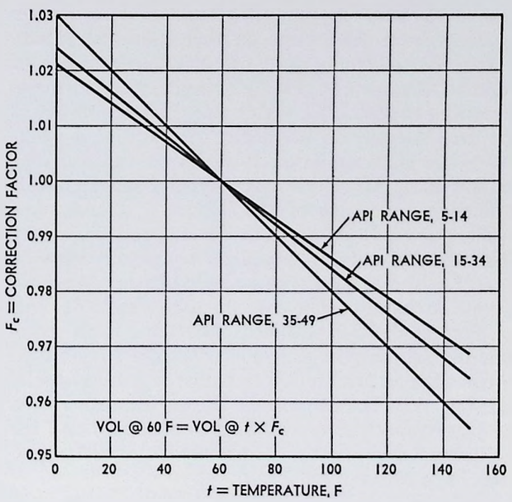


Fig. 6. Temperature-volume correction factor for fuel oil

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definition of deg API). Since the means of handling and particularly the burning equipment are usually designed for oil of a certain maximum viscosity, it is important to know the viscosity characteristics of the fuel oil. In addition, in some instances there are minimum viscosity limitations. To obtain the desired viscosity it may be necessary, because of variations in oil supply, to vary the temperature to which the oil is heated. The viscosity of oil decreases with increase in temperature. With more viscous oils, higher temperatures are generally required. Also with some oils and certain types of burning equipment it may be necessary to heat the oil to a higher temperature than that required to obtain the desired viscosity for satisfactory combustion. It is important to hold the upper limit of the oil temperature below the flash point to prevent carbonization of the oil and to avoid possible loss of burner ignition from interruption of flow by vapor binding of the pumps.

If the viscosity of heavy oils is known at two temperatures, the viscosities at other temperatures can be closely predicted. From a chart standardized by the ASTM, viscosities at various temperatures can be determined with negligible error by drawing a straight line through the plots of the viscosities at two known temperatures. For light oils it is necessary to know the viscosity at only one temperature to determine the viscosities at other temperatures. Using the ASTM standardized chart, Fig. 7, the viscosities of light oils at various temperatures, which fall in the region of the No. 2 fuel oil, can be determined by drawing a line parallel to the viscosity lines for this grade through the plot of the viscosity of the light oil at the known temperature. Copies of this chart may be obtained from the ASTM.

Since the true difference in viscosities is not readily apparent from the Second Saybolt Universal values, a more direct comparison of relative viscosities of different oils can be made by converting to kinematic viscosity (centistokes), which very closely represents the true viscosity. A kinematic viscosity of 4 centistokes, for instance, is equivalent to a value of 39.1 SSU, while a kinematic viscosity of 2 centistokes is equivalent to a value of 32.6 SSU.

Compared with coal, fuel oils are relatively easy to handle and to burn. Heating is not required for the lighter oils, and even the heavier oils are relatively simple to handle. There is no ash-in-bulk disposal problem as there is with coal, and there is no objectionable ash discharge from the stack. In most oil burners the oil is atomized to a mist of small particles that mix with combustion air. In the atomized state, oil approaches the characteristics of a gas, and the explosion hazards are of the same order as for gaseous fuels referred to below.

Because of its relatively low cost compared with that of other weight oils, No. 6 fuel oil is the most

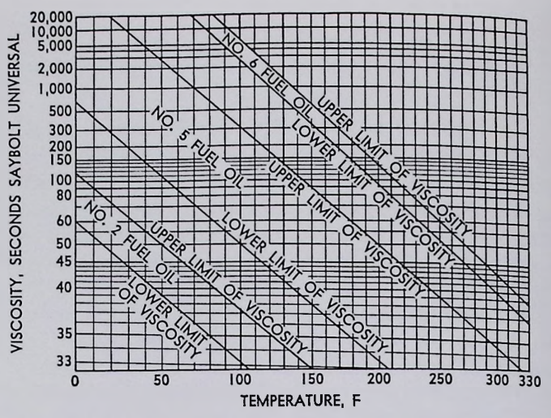


Fig. 7. *Viscosity-temperature curves for Nos. 2, 5, & 6 fuel oils from Chart A, ASTM Standard Charts D 341*

widely used for steam generation. It can be considered a by-product of the refining process. Its ash content ranges from about 0.01 to 0.50%, which is exceedingly low compared with that of coal. Despite this low content, however, the ash containing compounds of vanadium and sodium may be responsible for a number of serious operating problems to which sulfur also contributes (see Chapter 18).

Fuel oils generally available on the Eastern seaboard contain varying amounts of Venezuelan and Middle East crudes, depending upon the relative quantities of shipments and on the blending at the refineries to meet No. 6 fuel oil viscosity specifications. Fuel oils used on the Gulf Coast are produced largely from domestic crudes, although they may contain some Venezuelan crude. Because of this it is difficult, if not impossible, to identify the source of fuel oils as fired.

Natural Gas. Of all fuels, natural gas is considered the most desirable for steam generation. It is piped directly to the consumer, eliminating the need for storage at the consumer's plant. It is substantially free of ash, combustion is smokeless, and, because it is a gas, it mixes easily and intimately with air to give complete combustion with low excess combustion air. Although the total hydrogen content of natural gas is high, the amount of free hydrogen is low. Because of this, natural gas is not as easy to burn as some of the manufactured gases of high free-hydrogen contents. As with other gaseous fuels, its ease of mixing with air increases the liability of explosion. Great care in the design and operation of gas-burning equipment is essential to minimize explosion hazards.

Because of the high hydrogen content of natural gas compared with that of oil or coal, the water vapor formed in combustion is correspondingly greater and

is reflected in a lower efficiency of the steam-generating equipment. This can readily be taken into account in the design of the equipment and evaluated in comparing its cost with that of other fuels.

Typical analyses of natural gas from several of the U.S. fields are given in the Appendix, page 3-A3.

Oil Shale. Properties of typical oil shale from Colorado are given in the Appendix, page 3-A3. Heavy fuel oil produced from a refining process using oil from shale as feedstock might have properties very similar to residual fuel oil from petroleum.

Wood. Typical analyses and heating values of several types of wood (also analyses of wood ash) are given in the Appendix, page 3-A4. Wood, in common with all types of vegetation, is composed primarily of carbohydrates and consequently has a relatively low heating value compared with that of bituminous coal and oil.

Wood bark may pick up impurities during transportation. It is common practice to drag the rough logs to central loading points in the logging area, during which the bark will pick up considerable sand. Where the logs are salt-water-borne, the bark will absorb a considerable amount of sea water with its included salt. In burning dry bark, combustion temperatures may be high enough for these impurities to cause fluxing of refractory furnace walls and slagging of boiler heating surfaces, unless sufficient furnace cooling surface is provided. Sand passing through the boiler banks may cause erosion of boiler tubes, particularly if the sand loading of the flue gases is increased by the use of cinder catchers for returning collected material to the furnace. To reduce to acceptable limits the discharge of incompletely burned bark from the stack, such collectors may be required with some types of bark-burning equipment.

As mentioned above under preparation, the moisture content of wood and bark may be quite high. Wood or bark with moisture content of 50% or less burns quite well; however, as the moisture content increases above this amount, combustion becomes more difficult. With moisture content above 65%, a large part of the heat in the wood is required to evaporate the moisture, and little is left over for steam generation. Burning of this wet bark becomes a means of disposal rather than a source of heat energy.

Hogged wood and bark are very bulky and require relatively large handling and storage equipment. Uninterrupted flow from bunkers or bins through chutes is difficult to maintain, and available equipment for this purpose is not always satisfactory.

Bagasse. Mills grinding sugar cane commonly use bagasse for steam production. The mills normally operate 24 hr/day during the grinding season. In mills where refining of the sugar is not done, the supply of bagasse will easily equal that required to meet the plant steam demands. Consequently, where there is no other market for the bagasse, no particular effort is made to burn it efficiently, and burning equipment is provided which will burn the bagasse as received from the grinders. In plants where refining is done, supplemental fuels are required to provide the increased steam demands. In these plants greater efforts to obtain higher efficiency are justified. A typical analysis of bagasse as fired is given in the Appendix, page 3-A4.

Relative Costs of Fuels

Fuel Oils. As noted above, fuel oils are divided into five general classes according to properties, and the relative costs are given in the Appendix, page 3-A2. These costs are based on December 1960 prices in the New York area. Because of the proportionately great variations over periods of time and in different localities, these relative costs can only serve as rough approximations. In some parts of the country, steam-generating equipment is arranged to burn either coal, oil, or gas, the fuel selected depending upon the relative costs. Table 3 shows the "as-consumed" fuel costs for the year 1959 of steam-electric plants located in the Mideast Coast area. The "as-consumed" cost of natural gas is assumed to be the same as the delivered cost. The per cent of the total energy, on a Btu basis, supplied by each fuel is also indicated.

TABLE 3

As-Consumed Fuel Costs of Steam-Electric Plants, 1959*

	Cents/Million Btu			Per Cent of Total Btu		
State or City	Coal	Oil	Gas	Coal	Oil	Gas
Massachusetts	37.1	35.1	34.5	45	47	8
Connecticut	38.0	37.1	33.7	81	17	2
Rhode Island	37.0	37.7	37.2	74	25	1
New York City	37.7	34.9	34.3	56	23	21
New Jersey	35.8	35.2	31.2	45	37	18
Philadelphia	36.1	37.3	31.1	76	16	8
Delaware	34.4	38.7	34.2	77	3	20

* Source: *National Coal Association.*

Natural Gas. The cost of natural gas in the U. S. close to the source is about 12¢/million Btu and increases up to 70 cents farther from the source. Since a large percentage of natural gas is used for space heating, the demand for natural gas is considerably higher during the winter months. During summer months natural gas may be available at relatively low cost for steam generation, even in areas a considerable distance from the source. An ever increasing number of steam plants is being adapted to burn natural gas

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seasonally. Because the supply is much greater than the present pipeline capacity, plants for conversion of natural gas to gasoline are being built. As additional gas is so used and pipeline capacity is increased, the cost of natural gas may approach that of oil and coal, even in areas near its source. A wide range in comparative costs for three different fuels is shown by Fig. 8.

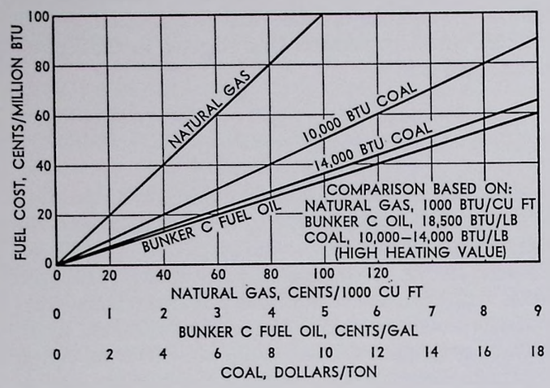


Fig. 8. *Comparative fuel costs in cents per million Btu*

Oil Shale. At the prevailing U.S. cost of producing oil from petroleum, it has not been commercially attractive to produce oil from oil shale. It is estimated that the cost of producing crude shale oil would be about $2/bbl, including operating costs, depreciation, taxes, and a 6% return on the investment. Continued increase in the cost of oil from petroleum may bring about the commercial production of oil from oil shale.

Other Fuels Sometimes Burned

Among other fuels used, those derived from coal are of the most importance. Combustible by-products from petroleum and by-products or wastes in industries processing woods or vegetation are also used as fuels.

Coke. When coal is heated in the absence of air or with a great deficiency of air, the lighter constituents are volatilized and the heavier hydrocarbons crack, liberating hydrogen and leaving a residue of carbon. Some of the volatilized portions crack on contact with the hot carbon, leaving an additional quantity of carbon. The carbonaceous residue containing the ash and a part of the sulfur of the original coal is called "coke." The amount of sulfur and the amount and nature of the ash in the coke obviously depend in large measure upon the coal from which it is produced and the coking process used. The principal uses for coke are in the production of pig iron in blast furnaces and the charging of iron-foundry cupolas. Because it is smokeless in combustion, considerable quantities are used for space heating.

At the steel mills, undersized coke called "coke breeze," usually passing a ⅝-in. screen, is unsuited for charging blast furnaces and is available for steam generation. Practically all of the coke breeze is consumed at the mills. A typical analysis of coke breeze appears in the Appendix, page 3-A4.

Coke production and consumption in millions of tons for the year 1958 are tabulated as follows:

Total coal used for coke production	76.8
Total annual coke production	53.6
Coke breeze produced	3.7
Coke breeze used for steam-raising	1.5
Coke breeze used for sintering iron ore	0.8

Approximately 5% by wt of the coal charged into slot-type coke ovens is recovered as breeze. Until the late 1940's, roughly ¾ of the breeze produced at oven-coke plants was used by the producing companies for steam-raising. In 1958 the use of breeze for steam-raising amounted to only 41% of the output. This reduction was, in large part, the result of increased requirements for coke breeze for smelting phosphate rock and for sintering iron ore in the many new beehive ovens that have recently been installed.

A portion of the coal tars produced may be burned in equipment similar to that used for heavy petroleum oil.

Char. Low-temperature (900 to 1100 F) carbonization processes have been developed to extract from coal valuable tars, bitumens, and gases for use as raw materials in the chemical industry. The by-product solid fuel from these processes is called char. Disco char, produced by low-temperature carbonization, has been sold for use in the domestic and small commercial market. The char which appears to have the greatest commercial significance is that produced by the fluidized-bed method. However, economics has curtailed the widespread use of these processes. A typical analysis of fluidized-bed char is given in the Appendix, page 3-A4.

Gaseous Fuels. A number of gaseous fuels are derived from coal. Typical analyses are given in the Appendix, page 3-A5. Their importance is diminishing because they have been displaced by natural gas and oil. A description of the tabulated gaseous fuels from coal follows:

Coke-Oven Gas. In the production of coke from coal, a considerable portion of the coal is released as gases or vapors. A number of valuable products are recovered from these gaseous portions, including ammonium sulfate, oils, and tars. The noncondensable portion is called "coke-oven gas." Its constituents

depend upon the nature of the coal from which the gas is produced and the coke procedure used. A part of the sulfur from the coal may be present as hydrogen sulfide and carbon disulfide gases. These may be removed by scrubbing. Coke-oven gas often contains other impurities that deposit in lines and burners. To reduce the possibility of trouble from plugging by these deposits, relatively large burner gas-port openings are used, and burners are arranged for easy access for cleaning. Coke-oven gas burns readily because of its high free-hydrogen content and, except for the build-up of deposits, presents no problems when used as fuel for steam generation.

Approximately 17% by wt of the coal charged into slot-type coke ovens is recovered in the form of a fuel gas. In 1958 a total of 790 billion cu ft of coke-oven gas was formed from 75.8 million tons of coal, averaging 10,420 cu ft per ton of coal carbonized. Of the gas produced in 1958, 35% was used to heat coke ovens; 55% was used in steel and allied plants and under boilers by the producing companies; 8% was sold for residential, commercial, and industrial heating; and 2% was wasted or unaccounted for.

Blast-Furnace Gas. The gas discharged from steel-mill blast furnaces is used at the mills in heating-furnaces, gas engines, and for steam generation. As shown by the analyses in the Appendix, page 3-A5, the gas is quite variable in quality but generally of low heating value and high carbon monoxide content.

The temperature of the gas as discharged from the furnace is around 500 F, and it contains 5 to 7 grains of entrained dust per cu ft. The gas as it comes from the furnace may be burned for steam generation, but at many mills the entrained dust is first removed by washing or by electrostatic precipitation. With many types of washers the wet washed gas will have a dust loading of 0.1 to 0.2 grains per cu ft, and, with high gas-pressure-drop water-scrubbing or cleaning by electrostatic precipitation, the gas may have a dust loading of the order of 0.02 grains per cu ft.

Both the dirty and the wet washed gas can cause trouble, when burned for steam generation, by plugging the gas lines and burner port openings and fouling the boiler heating surfaces. Because of the nature of the deposits, wet washed gas is often more troublesome than hot, dry, dirty gas. Electrostatically cleaned gas may or may not cause trouble, depending upon the collector efficiency maintained. Blast-furnace-gas deposits adhere very firmly, and special provisions are required for cleaning the boiler heating surfaces. For blast-furnace-gas firing, means must be provided for access to and cleaning of burners.

When blast-furnace gas is used in steam generation, special safety precautions are required because of fluctuations in the gas supply from variations or interruptions in blast-furnace operation. Also, where the steam production rate must be maintained, an alternate fuel must be immediately available.

Water Gas. The gas produced by passing steam through a bed of hot coke is known as water gas. The carbon in the coke combines with the steam to form hydrogen and carbon monoxide. This is an endothermic reaction which cools the coke bed. To maintain the latter at high temperature, the steam is periodically replaced by air for combustion of coke and elevation of bed temperature. While the air is passing through the bed, the discharged products of combustion are diverted to the atmosphere to avoid diluting the water gas.

Water gas is often enriched with oil by passing the gas through a checkerwork of hot bricks sprayed with oil which is cracked to a gas by the heat. Refinery gas is also used for enrichment. It may either be mixed with the steam and passed through the coke bed or mixed directly with the water gas. Such enriched water gas is called "carbureted water gas." It is piped for relatively short distances through city mains for industrial and domestic consumption. Where it is so used, it is cleaned at the source to remove sulfur gases and other impurities. In many areas, however, it has been replaced by natural gas.

Producer Gas. When coal or coke is burned with a deficiency of air and a controlled amount of moisture (steam), a gas known as producer gas is obtained. Because of its low heating value it is used close to the source, after removal of entrained ash and sulfur compounds.

In order to obtain energy in gaseous form for power production at the mine from coal occurring in seams too thin for mining, underground gasification by in situ combustion of the coal was accomplished on an experimental basis at Gorgas, Alabama, by the Bureau of Mines. Although underground gasification of coal is not economically feasible in the U.S., the U.S.S.R. has been obtaining producer gas by this means for power production.

Coke from Petroleum. The heavy residuals from the various petroleum-cracking processes are presently utilized in a number of ways to produce a higher yield of lighter hydrocarbons and a solid residue suitable for fuel. The characteristics of these solid residues vary widely, depending upon the process used. Typical analyses of some of these are given in the Appendix, page 3-A5. A description of the solid fuels from oil follows:

Delayed Coke. In the delayed-coking process, the residual is heated and pumped to a reactor to coke. The coke is deposited as a solid mass and is sub-

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sequently stripped, either mechanically or hydraulically, in the form of lumps of granular material. Some of these cokes are easy to burn and pulverize, while others are quite difficult.

Fluid Coke. In the fluid-coke process, the hot residual feed is sprayed onto externally heated seed coke in a fluid bed. The fluid coke is removed as small particles, which are built up in layers similar to an onion. This coke can be pulverized and burned, or it can be burned in the as-received size in the Cyclone Furnace. Both types of firing require some supplemental fuel to aid ignition.

Petroleum Pitch. This process is an alternate to the coking process and produces fuels of various characteristics. The melting points may vary considerably, and the physical properties may vary from soft and gummy to hard and friable. The low-melting-point pitches may be heated and burned like heavy oil, while those with higher melting points may be pulverized and burned or crushed and burned in the Cyclone Furnace.

Wood Wastes. There are several industries using wood as a raw material where combustible by-products or wastes are available as fuels. The most important of these are the pulp and turpentine industries. The nature and methods of utilization of the combustible by-products from the pulp industry are discussed in Chapter 20.

The residue remaining after the steam distillation of coniferous woods for the production of turpentine is usable as a fuel. In the distillation process, some of the more easily burned constituents are removed, with the result that the residue is somewhat more difficult to burn. Other than this, its properties are much the same as those of the raw wood, and the problems involved in its utilization are similar.

Vegetation Wastes. In the food and related industries there are numerous vegetable wastes that are usable as fuels. They include such materials as grain hulls, the residue from the production of furfural from corn cobs and grain hulls, coffee grounds from the production of instant coffee, tobacco stems, etc. The quantity of such fuels is a relatively insignificant part of the total heat-energy requirements.

4

PRINCIPLES OF COMBUSTION

In the field of power generation useful heat is produced by combustion, which may be defined as the rapid chemical combination of oxygen with the combustible elements of a fuel. The common fuels—coal, oil, gas, wood and their various derivatives—have only three elemental constituents, carbon, hydrogen, and sulfur, which unite with oxygen of the air to produce heat. For most fuels the elemental constituents are carbon and hydrogen, since the per cent of sulfur is so low as to be negligible in producing heat, although some of its compounds can have profound chemical effects (corrosion), which are considered elsewhere (see Chapter 11).

To achieve complete combustion, that is, the combination of the combustible elements and compounds of a fuel with all the oxygen which they can utilize, sufficient space, time, mixing or turbulence, and a temperature high enough to ignite the constituents must be provided.

The combustible elements and their compounds found in fuels used in the commercial generation of heat, as well as their molecular weights and combustion constants, including heat values, are listed in Table 1. The noncombustible elements and compounds involved in combustion are also listed in this table.

Oxygen combines with combustible elements and compounds in accordance with the laws of chemistry, and the various substances involved always combine in fixed proportions by weight. Typical chemical reactions frequently encountered in the commercial production of heat are given in Table 2. These chemical equations are based on the assumption that the reaction is complete and that the exact amount of oxygen required is utilized.

Principles Involved in Combustion

The principles involved in the development of heat by combustion, generally accepted as authoritative, were propounded by Berthelot. His "second law," as applied to combustion in furnace practice, is of particular interest and may be stated as follows:

In a boiler furnace (where no mechanical work is done) the heat energy evolved from the union of combustible elements with oxygen depends upon the ultimate products of combustion and not upon any intermediate combinations that may occur in reaching the final result.

A simple demonstration of this law is the union of 1 lb of carbon with oxygen to produce a specific amount of heat (about 14,100 Btu, Table 1). The union may be in one step to form the gaseous product of combustion, CO_2. Under certain suitable conditions the union may be in two steps, first, to form CO, producing a much smaller amount of heat (about 4000 Btu) and, second, the union of the CO so obtained to form CO_2, releasing 10,100° Btu. However, the sum of the heats released in these two steps equals the 14,100 Btu evolved when carbon is burned in one step to form CO_2 as the final product.

That carbon may enter into these two combinations with oxygen is of utmost importance in the design of combustion equipment. Firing methods must assure proper and complete mixture of the fuel and the oxygen. Failure to meet this requirement will result in appreciable losses in combustion efficiency and heat potentials because of unburned combustibles.

Measurement of the Heat of Combustion

In boiler practice the heat of combustion of a fuel, or the calorific value frequently referred to as the "fuel Btu value," is the amount of heat, expressed in Btu, generated by the complete combustion, or oxidation, of a unit weight (1 lb in the United States) of the fuel. The amount of heat so generated is a

° The National Bureau of Standards, Bulletin No. 500, gives 10,096 Btu per lb for graphitic C at 78 F burned from CO to CO_2.

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TABLE 1

Combustion Constants

No.	Substance	Formula	Molecular Weight	Lb per Cu Ft	Cu Ft per Lb	Sp Gr Air = 1.0000	Heat of Combustion Btu per Cu Ft Gross (High)	Heat of Combustion Btu per Cu Ft Net (Low)	For 100% Total Air Moles per mole of Combustible Required for Combustion O_2	For 100% Total Air Moles per mole of Combustible Required for Combustion N_2	For 100% Total Air Moles per mole of Combustible Flue Products CO_2	For 100% Total Air Moles per mole of Combustible Flue Products H_2O	For 100% Total Air Moles per mole of Combustible Flue Products N_2	Lb per Lb of Combustible Required for Combustion O_2	Lb per Lb of Combustible Required for Combustion N_2	Lb per Lb of Combustible Required for Combustion Air	Lb per Lb of Combustible Flue Products CO_2	Lb per Lb of Combustible Flue Products H_2O	Lb per Lb of Combustible Flue Products N_2
1	Carbon*	C	12.01	.....	.....	.....	...	...	1.0	3.76	1.0	...	3.76	2.66	8.86	11.53	3.66	...	8.86
2	Hydrogen	H_2	2.016	0.0053	187.723	0.0696	325	275	0.5	1.88	...	1.0	1.88	7.94	26.41	34.34	....	8.94	26.41
3	Oxygen	O_2	32.000	0.0846	11.819	1.1053	...	...	...	....	...	...	....	....	....	....	....	...	....
4	Nitrogen (atm)	N_2	28.016	0.0744	13.443	0.9718	...	...	...	....	...	...	....	....	....	....	....	...	....
5	Carbon monoxide	CO	28.01	0.0740	13.506	0.9672	322	322	0.5	1.88	1.0	...	1.88	0.57	1.90	2.47	1.57	...	1.90
6	Carbon dioxide	CO_2	44.01	0.1170	8.548	1.5282	...	...	...	....	...	...	....	....	....	....	....	...	....
	Paraffin series																		
7	Methane	CH_4	16.041	0.0424	23.565	0.5543	1013	913	2.0	7.53	1.0	2.0	7.53	3.99	13.28	17.27	2.74	2.25	13.28
8	Ethane	C_2H_6	30.067	0.0803	12.455	1.0488	1792	1641	3.5	13.18	2.0	3.0	13.18	3.73	12.39	16.12	2.93	1.80	12.39
9	Propane	C_3H_8	44.092	0.1196	8.365	1.5617	2590	2385	5.0	18.82	3.0	4.0	18.82	3.63	12.07	15.70	2.99	1.63	12.07
10	n-Butane	C_4H_{10}	58.118	0.1582	6.321	2.0665	3370	3113	6.5	24.47	4.0	5.0	24.47	3.58	11.91	15.49	3.03	1.55	11.91
11	Isobutane	C_4H_{10}	58.118	0.1582	6.321	2.0665	3363	3105	6.5	24.47	4.0	5.0	24.47	3.58	11.91	15.49	3.03	1.55	11.91
12	n-Pentane	C_5H_{12}	72.144	0.1904	5.252	2.4872	4016	3709	8.0	30.11	5.0	6.0	30.11	3.55	11.81	15.35	3.05	1.50	11.81
13	Isopentane	C_5H_{12}	72.144	0.1904	5.252	2.4872	4008	3716	8.0	30.11	5.0	6.0	30.11	3.55	11.81	15.35	3.05	1.50	11.81
14	Neopentane	C_5H_{12}	72.144	0.1904	5.252	2.4872	3993	3693	8.0	30.11	5.0	6.0	30.11	3.55	11.81	15.35	3.05	1.50	11.81
15	n-Hexane	C_6H_{14}	86.169	0.2274	4.398	2.9704	4762	4412	9.5	35.76	6.0	7.0	35.76	3.53	11.74	15.27	3.06	1.46	11.74
	Olefin series																		
16	Ethylene	C_2H_4	28.051	0.0746	13.412	0.9740	1614	1513	3.0	11.29	2.0	2.0	11.29	3.42	11.39	14.81	3.14	1.29	11.39
17	Propylene	C_3H_6	42.077	0.1110	9.007	1.4504	2336	2186	4.5	16.94	3.0	4.0	16.94	3.42	11.39	14.81	3.14	1.29	11.39
18	n-Butene	C_4H_8	56.102	0.1480	6.756	1.9336	3084	2885	6.0	22.59	4.0	4.0	22.59	3.42	11.39	14.81	3.14	1.29	11.39
19	Isobutene	C_4H_8	56.102	0.1480	6.756	1.9336	3068	2869	6.0	22.59	4.0	4.0	22.59	3.42	11.39	14.81	3.14	1.29	11.39
20	n-Pentene	C_5H_{10}	70.128	0.1852	5.400	2.4190	3836	3586	7.5	28.23	5.0	5.0	28.23	3.42	11.39	14.81	3.14	1.29	11.39
	Aromatic series																		
21	Benzene	C_6H_6	78.107	0.2060	4.852	2.6920	3751	3601	7.5	28.23	6.0	3.0	28.23	3.07	10.22	13.30	3.38	0.69	10.22
22	Toluene	C_7H_8	92.132	0.2431	4.113	3.1760	4484	4284	9.0	33.88	7.0	4.0	33.88	3.13	10.40	13.53	3.34	0.78	10.40
23	Xylene	C_8H_{10}	106.158	0.2803	3.567	3.6618	5230	4980	10.5	39.52	8.0	5.0	39.52	3.17	10.53	13.70	3.32	0.85	10.53
	Miscellaneous gases																		
24	Acetylene	C_2H_2	26.036	0.0697	14.344	0.9107	1499	1448	2.5	9.41	11.91	2.0	1.0	9.41	3.07	10.22	13.30	3.38	0.69	10.22
25	Naphthalene	$C_{10}H_8$	128.162	0.3384	2.955	4.4208	5854	5654	12.0	45.17	10.0	4.0	45.17	3.00	9.97	12.96	3.43	0.56	9.97
26	Methyl alcohol	CH_3OH	32.041	0.0846	11.820	1.1052	868	768	1.5	5.65	1.0	2.0	5.65	1.50	4.98	6.48	1.37	1.13	4.98
27	Ethyl alcohol	C_2H_5OH	46.067	0.1216	8.221	1.5890	1600	1451	3.0	11.29	2.0	3.0	11.29	2.08	6.93	9.02	1.92	1.17	6.93
28	Ammonia	NH_3	17.031	0.0456	21.914	0.5961	441	365	0.75	2.82	3.57	1.5	3.32	1.41	4.69	6.10	....	1.59	5.51
											SO_2						SO_2		
29	Sulfur*	S	32.06	.....	.....	.....	...	...	1.0	3.76	4.76	1.0	...	3.76	1.00	3.29	4.29	2.00	...	3.29
30	Hydrogen sulfide	H_2S	34.076	0.0911	10.979	1.1898	647	596	1.5	5.65	7.15	1.0	1.0	5.65	1.41	4.69	6.10	1.88	0.53	4.69
31	Sulfur dioxide	SO_2	64.06	0.1733	5.770	2.2640	...	...	...	....	....	...	...	....	....	....	....	....	...	....
32	Water vapor	H_2O	18.016	0.0476	21.017	0.6215	...	...	...	....	....	...	...	....	....	....	....	....	...	....
33	Air	.....	28.9	0.0766	13.063	1.0000	...	...	...	....	....	...	...	....	....	....	....	....	...	....

*Carbon and sulfur are considered as gases for molal calculations only.

Note: This table is reprinted from *Fuel Flue Gases*, 1941 Edition, courtesy of American Gas Association.
All gas volumes corrected to 60 F and 30 in. Hg dry.

constant for any given combination of combustible elements and compounds and, in accordance with Berthelot's second law, is not affected by the manner in which combustion takes place, providing it is complete.*

TABLE 2

Common Chemical Reactions of Combustion

Combustible	Reaction
Carbon (to CO)	$2C + O_2 = 2CO + Q$
Carbon (to CO_2)	$C + O_2 = CO_2 + Q$
Carbon monoxide	$2CO + O_2 = 2CO_2 + Q$
Hydrogen	$2H_2 + O_2 = 2H_2O + Q$
Sulfur (to SO_2)	$S + O_2 = SO_2 + Q$
Sulfur (to SO_3)	$2S + 3O_2 = 2SO_3 + Q$
Methane	$CH_4 + 2O_2 = CO_2 + 2H_2O + Q$
Acetylene	$2C_2H_2 + 5O_2 = 4CO_2 + 2H_2O + Q$
Ethylene	$C_2H_4 + 3O_2 = 2CO_2 + 2H_2O + Q$
Ethane	$2C_2H_6 + 7O_2 = 4CO_2 + 6H_2O + Q$
Hydrogen sulfide	$2H_2S + 3O_2 = 2SO_2 + 2H_2O + Q$

where Q = the heat of reaction

To determine the calorific value of any fuel, the most satisfactory and universally adopted method is by direct measurement, in a calorimeter, of the heat evolved during combustion. Fuel calorimeters and the methods of operation are described in detail by numerous authorities.

For solid fuels and most liquid fuels, calorimeters of the "bomb" type, in which combustible substances are burned in a constant volume of oxygen, give the most satisfactory results. With bomb calorimeters properly operated, combustion is complete, all of the heat generated is absorbed and measured, and heat from external sources can be excluded or proper corrections applied.

For gaseous fuels, calorimeters of the continuous, or constant-flow, type are ordinarily accepted as standard.

The accuracy of the determination of the heat value of a fuel by calorimeter, largely a question of the personal equation, depends upon careful manipulation of the instrument. With proper care the results should be accurate to within a fraction of 1%. In addition to the calorific value, it is customary (necessary for heat-balance calculations) to report the ultimate or the proximate analysis of the fuel.

For solid and liquid fuels, separate determinations of the calorific value of each specific fuel are necessary. For the calorific values of some solid and liquid fuels, refer to Chapters 2 and 3. For elements and combustible compounds comprising gaseous fuels, the heats of reaction have been determined by so many authorities that established values may be accepted as correct without determination. For gaseous fuels and various substances frequently encountered in combustion work, see Table 1.

While it is preferable, for accuracy, to determine the heat value of a fuel by actual test in a calorimeter, approximate heat values may be determined by computation from the ultimate chemical analysis of certain fuels. Dulong's formula, below, in general use as a routine check of values determined by calorimeter, gives reasonably accurate results (within 2 or 3%) for most coals:

$$\text{Btu/lb} = 14{,}544\,C + 62{,}028\left(H_2 - \frac{O_2}{8}\right) + 4050\,S \quad (1)$$

In this formula, the symbols represent the proportionate parts by weight of the constituents of the fuel—carbon, hydrogen, oxygen, and sulfur—as established by an ultimate analysis (as-fired or other basis); the coefficients represent the approximate heating values of the constituents in Btu per lb. The term $\frac{O_2}{8}$ is a correction applied to the hydrogen in the fuel to account for the hydrogen already combined with oxygen in the form of moisture. The formula is not generally suitable for calculating the Btu values of gaseous fuels.

In the case of gaseous mixtures where the proportionate parts by volume or weight are readily available, the heating value may be accurately computed from a table of the heat values of individual constituents, which values have been definitely fixed by numerous calorimetric experiments, as noted above.

High and Low Heat Values

Most commercial fuels contain hydrogen as one of the constituents, and water, H_2O, is formed as a product of combustion when the hydrogen is burned in air. This water may remain in the vapor state, or it may be condensed to the liquid state, giving a substantial difference in the heat value. In determining the heat given up by a unit of such fuel, two values may be reported—the high, or gross, heat value and the low, or net, heat value. For the high, or gross, heat value, it is assumed that any water vapor formed by the burning of the hydrogen constituent is all condensed and cooled to the initial temperature in the calorimeter at the start of the test. The heat of vaporization is therefore present in the reported value. For the low, or net, heat value, it is assumed that none of the water vapor condenses and that all of the products of combustion remain in a gaseous state.

*Actually, when substances react to form other substances, a definite amount of heat is either evolved or absorbed. The reaction of C with H_2O to form CO and H_2, in which about 5900 Btu per lb C is absorbed, is an example of heat absorption. Boiler practice in general, however, is concerned with combinations involving the evolution of heat.

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The high, or gross, heat value is given directly and accurately by proper and well-defined (ASTM) laboratory procedure, using prescribed equipment. Direct determination of the low heat value is difficult, and the low heat value, where this value is to be used, is usually calculated from the high heat value, thus:

$$Q_L = Q_H - 1040\ w \qquad (2)$$

where:

Q_L = low heat value of the fuel, Btu per lb
Q_H = high heat value of the fuel, Btu per lb
w = lb water formed per lb of fuel
1040 = a factor commonly used to reduce high heat value at 80 F and constant volume to low heat value at constant pressure

In the United States the practice in boiler combustion calculations is to use the high heat values, since these values are available directly from calorimeter determinations and also because of the established custom of buying fuel on a high-heat-value Btu basis. In Europe generally, low heat values are used.

Oxygen and Nitrogen in Air

Theoretically, it is easy to assume that sufficient oxygen is present for combination with the combustibles and that the temperature required to bring about the chemical reactions of combustion can be attained. However, even with proper temperature conditions, the most important and difficult problem in the burning of all fuels is the manner of the physical introduction of oxygen to the combustible substances to effect complete oxidation and at the same time to assure the utilization of all or a maximum proportion of the oxygen supplied.

Usually, in the generation of heat, the oxygen required for combustion is supplied by using air, and the other constituents of the air act as diluents. However, in certain instances, particularly when the products of combustion must be minimized because of disposal problems, pure or slightly diluted oxygen is used. In these designs the heat potentials are exceptionally high, since all of the heat of combustion is transferred to a greatly reduced quantity of combustion products.

Atmospheric air is a mixture—as distinguished from a chemical compound—of oxygen, nitrogen, and small amounts of carbon dioxide, water vapor, argon, and other inert gases. If N'_2 is taken to represent the aggregate of nitrogen (N_2) and the small amounts of CO_2, argon, and other inert gases, then air may be considered to be 23.15% O_2 and 76.85% N'_2 by weight. Since in most combustion work the rare inert gases are negligible in effect, it is sufficient to consider N'_2 as consisting wholly of nitrogen, N_2. Usually accepted values of O_2 and N_2 in air are given in Table 3.

Table 3

Per Cent Oxygen and Nitrogen in Air
(Rare inert gases included as N_2)

	Per Cent by Weight	Per Cent by Volume
Oxygen, O_2	23.15	21.0
Nitrogen, N_2	76.85	79.0

Because of the proportionate parts by weight of oxygen and nitrogen, 0.2315 and 0.7685, respectively, to supply 1 lb of oxygen for combustion it is necessary to supply $1/0.2315 = 4.32$ lb of air, and in this amount of air there are $4.32 \times 0.7685 = 3.32$ lb of nitrogen, which do not enter directly into the combustion process but remain present nevertheless.

Oxygen, with its strong affinity for the combustible constituents of the fuel under proper conditions of temperature, is separated from the oxygen-nitrogen mixture (air) and joins the combustible in chemical combination to form, with the nitrogen, the products of combustion. The heat liberated by combustion is transferred by radiation, convection, and conduction to the heat-exchange apparatus, and ultimately on leaving, the combustion products carry heat to waste (the stack). The quantity of heat so wasted is directly proportional to the temperature and the amount, at exit, of the products of combustion (commonly called flue gas at this point). Hence, the air used in modern combustion practice is generally as carefully measured as the fuel, and great care is taken to effect a tight setting to prevent the admission of air except in the zone where it unites with the fuel.

Ignition Temperatures

Every combustible substance has a minimum ignition temperature, which must be attained or exceeded, in the presence of oxygen, if combustion is to ensue under given conditions. This ignition temperature may be defined as the temperature at which more heat is generated by the reaction than is lost to the surroundings, and combustion thus becomes self-sustaining. Ignition temperatures depend to a considerable extent upon the surrounding conditions, such as pressure (generally lower as pressure increases). The term usually applies to rapid combustion in air at atmospheric pressure.

The ignition temperatures of combustible substances vary greatly, as indicated in Table 4, in which minimum temperatures and temperature ranges in air are given for fuels and for the combustible constituents of fuels commonly used in the commercial generation of heat.

The ignition temperatures of the gases of a coal vary considerably and are appreciably higher than the ignition temperatures of the fixed carbon of the coal. However, the ignition temperature of coal may be considered as the ignition temperature of its fixed carbon content, since the gaseous constituents are usually distilled off, but not ignited, before this temperature is attained.

Table 4

Ignition Temperature in Air*
(At Pressure of One Atmosphere)

Combustible	Formula	Temperature, F
Sulfur	S	470
Charcoal	C	650
Fixed carbon (bituminous coal)	C	765
Fixed carbon (semibituminous coal)	C	870
Fixed carbon (anthracite)	C	840-1115
Acetylene	C_2H_2	580-825
Ethane	C_2H_6	880-1165
Ethylene	C_2H_4	900-1020
Hydrogen	H_2	1065-1095
Methane	CH_4	1170-1380
Carbon monoxide	CO	1130-1215
Kerosene	—	490-560
Gasoline	—	500-800

* Rounded-out values and ranges from various sources; a guide only.

The chemical combinations resulting from the union of oxygen with the combustible elements and compounds of some fuels commonly used in the generation of heat are given in Table 2. From the chemical combinations for these and many other fuels used in the generation of heat (not listed in Table 2) and the use of the molecular weights and volumes of the elements and compounds involved, their proportionate parts by weight and volume in the reactions as well as the weight and volume of the products of combustion (flue products or "flue gases") have been computed as listed in Table 1. The amount of air required to supply the oxygen is also included in this table. Since the oxygen comes from the air, the amount of air and accordingly the amount of nitrogen are fixed when the amount of oxygen has been established. Computations may be demonstrated by the following examples with atmospheric air as the oxidant.

Oxygen Required for Combustion

From the second reaction in Table 2, it is evident that one molecule of carbon combines with one molecule of oxygen to form one molecule of carbon dioxide:

$$C + O_2 = CO_2$$

Using molecular weights, this reaction is represented by:

$$12 + 32 = 44$$

That is, when burning carbon to carbon dioxide, 12 parts by weight of carbon (the approximate molecular weight of C) combine with 32 parts by weight of oxygen (the molecular weight of O_2) to form 44 parts by weight of carbon dioxide (the approximate molecular weight of CO_2). By simple division, 1 lb of carbon plus 2.66 lb of oxygen yield 3.66 lb of carbon dioxide.

The equation, $C + O_2 = CO_2$, also fixes the proportions by volume of the elements and compound in this reaction, represented by the coefficients (1 in this case) of each chemical symbol. However, all the constituents must be considered as gaseous; hence, in the case of carbon (a solid) the fact is of academic interest only at this point. More suitable illustrations may be given:

$$2H_2 + O_2 = 2H_2O$$

where 2 volumes of hydrogen plus 1 volume of oxygen combine to form 2 volumes of water vapor. Also,

$$CH_4 + 2O_2 = CO_2 + 2H_2O$$

where 1 volume of methane plus 2 volumes of oxygen combine to form 1 volume of carbon dioxide and 2 volumes of water vapor.

Another example may be given:

$$2C_2H_2 + 5O_2 = 4CO_2 + 2H_2O$$

where 2 volumes of acetylene plus 5 volumes of oxygen combine to yield 4 volumes of carbon dioxide and 2 volumes of water vapor.

For the volume relationship to hold, all the gases involved must be at the same conditions of temperature and pressure.

It may be noted in the three illustrations above that in the first and third the volumes of the final products of combustion are less than the volumes of the original constituents and that in the second the volume of the final products of combustion is the same as the volume of the original constituents.

The Mole in Combustion Calculations

The volume relationship of gaseous combustion constituents is particularly important in calculations on the "mole" basis (see below) since, for the same conditions of temperature and pressure, the volume of the mole is constant for all so-called perfect gases. Hence, as noted in Table 1, the ratios of the elements and compounds to the combustibles in the reactions

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are numerically equal, whether expressed as cu ft per cu ft of combustible or as moles per mole of combustible.

Combustion calculations involving gaseous mixtures can be simplified by the use of the pound-mole unit of weight, generally known as the mole.* Since, from the fundamental gas laws, equal volumes of gases at any given pressure and temperature contain the same number of molecules, the weights of equal volumes of gases are proportional to their molecular weights. If M is the molecular weight of the gas, 1 mole equals M lb (or M grams, depending upon the system of units). For instance:

1 mole $O_2 = 32$ lb O_2
1 mole $H_2 = 2$ lb H_2
1 mole $CH_4 = 16$ lb CH_4

That the volume of 1 mole at a given pressure and temperature is approximately fixed and independent of the kind of gas may be demonstrated as follows (see Table 1):

At 60 F and atmospheric pressure (14.7 psia) the specific volume of oxygen is 11.819 cu ft per lb. Therefore, 1 mole of oxygen has a volume of $32 \times 11.819 = 379$ cu ft. Similarly, at 60 F and atmospheric pressure, the specific volume of hydrogen is 187.723 cu ft per lb, and 1 mole has a volume of $2.016 \times 187.723 = 379$ cu ft. This volume, 379 cu ft, therefore approximates the volume of 1 mole of any gas at the assumed standard pressure and temperature. The volume of the mole at atmospheric pressure is in direct proportion to the absolute temperature.

As noted above, molal ratios and volume ratios (gaseous) are numerically equal. By volume (see Table 3), air consists of 21 parts O_2 and 79 parts N_2. This relationship can be expressed as 21 volumes O_2 and 79 volumes N_2 per 100 volumes air or as 21 moles O_2 and 79 moles N_2 per 100 moles air.

As indicated by the following example for a fuel gas, molal calculations have a simple and direct application to gaseous fuels, where the analyses are usually reported in per cent on a volume basis.

Fuel Gas Analysis by Volume in Per Cent	
CH_4	85.3
C_2H_6	12.6
CO_2	0.1
N_2	1.7
O_2	0.3
Total	100.0

This analysis may also be expressed as 85.3 moles of CH_4 per 100 moles of fuel; 12.6 moles of C_2H_6 per 100 moles of fuel; etc.

*The validity of this method depends upon the fact that most gases follow rather closely the fundamental laws for so-called perfect gases.

Keeping in mind that the molecular form is taken as unity, the elemental breakdown of each constituent may also be designated in moles per 100 moles of fuel, as indicated in the following tabulation for the above analysis:

C in $CH_4 = 85.3 \times 1 =$	85.3	moles
C in $C_2H_6 = 12.6 \times 2 =$	25.2	moles
C in $CO_2 = 0.1 \times 1 =$	0.1	moles
Total C per 100 moles fuel =	110.6	moles
H_2 in $CH_4 = 85.3 \times 2 =$	170.6	moles
H_2 in $C_2H_6 = 12.6 \times 3 =$	37.8	moles
Total H_2 per 100 moles fuel =	208.4	moles
N_2 per 100 moles fuel = $1.7 \times 1 =$	1.7	moles
O_2 per 100 moles fuel = $0.3 \times 1 =$	0.3	moles

In general, for each mole of C, H_2, and S burned, one mole of CO_2, H_2O, and SO_2, respectively, will be formed.

An analysis of the flue gas (always made on a volume basis, dry, by Orsat or other gas analyzers) produced by burning a gas fuel of the composition given above will be, say, in per cent, 10.4 CO_2, 2.8 O_2, and 86.8 N_2. These percentages may also be expressed as 10.4 moles CO_2, 2.8 moles O_2, and 86.8 moles N_2, each per 100 moles of flue gas. Since, from the fuel analysis (above), there are 110.6 moles C per 100 moles fuel, the same number of moles CO_2 will be formed by the combustion of the 100 moles of fuel. Hence, the moles of flue gas per mole of CO_2 will be $100 \div 10.4 = 9.62$. Therefore, the 100 moles of fuel will yield $110.6 \times 9.62 = 1064$ moles of dry flue gas. Thus by the application of the mole method, an important value has been quickly determined through knowing only the flue gas analysis and fuel analysis. Assuming the molecular weight of flue gas to be approximately 30 (air is 29), the weight of 1064 moles of flue gas is approximately $1064 \times 30 = 31{,}920$ lb per 100 moles of fuel. The weight of 100 moles of the gas fuel is the sum of the products of the moles of each constituent in the fuel and its molecular weight. The weight of dry flue gas produced per lb of fuel can be determined by dividing the weight of 100 moles of flue gas by the weight of 100 moles of fuel.

For complete applications of the mole method in combustion calculations, including an exact evaluation of the molecular weight of flue gas and other quantity determinations, reference should be made to the examples in the Appendix.

Combustion Calculations—General

The purpose of combustion calculations is to determine a) the quantities of the constituents involved in the chemistry of combustion, b) the heat thus released, and c) the efficiency of the combustion process under ideal and under practical conditions. The results of these calculations are necessary for the determination of heat transfer and performance of heat transfer equipment, such as boilers and their related component parts (see Chapters 7 and 11).

The determination of the quantities of the constituents involved in combustion is necessary to establish the size and performance characteristics of equipment, such as fuel pumps or feeders, and fans and ducts for both air and combustion gases. The necessity of knowing the amount of heat released per unit of fuel or per unit of time is apparent.

The heat input and the losses in burning fuel in air must be determined and segregated by combustion calculations in order to establish the efficiency of heat transfer to the heat exchanger. Knowing the amounts of each of the various losses is particularly helpful in deciding how efficiency may be improved through the possible reduction of certain of the losses. For instance, only enough excess air should be supplied completely to burn the fuel, since more than this amount increases the heat rejected to the stack. A portion of the stack loss is, therefore, a controllable loss and should be held to a minimum by close control of excess air.

Another prime factor affecting efficiency and losses is, of course, temperature, particularly the temperature of combustion gases finally rejected to the stack. The temperatures throughout the boiler or heat exchange unit depend closely upon the manner of heat transfer, discussed in the above references (Chapters 7 and 11), and, therefore, combustion calculations are not generally concerned with these temperatures.

However, the adiabatic, or "theoretical," temperature (without gain or loss of heat), that might conceivably be attained in burning a fuel under certain conditions, is sometimes calculated. Even though this temperature never exists in actual practice, it is of value in estimating furnace temperatures and furnace heat absorption.

The theoretical, or adiabatic, temperature is the maximum gas temperature that can be obtained under certain conditions. With less excess air or higher air preheat temperature, the gas temperature will be higher. If the gas temperature exceeds about 3200 F, a phenomenon occurs in which the CO_2 and H_2O constituents of the combustion gases tend to split into their component parts. This process is called gas dissociation. The effect of this reversal in the combustion process is to reduce the availability of the heat of combustion and thereby reduce the heat producing high temperatures. However, since the furnace exit temperatures are usually not high enough to be affected by this phenomenon, gas dissociation is seldom considered in boiler combustion calculations.

Calculations by Weight and Mole Methods

The detailed steps in the solution of combustion problems are best illustrated by working examples. Four simple examples involving both solid and gaseous fuels are presented (see below) to show how calculations are made using the weight and the mole methods, respectively. It should be noted that in these examples the amount of moisture in the combustion air is taken as 0.0132 lb per lb of dry air, corresponding to conditions of 60% relative humidity and 80 F dry-bulb temperature. This value is often used as standard in combustion calculations.

The values in examples 1 to 4 were generally developed from the data in Table 1, and the procedure followed is apparent from the execution. The results are limited to the determination of the quantities of fuel and air constituents involved, and heat balances are not included in these examples. See *Combustion Air, Btu Method* below and the examples of combustion calculations appearing in the Appendix for suitable methods in heat balance determinations.

Example 1

Weight Method—Solid Fuel—Standard Air*

(Complete Combustion Assumed)

See Table 1 for Multipliers

Ultimate Analysis, lb/lb Fuel As Fired		Required for Combustion, lb/lb Fuel @ 100% Total Air: O_2	Dry Air
C	0.728×2.66; 11.53	1.936	8.394
H_2	0.048×7.94; 34.34	0.381	1.648
O_2	0.062	———	———
N_2	0.015	———	———
S	0.022×1.00; 4.29	0.022	0.094
H_2O	0.035	———	———
Ash	0.090	———	———
Sum	1.000	2.339	10.136
Less O_2 in fuel (deduct)		−0.062	−0.268†
Required (at 100% Total Air)		2.277	9.868

†Air equivalent of O_2 in fuel.

Required for Combustion, lb/lb Fuel @ 125% Total Air	O_2	Dry Air
O_2 and Air $\times$ 125/100, total	2.846	12.335
Excess air = 12.335 − 9.868	———	2.467
Excess O_2 = 2.846 − 2.277	0.569	———

Products of Combustion

	lb/lb Fuel @ 125% Total Air
CO_2 0.728×3.66	2.664
H_2O $0.048 \times 8.94 + 0.035 + 0.0132 \times 12.335$	0.624
SO_2 0.022×2.00	0.044
O_2 (excess)	0.569
N_2 $12.335 \times 0.7685 + 0.015$	9.494
Weight, wet	13.395
Weight, dry = 13.395 − 0.624	12.771

*Air at 60% relative humidity and 80 F dry bulb, or 0.0132 lb of moisture per lb dry air.

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EXAMPLE 2

Weight Method—Gaseous Fuel—Standard Air

(Complete Combustion Assumed)

See Table 1 for lb/cu ft and Multipliers

Fuel Analysis, % by Vol As Fired		lb/lb Fuel	Required for Combustion, lb/lb Fuel @ 100% Total Air O_2	Dry Air
CH_4	90.0	0.832 × 3.99; 17.27	3.320	14.370
C_2H_6	5.0	0.087 × 3.73; 16.12	0.324	1.402
N_2	5.0	0.081	---	---
Sum	100.0	1.000	3.644	15.772
Required (at 100% Total Air)			3.644	15.772

	Required for Combustion, lb/lb Fuel @ 115% Total Air O_2	Dry Air
O_2 and Air × 115/100, total	4.191	18.138
Excess air = 18.138 − 15.772	---	2.366
Excess O_2 = 4.191 − 3.644	0.547	---

Products of Combustion

		lb/lb Fuel @ 115% Total Air
CO_2	0.832 × 2.74 + 0.087 × 2.93	2.535
H_2O	0.832 × 2.25 + 0.087 × 1.80 + 0.0132 × 18.138	2.265
O_2	(excess)	0.547
N_2	18.138 × 0.7685 + 0.081	14.020
Weight, wet		19.367
Weight, dry	= 19.367 − 2.265	17.102

EXAMPLE 3

Mole Method—Solid Fuel—Standard Air

(Complete Combustion Assumed)

See Table 1 for Mol. Wt and Multipliers

Ultimate Analysis, lb/100 lb Fuel AF		Moles per 100 lb Fuel	Required for Combustion, Moles/100 lb Fuel @ 100% Total Air O_2	Dry Air
C	72.8	6.07 × 1.00; 4.76	6.07	28.89
H_2	4.8	2.38 × 0.5; 2.38	1.19	5.66
O_2	6.2	0.19	--	--
N_2	1.5	0.05	--	--
S	2.2	0.07 × 1.00; 4.76	0.07	0.33
H_2O	3.5	0.19	--	--
Ash	9.0	---	--	--
Sum	100.0	8.95	7.33	34.88
Less O_2 in fuel (deduct)			−0.19	−0.90°
Required (at 100% Total Air)			7.14	33.98

° Air equivalent of O_2 in fuel.

	Required for Combustion, Moles/100 lb Fuel @ 125% Total Air O_2	Dry Air
O_2 and Air × 125/100, total	8.93	42.48
Excess air = 42.48 − 33.98	--	8.50
Excess O_2 = 8.93 − 7.14	1.79	--

Products of Combustion

		Moles/100 lb Fuel	% by Vol, Dry Basis
CO_2	6.07 × 1	6.07	14.60
H_2O	2.38 × 1 + 42.48 × 0.021† + 0.19	3.46	---
SO_2	0.07 × 1	0.07	0.17
N_2	42.48 × 0.79	33.56	80.91
O_2	(excess)	1.79	4.32
Wet		44.95	100.00
Dry	= 44.95 − 3.46	41.49	

†0.021 = (29 ÷ 18) × 0.0132

EXAMPLE 4

Mole Method—Gaseous Fuel—Standard Air

(Complete Combustion Assumed)

See Table 1 for Multipliers

Fuel Analysis, % by Vol, As Fired		Moles per 100 Moles Fuel	Required for Combustion, Moles/100 Moles Fuel @ 100% Total Air O_2	Dry Air
CH_4	90.0	90.0 × 2.0; 9.53	180.0	857.7
C_2H_6	5.0	5.0 × 3.5; 16.68	17.5	83.4
N_2	5.0	5.0	--	--
Sum	100.0	100.0	197.5	941.1
Required (at 100% Total Air)			197.5	941.1

	Required for Combustion, Moles/100 Moles Fuel @ 115% Total Air O_2	Dry Air
O_2 and Air × 115/100, total	227.1	1082.3
Excess air = 1082.3 − 941.1	---	141.2
Excess O_2 = 227.1 − 197.5	29.6	---

Products of Combustion

		Moles/100 Moles Fuel @ 115% Total Air	% by Vol, Dry Basis
CO_2	90 × 1 + 5 × 2.0	100.0	10.11
H_2O	90 × 2 + 5 × 3 + 1082.3 × 0.021‡	217.7	--
N_2	1082.3 × 0.79 + 5.0	860.0	86.90
O_2	(excess)	29.6	2.99
Wet		1207.3	100.00
Dry	= 1207.3 − 217.7	989.6	

‡0.021 = (29 ÷ 18) × 0.0132

For more complete combustion calculations involving heat balances, reference should be made to calculations on the mole basis (5 cases) and on the Btu basis (2 cases) in the Appendix. Reference curves of molal specific heats (Fig. 1) and enthalpies of flue gas and air (Figs. 2 and 3) for use with the calculations on the mole and Btu bases, respectively, will also be found in the Appendix.

General Formulas—Combustion Air

Since carbon, hydrogen, and sulfur are the only combustible elements found in the fossil fuels used for commercial steam generation, the lb of air (for zero excess air, 100% total air) theoretically required for the complete combustion of 1 lb of fuel is:

$$11.53\,C + 34.34\,(H_2 - O_2/8) + 4.29\,S \qquad (3)$$

where C, H_2, O_2, and S represent the proportionate parts by weight of carbon, hydrogen, oxygen, and sulfur, and the constants are those given in Table 1. The factor $O_2/8$ in the term $(H_2 - O_2/8)$ is a correction for the hydrogen already combined with the O_2 in the fuel to form water vapor.

With gaseous fuels, instead of breaking down the hydrocarbons into their constituent elements, it is simpler to use the amount of air for the various compounds directly as given in Table 1. For instance, for a gaseous fuel containing the combustible gases indicated in the expression below, the lb of air required per lb of fuel for complete combustion with theoretical air will be:

$$2.47\,CO + 34.34\,H_2 + 17.27\,CH_4 + 13.30\,C_2H_2 + 14.81\,C_2H_4 + 16.12\,C_2H_6 + 6.10\,H_2S - 4.32\,O_2 \qquad (4)$$

Again, the molecular symbols represent the proportionate parts by weight of the gaseous compounds and elements.

If, as is the usual custom, the analyses of gaseous fuels are given on a volumetric basis, the cu ft of combustion air required as given in Table 1 should be used. Thus, for a gaseous fuel containing the combustible gases indicated in the following expression, the cu ft of theoretical air required per cu ft of fuel for complete combustion will be:

$$2.38\,(CO + H_2) + 9.53\,CH_4 + 11.91\,C_2H_2 + 14.29\,C_2H_4 + 16.68\,C_2H_6 - 4.76\,O_2 \qquad (5)$$

where the molecular symbols represent the proportionate parts by volume of the gaseous compounds and elements.

It should be noted that the total air requirement will be reduced if oxygen is one of the constituents of the fuel.

The products of combustion can also be determined from the data given in Table 1. Assuming complete combustion with theoretical air of the fuels ordinarily used for commercial steam generation, the products of combustion in lb (including the nitrogen carried with the combustion air) per lb of fuel are:

$CO_2 = 3.66\,C$

$H_2O = 8.94\,(H_2 - O_2/8) + H_2O$°

$SO_2 = 2.00\,S$

$N_2 = 8.86\,C + 26.41\,H_2 + 3.29\,S + N_2$†

° Proportional weight of H_2O in the fuel as moisture.

† Proportional weight of N_2 in the fuel as nitrogen.

To this must be added the moisture carried in with the combustion air, to obtain the total weight of combustion products. The molecular symbols represent the proportionate parts by weight of the constituents in the fuel.

Combustion Air, Btu Method

In many combustion calculations the air required per lb of fuel can be established by using a convenient working tool, the theoretical air required per 10,000 Btu heat value of the fuel. The established values for the air requirements on this basis vary slightly for each particular fuel. Accurate values may be obtained from the formula:

$$\text{Lb theoretical air/lb fuel} = \frac{\text{Btu/lb fuel, as fired}}{10{,}000} \times (\text{values from Fig. 4 or Table 5 or 6}) \qquad (6)^{*}$$

* Approximate formulas sometimes used:

$$\text{Lb theoretical air/lb fuel} = \frac{\text{Btu/lb fuel, as fired}}{1300}$$

$$\text{Cu ft air (@ 60 F \& 29.92 in. Hg)/lb fuel} = \frac{\text{Btu/lb fuel, as fired}}{100}$$

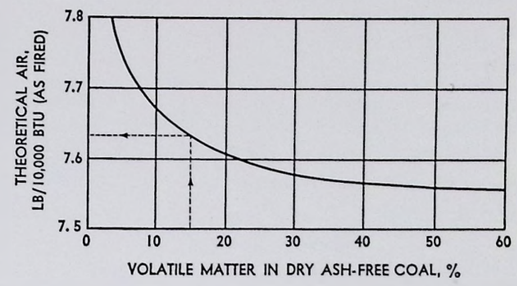


Fig. 4. *Theoretical air in lb per 10,000 Btu heat value of coal with a range of volatile*

For a typical heavy fuel oil† and a Texas natural gas‡, the theoretical air required and the moisture in the products of combustion in lb per 10,000 Btu (as-fired basis) are given in Table 5.

† Per Cent by Wt		‡ Per Cent by Vol	
S	1.16	CH_4	84.10
H_2	10.33	C_2H_6	6.70
C	87.87	CO_2	0.80
N_2	0.14	N_2	8.40
O_2	0.50	Sp gr	0.63
Btu/lb as fired,	18,400	Btu/cu ft as fired,	974

TABLE 5

Theoretical Air, Fuel, and Resulting Moisture per 10,000 Btu As Fired

Fuel	Theoretical Air§, lb/10kB	Fuel, lb/10kB	Moisture, lb/10kB
Fuel oil	7.46	0.544	0.51
Natural gas	7.19	0.496	0.93
Coal (prox anal.)	See Fig. 4	—	—
Coal (ult anal.)	See Table 6	—	—

§Dry air. To obtain wt of wet air required, moisture in air at standard conditions (0.0132 lb per lb dry air @ 60% relative humidity and 80 F dry bulb) must be added.

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The theoretical air requirements by the Btu method can be determined from Fig. 4 for coal if the proximate analysis is known and from Table 6 for any fuel if the ultimate analysis is available.

TABLE 6

Formula for Calculating Theoretical Air*

In lb per 10,000 Btu of Fuel As Fired
Ultimate Analysis of Fuel on As-Fired Basis, Per Cent by Weight

C = Carbon
H_2 = Hydrogen
O_2 = Oxygen
S = Sulfur
Btu/lb = Heat value of fuel

$$\text{Theoretical Air}\dagger,\ \text{lb (per 10kB)} = 144 \times \frac{8C + 24\left(H_2 - \frac{O_2}{8}\right) + 3S}{\text{Btu/lb}}$$

* This formula should be used only when the exact ultimate analysis and the correct heating value are given for the fuel.

† Dry air. To obtain wt of wet air required, moisture in air at standard conditions (0.0132 lb per lb dry air @ 60% relative humidity and 80 F dry bulb) must be added.

The following two simple examples illustrate the use of the Btu method to determine the amount of air required:

Bituminous Coal

Proximate Analysis As Fired, % by Wt	
Moisture	12.0
Volatile matter	25.8
Fixed carbon	46.2
Ash	16.0
Btu/lb	10,900

From this analysis, the volatile matter on a dry ash-free basis is $25.8 \times 100 \div (25.8 + 46.2) = 35.8\%$. Entering Fig. 4 with 35.8, the theoretical air in lb per 10,000 Btu as fired is 7.57, and the required total dry air (for 20% excess air) $= 7.57 \times 120/100 \times 10{,}900/10{,}000 = 9.90$ lb per lb of fuel.

From Table 5, for a heavy fuel oil of 18,400 Btu/lb, the theoretical air in lb/10,000 Btu as fired is 7.46, and the required total dry air (for 18% excess air) $= 7.46 \times 118/100 \times 18{,}400/10{,}000 = 16.19$ lb per lb of fuel.

The Btu method is entirely suitable for the rapid determination of complete heat balances and other data in combustion calculations, as will be noted by reference to the outline and examples in the Appendix.

Excess Air Held at a Minimum

As noted before and indicated in the preceding examples, it is necessary to use more than the theoretical air requirements to assure sufficient oxygen for complete combustion. Excess air would not be required if it were possible to have an ideally perfect union of the air and the fuel, and it must be held at a practical minimum to reduce the stack heat losses. Realistic values of excess air to the fuel-burning equipment (based on experience) required for burning various fuels are given in Table 7.

In most furnaces operating under suction there is also some leakage of air into the setting, and consequently the excess air leaving the furnace is greater than that entering the furnace as intended. This is not so in a pressure-fired unit.

Inherent and Avoidable Heat Losses

In the combustion of fuel for the commercial generation of heat, there are certain inherent heat losses over which no control is possible, as well as other heat losses which are subject to some control.

The inherent, or unavoidable, heat losses are the result of 1) the discharge of the products of combustion at a temperature higher than the temperature of the fuel, and 2) the moisture content of the fuel plus the combination of some of the hydrogen with the oxygen in the fuel.

The avoidable heat losses, or those which can be controlled to some extent by good design and careful operation, can be minimized by:

1. Careful control of excess air.
2. Tolerating virtually no unburned solid combustible matter in ash and refuse.
3. Permitting no unburned gaseous combustibles in the exit gases.
4. A well-insulated setting for the boiler and heat-exchanger components to reduce radiation.

The efficiency of combustion in a heat exchanger (boiler) is 100 minus the sum of the heat losses in per cent. This calculation is known as a heat balance and is illustrated in the combustion calculation examples in the Appendix.

TABLE 7

Usual Amount Excess Air Supplied to Fuel-burning Equipment

Fuel	Type of Furnace or Burners	Excess Air, % by Wt
Pulverized coal	Completely water-cooled furnace for slag-tap or dry-ash-removal	15-20
	Partially water-cooled furnace for dry-ash-removal	15-40
Crushed coal	Cyclone Furnace—pressure or suction	10-15
Coal	Stoker-fired, forced-draft, B&W chain-grate	15-50
	Stoker-fired, forced-draft, underfeed	20-50
	Stoker-fired, natural-draft	50-65
Fuel oil	Oil burners, register-type	5-10
	Multifuel burners and flat-flame	10-20
Acid sludge	Cone- and flat-flame-type burners, steam-atomized	10-15
Natural, Coke-oven, & Refinery gas	Register-type burners	5-10
	Multifuel burners	7-12
Blast-furnace gas	Intertube nozzle-type burners	15-18
Wood	Dutch-oven (10-23% through grates) and Hofft-type	20-25
Bagasse	All furnaces	25-35
Black liquor	Recovery furnaces for kraft and soda-pulping processes	5-7

Measurement of Total or Excess Air

To accomplish effective control over avoidable heat losses, the continuous measurement of the air used is a necessity in modern practice. The amount of air used can be determined by a) continuously indicated or recorded flue gas analysis and b) metering the combustion air or the flue gas.

Flue Gas Analysis

In the continuously recording or indicating flue gas analyzer, gas is continuously drawn from a selected location, and samples are analyzed at intervals of 1 minute or longer. Both the performance of the analysis and the indication or record of the results are automatic. Depending upon the type of instrument, the analysis may or may not include all the constituents of the products of combustion. Many of the continuously recording instruments give the per cent of CO_2 only; others only the per cent of O_2; and still others the per cent of unburned gases. Unburned gases, however, should not be present with proper combustion in the field of heat production.

The amount of O_2 in the flue gases is significant in defining the status of the combustion process. Its presence always means that more oxygen (excess air) is being introduced than is being used. Assuming complete combustion, low values of O_2 in the flue gases reflect moderate (nearly correct) excess air and reduced heat losses to the stack, while higher values of O_2 mean needlessly higher stack losses. The quantitative determination of total air (total air = 100 + % excess air) admitted to an actual combustion process requires a complete flue gas analysis for CO_2, O_2, CO, and N_2 (by difference) or the direct measurement of the air supplied by a suitable fluid meter (see below).

The approximate per cent total air from flue gas analyses may be determined from the curves of Fig. 5.

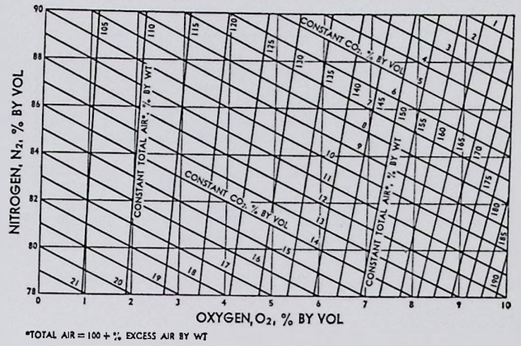


Fig. 5. *Chart for approximating total air from flue gas analysis*

A formula that has long been used for approximating the per cent excess air from an Orsat analysis is:

$$\%\ \text{Excess Air} = 100 \times \frac{O_2 - CO/2}{0.264N_2 - (O_2 - CO/2)}$$

The results are reasonably close for flue gases from hydrocarbon fuels of low nitrogen content; the error increases with increase in the per cent nitrogen in the fuel.

A convenient graph for the rapid conversion from CO_2 to excess air, and vice versa, is given in Fig. 6, using the appropriate factor from Table 8 according to the fuel burned.

TABLE 8

Fuel	Values of K
Coal, average	0.056
W. Va. (13,500 Btu/lb)	0.055
Kincaid	0.051
New River	0.047
Anthracite	0.019
Colorado lignite	0.033
Oil, range:	0.102—0.149
For 18,000 Btu/lb	0.123
Acid sludge	0.156
Coke-oven gas	0.487
Natural gas	0.331
Blast-furnace gas	−0.207
Petroleum coke	0.051
Coke, range:	0.004—0.008
Wood	0.016
Bagasse	0.008

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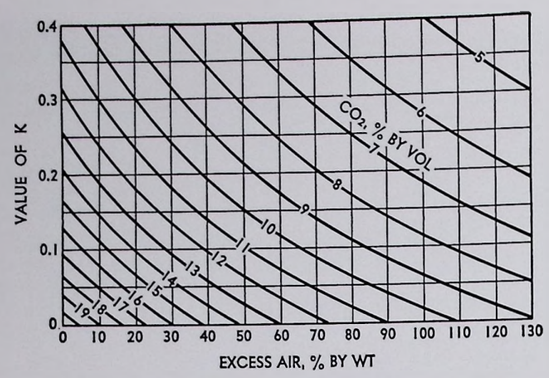


Fig. 6. *Excess air-CO_2 conversion curves*

Flue Gas Sampling. Great care should be taken to secure truly representative samples of the gas for analysis. The usual practice for a manually operated gas analyzer is to take successive samples from a number of points, laid out in "checkerboard" fashion over a cross section of the flue or area traversed by the gas. The number of sampling points and their position are best determined by preliminary trial analyses of gas samples from tentative locations. If the values from point to point vary widely, more sampling locations across the plane should be used.

Gas samples should be drawn at regular intervals over a relatively long period (during the entire period of a formal test). Unless operating conditions are exceptionally uniform, which is unlikely, a few samples drawn at irregular intervals are of little use in obtaining a true analysis.

Fixed-position samplers of the branched-pipe type, extending into the flue area, are liable to give misleading results, since the proportion of gas drawn into each branch may or may not correspond to the flows over the flue cross section. A better arrangement is to insert a single sampling element in the flue, at a location established by thorough preliminary analyses, from which samples can be drawn representing a fair average. Samples for the automatic mechanical gas analyzers are frequently drawn through a single sampling pipe carefully located in this manner.

When the temperatures exceed about 900 F, gas samples should be drawn through a water-cooled pipe to avoid the loss of O_2 from oxidation of pipe material at about this temperature. Noncooled pipes of special steels have been used with fair success, but there is still the uncertainty of the loss of some O_2 to the metal. A properly designed water-cooled sampler can be safely used to obtain gas samples from the hottest regions of a furnace (3000 F and above). The completeness of combustion progressively through the furnace is sometimes determined by the analysis of samples taken in this manner. Construction of a water-cooled sampler suitable for this duty is similar to that of the water-cooled portions of the HVT probe illustrated in Chapter 6.

Metering Combustion Air

Another method of establishing the amount of total air used in a boiler or other similar combustion heat exchanger is to meter the flow of air or the flow of combustion gases through the unit. Furthermore, for a given total air, the flow so measured has a nearly straight-line relationship to the Btu input. In a steam boiler, the steam output (and consequently the Btu output) bears a nearly constant relation to the Btu input. Metered steam is, therefore, a suitable index of Btu input, and the operator may thus proportion the combustion air to the fuel at any rate of steam output.

As a necessary adjunct to obtaining good boiler efficiency, instruments recording the air-flow steam-flow relationship are included in most modern boiler installations. The instrument in general use for this function is illustrated by Fig. 7 and is known as the Boiler Meter. Since the air quantity can be adjusted to the fuel quantity in the proper proportion at all times, manually by the operator or by automatic combustion-control equipment, it is not necessary to take and analyze flue gas samples in regular operation. In the initial setting of the Boiler Meter, however, it is necessary to take and to analyze such samples, and most boiler rooms are equipped to make periodical checks with an Orsat analyzer.



Fig. 7. *Boiler Meter for indicating and recording ratio of air used to Btu input*

5

STACKS, FANS, AND DRAFT

ATMOSPHERIC AIR is a mechanical mixture of gases. Its constituents are oxygen, nitrogen, and small amounts of carbon dioxide, water vapor, argon, and other inert gases. In steam boiler work, the "trace" gases in air are included in the nitrogen content, giving the generally accepted values (a) by volume: oxygen 21.0%, nitrogen 79.0% and (b) by weight: oxygen 23.15%, nitrogen 76.85%. On this basis, the molecular weight for air is commonly accepted as 28.9, or simply as 29.

From the established gas laws, the relationship between pressure, volume, and temperature of any weight of a "perfect" gas is expressed by:

$$pV_L = W_L R_g T \quad (1)$$

hence $$pv = R_g T \quad (2)$$

and $$pMv = MR_g T = RT \quad (3)$$

where:

p = absolute pressure, lb/sq ft
V_L = total volume of the gas, cu ft, at p and T conditions
W_L = total weight of gas, lb
T = absolute temperature, R = F + 460
R_g = specific gas constant, peculiar to each gas
v = specific volume of the gas, cu ft/lb, at p and T conditions
M = molecular weight of the gas

In (3), Mv = a constant, approx 379 cu ft at 60 F and 14.7 psia (the volume of 1 mole of any gas at these p and T conditions, see Chapter 4)

and $MR_g = R$, the "universal gas constant"

From (3): $MR_g = R = pMv/T = 1544$ (approx) (4)

The value of the specific gas constant, R_g, may be computed from equation (4) for gases which can be considered as obeying the gas laws:

$$R_g = 1544/M \quad (5)$$

For practical purposes, within the range of ordinary pressures and temperatures used, air may be considered as complying with these requirements.

Since the volume of a gas is a function of both temperature and pressure, it is necessary, as a suitable basis for comparison, to refer volumes to a standard set of conditions, now ordinarily accepted as pressure of 14.7 psia and variously at temperatures such as 32 F, 60 F, 70 F, and 80 F, depending upon the field of application. The specific volume and the specific weight of air at atmospheric pressure and various temperatures are given in Table 1. The specific gravities of the gases in boiler flue products are given in Table 1, Chapter 4.

The Flow of Air and Gases

In nature, a force varying in intensity from a light breeze to a gale or hurricane moves very large quantities of air with velocities up to 150 mph. This force is caused by the earth's rotation and by differences in atmospheric pressures in different locations on the earth's surface and in the stratosphere. Differences in pressures are caused primarily by temperature and moisture in the atmosphere. These unrestricted movements of the earth's atmosphere represent the expenditure of tremendous amounts of energy.

On the other hand, in steam power boilers the flow of air and combustion gases is confined to ducts, boiler settings, flues, and stacks. To supply the correct amount of air for combustion and to remove the gases from the boiler setting, this flow must be established and controlled. This can be done by using a stack or mechanical devices, such as fans which act as pumps. Either the stack or fans or a combination of stack and fans produces a difference in pressure, causing gas flow through the boiler unit.

Draft. Draft is a term commonly used to designate a static pressure in a furnace, gas passage, flue, or stack. If the gas within a stack is hot, its specific weight will be less than that of the cool air outside the stack. Therefore, the unit pressure at the base of

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the stack resulting from the weight of the column of hot gas within the stack will be less than that of the column of external cool air. This difference in pressure will cause a flow of gas through an opening in the base of the stack. In the customary boiler application, the cool air on its way to the stack must pass through the furnaces of the operating boilers connected to the stack. In the process of combustion, the air is converted to heated flue gas. This heated gas will be displaced by more incoming cool air. The action becomes continuous, and flow is established.

Gases always press or push but never pull. To avoid confusion of terms, the value of drafts or draft pressures is referred to atmospheric pressure at the same elevation, and the plus or minus sign is used to designate whether the value is above or below the atmospheric pressure. A draft differential is the difference between draft pressures measured at two points. Draft readings in absolute pressure do not throw any more light upon the flow of gases than do readings taken with an atmospheric pressure datum of reference.

Forced Draft. When air or the products of combustion are caused to flow to or through a unit by maintaining them at a pressure above atmospheric, the system is said to be under forced draft. The fans that effect this condition are called forced-draft fans.

Induced Draft. When air or the products of combustion are caused to flow to or through a unit by maintaining them at a progressively increasing subatmospheric pressure, the system is said to be under induced draft. This condition is effected by stacks alone, which furnish so-called natural draft for low differential pressures, or by stacks supplemented by fans, known as induced-draft fans, where higher suctions are required.

Balanced Draft. Balanced draft is a term used in referring to a particular point in a furnace where the draft is zero. Usually in boiler operation the furnace is operated at slightly less (—0.1 in. of water) than atmospheric pressure. In present practice, however, many furnaces operate under positive pressure (see *Pressure Furnaces,* page 5-16).

Velocity Pressure. The pressure corresponding to the velocity of flow is known as the velocity pressure and is a measure of the kinetic energy of the fluid. It is equivalent to the dynamic head, or the pressure producing the flow, and can be measured by a pitot-static tube placed in the path of the gas. The relation between flow velocity and dynamic head is expressed, as in Chapter 8, by the fundamental formulas of fluid flow:

TABLE 1

Specific Volume and Specific Weight of Air

At Atmospheric Pressure

Temp, F	Sp Vol, cu ft/lb	Sp Wt, lb/cu ft	Temp, F	Sp Vol, cu ft/lb	Sp Wt, lb/cu ft	Temp, F	Sp Vol, cu ft/lb	Sp Wt, lb/cu ft
32	12.36	0.0809	160	15.58	0.0642	425	22.24	0.0450
35	12.44	0.0804	170	15.83	0.0642	450	22.87	0.0437
40	12.56	0.0796	180	16.08	0.0622	475	23.50	0.0425
45	12.69	0.0788	190	16.33	0.0612	500	24.10	0.0415
50	12.81	0.0781	200	16.58	0.0603	525	24.75	0.0404
55	12.94	0.0773	210	16.84	0.0594	550	25.38	0.0394
60	13.06	0.0766	220	17.09	0.0585	575	26.01	0.0385
65	13.19	0.0758	230	17.34	0.0577	600	26.64	0.0376
70	13.32	0.0751	240	17.59	0.0568	625	27.27	0.0367
75	13.44	0.0744	250	17.84	0.0561	650	27.90	0.0359
80	13.57	0.0737	260	18.09	0.0553	675	28.53	0.0351
85	13.69	0.0730	270	18.34	0.0545	700	29.15	0.0343
90	13.82	0.0724	280	18.60	0.0538	725	29.78	0.0336
95	13.94	0.0717	290	18.85	0.0531	750	30.40	0.0329
100	14.07	0.0711	300	19.10	0.0524	775	31.04	0.0322
110	14.32	0.0699	320	19.60	0.0510	800	31.67	0.0316
120	14.57	0.0686	340	20.10	0.0498	825	32.30	0.0310
130	14.82	0.0675	360	20.61	0.0485	850	32.93	0.0304
140	15.08	0.0664	380	21.11	0.0474	875	33.55	0.0298
150	15.32	0.0653	400	21.61	0.0463	900	34.18	0.0293

$$V = \sqrt{2gH} \tag{6}$$

and
$$H = V^2/2g \tag{7}$$

where:

V = velocity of the fluid, ft/sec
H = total head causing flow, ft of fluid flowing (at the same specific weight as the fluid)
g = gravitational constant, 32.2 ft/sec²

It is necessary clearly to understand the meaning of H in equations (6) and (7). It is helpful to visualize this H, or head in feet of fluid causing flow, by recalling that the pressure exerted by a stream of water, moving at a velocity V, on the open end of a pitot-static tube will support a column of water of a height in feet equal to the H in equations (6) and (7). Similarly, the pressure exerted by a stream of mercury, for instance, moving at a velocity of V, on the open end of a pitot-static tube will support a column of mercury of a height in feet equal to H in equation (6) or (7). If the velocity of the stream is the same, then the height, H, of the supported column will be the same whether the fluid is water or mercury.

While it is not so easy to visualize, the pressure exerted on the open end of a pitot-static tube inserted in a moving stream of air will also support a similar column of air of the same height, H, as the supported columns of water or mercury, if the velocity of the stream is the same in each case. For a velocity of 10 fps, for instance, it is evident from equation (7) that $H = 100/64.4 = 1.55$ ft of the fluid flowing, whether that fluid is water, mercury, or air. Thus, if H in equation (6) or (7) is always considered as being of the same physical composition as that of the corresponding moving fluid, it will not be difficult correctly to apply these fundamental formulas.

In the case of air, or gases generally, it is impractical to observe or read H in feet of gas. The head, however, may be conveniently observed as the equivalent height, in inches, of a column of visible liquid in the common U tube or draft gage. If water is used as the visible liquid, the following relation* applies:

$$H_w = 12H\,\gamma_g/\gamma_w \tag{8}$$

where:

H_w = head in in. of water
H = as in equations (6) and (7)
γ_g = sp wt of the gas flowing, lb/cu ft
γ_w = sp wt of water, lb/cu ft, at the temperature of the U tube (usually taken as 62.4 lb/cu ft)

Air flow or gas flow as used in boiler practice is the actual quantity of material flowing through an enclosure per unit time. The term is usually expressed either as pounds per hour or cubic feet per minute. Flow is produced by a differential draft or differential pressure, but a differential draft does not necessarily produce a flow (as in a boiler setting closed against entrance of air but connected to an operating stack).

Air flow is related to velocity by the expression:

$$Q = AV \tag{9}$$

where:

Q = quantity flow rate, cfm
A = cross-sectional area of flow channel, sq ft
V = velocity, fpm

Draft Loss. The term "draft loss" is commonly applied to energy and friction losses caused by gas flow between any two points in a boiler setting. Gas flow in closed conduits always involves or is accompanied by draft loss.

Draft Differential. Draft differential on the other hand is a more inclusive term. It is simply a difference in pressure between two points. It may be used to denote a draft loss caused by flow or to indicate the difference in pressure between two points because of their relative position above a certain datum (see *Stack Effect* below). It may also be used to signify the difference in pressure between two points resulting from a combination of draft loss and stack effect.

Stack Effect. Stack effect, or chimney action, is the difference in pressure caused solely by the difference in elevation between two stations in vertical ducts or passages conveying heated gases. It is independent of gas flow losses. The intensity and distribution of this draft or pressure depend upon the vertical height and arrangement of the various ducts or passages and upon the gas temperature in each. Stack effect is of particular significance in a series of boiler passes and flues with the direction of gas flow variously up and down. Stack effect cannot be determined from simple draft gage readings. Such readings combine the stack effect and flow losses. Stack effect must be separately calculated. Fig. 1 illustrates the general method of its determination.

Three passages of heights AB, BC, and CD are shown. The passages contain heat-absorbing surface. Gases at different average temperatures, as indicated, are assumed to be momentarily brought to rest, as by a closure at A. The top of passage CD, at D, is in a region of atmospheric pressure.

In analyzing the stack effect for the arrangement shown in Fig. 1, it is best to establish the effect in each passage separately and in succession, assigning minus values when proceeding downward and plus values when proceeding upward. Thus, for the conditions indicated in the sketch and using the values in Table 2 for an ambient temperature of 80 F, the stack effect in in. of water, referred to atmosphere, in each component is as follows:

* See also Chapter 8.

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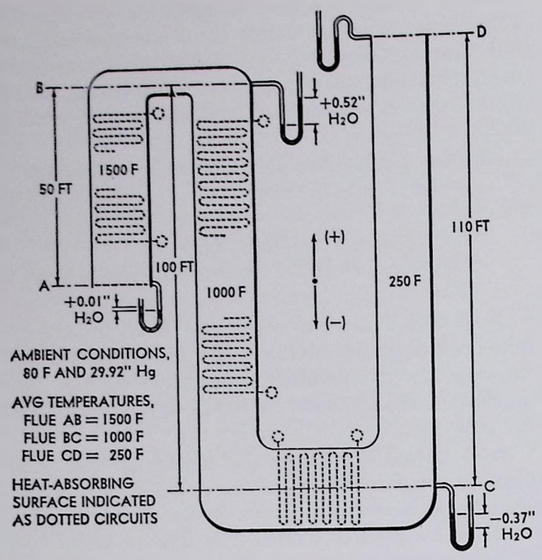


Fig. 1. *Diagram illustrating stack effect, or chimney action, in three vertical gas passes arranged in series*

Stack effect D to C = $110 \times 0.0034 = -0.37$ in.
Stack effect C to B = $100 \times 0.0089 = +0.89$ in.
Stack effect B to A = $50 \times 0.0102 = -0.51$ in.

If draft gages, one end open to atmosphere, were placed as indicated at A, B, and C, the theoretical zero-flow draft readings in in. of water would be:

Draft at A = $-0.37 + 0.89 - 0.51 = +0.01$ in.
Draft at B = draft at A + stack effect A to B
= $+0.01 + 0.51 = +0.52$ in.
Draft at C = draft at B + stack effect B to C
= $+0.52 - 0.89 = -0.37$ in.

For the conditions in the above example there is no net head available to produce flow up the component CD, even though the outlet lies above all parts of the arrangement. The net stack effect of the system in this case is detrimental rather than helpful in causing a flow of gas through the unit. Fans or stack height must, therefore, be selected not only to provide the necessary draft to overcome friction losses through the unit but also to allow for the stack effect of the system.

In some boiler settings, gases leak from the upper portions into the boiler room when the unit is operating at very low rates of output or when taken out of service, even though the outlet flue may show a substantial negative draft. The reason is evident from the above example, where there is a positive pressure of 0.52 in. of water at B and a negative pressure of 0.37 in. of water at C.

Barometric Pressure. Barometric pressure is the total deadweight of a unit column of air of the earth's atmosphere from the point of measurement to the limit of the atmosphere. At sea level, the normal barometric pressure is 29.92 in. of mercury, or 33.9 ft of water, or 14.7 psia. Values for barometric pressure at various elevations above sea level are given in Table 3.

TABLE 2

Stack Effect or Pressure Difference, In. of Water

For Each Foot of Vertical Height
(Barometer = 29.92 in. Hg)

Avg Temp in Flue,	Temperature of Air Outside Flue, F			
F	40	60	80	100
250	0.0045	0.0039	0.0034	0.0029
500	0.0073	0.0067	0.0061	0.0056
1,000	0.0101	0.0095	0.0089	0.0084
1,500	0.0114	0.0108	0.0102	0.0097
2,000	0.0122	0.0116	0.0110	0.0105
2,500	0.0127	0.0121	0.0115	0.0110

TABLE 3

Barometric Pressure
(Effect of Altitude)

Ft Above Sea Level	Pressure,* In. Hg
0	29.92
1,000	28.86
2,000	27.82
3,000	26.81
4,000	25.84
5,000	24.89
6,000	23.98
7,000	23.09
8,000	22.22
9,000	21.38
10,000	20.58

*Values from Bulletin No. 110, 1952, Air Moving and Conditioning Association, Inc.

Pressure. Pressure is the term normally used to indicate the condition existing within those parts of the air and gas system of a boiler unit that operate above atmospheric pressure. This term would apply throughout a pressure-fired unit and to that part of a forced- and induced-draft unit from the forced-draft fan to the point of balanced draft.

The Chimney or Stack

All the early boilers operated with natural draft. The draft necessary to produce the required air flow was provided by stacks. Smaller modern boilers may also use natural draft. However, for larger boilers equipped with superheaters, economizers, and air heaters, where the total draft loss is necessarily large, it is not practical or economical to obtain sufficient induced draft by use of a stack alone. In these units the stack serves as an aid to a mechanically driven draft fan.

The required height and diameter of a chimney, or stack, for natural-draft boilers depend upon:

1. Draft loss through the boiler from the point of balanced draft to the gas entrance to the stack. For natural-draft units (without forced-draft fan) the entire unit will be under suction, or induced draft.
2. The average temperature of the gases passing up the stack and the temperature of the surrounding atmosphere.
3. The quantity of gases carried by the stack.
4. The barometric pressure – which is substantially influenced by altitude (see Table 3).

There is no satisfactory formula covering all of the many factors involved in the determination of the height and diameter of the stack. For practical purposes, however, a sufficiently comprehensive and accurate method has been developed which covers the most important factors affecting stack draft: (a) the temperature of the surrounding atmosphere and the temperature of gases entering the stack; (b) the drop in temperature of gases within the stack because of heat loss to atmosphere and air infiltration; and (c) the stack flow loss from friction and exit losses.

(a) Theoretical Stack Draft. The suction that would be produced at the base of a stack with no flow losses is the theoretical stack draft. It results from the difference between the specific weights of the surrounding atmospheric air and the stack gases. As an example, if the atmospheric temperature is 60 F and the average temperature of the gases in the chimney is 500 F, the difference between the weights of the external air and the internal flue gases per cu ft is:

Weight of air at 60 F = 0.0766 lb/cu ft
Weight of gases at 500 F = 0.0415 lb/cu ft
Difference = 0.0351 lb/cu ft

If the stack is 100 ft high, the pressure exerted at the base is $0.0351 \times 100 = 3.51$ lb/sq ft. One lb/sq ft is equivalent to 0.192 in. of water. Under the above temperature conditions, the 100-ft stack would, therefore, show a theoretical draft of:

$3.51 \times 0.192 = 0.674$ in. of water

From the principles outlined above, the following formula may be derived:

$$\text{Stack draft} = 0.192L\ (\gamma_a - \gamma_g) \qquad (10)$$

Since the weight per cu ft of air (gas) varies inversely as the absolute temperature, the equation may also be written:

$$\text{Stack draft} = 0.52Lp\left\{\frac{1}{T_a} - \frac{1}{T_g}\right\} \qquad (11)$$

where:

Stack draft = induced draft, in. of water
L = stack height above gas entrance, ft
γ_a = sp wt of outside air, lb/cu ft
γ_g = sp wt of stack gases, lb/cu ft
p = atmospheric pressure, psia
T_a = temperature (absolute) of outside air, R = F + 460
T_g = average temperature (absolute) of stack gases, R = F + 460

Because the atmospheric, or barometric, pressure is usually listed in in. of mercury, the stack draft formulas may be written:

$$\text{Stack draft, in. of water} = 0.256Lp'\left\{\frac{1}{T_a} - \frac{1}{T_g}\right\} \qquad (12)$$

where:

p' = atmospheric pressure in in. of mercury

In developing formulas (11) and (12), as a simplification the sp wt of the flue gases is taken as equal to that of air at the same temperature, without appreciable error.

Under the temperature conditions of the preceding example, the 100-ft stack at an atmospheric pressure of 22.22 in. of mercury, corresponding to 8000 ft above sea level, would produce a theoretical draft of:

$$\text{Stack draft} = 0.256 \times 100 \times 22.22\left\{\frac{1}{520} - \frac{1}{960}\right\}$$

Stack draft = 0.50 in. of water

(b) Average Stack Temperature. The stack temperature used in the above example is an arithmetic average of the temperatures entering and leaving the stack. Gases passing through a stack lose some heat through the shell of the stack to the outside atmosphere. By the infiltration of cold air, stack temperatures may be lowered an appreciable amount. The loss in temperature depends upon the type of stack, stack diameter and its height, and a number of variables influencing the outside atmosphere. The approximate temperature change for stacks is indicated by the curves in Fig. 2.

(c) Stack Flow Loss. In the flow of gases through the stack there are friction and stack exit losses. Obviously, the available induced draft is the difference between the theoretical draft as determined by the draft formula and the flow loss in the stack proper. The friction flow loss may be determined by the first term of equation (13), which is based on the Fanning formulation (see Chapter 8, page 8-9). The second term of the formula represents the exit loss.

$$\text{Stack flow loss} = 0.0942fT_g\frac{L}{D_i^5}\left(\frac{w}{100{,}000}\right)^2 + \frac{0.0942T_g}{D_i^4}\left(\frac{w}{100{,}000}\right)^2 \qquad (13)$$

Stack flow loss =

$$0.0942\frac{T_g}{D_i^4}\left(\frac{w}{100{,}000}\right)^2\left(\frac{fL}{D_i} + 1\right) \qquad (14)$$

where:

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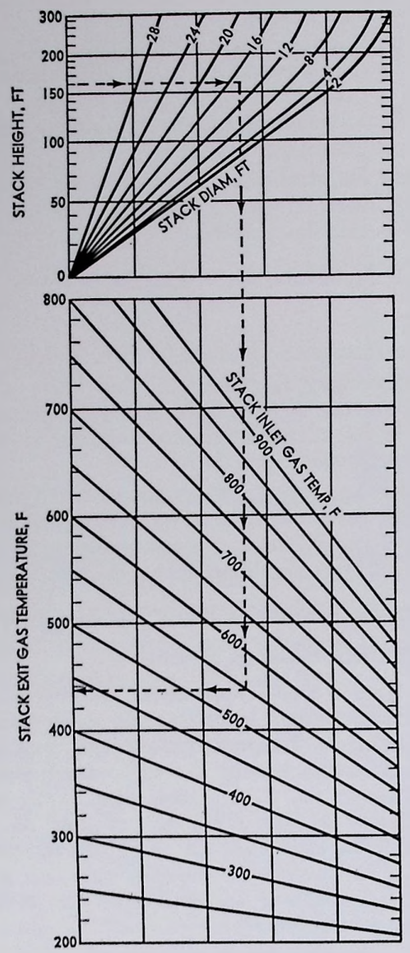


Fig. 2. *Approximate relationship between stack exit gas temperature and stack dimensions*

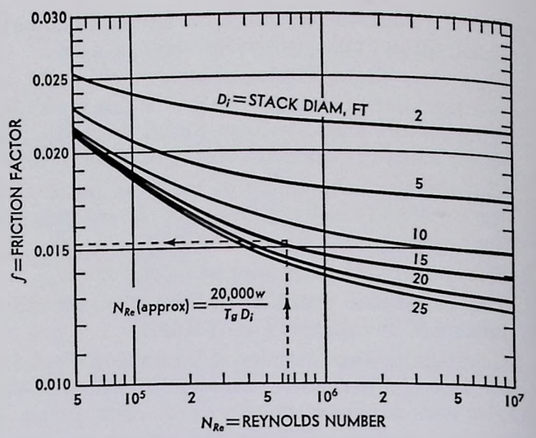


Fig. 3. *Friction factor, f, as related to Reynolds number and stack diameter, for use in equations (13) and (14)*

Stack flow loss is measured in in. of water

w = weight flow of gases, lb/hr
D_i = internal diameter of stack, ft
L = height of stack above gas entrance, ft
f = friction factor from Fig. 3
T_g = avg gas absolute temp, R = F + 460

The available draft is the theoretical draft, equation (12), minus the stack flow loss, equation (14). The total stack flow loss is usually less than 5% of the calculated draft. Of the total stack flow loss, the stack exit loss is much greater than the loss from friction—in the order of 3 to 7 times, depending upon the height and diameter of the stack.

Selection of Stack Size

In selecting a stack suitable for the requirements of a particular installation, it should be kept in mind that the construction cost for some one combination of diameter and height will be less than that for any other combination. The builder of stacks should be consulted before these dimensions are fixed. For convenience in making a tentative selection of optimum stack dimensions, the necessary data are given graphically in Figs. 2, 4, and 5 for sea level and 80 F atmosphere. The use of these graphs to establish stack dimensions is illustrated by the following example:

General Data

Type of Firing	Approx Gas Weight, lb/lb of Steam
Oil or gas	1.15
Pulverized coal	1.25
Stoker	1.50

Specific Data

Fuel	Pulverized coal
Steam generated, lb/hr	360,000
Stack gas flow, lb/hr	450,000
Stack inlet gas temp, F	550
Stack exit gas temp (assumed), F	450
Stack draft required (from point of balance to stack gas entrance), in. of water	1.0
Altitude of plant	Sea level

Find Economical Dimensions of Stack

1. Diameter of stack to nearest 6-in. increment, ft
2. Active height of stack, ft
3. Stack exit gas temperature, F
4. Total height of stack (add to item 2 as required), ft

Diameter. Enter graph, Fig. 5, with a stack gas flow of 450,000 lb/hr. This will give a stack diameter (to nearest 6-in. increment) of 14 ft 6 in.

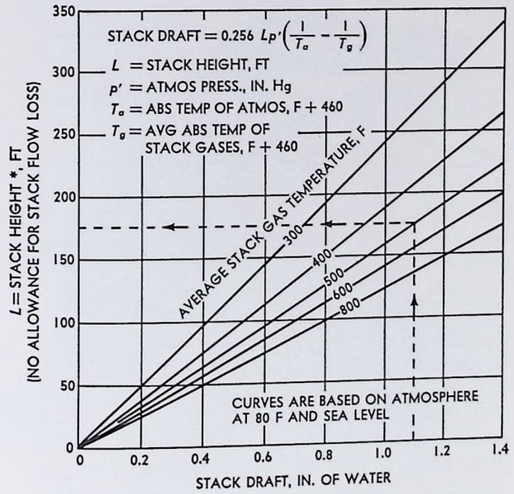
Height (approx). Enter graph, Fig. 4, with the required stack draft of 1 in. of water and an average gas temperature of (550 + 450)/2 = 500 F. This will give an approximate stack height of 160 ft.

Exit Gas Temperature. Enter graph, Fig. 2, with the approximate stack height of 160 ft and the diameter of 14 ft 6 in. This will give a stack exit temperature of 436 F, and the average stack temperature will be (550 + 436)/2 = 493 F.

Height (actual). Enter graph, Fig. 4, with the stack draft required increased by 10% as a factor of safety, and, using the average stack temperature of 493 F, determine a new stack height based on zero stack flow loss = 177 ft. For a stack flow loss taken at the usual arbitrary value of 5%, the final actual stack height = 177/0.95 = 186 ft. This is the active height above flue entrance, to which must be added any inactive section required from foundation to flue entrance.

The stack flow loss may be checked by using the above values for diameter, height, average gas temperature, and gas flow in equation (14). A check of the net draft available, by using equations (12) and (14), indicates that the 1-in. draft requirement is amply met. For economic reasons, the stack flow loss should be about 5% of the available draft, and the diameter, as the dominating factor in this loss, may be modified as necessary to suit this condition.

If the plant is not located at sea level, the stack draft required should be increased by the altitude factor, approx 30/B, where B equals the normal barometer reading, in in. of mercury, at the boiler site (see Table 3).



*STACK HEIGHT IS DEFINED AS THE HEIGHT ABOVE THE FLUE ENTRANCE.

Fig. 4. *Stack height required for a range of stack drafts and average stack gas temperatures*

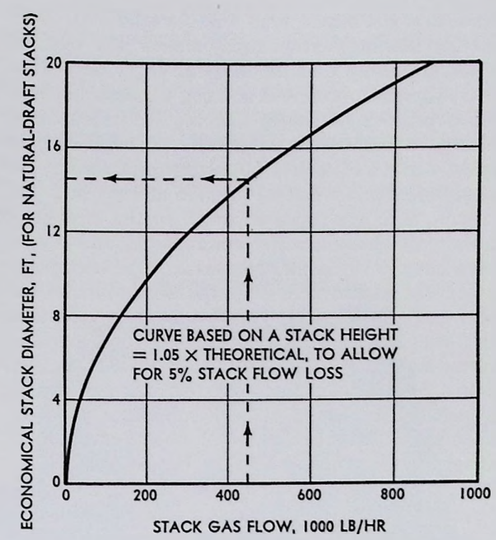


Fig. 5. *Economical stack diameter for a range of gas flows*

Stack Types

Normally a stack is thought of merely as a long cylinder extending high into the air, although there are other forms with which improved performance is sometimes possible in specific applications. In the evasé stack, a venturi design, a part of the velocity or kinetic energy of the gases is recovered before discharge into the atmosphere. In another type of stack, a fan is used as an integral part of the evasé feature to discharge air or gases at high velocity into the throat of the venturi section. In this way the gases are withdrawn from the setting by aspiration. Depending upon the application, the fan handles only a portion of the gases.

As one of the essential elements of any steam power plant, the stack is a symbol of industry. As noted, it may be used for moving the products of combustion through the boiler. In every instance it is used to convey the flue gases to the atmosphere. In many locations one of its purposes is to discharge otherwise noxious gases at a height sufficient for dispersion and dilution.

The wind pressure on a stack is a major consideration in its design. For stability, the diameter required for a tall stack is so great that the large volume in its base, up to the elevation of gas entry, has been economically used for material storage, condensate storage, air-raid shelters, and control rooms.

Air Pollution

While the stack is used successfully for dispersing the suspended dust particles over a large area, nevertheless, in locations where there is a concentration

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of industry, the atmosphere will often be overloaded with the discharge from many stacks. The effect is particularly objectionable on damp foggy days when the combination of smoke and fog blankets the area. Such a mixture of smoke and fog is termed *smog*, which is a suspension of minute particles of condensed vapor and carbon (the latter produced by the incomplete combustion of organic matter) in a gas, a vapor, or a gas-vapor mixture. In the interest of comfort and health, many communities and several states have established ordinances to restrict pollution of the atmosphere by requiring industrial furnaces and power plants to meet rigid specifications controlling the quantity and size of dust particles emitted from the stack. To comply with such regulations, dust collectors or precipitators are usually installed for coal-fired units. Some rather wide differences still exist in the requirements of the ordinances in different locations. An interim guide is the *Example Sections for a Smoke Regulating Ordinance* appearing in the ASME Information Bulletin, May 1949. A continuing committee of the ASME is working on a final code similar in purpose, in its field, to the ASME *Boiler and Pressure Vessel Code.*

Some power plants, located near airports, are prohibited from using stacks sufficiently high to discharge the gases for proper dispersion to eliminate excessive pollution. To approximate the dispersive effect of a higher stack, the stack may be necked down at the top to increase the discharge velocity. When this is done, the discharge velocity in fps from the stack is numerically the same as the number of ft that would have to be added to the stack to obtain the same dispersive effect. This rule of thumb applies with a 10-mph wind at the top of the stack. If the wind velocity increases, the effect of the high stack-discharge velocity will decrease. Naturally, necking down the stack adds an appreciable amount of stack resistance and can only be used in connection with a mechanical-draft system.

The Fan

A fan is a "pump" or volumetric machine (see Figs. 6 and 7) which can move a quantity of air or gas by imparting to it enough energy to set it into motion and to overcome all resistance to flow in its path. The fan usually consists of a bladed rotor, or impeller, which does the actual work, and a housing which collects and directs the gas discharged by the impeller. The power required depends upon the volume of gas moved, the head against which the gas is delivered, and the efficiency of the fan and its drives. The power may be expressed on several bases, e.g., shaft horsepower, input horsepower to motor terminals if motor driven, or theoretic horsepower computed by thermodynamic methods. Each basis has its own merits. As far as the fan itself is concerned, the vital figures are the power input to the shaft and the theoretic power dictated by thermodynamic calculations.

Air Horsepower and Fan Efficiency. The theoretic work can be evaluated from the pressure-volume diagram of the fan cycle, Fig. 6, where the suction phase is 1-2, compression phase is 2-3, and delivery phase is 3-4. The change in sp wt of gas between fan inlet and discharge is usually so small (less than 7% for a pressure difference of 1 psi) as to be negligible. As a consequence the gas can be considered incompressible, and the phase 2-3 is shifted to the vertical line 2-3′ (cf Fig. 8, Chapter 10). The theoretic work is then effectively represented by the rectangular area 1-2-3′-4, and its evaluation is by the equation:

$$\text{Work} = (p_{s'} - p_s) V_{Ls} = \triangle p \times V_{Ls} \qquad (15)$$

where:

$p_{s'}$ = delivery pressure, psfa
p_s = inlet pressure, psfa
V_{Ls} = volume, cu ft
$\triangle p = (p_{s'} - p_s)$, the pressure difference across the fan, psf

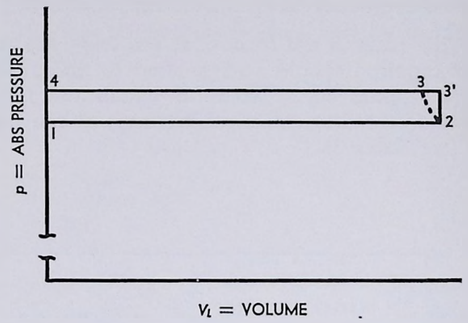


Fig. 6. *Pressure-volume diagram of the fan cycle*

The pressure difference is generally measured on a manometer in in. of water and identified as head, H_w. If the volume is given as the rate of flow, or capacity, in cfm, then substitution in equation (15) gives the theoretic or air horsepower:

$$\text{ahp} = \frac{H_w \times 5.2 \times \text{cfm}}{33{,}000} = 0.0001573 \text{ cfm} \times H_w \qquad (16)$$

where 5.2 (= 62.4/12) is the pressure in psf for a column of water 1 in. high and 33,000 ft-lb/min defines a horsepower.

In the application of equation (16), two interpretations are permissable for the value of the head. The static head, H_{sp}, or the larger value, total head, H_{TP}, which includes the velocity head, may be used. The results are:

$$\text{ahp (static)} = 0.0001573 \text{ cfm} \times H_{sp} \qquad (17)$$
$$\text{ahp (total)} = 0.0001573 \text{ cfm} \times H_{TP} \qquad (18)$$

If these values of air horsepower are referred to the shaft-horsepower input to the fan, the result is fan efficiency or mechanical efficiency (static or total):

$$\text{Mechanical (static) eff} = \frac{\text{ahp (static)}}{\text{shp}} \qquad (19)$$

$$\text{Mechanical (total) eff} = \frac{\text{ahp (total)}}{\text{shp}} \qquad (20)$$

These several equations can be applied to give the complete fan characteristics as plotted in Fig. 7.

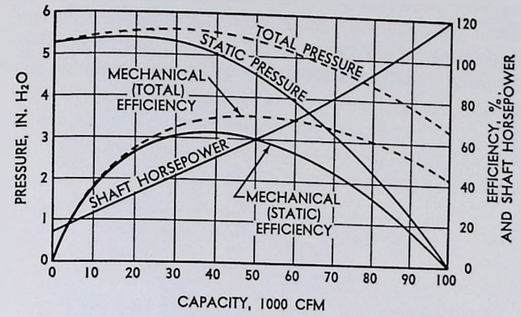


Fig. 7. *Typical fan performance curves*

In the application of fans to boiler draft service, it is customary to use static pressure and static efficiency rather than total pressure and total efficiency because the velocity is of negligible use in overcoming flow resistances. Figs. 13 and 14 show a variety of fan characteristics as encountered in boiler service.

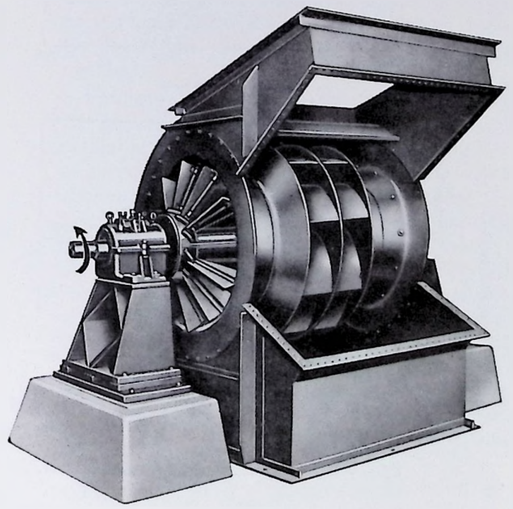


Fig. 8. *Fan with backward-curved blades and vane-control inlet, suitable for forced draft handling room air*

Fan Performance. Stacks seldom provide the draft required by modern boilers. A 100-ft-high stack, with gas at 400 F, will develop a draft of approximately 0.5 in. of water, whereas gas flow resistances as high as 50 in. of water may be encountered. Fans are generally used to supply the required mechanical draft. They are built in a variety of designs and types so that the user has adequate selection opportunity.

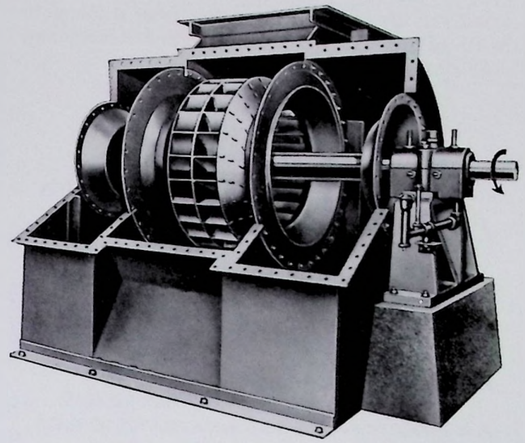


Fig. 9. *Fan with radial-tip blades and double inlet, arranged for induced-draft service*

All fans are basically fluid-acceleration devices. Their performance can be altered by changing one or many of a multitude of design variables. The details of such variations and the underlying theory are not properly subjects for treatment in this book.

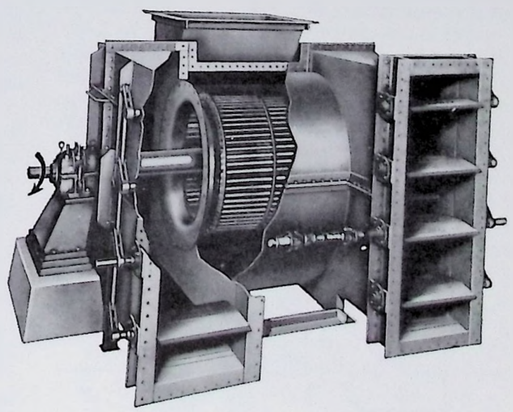


Fig. 10. *Multibladed fan with double inlet and louver-damper control, arranged for handling hot air or gases*

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The reader can find detailed information in standard references. Here the objective is better to acquaint the boiler user with those features of the fan art which are particularly pertinent to the generation of steam. Some of the types of fans in use are illustrated by Figs. 8, 9, and 10.

Two essentially different kinds of fans are distinguishable:

1. The centrifugal fan, in which the gas is accelerated radially outward in a rotor from the heel to the tip of the blades and delivered to a surrounding scroll casing (Fig. 11).

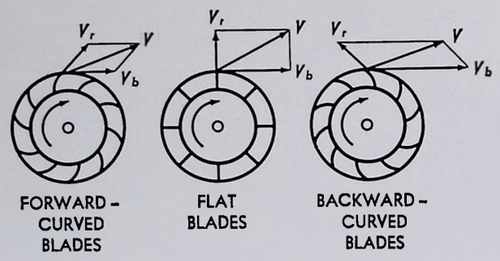


Fig. 11. *Three general types of centrifugal fans showing relative order of tip speed required to obtain same fluid velocity leaving blades*

2. The axial-flow fan, in which the fluid is accelerated parallel to the fan axis, as in the customary desk fan, but with a casing added for development of static pressure (Fig. 12).

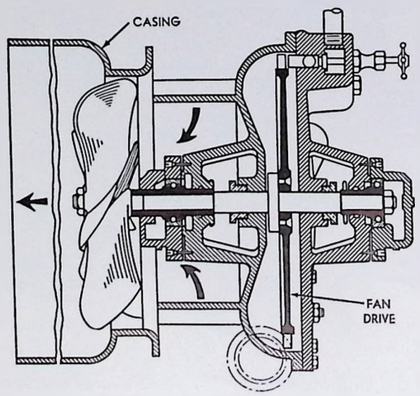


Fig. 12. *Simple type of axial-flow fan*

The performance of a fan is a complicated functional relationship which is best expressed in graphical form, as in Fig. 13. These graphs are called the characteristic curves. In the figure, the capacity in cfm is plotted horizontally as the independent variable, and the head (static pressure), the shaft horsepower, and the static efficiency are plotted vertically as the dependent variables. The rotative speed (rpm) is constant. The basic significance of the characteristic curves must be appreciated. They rigorously define the performance of the fan. If the head required for a given capacity is less than that indicated on the fan characteristic curve, additional system resistance, such as a damper to serve as a throttle, must be interposed. If this increase in resistance to flow is not interposed, the fan will supply additional air until the flow rate establishes a system resistance which complies with some point on the characteristic curve. There is no alternative; the fan must operate at a point on the characteristic where a balance is established. This concept is fundamental to an appreciation of the performance of fans and, in fact, all other fluid-acceleration machines.

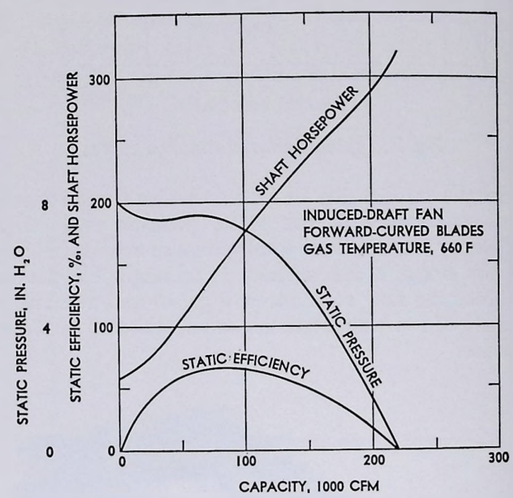


Fig. 13. *Characteristic curves for two types of centrifugal fans operating at 5500-ft elevation and 965 rpm*

For a given fan, the numerical performance values of the characteristics can be changed by varying the operating speed, rpm, to yield a family of curves, as shown in Fig. 14. The nature of the curves remains, however, substantially unaltered. The performance at the different speeds can be related, for the same point on the efficiency curve, by the rules:

1. Capacity is directly proportional to speed (rpm).
2. Head is directly proportional to speed squared $(rpm)^2$.
3. Power input is directly proportional to speed cubed $(rpm)^3$.

Fan performance is based on standard air at 70 F with a sp wt of 0.075 lb/cu ft, which corresponds to dry air at 29.92 in. Hg absolute pressure[6]. If a gas of some other sp wt is to be handled, the performance can be estimated by using the characteristic curves as plotted. For any given volume, the head, measured in in. of water, and the horsepower will each change directly with the sp wt.

System resistance may be plotted on the same chart as the characteristic curves of the fan, as shown in Fig. 14. If the fan runs at constant speed, then, for any output less than that shown at the intersection of the system resistance and rpm curves, the excess head must be throttled. If variable speed can be used, then the waste of energy by throttling is avoided.

Variations in the designs of fans are indicated by the diversity of characteristics in those which are available in the market. By plotting characteristics on a percentage basis, some of these variations are shown in Fig. 24, Chapter 10. The rating point is taken for the conditions prevailing at the point of maximum efficiency. Some fans give steep head characteristics, others give flat head characteristics. Some horsepower characteristics are concave upward; others are concave downward. The latter have the advantage of being self-limiting, so that there is little danger of burning out the driver or little need for the alternative of overmotoring. The power requirement at lighter loads, however, is greater than that for the horsepower characteristic with concavity upward.

The steep head characteristic has merit in that

[6] Bulletin No. 110, 1952, Air Moving & Conditioning Association, Inc.

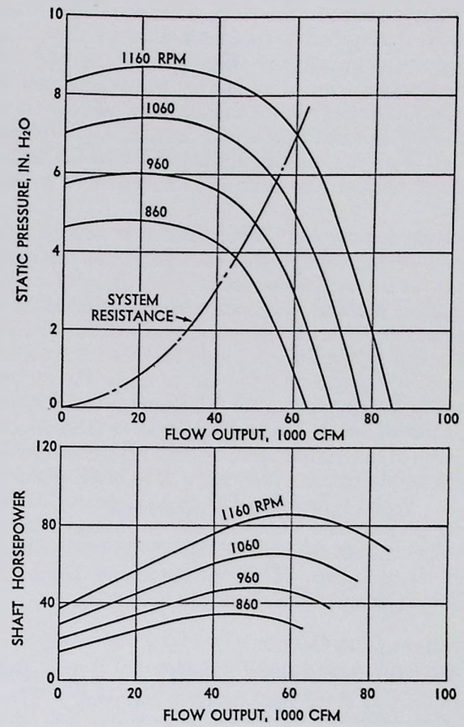


Fig. 14. *Graph to show how desired output and static pressure can be obtained economically by varying fan speed to avoid large throttling losses*

there is less possibility of error in the selection and a better opportunity to assure equalization of load in parallel operation. Those fans, on the other hand, which offer the flat head characteristic have the merit of handling larger volumes for a given wheel diameter and head and are usually identified as volume fans. The fans with steep head characteristics are the high-speed designs calling for the maximum speeds in the generation of a specified head. In the selection of centrifugal fans for draft service, it is customary to find the backward-curved-blade designs used for forced draft and the radial-tip or forward-curved-blade designs preferred for induced draft. The steel-plate types are sometimes used for induced draft when there are heavy dust loadings in the gas. Likewise, exhausters for pulverizing mills are generally of the steel-plate type. Axial-flow and vane-axial designs are found in forced-draft service where maximum speed and minimum bulk are prime objectives, as is the case with marine applications.

Fan Testing. It is difficult to obtain consistent results from field tests of fans installed in duct systems. For

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air measurement in the field, there is seldom available in the system a straight section of duct long enough to eliminate the effect of bends or dampers before entering or after leaving the section. Structural arrangements at the fan entrance and discharge will also materially affect the results of field tests of fans. Fans are, therefore, guaranteed by the manufacturer only on the basis of factory test on a "test block."

In the factory test setup, there is a long straight section of duct at the fan discharge, with a throttling device at the end of the duct to vary the output. The long duct permits stabilization of velocity and static pressure profiles, whereas in most commercial installations a bend or a change in duct cross section is usually too close to the fan to allow the uneven velocity distribution at its discharge to stabilize before the air enters the modified section. The characteristics of fan performance established by tests under ideal conditions are known as "test block performance." When ordering a fan on the basis of its test block performance, adequate allowance must be made for the effect of field conditions on fan performance, and the guidance of the manufacturer should be followed because deviations can be severe.

Control of Fan Output

Very few applications permit the fan to operate continuously at the same pressure and volume. Therefore, to meet the requirements of the system, some convenient means of varying the fan output is necessary. The common methods of controlling fan output are:

Damper Control. A damper is a simple control. It acts as a throttle valve, introducing enough resistance into the system to restrict the fan output to any desired quantity. Damper control, however, is wasteful because the fan develops excess pressure energy which must be dissipated by throttling.

The advantages of damper control are:

1. Lowest first cost of all types of control.
2. Ease with which it can be operated or adapted to automatic control.
3. The least expensive type of fan drive, a constant-speed induction-type a-c motor, may be used.
4. It is a continuous rather than a step type of control and is therefore effective throughout the entire range of fan operation.

Variable-Speed Control. The most efficient method of controlling output, as far as power consumption of the fan is concerned, is by varying the speed. From the relationship of speed to capacity, pressure, and power (see page 5-11), it follows that if the speed is cut to ½, the output of the fan will be cut to ½, the pressure to ¼, and the horsepower to ⅛. In using speed control there is some loss in efficiency, since no variable-speed driver is as efficient throughout the entire fan-load range as a direct-connected constant-speed a-c motor. The amount of the loss in effectiveness depends upon the type of speed variation selected.

A number of commonly used variable-speed arrangements are: (a) magnetic coupling; (b) hydraulic coupling; (c) special mechanical drives; (d) variable-speed d-c motor; (e) variable-speed a-c motor; and (f) variable-speed steam turbine. The magnetic coupling consists of two windings in a housing with a variable field. A change in field strength varies the slip and consequently the speed of the fan. An hydraulic coupling is illustrated in Fig. 15. Among the special mechanical drives are the variable-pitch V-belt and the variable-speed planetary transmissions. In general, all of these methods of speed variation call for a higher investment than the less efficient damper control. An economic balance of first cost and power consumption must, therefore, be effected in selecting the type of drive.

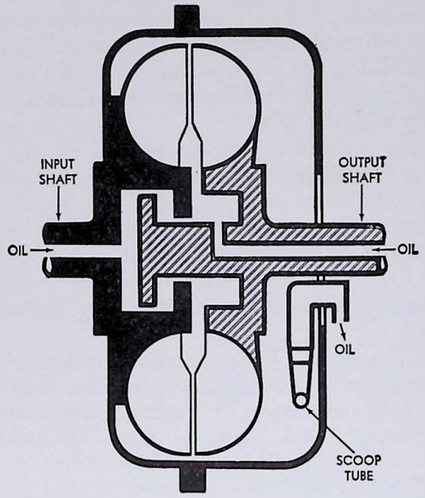


Fig. 15. *Hydraulic coupling (schematic) enables wide variation in fan speed with constant-speed driver*

Two-Speed A-C Motors. Two-speed a-c motors are often used to supplement damper control. A two-speed motor is less expensive than the variable-speed arrangements with a-c drives and will give an improvement in the fan efficiency with simple damper control. The spread between upper and lower speeds should not be too great. Thus, if the top speed is 1200 rpm, the second speed should not be less than 900; if the top speed is 900, the lower speed may be 720 rpm.

Inlet-Vane Control. The inlet-vane-control method in regulating the air flow through a fan requires less horsepower at fractional outputs than outlet-damper control. The inlet vanes (see Fig. 16) give the air a varying degree of spin in the direction of fan rotation. This enables the required head to be produced at a proportionately lower power input. Inlet-vane control is more expensive than damper control but less expensive than speed variation. It is most effective for a moderate degree of change at high fan output but offers considerable saving in efficiency over damper control throughout most of its range.

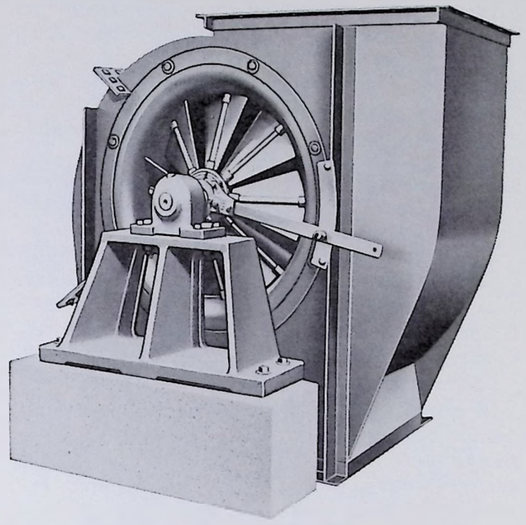


Fig. 16. *Inlet vanes control the output of this forced-draft fan handling room air*

Vane leakage often makes it difficult to reduce the air flow below ¼ of full fan output when a single-speed motor is used for the fan drive. Therefore, when vane control is used for mechanical-draft fans and where a wide load range is required, it is advisable to use a two-speed-drive motor or a supplementary damper to increase the control range of the vanes.

The effect upon the power input required by the fan drives for various types of fan control is indicated by the curves in Fig. 17.

Fan Drives

Electric motors are normally used for fan drives because, when properly selected, they are less expensive and more efficient than any other type of drive. For fans of more than a few horsepower, squirrel-cage induction motors are the usual choice. This type of motor is relatively inexpensive, reliable,

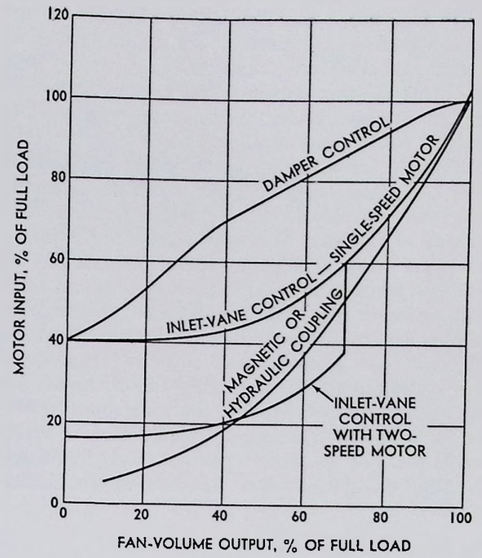


Fig. 17. *Fan-drive power required for several types of volume control*

and high in efficiency over a wide load range. In large sizes, it is frequently used in conjunction with a magnetic or hydraulic coupling where variable speed is required. For some variable-speed installations, particularly in the smaller sizes, wound-rotor (slip-ring) induction motors are used. If a d-c motor is required, the compound type is usually selected. For fan drives, the steam turbine is more expensive than a squirrel-cage motor, but it is generally less expensive than any of the variable-speed electric-motor arrangements in sizes over 50 hp. The steam turbine is used where an uninterrupted supply of electricity is not assured. This type of drive may prove more economical than the electric drive in plants where the exhaust may be used for process work or feed heating.

Fan Safety Factors

To insure that a boiler unit will not be limited in performance by the fans, it is necessary to add safety factors to the calculated or net fan requirements to arrive at satisfactory test block or fan design specifications. These factors are intended to cover conditions encountered in operation that cannot be evaluated. The usual factors are in the order of 15 to 20% increase in net weight of air or gas, 15 to 25% increase in net head, and 25 F increase in temperature of air or gas to the fans. These values have been developed from experience. Because of the space required, it is not economically justifiable to arrange duct work so that a fan will operate under ideal conditions. For

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boilers that use a fuel containing ash, there is the possibility of a variation in ash characteristics or unusual operating conditions that might cause fouling of the heating surfaces and require additional draft. Stoker-fired boilers may have to burn coal improperly sized, thus requiring more pressure than normally to force the air through the fuel bed. In order to increase rapidly from partial to maximum load, it is necessary to overfire, which requires overcapacity of the fans. When an emergency arises, as may occur when a boiler is suddenly lost, it may be desirable to operate other boilers at overload for a short period of time; this requires more volume from the fans and higher pressure than for the calculated net conditions. These as well as other minor variations make the addition of safety factors to net fan requirements imperative for the satisfactory operation of a boiler. Unnecessary pyramiding of safety factors must be avoided to prevent costly initial and operating charges and less desirable control characteristics.

The Forced-Draft Fan

Fan Capacity. In a boiler installation with both forced and induced draft, the forced-draft fan pushes the air through the combustion-air-supply system to the furnace. It must have a discharge pressure high enough to equal the total resistance of the air ducts, air heater, burners or fuel bed, and any other resistance between the fan discharge and the furnace. This makes the furnace the point of balanced draft, since, when the air reaches it, the pressure has dropped to zero. The volume output of the fan must equal the total quantity of air required for combustion. In many boiler installations, greater reliability is obtained by division of fan capacity between two fans operating in parallel. If one fan is out of service, the remaining fan will generally be able inherently to carry approximately two-thirds load on the boiler.

To establish the characteristics of the forced-draft fan, the system resistance from fan to furnace is calculated for the actual weight of air required by the combustion equipment plus the expected leakage from the air to the gas side of the air heater. For calculation purposes in boiler work, the air temperature entering the fan is usually taken as 80 F. Having thus established the net requirements, the test block specifications are arrived at by adding factors of safety to the weight, total head, and temperature.

Fan Requirements. A forced-draft fan for boiler service must operate under far more stringent conditions than an ordinary ventilating fan. The important requirements for a forced-draft fan are:

1. *Reliability.* Modern boilers must operate continuously for long periods (up to 18 months in some instances) without shutdown for repairs or maintenance. To satisfy this condition, the fan must have a rugged rotor and housing and conservatively loaded bearings. The fan must also be well balanced, and the blades should be so shaped that they will not collect dirt to disturb this balance.
2. *Efficiency.* Since many boilers are operated under varying load conditions, it is desirable for the fan to have a high efficiency over a wide range of output.
3. *Pressure.* Fan pressure should vary uniformly with output, giving a smooth curve over the range of capacity. This permits easy application of damper control and assures minimum disturbance of air flow when the system resistance changes because of minor adjustments to the fuel-burning equipment.
4. *Capacity.* When two or more fans operate in parallel, the pressure-output curves should drop off at a constant rate from near the shutoff condition, so that each fan shares the load equally.
5. *Horsepower.* If the fan is motor driven, it is desirable to have a self-limiting horsepower characteristic, so that the driving motor cannot be overloaded. This means that the horsepower should reach a peak and then drop off near the full-load fan output (see Fig. 14).
6. *Space.* To reduce space requirements, since relatively clean air is handled, a high-speed type of forced-draft fan may be used.

In general, the requirements for a forced-draft fan are best satisfied by the backward-curved centrifugal designs. For highest-pressure service, the centrifugal compressor (Fig. 18) is favored. For small boiler units, however, a single propeller-type fan, mounted directly upon the burner windbox, is frequently used because of its low cost and economy of space. Vane-axial fans are used for some special high-pressure forced-draft applications.

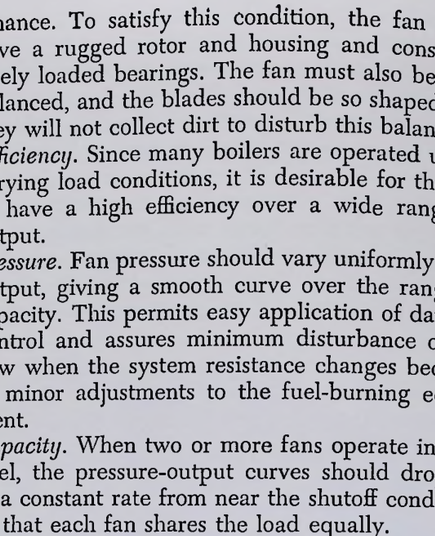


Fig. 18. *Cross-sectional view of centrifugal compressor*

The Induced-Draft Fan

Fan Capacity. The induced-draft fan must create a sufficient pressure differential to cause the required quantity of gas at full load to pass through the boiler against the draft loss resulting from flow over all the heating surface and through the gas passages between the furnace and the fan. Normally the gases pass into a stack or discharge from the fan at substantially atmospheric pressure.

The weight of gas used to calculate the draft loss is equal to the weight of the combustion gases plus the weight of air infiltration to the boiler setting from the room and from the air side to the gas side of the air heater. The gas temperature used in arriving at the fan size is the temperature calculated for estimating boiler performance. To establish the test block specifications for the fan required, safety factors are added to the calculated quantities.

Fan Requirements. An induced-draft fan has the same requirements as a forced-draft fan except that it must handle higher temperature gas which may contain erosive ash. Flat, forward-curved, and occasionally backward-curved blades are variously used. The curvature of backward-curved blades is usually less than that in forced-draft fans. The lesser curvature blade, by the reduction in rotative speed, tends to diminish the erosive effect of the ash particles, and, because of the shape, the deposit of dirt upon the back of the blades is reduced. In handling particularly dirty or erosive gases, a flat-bladed centrifugal fan of lower speed may be used. To avoid excessive maintenance from erosion, the casing and blades may be protected by replaceable wear strips. Bearings are usually water cooled and have radiation shields on the shaft between the fan and bearing to avoid overheating.

Fan Selection

The following example shows the method of establishing the characteristics of fans suitable for a boiler, based on data obtained from the calculated performance of the oil-fired unit of Fig. 19.

Data, Assigned or from Calculated Performance

(Boiler Unit illustrated in Fig. 19)

Steam flow, lb/hr	325,000
Output at station bus, kw	40,000
Input to furnace, Btu/hr	416,000,000
Air flow through unit, lb/hr	
Theoretical air for combustion	310,340
Excess air required at burner, 5%	15,520
Leakage, furnace and boiler, 10%	31,030
Leakage, air to gas in air heater, 2%	6,200
Fuel consumption, lb/hr	22,500
Temperatures, F	
Air entering air heater	80
Gas leaving air heater	335
Atmospheric pressure, in. of mercury	29.9
Air resistance, in. of water (includes stack effect)	
Air heater	2.84
Ducts	0.86
Burners and windbox	3.15
Total air resistance	6.85
Draft loss, in. of water (includes stack effect)	
Furnace	0.10
Boiler and superheater	5.10
Air heater	2.08
Flues	0.40
Total draft loss	7.68

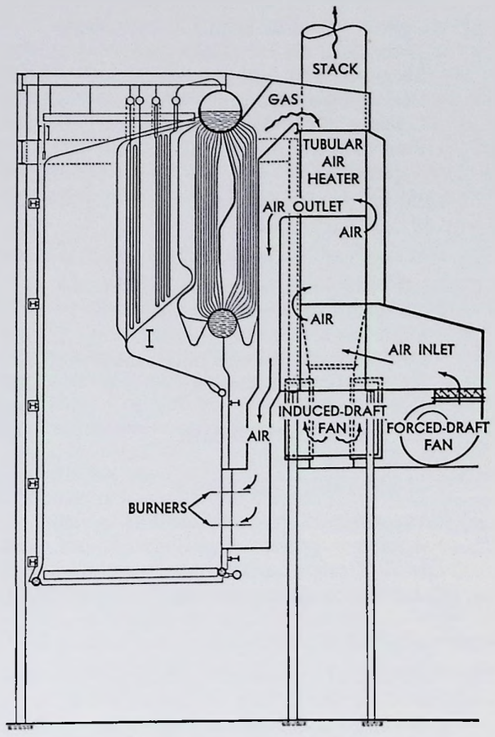


Fig. 19. *Diagrammatic arrangement of forced- and induced-draft fans and stack for 325,000 lb steam per hr boiler unit*

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Selection of Forced-Draft Fan

a) Weight of air to be handled by fan, lb/hr

Theoretical air for combustion	310,340
Excess air required at burner	15,520
Leakage, air to gas in air heater	6,200
Total net wt requirement	332,060
Add 20% safety factor	66,410
Test block wt requirement	398,470

b) Static pressure at fan discharge, in. of water

Total air resistance, fan to furnace	
Air resistance in air heater	2.84
Air resistance in ducts	0.86
Air resistance in burners and windbox	3.15
Total net pressure requirement	6.85
Add 25% safety factor	1.71
Test block pressure requirement	8.56

c) Temperature of air to be handled by fan, F

Net temp requirement, air at fan	80
Add 25 F safety factor	25
Test block temp requirement	105

Selection of Induced-Draft Fan

a) Weight of gas to be handled by fan, lb/hr

Theoretical air for combustion	310,340
Excess air required at burner	15,520
Leakage, furnace and boiler	31,030
Leakage, air to gas in air heater	6,200
Fuel (assume no ash) all in flue gases	22,500
Total net wt requirement	385,590
Add 18% safety factor	69,410
Test block wt requirement	455,000

b) Draft to be provided by fan, in. of water

Total draft loss, furnace to fan	
Draft loss in furnace	0.10
Draft loss in boiler and superheater	5.10
Draft loss in air heater	2.08
Draft loss in flues	0.40
Total net draft requirement	7.68
Add 25% safety factor	1.92
Test block draft requirement	9.60

c) Temperature of gas to be handled by fan, F

Net temp requirement, gas lv air heater	335
Add 25 F safety factor	25
Test block temp requirement	360

Pressure Furnaces

In a number of large steam-generating units and in almost all marine boilers, the induced-draft fan is omitted and the entire boiler setting is under positive pressure. To prevent the escape of noxious gases into the boiler room, it is essential that the leakage, normally expected in a conventional setting, be eliminated. In a pressure setting, all the work of creating gas flow through the unit is done by the forced-draft fan, and operating cost is reduced in two ways:

1. Since there is no air infiltration into the setting to increase the stack loss, the boiler efficiency may be improved by 0.25 to 0.75%. A mean value of 0.50% may be used in most calculations.

2. Since there is no air leakage into the boiler setting, the forced-draft fan handles a lesser weight of cool air at smaller volume and higher efficiency than would a fan handling hot gases.

The following example illustrates the total saving that can be effected when operating the boiler unit, shown by Fig. 19, with forced draft only (pressure furnace) instead of with a combination of forced and induced draft (suction furnace). An over-all efficiency of 70% is used for fan and motor.

1) Suction-Furnace Operation

a) Forced-draft fan

Air weight, lb/hr	332,060
Air temperature, F	80
Static pressure, H_{SP}, in. of water	6.85
Air sp wt at 80 F, lb/cu ft	0.0737
Air volume, cfm $= \frac{332{,}060}{0.0737 \times 60}$	75,093
Air hp, Eq (17) $= 0.0001573 \times 75{,}093 \times 6.85$	80.91
Motor input (combined efficiency of 70%), hp $= 80.91/0.70$	115.6
Motor input, kw $= 115.6 \times 0.746$	86

b) Induced-draft fan

Gas weight, lb/hr	385,590
Gas temperature, F	335
Static pressure, H_{SP}, in. of water	7.68
Gas sp wt at 335 F, lb/cu ft	0.0490
Gas volume, cfm $= \frac{385{,}590}{0.0490 \times 60}$	131,143
Air hp, Eq (17) $= 0.0001573 \times 131{,}143 \times 7.68$	154.8
Motor input (combined efficiency of 70%), hp $= 154.8/0.70$	221.1
Motor input, kw $= 221.1 \times 0.746$	165

c) Total fan power required, forced + induced

Total motor input, hp $= 115.6 + 221.1$	336.7
Total motor input, kw $= 86 + 165$	251

2) Pressure-Furnace Operation

a) Forced-draft fan only

Air weight, lb/hr	332,060
Air temperature, F	80
Static pressure, H_{SP}, in. of water (see below)	13.37
Air sp wt at 80 F, lb/cu ft	0.0737
Air volume, cfm $= \frac{332{,}060}{0.0737 \times 60}$	75,093

Air hp, Eq (17) $= 0.0001573 \times 75{,}093 \times 13.37$	157.2
Motor input (combined efficiency of 70%), hp $= 157.2/0.70$	225
Motor input, kw $= 225 \times 0.746$	168

3) Saving of Pressure-Furnace over Suction-Furnace Operation

Fan-power saving, kw $= 251 - 168$	83
Fan-power saving, % of output (more salable power) $= 83 \times 100/40{,}000$	0.21
Improved boiler efficiency, % (0.5% less fuel burned, see 1. under *Pressure Furnaces* above)	0.50
Approximate total saving, %	0.71

In the pressure-furnace arrangement with no leakage from the room into the setting, the calculations (see Chapter 8) to determine the gas pressure drop through the unit are based on the following gas weights (lb/hr), differing somewhat from those used in determining the draft loss for the conventional arrangement:

Theoretical air for combustion	310,340
Fuel (100% into flue gases)	22,500
Excess air at burners	15,520
Gas through boiler and superheater, lb/hr	348,360
Gas through boiler and superheater	348,360
Leakage, air to gas in air heater	6,200
Gas through air heater, lb/hr	354,560

The calculated pressure drop (in. of water) through the unit with a pressure setting, used above as the required static pressure at the fan discharge, may be recapitulated as follows:

Air resistance, fan to furnace	6.85
Furnace (assumed)	0.10
Boiler and superheater	4.32
Air heater	1.76
Flues	0.34
Total net head on fan, in. of water	13.37

It will be noted that the decrease in operating cost of the unit with a pressure setting is substantial. There is an added saving from the elimination of the maintenance of the induced-draft fan, which must handle hot and frequently erosive and corrosive gases. The increased cost of a completely airtight boiler setting must be weighed against these savings.

Operation and Maintenance

Stacks. All idle entrances leading to the stack should be tightly sealed to prevent cold-air leakage to its base. Cold-air leakage not only lowers the stack temperature and available draft but it also increases the quantity of gas that the stack must handle. Some plants find it expedient to dispose of factory wastes into the boiler stack for dispersion high into the atmosphere. Careful consideration should be given to such use of the stack since it may increase maintenance cost.

The stack is subject to the erosive action of ash in the flue gases, to acid corrosion by sulfur products, and to deterioration incident to continuous exposure to the weather. The erosion or wastage of the stack material normally occurs at stack entrance or at throats and necked-down sections, where there is a change in the direction or the velocity of the gas. These sections may require replacements at long-period intervals. The use of premium-priced abrasive-resistant materials is justified by the reduction in stack maintenance.

Low exit gas temperature from a boiler unit is desirable from an efficiency standpoint. Too low a temperature, however, may result in the condensation of sulfuric acid, which will cause corrosion of steel or steel-lined stacks. The limit to which flue gases can be cooled before condensation occurs depends upon the sulfur trioxide and moisture content of the flue gas.

Acid corrosion has destroyed steel stacks in as short a period as a few months. Corrosion-resistant metals, refractory coatings, and high-temperature acid-resistant paints are used to prolong the life of steel stacks. Refractory stacks are subject to deterioration from the chemicals and water vapor in the gases. Long exposure in wet weather (during out-of-service periods) and insufficient drying on starting-up contribute to increased maintenance cost.

Fans. The service of fans as components of a boiler unit is a fairly simple and light duty, but their operation, like that of all running equipment, requires frequent inspection to detect and correct irregularities that might cause trouble. The period of continuous operation of fans should be long compared with that of other power plant equipment. To assure this continuity of operation, proper lubrication and cooling of fan shafts, couplings, and bearings are essential.

Fan bearings and couplings must be carefully aligned with the driving equipment. Proper allowance should be made for the small vertical movement of the fan rotor caused by temperature rise of the bearing supports from cold to running conditions. Additional end clearances must be provided for oilslingers at the floating bearing and at the coupling to compensate for the increase in length of the shaft from heat conduction. The bumping action of a poorly aligned bearing becomes progressively worse with each revolution and rapidly destroys the bearing.

To assure smooth and lasting service, a fan should be properly balanced both statically and dynamically. The weight of the rotor should be equally distributed about the center of rotation, and the load should be

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equally distributed around the fan rotor and uniformly outward from the shaft. The balance should be checked, after each shutdown for maintenance, by running the fan at full speed 1) with no air flow and 2) with full air flow. A skillful mechanic can usually balance the fan rotor by trial-and-error methods. Special instruments may be used, especially with larger fans, to establish the amount and location of weight to be added for static balance. Fans can be thrown out of balance by the uneven shedding of deposits from the blades. This condition is usually self-correcting, but, to guard against serious mechanical damage to the equipment, operation while out of balance should be under careful observation.

In fans handling gases with entrained abrasive dust particles, the blades and the housing near the discharge are subject to erosion. Abrasive-resistant materials and liners are used to reduce such wear. Some operators apply beads of welding to repair and to lengthen the life of fan blades of induced-draft fans subject to erosion.

6

TEMPERATURE MEASUREMENT

In the measurement of temperatures in the boiler art, good instruments will give good readings only when properly applied. This chapter recites features of good instruments, but, more importantly, it stresses the techniques for the correct application of these instruments.

Early in life everyone learns to distinguish between hot and cold. This sensory experience, supplemented by familiarity with such phenomena as the freezing and boiling of water, the cooking of food, and the flow of hot metals, gives rise to a general concept of temperature but does not provide any quantitative measure of degree of hotness—in other words, a scale of temperature. One of the great contributions of thermodynamics was the establishment of a usable temperature scale (thermodynamic scale), completely independent of any actual substance, which forms the basis of scales of temperature used in the quantitative measurement of degree of hotness.

Temperature Scales

Before the theory of temperature had been fully developed, various arbitrary temperature scales were devised. Two of these, Fahrenheit and Celsius (centigrade), are firmly established. The Fahrenheit scale is fixed by the freezing point (32 F) and the boiling point (212 F) of water at standard atmospheric pressure, with 180 equal degrees between the freezing and boiling points. The centigrade scale is based on the same freezing point (zero C) and boiling point (100 C) of water at standard atmospheric pressure, with 100 equal degrees between. In the conversion from one to the other, $F = (C \times 9/5) + 32$, and $C = (F - 32) \times 5/9$.

Whenever energy is added to a substance, the internal energy is increased, causing temperature rise, melting, or vaporization. Similarly, when energy is removed, the decrease in internal energy causes lowering of temperature, freezing, or condensation. Classical thermodynamics demonstrates that a temperature of zero exists on the thermodynamic scale; this point is called absolute zero and corresponds to —273.2 C or —459.7 F.

Following the principle of the thermodynamic scale, which is based on the Carnot cycle and is independent of the substance used, equal temperature intervals would also correspond to equal changes in internal energy for the perfect gas. Actual gases are assumed to follow the behavior of a theoretically perfect gas, subject to specific adjustments.

Two additional scales useful in scientific work are the absolute temperature scales starting at absolute zero. The absolute scale using centigrade degrees as the temperature interval is called the Kelvin scale (K); the absolute scale using Fahrenheit degrees is called the Rankine scale (R). For engineering work using round figures for absolute zero, the values of K and R and the conversion from one to the other are: $K = C + 273 = R/1.8$, and $R = F + 460 = 1.8\ K$.

For the purpose of calibrating thermometers, the fusion or vaporization temperatures of various pure substances have been carefully determined and are known as "fixed points." Six fixed points form the basis of the International Temperature Scale. These, all at atmospheric pressure of 29.92 in. Hg, are:

Oxygen boiling	— 182.97 C	— 297.35 F
Water freezing	0.00 C°	+ 32.00 F°
Water boiling	+ 100.00 C°	+ 212.00 F°
Sulfur boiling	+ 444.60 C	+ 832.28 F
Silver melting	+ 960.8 C	+ 1761.4 F
Gold melting	+ 1063 C	+ 1945 F

° Exact, by definition.

Importance of Temperature Determination

Heat exchange, heat balances, and a multitude of other problems involving temperature are important in all phases of power generation. Means of indicating, recording, and controlling temperatures are important in design, fabrication, operation, and testing

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power-generating and -utilizing equipment and associated apparatus. Methods of measuring temperatures accurately are therefore of prime interest to power engineers. Great strides have been made in the past in developing suitable instruments, and progress is continuing. Both test and commercial operating instruments are constantly being improved and simplified. In recent years thermocouples have found widespread acceptance and are extensively used in the power-generating field.

Methods of Measuring Temperature

Many properties of substances affected by heat, such as change of state, thermal expansion, radiation, and electrical effects, have been applied to temperature indicators suitable for commercial use. Instruments of this kind, usually of a high order of accuracy, are available for a great variety of applications. Accuracy in temperature measurement will depend upon care in application and an appreciation of the limitations of the instrument used. Before readings are taken, it is the engineer's responsibility to make certain that every application is correct and that the results are not affected by any extraneous factors.

Changes of State

Fusion. For a pure chemical element or compound, such as mercury or water, fusion, or change of state from solid to liquid, occurs at a fixed temperature. The melting points of such materials are therefore suitable fixed points for temperature scales.

The fusion of pyrometric cones is widely used in the ceramic industry as a method of measuring high temperatures in refractory heating furnaces. These cones, small pyramids about 2 in. high, are made of selected mixtures of oxides and glass which soften and melt at established temperatures. In the power industry, use of the pyrometric cone is generally limited to the laboratory and is suitable for a temperature range from 1100 to 3600 F.

Fusion pyrometers are also made in the form of crayon, paint, and pellets, which indicate a range of established temperatures up to 2000 F. When the crayon or paint is applied to a cold surface, a dull-finish mark is made, which melts and changes to a glossy finish as the surface is heated to the specified temperature. These marks, therefore, indicate whether or not the surface temperature has reached or exceeded a selected value. The pellets begin to melt at specified temperatures when in contact with a surface.

Vaporization. The vapor pressure of a liquid depends upon its temperature. When the liquid is heated to the boiling temperature, the vapor pressure is equal to the total pressure above the liquid surface. Therefore, the boiling points of various pure chemical elements or compounds at standard atmospheric pressure (29.92 in. Hg) can be used as thermometric fixed points.

The change of vapor pressure with temperature is utilized in the vapor-pressure thermometer, illustrated schematically in Fig. 1, which consists of a bulb partially filled with liquid and a capillary tube leading from the bulb to a pressure gage calibrated to read temperature directly. If the space between the liquid and the pressure gage is filled only with vapor from the liquid, the pressure will vary directly with the temperature of the liquid in the bulb. The capillary tube may be of considerable length, and its temperature does not affect the reading, but the accuracy of the instrument is affected by variations in atmospheric pressure and by elevation of the bulb above or below the gage. The temperature scale is nonuniform. To prevent partial and indeterminate condensation of the vapor in the capillary, the bulb temperature must be somewhat above or below that of the capillary. The working range of a given instrument is limited to several hundred degrees and usually lies between minus 20 and plus 700 F.

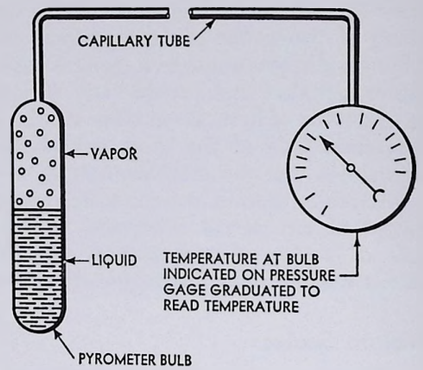


Fig. 1. *Schematic assembly of vapor-pressure pyrometer*

Expansion Properties

Most substances expand when heated, and in many cases the amount of expansion is almost directly proportional to the change in temperature. This effect is utilized in various types of thermometers using gases, solids, or liquids.

Gases. The expansion of gases follows the relationship:

$$pv = RT/M$$

where:

p = absolute pressure, lb/sq ft
v = specific volume, cu ft/lb
R = universal gas constant, 1544
T = absolute temperature, R = F + 460
M = molecular weight of the gas

At high pressures or near the condensation point, gases deviate considerably from this relationship. Under other conditions, however, the deviation is small.

Two types of gas thermometers are based on this relationship. In one a constant gas volume is maintained, and changes in pressure are used to measure changes in temperature. In the other a constant pressure is maintained, and changes in volume are used to measure changes in temperature. Instruments of this type, of highest accuracy, have been developed for laboratory work. The constant-volume type is widely used commercially.

Nitrogen is commonly used for the gas-filled thermometer in industrial application. It is suitable for a temperature range of minus 200 to plus 1000 F. The construction is similar to the vapor-pressure pyrometer shown by Fig. 1, with nitrogen gas replacing the liquid and vapor. Expansion of the heated nitrogen in the bulb increases the pressure in the system and actuates the temperature indicator or recorder. The temperature scale is uniform, and the capillary tube may be of any length. Changes in temperature of the capillary tubing will introduce small errors.

Liquids. The expansion of liquids is used in a thermometer similar in design to the vapor-pressure instrument, shown by Fig. 1, except that the bulb and capillary tube are completely filled with liquid. Mercury is suitable for a temperature range of minus 38 to plus 1000 F, and alcohol may be used for a range of minus 50 to plus 300 F. The readings of instruments of this type are subject to error if the capillary tubing is affected by temperature changes.

Liquid-in-glass thermometers, also based on the expansion of liquids, are so widely used that they are discussed in a separate section (see page 6-5).

Solids. The actuating medium in thermometers based on the expansion of solids when heated is the bimetallic strip. Flat ribbons of two different metals with unlike coefficients of thermal expansion are joined face to face by riveting or welding to form a bimetallic strip. When the strip is heated, the expansion is greater for one side of the double layer than for the other, and the strip bends, if originally flat, or changes its curvature if initially in the spiral form frequently used. Bimetallic strips are seldom used in power plant thermometers but are widely used in inexpensive household thermometers, many designs of thermostats, and a variety of temperature-control and -regulating equipment. They are particularly useful for automatic temperature compensation in the mechanisms of other instruments.

Radiation Properties

All solid bodies emit radiation. The amount is very small at low temperatures and large at high temperatures. The values for radiation may be calculated by the following formula, developed by Stefan-Boltzmann:

$$W = \sigma \epsilon T^4$$

where:

W = radiant energy per unit area per unit time, Btu/sq ft, hr
σ = Stefan-Boltzmann constant, 1.71×10^{-9} Btu/sq ft, hr, T^4
ϵ = Emissivity of the surface, a dimensionless number between 0 and 1 (usually about 0.90 for steel)
T = absolute temperature, R = F + 460

At low temperatures the radiation is chiefly in the infrared range, invisible to the human eye. As the temperature rises, an increasing proportion of the radiation is in shorter wavelengths, becoming visible as the dull-red glow of bodies at about 1000 F and passing through yellow toward white at higher temperatures. The temperature of hot metals (above 1000 F) can be estimated by their color. For iron or steel, the color scale is roughly as follows:

Dark red	1000 F
Medium cherry red	1250 F
Orange	1650 F
Yellow	1850 F
White	2200 F

Two types of temperature-measuring instruments, the optical pyrometer and the radiation pyrometer, are based upon the radiating properties of materials.

Optical Pyrometers. By sighting an optical pyrometer on a hot object, brightness of the latter can be compared visually with the brightness of a calibrated source of radiation within the instrument, usually an electrically heated tungsten filament. A red filter may be used to restrict the comparison to a particular wavelength. This instrument is designed for measuring the temperature of surfaces with an emissivity of 1.0, known in heat transfer parlance as "blackbodies."° When accurately calibrated, it will give excellent results above 1500 F, provided its use is restricted to the application for which it is designed. Measurement of the temperature of the interior of a uniformly heated enclosure, such as a muffle furnace, is such an application. When used to measure the temperature of a hot object in the open, the optical pyrometer will always read low, the error being small (20 F) for high-emissivity bodies such as steel ingots and considerable (200 to 300 F) for unoxidized liquid steel or iron surfaces.

The optical pyrometer has a wide field of application for temperature measurements in heating fur-

°By definition, a "blackbody" absorbs all radiation incident upon it, reflecting and transmitting none (see Chapter 7, page 7-6).

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naces and around steel mills and iron foundries. It is of no value for the commercial measurement of gas temperature, since clean gases do not radiate in the visible range.

Radiation Pyrometers. In the radiation pyrometer, all radiation from the hot body, regardless of wavelength, is absorbed by the instrument. The heat absorption is measured by the temperature rise of a delicate thermocouple within the instrument, calibrated to indicate the temperature of the hot surface at which the pyrometer is sighted, assuming that the surface emissivity equals 1.0. The hot surface must fill the entire field of view of the instrument.

The radiation pyrometer has been developed into a laboratory research instrument of extreme sensitivity and high precision over a wide range of temperature. The usual industrial-type instrument gives good results above 1000 F when used to measure temperatures of high-emissivity bodies, such as the interiors of uniformly heated enclosures. Since operation is independent of human judgment, radiation pyrometers may be used as remotely operated indicators or recorders, or in automatic-control systems. Errors in measuring the temperatures of hot bodies with emissivities of less than 1.0, especially if they are in the open, are extremely large. The radiation pyrometer is not suitable for the commercial measurement of gas temperatures.

Electrical Properties

Two classes of widely used temperature-measurement instruments, the thermocouple and the electrical-resistance thermometer, are based in principle on the relation of temperature to the electrical properties of metals.

Because of its versatility, convenience, and ruggedness, the thermocouple is of particular importance in power plant temperature measurements. A discussion of the thermocouple and its application will be found in separate sections (see page 6-6 ff).

Electrical Resistance. The electrical-resistance thermometer, used over a range of temperature from minus 400 to plus 1800 F, depends upon the increase in electrical resistance of metals with increase in temperature, which is almost in direct proportion. Therefore, if the electrical resistance of a wire of known and calibrated material is measured (by a Wheatstone bridge or other device), the temperature of the wire can be determined.

In the simplest form of this instrument, shown by sketch (a), Fig. 2, the readings would be the sum of the resistances of the calibrated wire and the leads connecting this wire to the Wheatstone bridge. These readings would be subject to error from temperature changes of the leads. By using a slightly more complicated circuit, shown by sketch (b), Fig. 2, the resistance of the leads can be eliminated from the instrument reading. In order to localize the point of temperature measurement, the resistance wire may be made in the form of a small coil. From room temperature to 250 F, commercial instruments usually have resistance coils of nickel or copper. Platinum is used for higher temperatures and in many high-precision laboratory instruments over a wide range of temperatures.

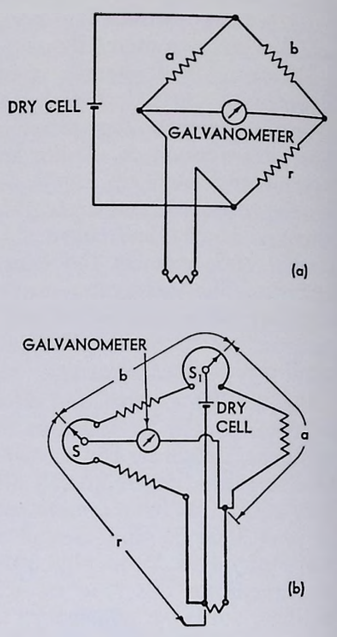


Fig. 2. *Diagrams of electrical circuits for resistance thermometers, from Leeds & Northrup*

The electrical-resistance thermometer does not require human judgment and can therefore be used for the remote operation of indicating, recording, or automatic-control instruments. If proper precautions are taken in its use, it is stable and accurate. However, it is less rugged and less versatile than a thermocouple.

Other Properties

There are other properties of materials which vary with temperature and which could be used as the bases for temperature-measuring instruments. For example, the velocity of sound in a gas varies directly as the square root of the absolute temperature. While this phenomenon has been used in research, it has not been commercially applied because of difficulties in instrumentation. By using the flow of gas through two orifices in series, a method has been developed for measuring gas temperatures up to 2000 F commercially and up to 4500 F in the laboratory. Temperatures in the range from absolute zero to 5000 F have been determined with considerable accuracy by the use of an electronic laboratory instrument that measures the motion of electrons. High-temperature research has been stimulated by gas-turbine and jet-propulsion problems, and the development of new and improved temperature-measuring instruments is to be expected.

Liquid-in-Glass Thermometers

The liquid-in-glass thermometer is a simple, direct-reading, and conveniently portable instrument, widely used in many activities requiring the determination of temperature. Low-precision thermometers are inexpensive, and instruments of moderate precision are available for laboratory use.

This type of thermometer is usually made with a reservoir of liquid in a glass bulb connected directly to a glass capillary tube with graduated markings or with a scale attached. Mercury, the most commonly used liquid, is satisfactory from minus 38 F, its freezing point, up to about 600 F if the capillary space above the mercury is evacuated, or up to 900 F or higher if this space is filled with nitrogen or carbon dioxide under pressure. In the range from minus 95 to plus 150 F, alcohol may be used, and in the range from minus 300 to plus 70 F, pentane has been found suitable.

Use of unprotected glass thermometers is restricted to laboratory or field research applications. For more rugged service there are various designs of "industrial" thermometers, with the bulb and stem protected by a metal casing and usually arranged for screwing into a thermometer well. Obviously, the accuracy of this type is not as high as that of the unprotected laboratory-type instruments, and there is a greater time lag if temperatures are changing.

Correction of Errors

When high precision is desired, correction must be made for several possible sources of instrument error, which may be outlined as follows:

If the capillary bore is not uniform, equal increments of temperature will produce unequal changes in the length of the liquid column, hence the readings will not represent the true temperature.

The markings of the fixed and intermediate points on the scale may be in error.

The temperature readings will be slightly different when taken in the vertical and in the horizontal position of the thermometer because of the change in volume of the bulb with the hydrostatic pressure of the liquid column. Any change in the volume of the bulb as a result of external pressure will also, of course, affect the readings.

When the temperature of a thermometer is changed over a wide range, the glass will slowly "creep." This is known as thermal hysteresis. It may take a day to attain equilibrium.

A correction is required for the portion of the stem of the thermometer not exposed to the temperature being measured. This is known as the emergent-stem correction. Many laboratory thermometers are calibrated with the entire thermometer at uniform temperature (total immersion). If a total-immersion thermometer is only partially immersed in a hot fluid, with a portion of the stem exposed to the atmosphere, some of the liquid in the capillary will be at a reduced temperature and will expand less than with total immersion. A correction for partial immerson may be made by using an auxiliary thermometer affixed to the first so that the bulb of the auxiliary thermometer is next to the emergent stem of the first thermometer and one fourth of the way up between the liquid surface and the top of the liquid column in the first thermometer. The correction factor for mercury-in-glass thermometers will then be:

$$K = 0.00009\,N\,(t - t')$$

where:

K = correction to be added when the stem is cooler than the bulb, F

N = number of degrees on scale of first thermometer not immersed

t = temperature indicated by the partially immersed thermometer, F

t' = temperature indicated by the auxiliary thermometer, F

Some thermometers are calibrated for partial immersion to a specified point marked on the stem.

If a separation appears in the liquid column in the capillary, the column should be reunited before the thermometer is used. This is usually done by cooling the bulb to a very low temperature and shaking or rapping the thermometer so that the separated sections of the column rejoin the main body of the liquid.

When the temperature of the thermometer surroundings is changed, readings should not be taken until thermal equilibrium has been re-established.

Thermometers used at high temperature (above 300 F) may undergo permanent change of bulb volume and, therefore, require recalibration.

In reading the instrument, care must be taken to avoid an error from parallax (viewing the scale at an angle).

In industrial measurements, the largest source of error usually results from the difficulty of keeping the immersed portion of the thermometer at the same temperature as the fluid in which it is immersed. Cooling or heating effects of conduction or radiation from nearby materials must be eliminated or minimized, if accurate results are to be achieved. It is also well to bear in mind that a strong source of

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illumination near the thermometer may change its temperature as a result of radiation effects.

The Thermocouple

A thermocouple consists of two electrical conductors of dissimilar materials joined at their extremities to form a circuit. If one of the junctions is maintained at a temperature higher than that of the other, an electromotive force is set up which will produce a flow of current through the circuit (see Fig. 3).

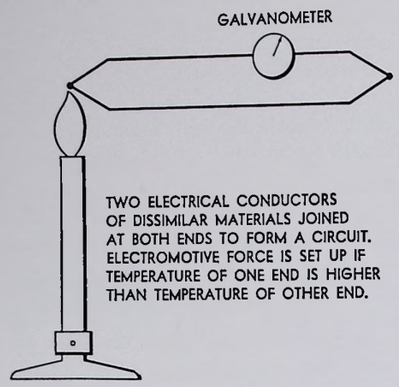


Fig. 3. *Principle of the thermocouple illustrated*

The magnitude of the net electromotive force depends upon the magnitude of the difference between the temperatures of the two junctions and the materials used for the conductors. No unbalanced or net electromotive force will be set up if the two junctions of dissimilar materials are at the same temperature, or if the conductors are of the same material even though the two junctions are at different temperatures. If one junction of dissimilar materials, which have been calibrated, is maintained at a known temperature, the temperature of the other junction can be determined by measuring the net electromotive force produced, since this force is proportional (almost directly) to the difference in temperature of the two junctions (see Fig. 4).

The principal advantages of the thermocouple in measuring temperatures are: versatility of application, rapidity of response through wide ranges of temperature, high degree of accuracy, durability, accurate reproducibility at relatively low cost, convenience of centralized reading or recording from one or many remote points, and simplicity of application to equipment for control and regulation of temperature.

Temperature Measurement

As a temperature-measuring device within its use limits, a thermocouple will accurately indicate the

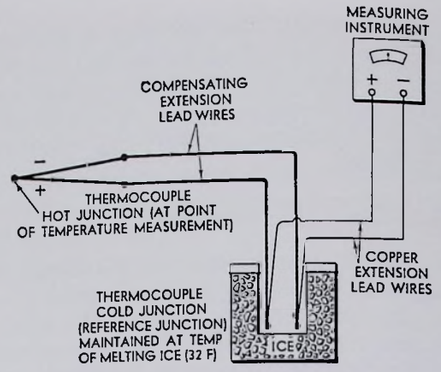


Fig. 4. *Simple thermocouple circuit and extension leads*

temperature at any accessible point. The net or unbalanced emf developed in two dissimilar electrical conductors joined at both ends is proportional to the difference in temperature between the point of application ("hot junction") and the "cold junction," or "reference junction," of the circuit. The latter is at a point of known temperature, as the room or area where the reading is taken, or it may be an ice bath. A commonly used and generally accurate method is illustrated diagrammatically by Fig. 5.

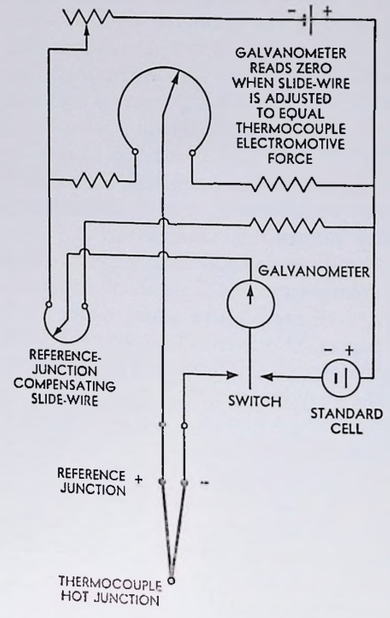


Fig. 5. *Thermocouple circuit with manually adjusted reference-junction compensation*

The relationship between the electrical unit (mv) for emf and the equivalent corresponding temperature difference between the two junctions has been carefully established by laboratory tests throughout the temperature-range limits for various thermocouple conductors in common use (see Tables, page 6-A1 ff, Appendix, for detailed data). The electrical readings are therefore readily converted to the equivalent temperature. For example, assume that A represents the unknown temperature at the point of application (hot junction), that B is the known temperature at the cold junction from which the leads are taken to the point of reading, and that the thermocouple conductors are copper to constantan. By reference to the table for these conductors: if the electrical reading is 14.71 mv and the temperature at B is 80 F corresponding to 1.73 mv, the mv corresponding to the difference in temperature from zero to A will be 14.71 + 1.73 or 16.44 mv, equivalent to a temperature of 600 F for A. To make direct readings possible, the electrical instruments now usually have scales in degrees of temperature calibrated for the various thermocouple conductors commonly used.

In considering the temperature-emf characteristics of different pairs of materials, one conductor is said to be positive to the other when the emf at its cold-junction terminal is electrically positive to that of the companion material at its cold-junction terminal. In referring to thermocouple pairs, it is customary to mention the positive conductor first. The relation of electromotive force (in millivolts) to temperature, for the various thermocouple conductors normally used and for their normal temperature-use limit, is shown by Fig. 6.

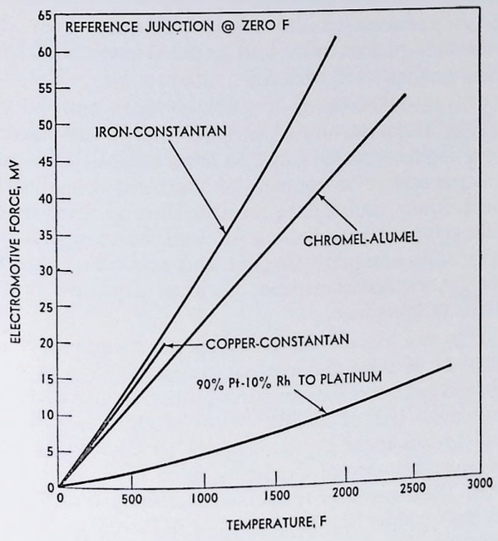


Fig. 6. *Relation of electromotive force to temperature produced by several commonly used thermocouples*

Reference Junction. For precise laboratory work or where exacting accuracy is required, it is customary to maintain the cold junction (or reference junction) at a controlled temperature, such as the fixed point of melting ice (0 C or 32 F). For less exacting work, the temperature of the reference junction may be measured by an auxiliary thermocouple or thermometer and the potentiometer readings corrected accordingly, or the instrument may be adjusted to give the true readings directly. Many popular types of indicating or recording potentiometers are equipped with compensating coils which, by a change of resistance with temperature, automatically adjust the readings to suit any variation from the cold-junction temperature at which the materials of the thermocouple are calibrated.

Measuring Circuits. In the simple thermocouple circuits illustrated by Figs. 4 and 5, the thermocouple wires of dissimilar materials extend from the hot-junction to the cold-junction (or reference-junction) terminals. In the potentiometer circuit, copper leads may be run from the reference-junction terminals to the measuring instrument without affecting the net emf of the thermocouple. The pair of similar copper leads acts merely as electrical connectors to transfer the emf from the reference junction to the copper terminals of the potentiometer. If the instrument is at a uniform temperature throughout, no emf is set up between the copper conductors and slide-wire materials of different composition within the potentiometer itself. However, if temperature differences do exist within the instrument, there will be disturbing emfs in the circuit, and the readings will be affected.

Multiple Circuits. If two or more thermocouples are connected in series, the total net emf at the outside terminals is equal to the sum of the emf developed by the individual couples. Where all of the individual hot junctions and all of the individual cold junctions are maintained at the same respective temperatures, as in the device known as the "thermopile," this multiplied value of emf makes it possible to detect and measure extremely small variations in temperature of the hot junctions.

In a similar series connection, if the individual hot junctions are at different temperatures (as when located at several points of a gas stream or of an object that is not uniform in temperature) while the cold junctions are maintained at a constant temperature, the total net emf will likewise be the sum of the individual emfs. The total net emf divided by the number of thermocouples in the series connection will give an arithmetical mean of the temperature of the gas stream or object. Use of the series circuit for

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measurement of average temperature is usually impractical, because the multiplied value of emf at the terminals is liable to be beyond the range of the instrument normally available for measurement of data from single thermocouples.

Two or more thermocouples may be connected in parallel for the purpose of obtaining a single reading of average temperature. In this case, the resistances of leads of all individual thermocouples must be equal. The emf across the terminals of such a circuit is the average of all the individual emfs and may be read on a potentiometer normally used for single thermocouples.

Two thermocouples may be connected in series with opposed or "bucking" polarity to measure the difference in temperature between their hot-junction contacts. It is advisable in this case to provide leads for the measurement of both individual temperatures as well as for the differential, in order to establish the proper range of emf-per-degree characteristics of the conductors employed.

To prevent short circuit or current flow between points of differing potential when two or more thermocouples are connected in series or parallel, it is important that both the hot-junction and cold-junction terminals of the individual couples be electrically insulated from one another. Also, for multiple-type circuits the simple conversion of average emf to temperature equivalent is strictly true only if the emf-per-degree for the thermocouple materials is constant through the range of temperature of all thermocouple positions.

General Principles of Application

For successful use of thermocouples in practical application, certain general principles and their corollaries, as outlined below, must be kept in mind.

There may be a difference between the temperature of the hot junction and that of the object or substance under investigation. This may be caused by inadequate thermal contact between thermocouple and object, by localized conduction of heat to or from the hot junction resulting from exposure of the thermocouple wires to media of higher or lower temperature, or by radiation effects of surrounding objects. The application of the thermocouple itself may in some cases distort the normal conditions of heat flow or temperature pattern within the object being studied. These factors should be minimized as far as possible by means that become apparent on consideration of the problem. Beyond that, the readings may sometimes be corrected by evaluating the possible errors by calculation or by reasonable estimate and comparison with correlated data.

If the composition of the thermocouple conductors is not homogeneous or a third material is present in the circuit, then, if there is also a temperature difference in the region of these conditions, the emf at the terminals will be affected. Intermediate or parasitic couples will be set up within the circuit, which may raise or lower the net emf developed by the primary couple. It is therefore clear that the use of brazing or soldering material as a bond between two thermocouple wires at the hot junction would affect the readings unless all these materials were at the same temperature. The same is true if thermocouple wires, in close proximity to one another, are separately attached to an object for the purpose of determining its temperature. The object then becomes the hot junction, and any temperature gradient in it between the thermocouple wires affects the accuracy of the measurement. If the third material at each point of contact with the conductors is at the same temperature as the conductors, the net thermal emf at the cold-junction terminals will not be altered. By similar reasoning, it is wrong to assume that the structural parts of a boiler and its supports may be utilized without error as a common iron conductor, as is sometimes done for connecting a group of iron-constantan couples to a multipoint recorder or selector switch.

Faulty readings may be obtained from a portable potentiometer when it is taken from cold surroundings into a warm room, or vice versa, and used for temperature measurements before all parts of its circuit have reached a uniform temperature.

Spurious emfs may be developed in the thermocouple circuit by inductive effects of unshielded a-c power leads or by electrolytic activity in damp insulation or in damp deposits in contact with the conductors. Similarly, the terminal emf may be affected by short circuit of the conductors caused by deterioration of insulation and physical contact of lead wires or deposited material.

The service life of a thermocouple can be no greater than the endurance of any of its several components—the hot-junction contact, the lead wires, and the splices to contacts of the measuring instrument. Many useful data are lost through careless or unimaginative installations of lead wire, which fail to provide adequate support and protection against chafing, expansion strains, or local exposure to extreme temperature.

The emf developed by a thermocouple is independent of wire size and of change in size of conductors between the measuring junction and recorder terminals. Use of smaller wires at the hot junction will give a more rapid response to change of temperature because of the lesser mass of the hot junction. Their use also minimizes the disturbing effects of heat conduction in the wires adjacent to the hot junction. Offsetting these advantages, however, are the greater vulnerability of the smaller wires to physical damage and the shorter service life if subjected to oxidation at high temperature or to corrosive atmospheres. Also, in using smaller wires, if long leads are employed it is more difficult to balance the potentiometer because of the reduced sensitivity of the thermocouple-potentiometer assembly, since the current flow for a given emf is less because of the increased electrical resistance.

Practical Use of Thermocouples

Selection of Materials. While many combinations of dissimilar metals might be employed to form thermocouple pairs, it is important for satisfactory use in temperature measurement that they have the following characteristics: a thermal emf which increases continuously and uniformly with increase in temperature and is of sufficient magnitude to be measured with accuracy; physical strength; resistance to oxidation or corrosive attack; availability in wire form; and composition or emf characteristics unaffected by fabrication or by reaction with atmospheres and contact with other materials.

Combinations of metals and alloys most frequently used for thermocouples are listed in Table 1 with their general characteristics and useful temperature range. Selection depends largely upon ability to withstand oxidation attack at the maximum service temperatures expected. Durability will depend upon size of wire, use or omission of protection tubes, and nature of the atmosphere. For heavy-duty service B&S 8-gage (0.128-in. diam) wire is generally used. Smaller diameters, also available, are more convenient for test work and cost less.

All thermocouple materials will deteriorate when exposed at the higher temperatures of their useful range to air or other atmospheres and to contact with other materials. Platinum in particular is affected by metallic oxides and by carbon and hydrocarbon gases when used at temperatures above 1000 F and, in the course of time, is subject to calibration drift. From Table 1 it will be noted that the upper limit of the useful range in temperature for all thermocouple materials listed is far below the maximum temperature at which rapid deterioration sets in.

For high-temperature duty in a permanent installation or where destructive contact is extreme, service life may be extended, at a sacrifice in rapidity of response to temperature changes, by using closed-end protection tubes of alloy or ceramic material. The arrangement should permit removal of the thermocouple from the protection tube for calibration and renewal when necessary. For duty during short periods of time, as in some test work, protection tubes may be omitted if calibration is frequent, permitting correction of the data as required. For normal duty within the useful range of the thermocouple selected, the correction for change of calibration is usually negligible.

Thermocouple and Lead Wire. There are two classes of wire for thermocouples, the closely standardized and matched "thermocouple wire" and the less accurate "compensating lead wire." For thermocouples of noble metal, extension leads of copper and copper-nickel alloy, which have an emf characteristic near that of the noble-metal pair, are used to save cost. For thermocouples of base metal, the extension leads in general use, while of the same nominal composition as the thermocouple wires, are less expensive since the control in manufacture and in subsequent calibration and matching of pairs need not be as rigorous. Wires of these two classes are made by several manufacturers in a variety of sizes and with insulating coverings of various materials in both the single-conductor and "duplex" arrangements of paired wires.

For accuracy, the matched "thermocouple wire" should be used at the hot junction and continued through the zone of greatest temperature gradient to a point substantially at room temperature, where "compensating lead wire" may be spliced on for extension to the reference junction. The reference junc-

TABLE 1

Types of Thermocouples in General Use

Type of Thermocouple°	Useful Temp Range, F	Maximum Temp, F	Millivolts at 500 F†	Magnetic Wire
(+) Copper to Constantan (—)	— 300 to 650	1100	13.24	——
(+) Iron to Constantan (—)	0 to 1400	1800	15.01	Iron (+)
(+) Chromel to Alumel (—)	0 to 2300	2500	11.24	Alumel (—)
(+) 90% Pt—10% Rh to Platinum (—)	900 to 2600	3190‡	2.048	——

° Nominal composition: constantan, 55% Cu, 45% Ni; chromel, 90% Ni, 10% Cr; alumel, 95% Ni, 5% Al, Si, & Mn.
† Reference junction, zero F.
‡ Melting point.

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tion is usually located at the recorder or central point of observation, but where a number of thermocouples are used it is sometimes economical to establish a "zone box" (see Fig. 7), from which copper conductors of lower cost are extended to the recorder panel.

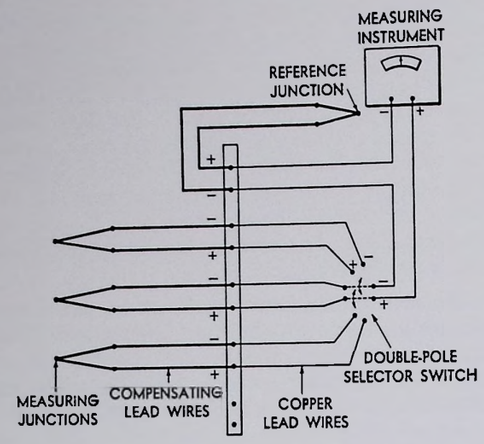


Fig. 7. *Thermocouple and lead-wire arrangement to measuring instrument through selector switch*

Splicing of extension lead wires may be done by twisting the ends of the wires together, by the use of screw-clamp connectors, or by fusion welding, soldering, or brazing. The wire ends should be thoroughly cleaned to provide good electrical contact.

Care should be taken to maintain correct polarity by joining together wires of the same composition. Polarity is usually identified by color code or tracers in the wire covering. Where one wire of a thermocouple is magnetic, the correct polarity may be confirmed by using a magnet to detect the magnetic wire of the pair. As a check, the completed thermocouple should be connected to a potentiometer circuit, and each of the intermediate junctions should be heated. If the wires are correctly paired, the galvanometer will not indicate any deflection unless the thermocouple hot junction is heated.

Upon completing an installation of thermocouples and extension leads, each circuit may be checked for poor contact in splices and broken or grounded conductors by measuring its over-all electrical resistance, using a portable ohmmeter. This record will also serve as a reference in subsequent checking, when deterioration of junctions may be suspected. Normal resistances of various thermocouple wires are shown in Table 2.

Application of Hot Junction

A number of practical applications of the hot junction of a thermocouple are illustrated in Figs. 8 through 13.

Welded Junction. Wire and pipe-type elements may be purchased commercially or may be made from stock thermocouple wire, as shown by Fig. 8. These types are useful for direct immersion in a gas stream, for insertion into thermometer wells, or for thermal contact with solid surfaces. The junction contact between wires is preferably made by fusion welding in a carbon arc or, for base-metal couples, in a neutral to oxidizing flame, with sparing use of borax or other flux suited to the material. If the ends of very fine wires (28 gage) are joined by a short twist, a welded bead may be formed by making arc contact with mercury as an electrode, submerged in silicone or mineral oil to prevent oxidation. Where temperature and other limitations permit, the junction may be soldered or brazed.

The effective hot junction occurs where the conductors make electrical contact (nearest to the potentiometer), which may be in advance of the terminal weld. If the entire twist and bead are at uniform temperature, no error is introduced. Precautions should be taken to prevent a temperature gradient in the vicinity of the measuring junction.

TABLE 2

Thermocouple Wire Resistance

Ohms/ft at 68 F

B&S Gage	Iron	Constantan	Copper	Chromel	Alumel	Platinum	90% Pt-10% Rh
8	0.00364	0.0178	0.000628	0.0257	0.0107	0.0041	0.0081
14	0.01465	0.0716	0.00253	0.1035	0.0431	0.0165	0.0328
16	0.02328	0.1138	0.00402	0.1645	0.0685	0.0262	0.0519
20	0.06890	0.2880	0.0102	0.4163	0.1730	0.066	0.131
22	0.09359	0.4575	0.0161	0.6614	0.2754	0.105	0.208
24	0.1489	0.7277	0.0257	1.052	0.4381	0.167	0.332
28	0.3763	1.840	0.0649	2.66	1.108	0.422	0.840

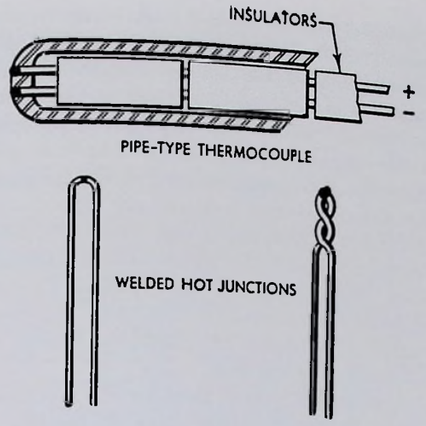


Fig. 8. *Thermocouple elements and hot junctions*

Peened Junction. Where the temperature of a metallic surface is to be measured, one of the most useful and versatile types of hot junction is formed by peening the wires separately into holes drilled into the surface of the metal. Steps in the procedure for making this type of junction are illustrated in Fig. 9.

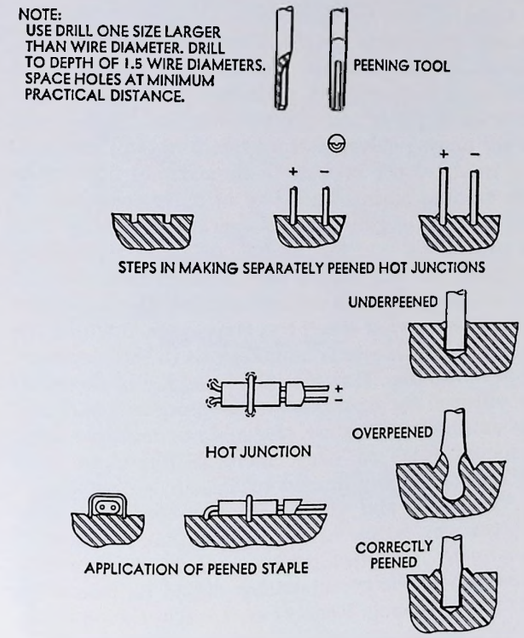


Fig. 9. *Procedure for installing a peened-junction thermocouple to measure surface temperature of metal*

The peened junction has the advantage of being completely mechanical in application, with tight and remarkably strong attachment and minimum interference with the normal temperature of the object. The temperature indicated is essentially that of the metallic surface, which is the first point of contact with the conductors. The depth of the drilled hole has no effect other than for mechanical strength. The points of junction contact are subject to the effects of thermal conduction in the wires. Where this type of error is significant, precautions should be taken to prevent temperature gradients in the wires. Drilled holes for reception of the wire ends should allow for a snug fit before peening. The wire ends should be thoroughly cleaned and dressed to a flat end by filing.

The mechanics of forming the peened hot junction lend themselves to making a convenient type of anchorage, referred to as the "peened staple," for giving physical support to thermocouple and lead wires along a metal surface. The peened staple is illustrated in Fig. 9.

Millivolt Potentiometer

The temperature-emf relationships of standardized thermocouple wires, as established by the Leeds & Northrup Co. for the four thermocouple pairs in Table 1, are given in tables in the Appendix, page 6-A1 ff. By the use of these tables, readings taken on a millivolt potentiometer may be converted to values of equivalent temperature. Tabulated values represent the net emf impressed on the potentiometer terminals when the reference junction is at zero F and the measuring junction is at the temperature shown. The millivolt potentiometer may be employed with any type of thermocouple and is frequently used in practice when several different types of couples are installed in the test setup.

It is, of course, impractical to maintain the reference junction at zero F while taking readings. Correction for any reference-junction temperature can be made by adding to each observed reading of emf the value of the emf corresponding to the temperature of the reference junction and then entering the table with this sum to find the equivalent temperature of the measuring junction. Most millivolt potentiometers are equipped with a compensator, which may be adjusted to correct for the actual reference-junction temperature by setting the compensator at the value of emf from the table corresponding to the actual temperature of the reference junction. Direct readings will then conform to the values in the table. Even when this is so, the previous method is recommended and may be followed by setting the compensator at zero. This method is preferable, since it is simpler and eliminates the possibility of incorrect compensator setting. Also, in using this method, compensation may be made at a later time under good working conditions and can be independently checked by others at any time.

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Direct-Reading Potentiometer

When calibrated for use with specific types of thermocouples, a potentiometer may be graduated to read directly in terms of temperature of the hot junction instead of millivolts. In this case, the reference-junction temperature is usually compensated automatically by resistor coils, built into the instrument circuit, which respond to deviations from a standard room-temperature basis. It is, of course, necessary that compensating lead wires, or their equivalent, be extended to the instrument terminals, so that the reference junction is located at, or is at the same temperature as, the instrument. Characteristics of the compensating coils are specifically adapted to the type of thermocouple material for which the potentiometer is calibrated.

While not recommended as good practice, circumstances sometimes make it expedient to use a direct-reading potentiometer, calibrated for one type of couple, in combination with thermocouples of different emf characteristics. Equivalent corrected temperatures may be determined by conversion of scale readings, through use of the tables previously mentioned, with proper recognition of the effect of automatic reference-junction compensation. The following example of such conversion may serve as a pattern for the procedure for other combinations of types.

Assuming an I-C (iron to constantan) thermocouple connected to a potentiometer calibrated for a C-A (chromel to alumel) thermocouple:

(a) Read the potentiometer scale in F.
(b) Measure reference-junction temperature in F.
(c) From the table for C-A, take the mv corresponding to the temperature in (a).
(d) From the table for C-A, take the mv corresponding to reference-junction temperature in (b).
(e) Subtract (d) from (c).
(f) From the table for I-C, take the mv corresponding to reference-junction temperature in (b).
(g) Add (e) and (f).
(h) In the table for I-C, the temperature corresponding to (g) will be the measuring-junction temperature.

Temperature Recorders

In temperature recorders, a power-operated mechanism adjusts the slide-wire of a potentiometer circuit to balance the impulse received from the thermocouple. The action of the slide-wire is coordinated with the movement of a recording pen or print wheel that is drawn over, or impressed on, a temperature-graduated chart that moves at a given speed.

In the earlier types of recorders (many are still in use), the deflection of a moving galvanometer coil (and its stylus), which receives the primary impulse of a change in thermocouple emf, monitors the independently driven power mechanism that adjusts the slide-wire until the emf is balanced and the galvanometer is restored to neutral or null position. In this type of recorder, the only load imposed upon the feeble impulse of the thermocouple is that of deflecting the galvanometer, and at the null or balanced position this load is essentially zero. To insure sensitivity, the galvanometer coil and its suspensions are necessarily light and require accurate mechanical balancing. Some provision is usually made for cushion mounting of the recorder to minimize disturbances from mechanical shock or vibration, which might cause deflection of the coil.

The more recent type of recorder has a completely electrically balanced circuit and is therefore not subject to the limitations imposed by the moving galvanometer. In this circuit, a change of thermocouple emf sets up an electrical imbalance which, upon being amplified, is used to drive a reversing motor that adjusts the slide-wire in the proper direction to restore the circuit to balance. Response to electrical imbalance is immediate, and extremely fast action of the driving mechanism is possible, so that an essentially continuous record may be traced for one thermocouple or a rapid scanning of many thermocouples may be recorded.

Checking Recorder Calibration. Compared with the manually operated potentiometer, all recorders are inherently less accurate because of wear and lost motion in driving gears, shrinkage or printing inaccuracies of paper and chart scales, and, in the case of the moving-galvanometer type, some lag in reaching exact balance because of diminishing impulse as this point is approached. Use of extremely long or unduly small-gage lead wires aggravates the lag. This is particularly true for the galvanometer type of instrument because of the very small current flow as the unbalanced emf is reduced to zero. The resistance of long leads is of much less consequence where the energy of imbalance is amplified, as in the electronic type of recorder. However, the accuracy of recorders is sufficient for most commercial purposes and may be enhanced by the use of double or multiple-range circuits, which in effect increase the chart scale dimensions to any desired extent.

Calibration and adjustment of recorders may be checked by means of a portable potentiometer of known accuracy. In this operation, the thermocouple terminals of the potentiometer should be connected, by compensating lead wires, to the same-polarity thermocouple terminals of the recorder from which the thermocouple was disconnected. After checking both instruments for standard-cell adjustment, the recorder should be started, the slide-wire of the portable instrument set on some desired reading, and the thermocouple switch engaged. The recorder will

then operate to balance the emf imposed on it by the service cell of the portable potentiometer, and if it is in correct adjustment, the printed record will correspond to the reading set manually on the scale of the portable instrument.

Temperature of Fluids Inside Pipes

The temperature of a fluid (liquid, gas, or vapor) flowing under pressure through a pipe is usually measured by a glass thermometer, an electrical-resistance thermometer, a thermocouple inserted in a thermometer well projecting into the fluid, or a thermocouple attached to the outside of the pipe wall. While the latter will give accurate results if all the necessary precautions are observed, the thermometer well is generally preferred, since the probability of error is less.

A metal tube closed at one end, screwed or welded into a pipe wall, and projecting into the fluid serves as a thermometer well (see Fig. 10). It must have the mechanical strength and rigidity to withstand hydrostatic pressure, bending caused by its resistance to fluid flow, and vibration. To minimize conduction of heat to the surroundings and to give rapid response to temperature changes, the dimensions of the well

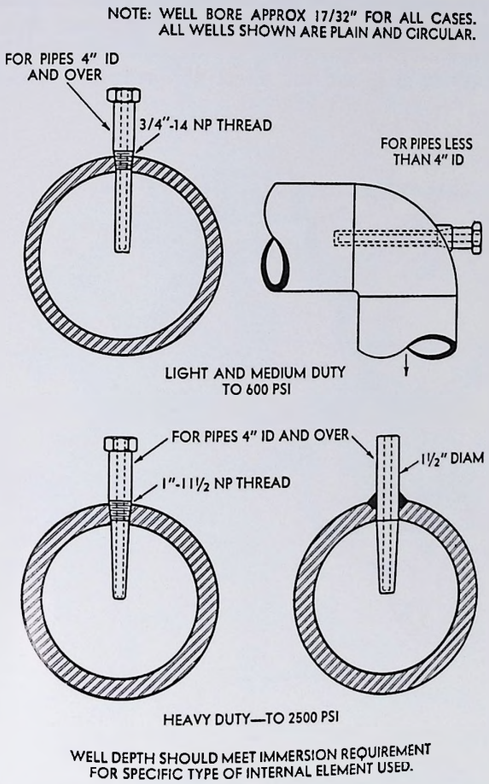


Fig. 10. *Thermocouple wells and methods of installation*

should be as small as compatible with the strength requirements. The part outside the pipe should be as small as possible. The material of the well must resist the erosive or corrosive action of the fluid. Since the temperature may be locally depressed by acceleration through a constriction in a pipe carrying a compressible fluid, the well should not be placed near a constriction. Projecting parts of the thermometer well and the pipe wall for some distance from the well should be thoroughly insulated against heat loss. The space between the outer end of the well and the thermal element should be tightly packed with insulating material to prevent circulation of air into the well; omission of this packing could introduce a substantial error.

The details of thermometer-well designs vary. If the fluid is a liquid or a saturated vapor, good heat transfer is assured, and a plain well is satisfactory. If the fluid is a gas or superheated vapor, a finned well is sometimes used. The method of attachment to the pipe wall, either by screwing into the wall or boss or by welding or brazing, is optional with the designer provided safety-code requirements are met.

The thermometer well will tend to give an average temperature even if there is temperature stratification in the fluid, since thermal conduction along the well will serve to equalize the temperature.

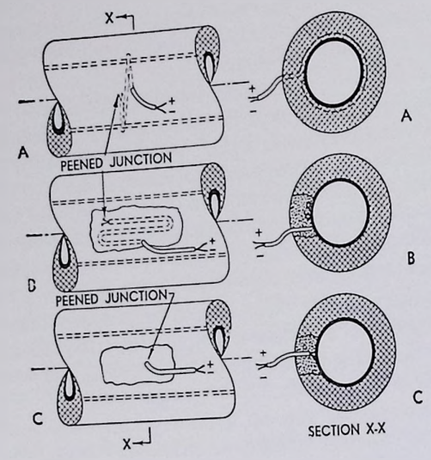
Thermocouples may be attached to the outside of pipes by brazing, welding, or peening. Peening the thermocouple wires into holes drilled in the surface of the pipe is preferred as an easy and reliable method of installation. The thermocouple readings will give the temperature of the pipe wall at the exact point of contact with the wires, and any factor which causes the pipe-wall temperature at that point to differ from the temperature of the fluid inside the pipe will introduce an error.

Heat flow from the fluid to the surroundings will cause a temperature drop through the fluid film inside the pipe and through the pipe wall. Conversely, a temperature rise will occur if heat flows from the surroundings through the pipe wall to the fluid. If the pipe is not insulated, these temperature differences will usually be great enough to give incorrect readings for the fluid temperature. However, if the pipe is properly insulated, such changes in temperature will be negligible, and the readings will accurately indicate the temperature of the fluid if that temperature has been constant or if, after change, time has been allowed for the pipe wall to reach thermal equilibrium. Insulation restricted to the immediate vicinity of the thermocouple, with the remainder of the pipe bare, is not sufficient, since thermal conduction along the pipe wall will affect the readings.

Conduction of heat along the thermocouple wires is an important source of possible error. This can be

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prevented if the thermocouple wires are wrapped once or twice tightly around the pipe and held in place before the pipe insulation is applied. In this way the wires are brought to pipe-wall temperature in the region of the thermocouple junction, and thermal equilibrium is established. Extending the wires back and forth against the pipe wall will accomplish the same result. Both methods are illustrated in Fig. 11. In either case, fine wire will permit less heat conduction than heavy wire.



A- GOOD PRACTICE—THERMOCOUPLE WIRE WRAPPED AROUND PIPE UNDERNEATH PIPE INSULATION.

B- GOOD PRACTICE—THERMOCOUPLE WIRE LAID ALONG PIPE WALL BEFORE REINSULATION.

C- BAD PRACTICE—THERMOCOUPLE WIRE LED DIRECTLY TO OUTSIDE.

NOTE: ANY OPENING MADE IN PIPE INSULATION FOR ACCESS TO PIPE WALL FOR INSTALLATION OF THERMOCOUPLE MUST BE CAREFULLY REINSULATED WITH DRY INSULATION FIRMLY HELD IN PLACE.

Fig. 11. *Thermocouple wires extending from hot junction disposed on pipe wall before leading outside*

An additional source of error is the influence of "through steel"—uninsulated connections or valves which extend through the insulation and cause local cooling of the pipe wall. Locations remote from "through steel" should therefore be selected for the installation of thermocouples to determine the temperature of fluids inside pipes.

Tube-Temperature Measurement

In steam boiler practice, it is frequently desirable to know accurately the metal temperature of tubes in different classes of service, such as furnace wall or boiler generating tubes that are cooled by water and steam at saturation temperature, economizer tubes cooled by water below the temperature of saturation, or superheater and reheater tubes cooled by steam above saturation temperature. Such measurements may be for the purpose of determining the safety of pressure parts, for determining uniformity or unbalanced condition among tubes in parallel-flow circuits, for measurement of temperature increase of the fluid between inlet and outlet conditions, or for similar functional problems. In determining the temperature of the tube metal itself, measurement by thermocouple is a very practical and reliable method and is now used almost exclusively.

The peened type of hot junction, as illustrated by Fig. 9, is satisfactory in many cases and is probably the simplest form of thermocouple application. This couple measures essentially the temperature of the tube-metal surface, subject to errors caused by thermal conduction in the thermocouple wires to or from the junction contact. Conduction errors may be minimized as previously described and by shielding the wires from heat input or loss by wrapping with asbestos or other insulating material.

Furnace Wall Tubes

For satisfactory results and durability of the thermocouple, special protection is obviously necessary in the determination of the temperature of water-cooled furnace wall tubes that are exposed to extremely high temperatures, the deposit and shedding of ash or slag, and the effect of destructive atmospheres. If a simple peened couple is used, the results may be of doubtful accuracy because of error from conduction, and in most cases the service life would be short because of physical damage or deterioration of the wires from overheating. Cover plates welded to the tubes have been used to protect the wires, but these plates interfere with the normal heat flow and may be the cause of local ash deposits that create abnormal conditions at the point of temperature measurement.

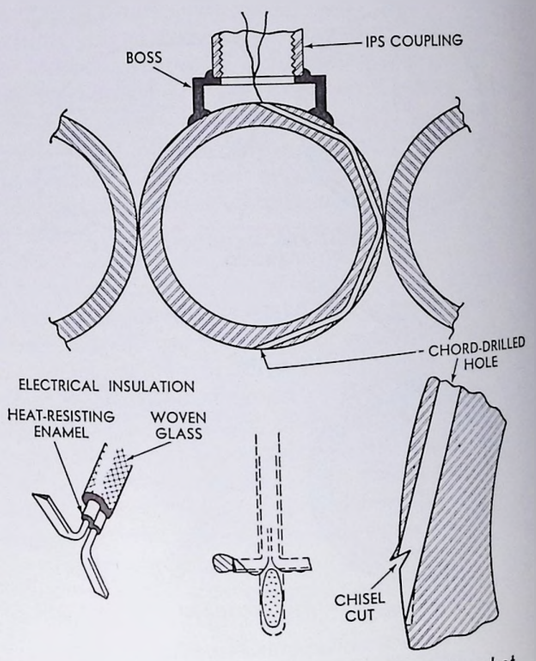


Fig. 12. *Thermocouple wires extending from hot junction through chord-drilled holes in tube wall*

The use of chord-drilled holes through which the thermocouple wires are laced, as shown in Fig. 12, was developed by B&W and has been found to be a very satisfactory method of application for furnace wall tubes. The tube surface is free from projections, the wires are protected, and the effect of thermal conduction at the hot junction is minimized because the wires pass through an essentially isothermal zone before emerging into cooler surroundings at the rear of the tube. To minimize the effect on the heat-flow pattern within the tube metal, the chord-drilled holes should be as small as possible. The effect of these holes upon the strength of the tube is small. It is least in the critical direction of hoop stress and is readily tolerated in the direction of longitudinal stress. Use of chord-drilled holes in the thicker-wall tubes for high steam pressure is therefore justified and practicable.

Gradient Thermocouples. For measuring the temperature gradient in the tube metal and for determining the heat flow rate or the thermal resistance of internal scale deposit, two thermocouples may be applied within the thickness of the tube wall, spaced at different depths below the tube surface. The chord holes for the thermocouples are spaced at a nominal distance from one another along the length of the tube to avoid physical or thermal interferences. As a check and to detect the possible presence of localized or nonuniform external ash deposits that might cause individual gradient couples to vary, it is advisable to provide two sets of each type of couple. In practice, couples of this type may be applied to a separate short section of tube, which may then be welded into a furnace tube at the desired location. Gradient couples are shown in Fig. 13, and characteristic data obtained from their use are shown in Fig. 14.

When heat is conducted radially through a cylindrical wall, the heat flow area decreases as the inner wall surface is approached. Consequently, the temperature does not vary in proportion to the actual wall thickness, since there is a change in area. A plot of temperature versus wall thickness will not be a straight line. In analyzing test data from tube-wall thermocouples, it is convenient to make use of an

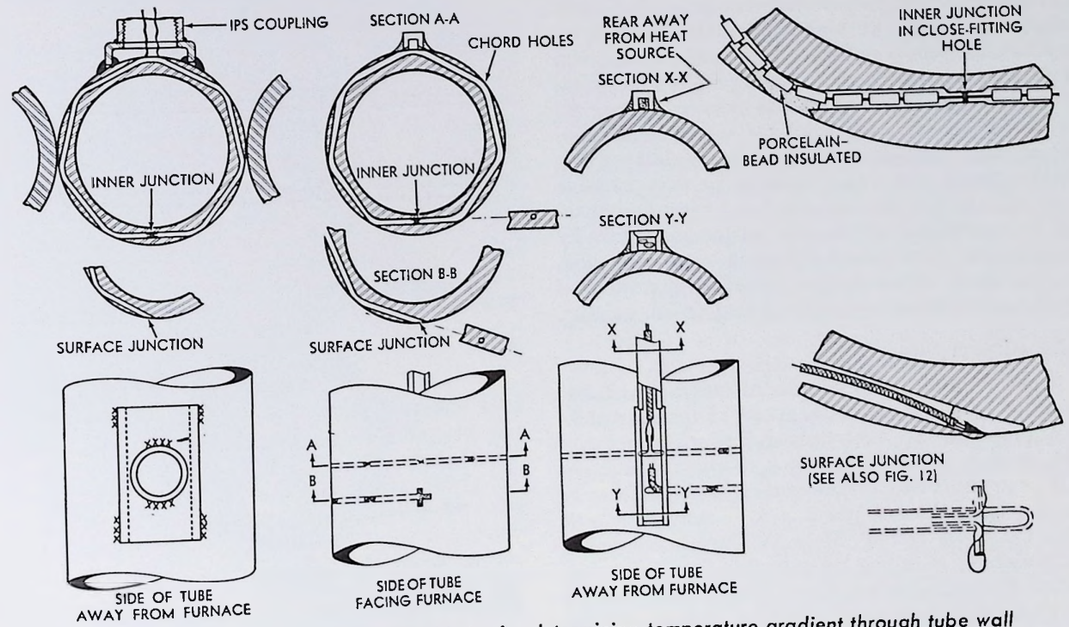


Fig. 13. *Thermocouples in chord-drilled holes for determining temperature gradient through tube wall*

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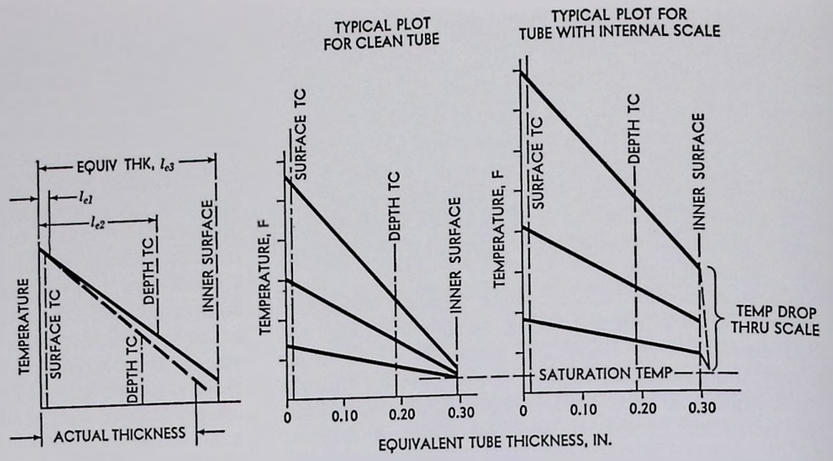


Fig. 14. *Typical observed temperature gradients in tube wall for analysis and determination of rate of heat flow*

"equivalent thickness," as shown in the illustration. An equivalent thickness for any depth can be determined by use of the following equation:

$$l_e = 0.5 d_o \log_e (d_o/d_i)$$

where:

l_e = equivalent thickness, in.

d_o = outside diam of tube, in.

d_i = diam of inner-junction circle at center line of thermocouple wire, in.

When temperature is plotted versus equivalent thickness, the points lie on a straight line, which may be extended to the tube-wall surfaces.

Lead-Wire Protection. If proper care is taken, thermocouples may be placed on steam-generating tubes completely exposed to high-temperature gas flow, as in a furnace screen or boiler tube bank. Lead wires of suitably small dimension, having high-temperature Fiberglas covering or porcelain insulators, may be protected by extending them through a steel channel conduit attached by continuous welding along the length of the water-cooled tube to a suitable point of exit from the furnace. Lead wires and channel are maintained within safe temperature limits by good thermal contact with the water-cooled tube and, in most cases, obtain further shielding or protection by the accumulation of external slag or ash coating.

Superheater Tubes

Application of thermocouples to superheater tubes in active heat transfer zones is one of the most difficult problems of tube-temperature measurement. This is true primarily because of the difficulty in protecting lead wires from excessive temperature in passing from the hot junction to a convenient exit from the boiler setting. Since the superheater tubes are essentially at higher temperatures than the water-cooled generating tubes, less cooling is available for the protecting channels. Continuous welding of these channels is usually not permissible because of thermal expansion stresses set up in the tubes. Short channel sections with intermittent welds have sometimes been used with success. Such an arrangement for the protection of thermocouple and lead wires is illustrated by Fig. 15.

In several instances, where the effort has been justified by the importance of tube-temperature measurement, the method illustrated by Fig. 16 has been used with good results. A tube of small dimensions, extending through the interior of the superheater tube and attached at each end to the wall of the tube with pressure-tight welds, serves as a conduit

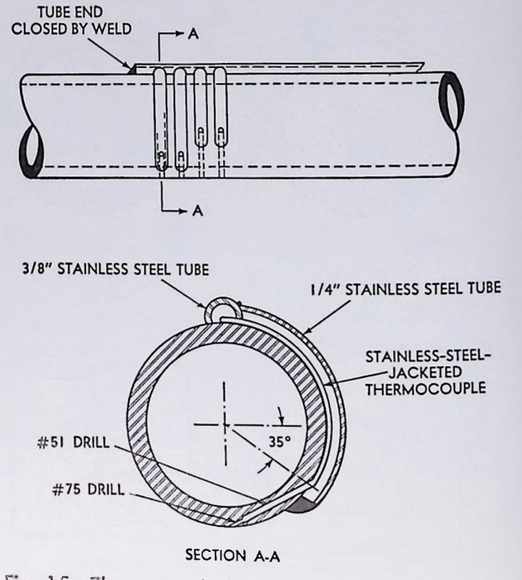


Fig. 15. *Thermocouple lead wires in high-temperature service protected by external channels*

for the thermocouple wires. Chord-drilled holes at the hot-junction end provide thermal and physical protection of the wires, and the wires emerge outside of the gas flow zone where the extension lead wires may be spliced on without difficulty. In passing through the coolest medium (steam), the thermocouple wires are given the maximum possible protection. The conduit is made sufficiently flexible so that its terminal welds are not unduly stressed by differential expansion between the materials of conduit and superheater tube. Careful attention is given to the metallurgical and welding problems involved in making these attachments.

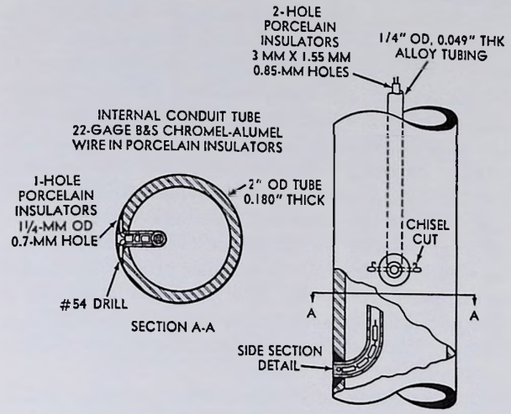


Fig. 16. *Thermocouple wires in high-temperature service protected by internal conduit*

For the purpose of determining when nondrainable superheater tubes are clear of condensate during pressure-raising procedures, many operators of high-pressure high-temperature units equip superheater tubes with permanent thermocouples installed out of the gas stream (see Fig. 17). These couples are connected to temperature recorders in the control room, which provide an operating guide and a permanent record. A second and equally important function of such installations is a monitoring of tube-metal temperatures during normal operation to permit rapid adjustment should an upset distribution cause overheating.

Gas-Temperature Measurement

Gas temperature is one of the most important items of the data required in testing or recording the performance of steam-generating units. It is also of equal importance in safe and efficient operation. In the measurement of gas temperature, therefore, care must be taken, first, to make certain that the instrument used indicates the temperature correctly, and second, to interpret the temperature readings to give a true average temperature of the gas stream, which is usually not uniform.

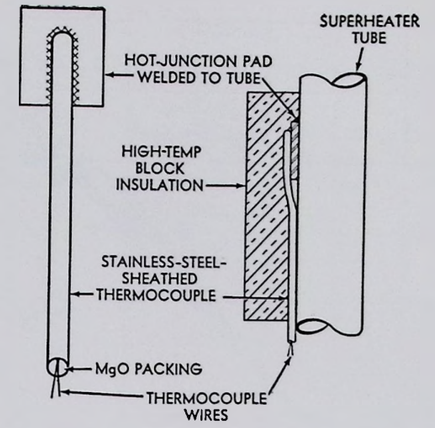


Fig. 17. *Permanent sheathed thermocouple for measuring metal temperatures out of the gas stream*

In all cases of gas-temperature measurement, the temperature-sensitive element approaches a temperature in equilibrium with the conditions of its environment. While it receives heat primarily by convection transfer from the hot gases in which it is immersed, it is also subject to heat exchange by radiation to and from surrounding surfaces and by conduction through the instrument itself, which should be reduced to negligible values. If the temperature of the surrounding surfaces does not differ from that of the gas (air flowing through an insulated duct), the temperature indicated by the instrument should represent quite accurately the temperature of the gas. If the temperature of the surrounding surfaces is higher or lower than that of the gas, the temperature indicated by the instrument will be higher or lower, respectively, than the temperature of the gas.

The amount of variation from the true temperature of the gas depends upon the temperature and velocity of the gas, the temperature of the surroundings, the size of the temperature-measurement element, and the physical construction of the element and its supports. To correct for the errors in temperature caused by the surroundings in any particular installation, it is best to calibrate the instrument against a known and reliable determination of the gas temperature.

When gas temperatures below 1000 F are to be measured, a bare thermocouple, resistance pyrometer, mercury-in-glass thermometer, or one of the various bulb-type pyrometers may be used.

When a 22-gage bare thermocouple is used to measure the gas temperature in boiler, economizer, and air heater cavities with surrounding walls colder than the gas, the magnitude of the error in the observed readings is indicated in Fig. 18.

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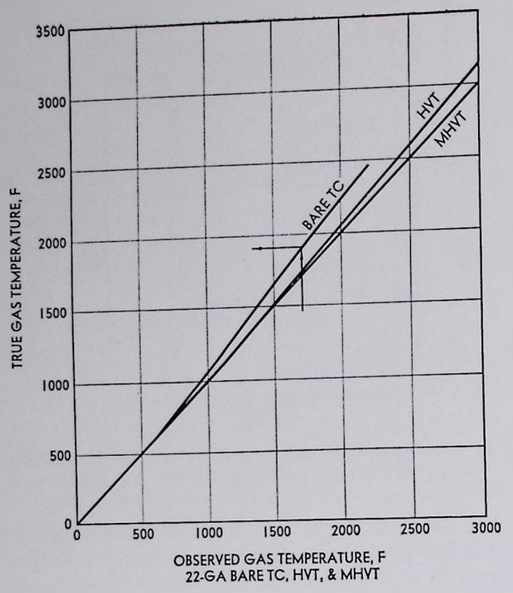


Fig. 18. *General magnitude of error in observed readings when measuring gas temperature in boiler cavities with thermocouples*

If gas temperatures exceed 1000 F, the errors mentioned above become more pronounced. The high-velocity and multiple-shield high-velocity thermocouples (see below), developed to correct these errors, are the best instruments available for the measurement of high gas temperature in colder surroundings or low gas temperature in hotter surroundings. A comparison between the readings for the high-velocity and for the multiple-shield high-velocity thermocouples in typical boiler furnaces and cavities is shown in Fig. 18. The optical pyrometer and the radiation pyrometer are not designed to measure gas temperatures and should not be used for this purpose, since the result may be extremely misleading.

High-Velocity Thermocouple

Since the transfer of heat by radiation is proportional to surface area, emissivity, and the difference of the fourth powers of the absolute temperatures of source and receiver, it follows (the gases being considered as nominally transparent to the passage of radiant energy) that the effects of radiation increase as the temperature difference between the thermocouple hot junction and the surrounding surfaces increases. Also, since transfer of heat to the thermocouple by convection is proportional to a power of gas mass flow and to the first power of the temperature difference between gas and thermocouple, it follows that the temperature of the junction may be brought more nearly to the true temperature of the gas by increasing the rate of mass flow and convection transfer while shielding the junction from the influence of radiation transfer.

A high-velocity thermocouple (HVT) for measuring high gas temperatures in boiler units is illustrated in the sketch, Fig. 19. This portable assembly is used primarily for making test traverses in high-duty zones by insertion through inspection doors or other test openings in the setting.

The thermocouple is supported by a water-cooled holder consisting of concentric tubes of suitable length to span the traverse distance. The measuring junction is surrounded by a tubular porcelain radiation shield through which gas flow is induced at high velocity by an aspirator attached to the external connections. The gas aspiration rate over the thermocouple can be checked by an orifice incorporated with the aspirator and connected to the probe by a length of hose. The gas mass flow over the thermocouple junction should be not less than 15,000 lb/sq ft, hr (see Fig. 20). Heat transfer to junction and shield by convection is simultaneous, so that both approach the temperature of the gas stream. Radiation transfer at the junction is diminished by the shield. However, since the shield is exposed externally to the radiation effect of the surroundings, it may gain or lose heat, and its temperature may be somewhat different from that of the junction.

Fig. 19. *Rugged water-cooled high-velocity thermocouple (HVT) for determination of high gas temperatures*

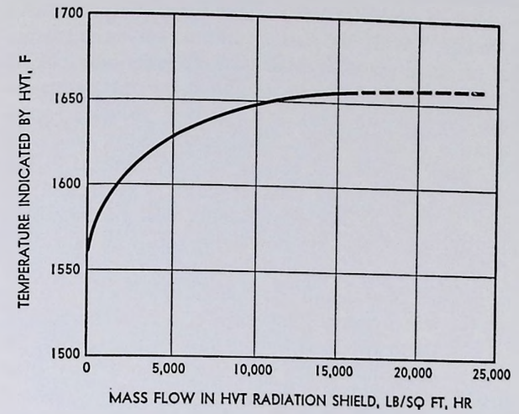


Fig. 20. *Gas mass flow over high-velocity-thermocouple junction greatly affects indicated temperature*

Recommended sizes for water-cooled high-velocity thermocouples are noted in Table 3. The fittings (preferably of brass, though malleable iron can be used) indicated in Fig. 19 for water outlet and inlet, aspirator suction, bushings, and terminal packed gland are of standard commercial size suitable for machining to telescope and seal the tubing.

By using multiple shields around the junction, all of which receive heat by convection induced by the high rate of gas flow, successive effects of the transfer of heat by radiation are so reduced that there is virtually no exchange of heat between junction and innermost shield. Measurements with the multiple-shield high-velocity thermocouple (MHVT) will closely approach the true temperature of the gas. Cross sections through single and multiple-shield high-velocity thermocouples developed for use in boiler testing are shown in Fig. 21. Because of the small flow areas which rapidly become clogged by ash, practical use of the MHVT is limited to clean-gas conditions.

The surfaces (water-cooled walls, superheater or boiler tube banks, etc.) surrounding the usual location of a gas traverse are cooler than the gases. Consequently, the readings from a bare unshielded thermocouple will indicate lower temperatures than those obtained with a single-shield HVT. For the same reason, a single-shield HVT will generally indicate lower values than an MHVT. Where traverses are taken in dust- or slag-laden gases, it is usually necessary to use the single-shield HVT and correct

TABLE 3

Recommended Sizes for Water-Cooled HVT

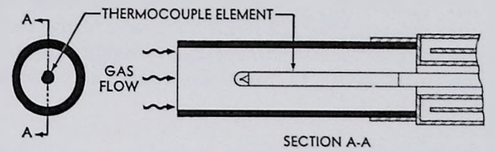
(See Fig. 19)

Probe Sizes, Nominal	Tubing Sizes, Stainless Steel or Brass	Water Flow Annulus Area, Sq In. Outside	Inside
1 in.°	1-in. OD, 18 BWG ¾-in. OD, 20 BWG ½-in. OD, 20 BWG	0.20	0.17
1¼ in.†	1¼-in. OD, 18 BWG 1-in. OD, 20 BWG ⅝-in. OD, 20 BWG	0.26	0.37
1½ in.‡	1½-in. OD, 18 BWG 1¼-in. OD, 20 BWG ⅞-in. OD, 20 BWG	0.91	0.79

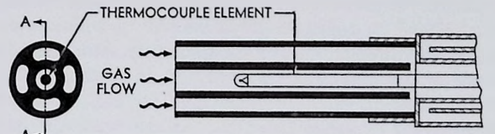
° Seldom used over 12 ft long, and at this length subject to overheating above 2200 F.

† Best for all-round use for lengths to 15 ft except in primary furnaces.

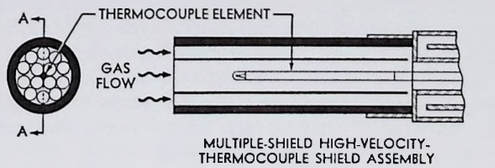
‡ Used for primary furnaces or lengths to 20 ft.



HIGH-VELOCITY-THERMOCOUPLE SHIELD ASSEMBLY



TYPE E BUREAU OF MINES HIGH-VELOCITY-THERMOCOUPLE SHIELD ASSEMBLY



MULTIPLE-SHIELD HIGH-VELOCITY-THERMOCOUPLE SHIELD ASSEMBLY

Fig. 21. *Shield assemblies for high-velocity thermocouple (HVT) and mulitple-shield high-velocity thermocouple (MHVT)*

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the readings by comparison with the results from an MHVT obtained under clean-gas conditions. The magnitude of these corrections is indicated in Fig. 18.

For temperatures exceeding 2200 F, noble-metal thermocouples are required, and it is important to protect the couple from contamination by the gases or entrained ash. The porcelain tube covering, shown in Fig. 17, provides some protection for the wires, especially when fouling occurs from molten slag at temperatures above 2400 F. When platinum couples are used in gases ranging from 2600 to 3000 F or higher, appreciable calibration drift may occur while taking measurements requiring only several minutes of exposure time. Under these conditions, the thermocouple elements should be checked periodically and corrections applied to the observed readings. When the magnitude of error reaches 40 to 60 F, the contaminated end of the couple should be removed and a new junction of the sound portion of the wire made. Calibration of the service couple may be checked by comparing its readings with those of a standard couple of the same material when both couples, bound together, are inserted in a specially arranged tubular resistance-wound electric furnace, taking care that the temperature gradient from the hot junction to the sound section of the service couple is the same as during its use.

Evaluation of Gas Temperatures

If the entire gas stream is at a uniform temperature, a single temperature reading at any point in the stream will be representative. However, such a condition is unusual, and it is generally necessary to make a traverse, taking readings at a number of points distributed across the gas stream. In order properly to evaluate the temperature readings, it is necessary to know, or to estimate, the direction and velocity of the gas flow at the various points in the gas stream where the temperatures are measured. Sometimes the gas velocity will be constant at all points, but there will be temperature stratification in the gas stream. More frequently, both gas temperature and gas velocity will be stratified. Occasionally there is recirculation, with some of the gas flowing through the traverse plane in one direction and some in the opposite direction.

The degree of accuracy required in the determination of the temperature of the gas at any location in the unit will depend upon the use to which the data are to be applied. For instance, an arithmetical average of the gas temperatures in the region of the soot blower may be entirely satisfactory as the basis for a study of the service life of a soot blower. On the other hand, if a heat balance is to be determined for a section of the unit, such as the furnace, the average temperature required must be the one which, in combination with the total gas weight, will give a true measure of the quantity of heat leaving the section. In this case a higher degree of accuracy is necessary, and the correct procedure (though often in practice it can be only approximated) for the determination of an average temperature is as follows:

Select an arbitrary imaginary surface (a surface in one plane is often, but not always, the most convenient) in which the temperature and gas-velocity traverses are to be made. Measure gas temperature, gas-flow direction, and gas velocity at a sufficient number of points of this surface for a comprehensive coverage. Divide the surface into a convenient number of small areas (which may be unequal). From the measurements taken, establish the gas temperature, gas-flow direction, and gas-flow velocity representative of each of the small areas. Next, calculate the vector component of gas velocity normal to the surface of each small area. If there is recirculation, some of these vector components will be negative. For each small area set down separately the product of temperature, area, and component of velocity, and the product of area and component of velocity. To obtain the true average temperature, divide the sum of the first set of products by the sum of the second set of products. Should the gas temperature vary over so wide a range that the assumption of a constant average value of specific heat for all parts of the gas stream is not warranted, gas enthalpy must be substituted for gas temperature. This procedure is based on a general mathematical theorem (Gauss's theorem) which finds wide application in various branches of physics.

When bulb-type or resistance pyrometers are installed permanently to measure gas temperature in a flue or duct, it is common practice to extend the bulb or resistance element from one side of the gas pass to the other in order to obtain a representative average temperature.

To insure against overheating of superheater or reheater tubes during a pressure-raising operation before steam is flowing through the tubes, it is satisfactory to use bare thermocouples temporarily installed in the gas stream immediately before the tubes involved and in sufficient number or proper location to insure indicating the highest gas temperature. The normal procedure is to remove these thermocouples after steam flow through the tubes is adequate to cool them. Gas temperature may then be increased. Future pressure-raising procedures can be based on this established rate or, if close control over the procedure is desired, gas temperatures can be monitored and recorded for each procedure. On units designed for remote operation, it is possible to retract and insert these thermocouples remotely, using a retractable soot blower carriage, as illustrated in Fig. 22.

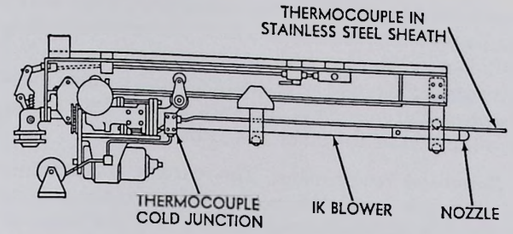


Fig. 22. Retractable thermocouple for measuring furnace exit gas temperatures as used for regulating firing rates during boiler pressure-raising

Insulation and Casing Temperature

Inner Surfaces. The thermocouple is particularly well suited for the measurement of temperatures at interfaces between successive layers of insulation, or at other points inside insulation. The temperature-sensitive hot junction may be buried out of sight, and the temperature may be read remotely. Furthermore, the hot junction and lead wires of the thermocouple are of small dimensions, which is important since it is imperative that the temperature-sensitive elements should be accurately located because of the steep temperature gradient through insulation.

Because of its low thermal conductivity, the rate of heat transfer through insulating material is low. This accentuates the effect of any heat conduction along the thermocouple lead wires, which is therefore a potential source of large error. To minimize or to eliminate this conduction error, the leads must extend for some distance in a constant temperature zone, and small-diameter wire should be used. If, for instance, in a composite wall the temperature of the interface between high-temperature block and 85% magnesia block is to be determined, the hot junction should be located between the layers of block, and the lead wires should be extended at least one foot between the layers before being brought to the outside (see Fig. 23). Since mechanical support is assured, fine wire (B&S 22 or 28 gage) may be used. The block surfaces should be grooved sufficiently to accommodate the wire without being separated by it.

Outer Surfaces. To avoid errors of considerable magnitude in the measurement of the temperature of the outer surface of insulation, thoughtful attention and careful workmanship are essential. Portable contact thermocouple instruments, designed to be pressed against a surface to measure its temperature, are unsatisfactory for this case and will not read correctly when applied to hot uncased insulation. Such instruments cool the surface at the point of contact, and the low rate of heat transfer through the insulating material prevents the restoration of normal surface temperature by heat flow from the surrounding areas.

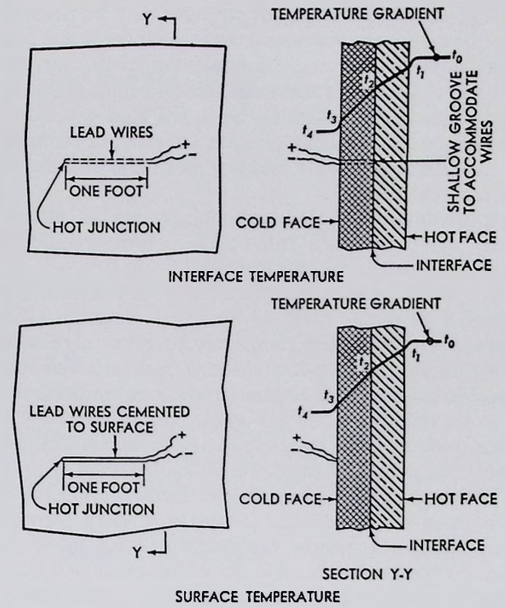


Fig. 23. Thermocouples for measuring surface or interface temperature of insulation

For satisfactory results, the means of attachment of the termocouple at the surface of the insulation must not appreciably alter the normal rate of heat transmission through the insulation and from the surface of the insulation to the surroundings. If serious errors from conduction are to be avoided, it is essential that the lead wires be maintained at the temperature of the surface. Fine wires can be attached more easily than heavy wires. If the insulation is plastic at the time of application, it may be possible to press the thermocouple junction and several feet of lead wires into the very surface of the insulation, with assurance that the thermocouple will adhere when the insulation hardens. If the insulation is already hard and dry, it may be possible to cement the junction and lead wires to the surface, using a minimum amount of cement (see Fig. 23). Fastening the wires to the surface with staples introduces conduction errors; covering the wires with scotch tape or friction tape changes the heat transfer characteristics of the surface and imposes an undesirable insulation layer between the wires and the ambient air. Each specific installation will tax the ingenuity and heat transfer knowledge of the test engineer.

Steel Casings. The temperatures of steel casings of boilers may be measured with greater ease and accuracy. Portable contact thermocouple instruments give good results, since lateral heat flow from adjoining metal areas quickly compensates for the small quantity of heat drawn by the instrument from the

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point of contact. Thermocouple wire may be peened into or fused onto the metal surface to form the hot junction, and leads may be cemented to the surface for several feet. It is most desirable to have the best possible thermal contact between the lead wires and the surface and the least possible covering of the wire or disturbance of heat transfer from the surface to the ambient air.

Approximate surface temperatures may be conveniently measured with the fusion paints or crayons mentioned on pages 6-2 and 25-17.

In power plants, thermometers are sometimes fastened to metal surfaces by wads of putty. This method of measuring temperatures gives a reliable approximate surface-temperature indication if the metal is massive and not far from room temperature. It is not recommended for boiler-casing temperature measurement and is completely unsatisfactory for insulation surface-temperature measurement.

"Through steel," such as metal ribs imbedded in insulation, studs or door frames extending through insulation, etc., causes considerable local upsets in surface temperatures, and its influence may spread laterally for some distance along a metal casing. These effects must be given careful consideration in the planning and interpretation of surface-temperature measurement.

Miscellaneous Applications

It is apparent that the several features of the thermocouple may be combined to advantage in numerous unique applications in the measurement of temperature. A few of these are mentioned below to illustrate the peculiar appropriateness of the thermocouple for special applications.

Coal-Pile Temperatures. Raw coal in storage undergoes slow oxidation, which may develop zones of elevated temperature and spontaneous combustion. As the coal storage pile is being laid, thermocouples may be placed at strategic points with lead wires extended to the border of the pile. Thus a check may be maintained of the trend of temperature to give a timely warning of dangerous conditions. Exploratory readings of temperature in a coal pile or bunker may be taken by probing with a thermocouple lance. For this service, the pipe-type thermocouple is sufficiently sensitive and may be made very rugged for driving into the pile or for other rough handling.

Stoker-Link Temperatures. The extreme temperature conditions to which the moving links of a chain-grate stoker are sometimes subjected cause decrease of stoker service life and increase in the cost of maintenance. By charting the temperature history of chain-grate links during their travel through the furnace, much useful information can be obtained about fuel-bed conditions and the proper adjustment of zoning dampers for different types of coal. A thermocouple may be peened to the end of a stoker link as it passes around the feed-end sprocket, and, by marking off the lead wire to correspond to linear distances of travel, temperature readings may be made at successive positions of the junction. As the chain moves around the sprocket, the lead wire is guided into the clearance spaces between adjacent links. When the cycle of chain travel is completed, the leads may be detached and the procedure may be repeated for check readings in several successive cycles until the couple is damaged.

Drum Temperatures. In determining thermal stresses in thick-shelled drums, it is possible to obtain actual inside surface temperatures by using permanent thermocouples welded to the inside shell and brought to the outside through a *Conax* fitting. Another method of obtaining similar information is to install "depth" thermocouples by drilling externally to within a fraction of an inch of the inside surface, the depth of this hole being determined by the amount of pressure stress on the remaining metal. Installations of such thermocouples are illustrated by Fig. 24. With permanent thermocouples either welded or peened to the outside surface of the drum in the area of the inside surface thermocouples, an accurate temperature gradient through the shell can be obtained by comparing inside and outside temperatures.

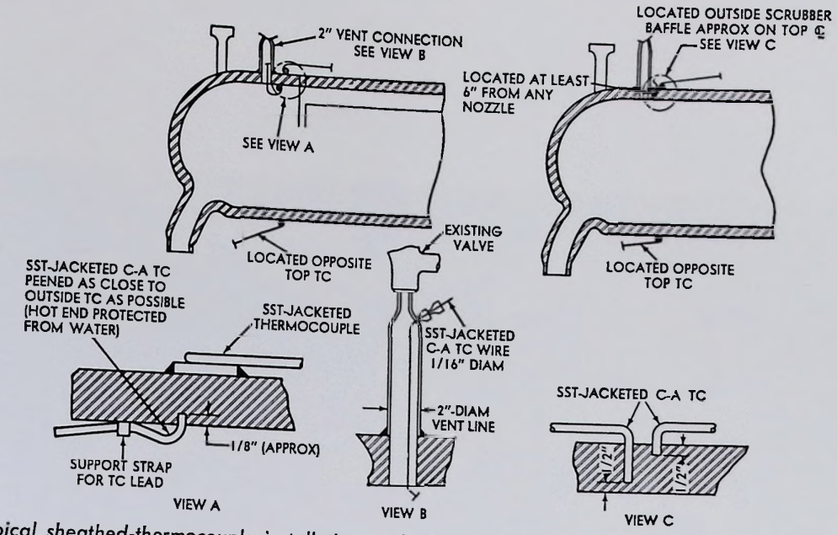
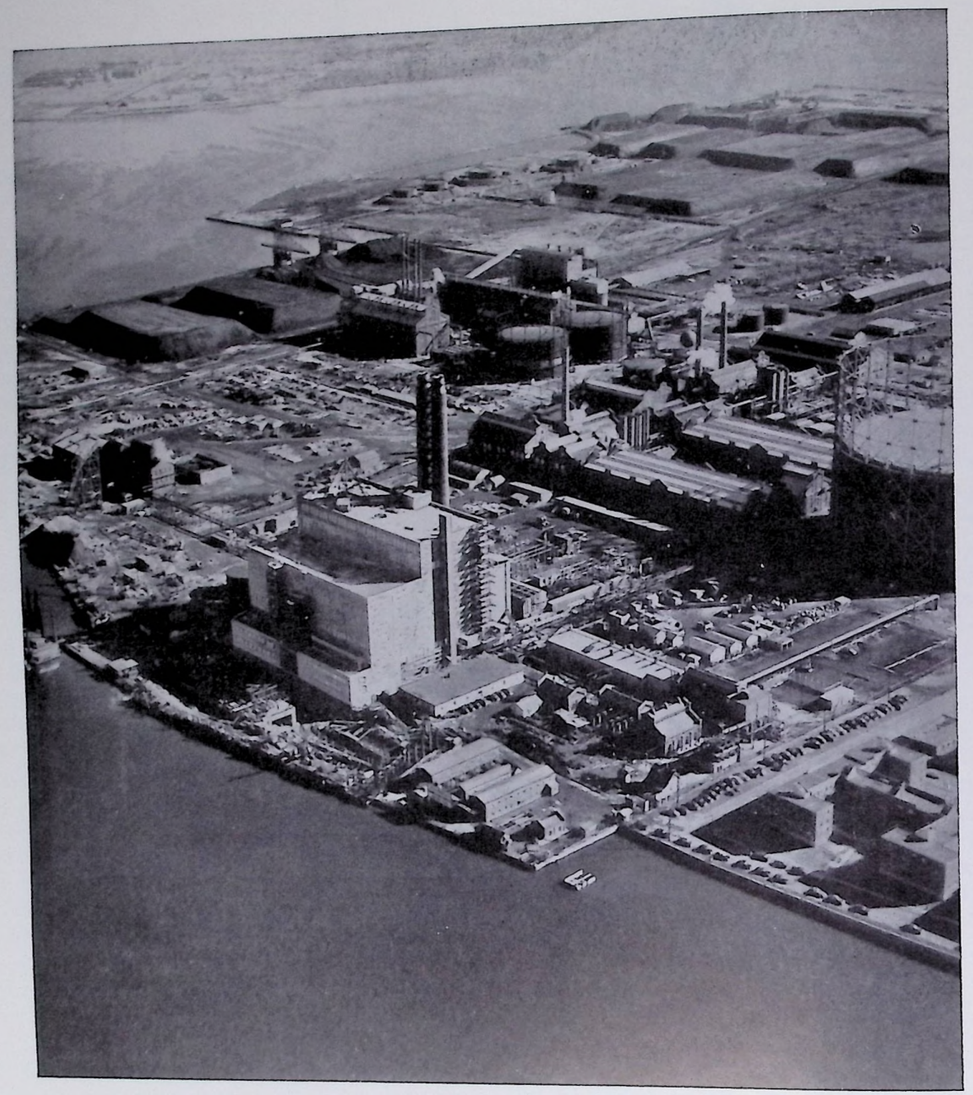


Fig. 24. *Typical sheathed-thermocouple installation on boiler steam drum for measuring inside shell temperatures*

The examples and instructions on temperature measurement, as given in this chapter, depict the severity of the problems of instrument application. Progress and development can be expected with the exercise of skill, the display of ingenuity, and the accumulation of experience.

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CALCULATIONS OF HEAT TRANSFER IN STEAM-GENERATING UNITS CALL FOR THE SKILLFUL USE OF THE TOOLS OF SCIENCE AND TECHNOLOGY

7

HEAT TRANSFER

HEAT transfer is a subject which deals with the transmission of thermal energy, the source of which may be the combustion of fuels, the fissioning of material, or the inevitable losses, such as pressure drop and friction, encountered in the transmission or the transformation of other types of energy. Heat transfer is demonstrated in countless aspects of everyday life, as in cooking, heating, refrigeration, and clothing.

While the fundamentals of heat transfer are simple, practical applications may become extremely complicated because of irregular configurations of equipment, simultaneous operation of several different modes of heat transfer, and changes in conditions from moment to moment. In the applications of heat transfer it is, therefore, necessary to combine basic scientific principles with empirical information derived from experience.

Application of heat transfer data to the design of boiler surface, economizers, superheaters, and air heaters is covered by specific examples in Chapter 11. Making due allowance for the great value of the theoretical and practical concepts of heat transfer, it is clear from these examples that the exercise of judgment based on experience is paramount for successful results.

Three Usual Modes of Heat Transfer

There are three usual modes of heat transfer—conduction, radiation, and convection. One or another or a combination of these modes enters into all the many varied phases of heat transfer. The three modes of heat transfer may be defined as follows:

Conduction. The transfer of heat from one part of a body to another part of the same body, or from one body to another in physical contact, without appreciable displacement of the particles of the body or bodies.

Radiation. The transfer of energy between bodies by electromagnetic vibrations without dependence upon the presence of matter in the intervening space. Thermal radiation is the transfer of heat by radiation. All matter radiates, and the transfer of heat is one important manifestation of this phenomenon. The intensity of the emission and the distribution among various wavelengths depend upon the temperature and the nature of the matter. When radiation impinges on a body, some of it may be reflected, some of it may be transmitted through the body, and the remainder will be absorbed. The portion absorbed is commonly converted into heat.

Convection. The transfer of heat within a fluid (gas or liquid) from one point to another by the mixing of one part with another through the movement of the fluid, or between one fluid and another by the mixing of the fluids, or between a fluid and a solid through relative motion between them.

(a) It is called natural or free convection where the movement of the fluid or fluids is caused solely by differences in density resulting from differences in temperature within the fluid or fluids.

(b) It is called forced convection where mechanical force, density difference from change of state of the fluid, or stack induction is used to give motion to the fluid.

Over-all or Combined Conductance

Actual transfer of heat energy usually involves more than one of these modes. When two modes of heat transfer occur simultaneously and independently, such as radiation and convection, the combined conductance, U, is the sum of the individual conductances U_r and U_c, which in effect are in parallel. When the heat flow paths are in series, the resistances, not the conductances, are additive. In all cases the potential force causing the transfer is a temperature difference.

The general equation for heat flow rate by these modes, singly or in combination, may be written in the form:

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$$q = U S \triangle t = S \triangle t / R \qquad (1)$$

where:

q = rate of heat flow, Btu/hr
U = over-all or combined conductance, Btu/sq ft, hr, F
S = surface involved in heat transfer, sq ft
$\triangle t$ = temp difference causing heat flow, F
$R = 1/U$ = combined resistance of the heat flow path, sq ft, hr, F/Btu

In the following, each mode of heat transfer is treated separately.

Heat Transfer by Conduction

If a flat plate is heated on one side and cooled on the other, heat will flow through the plate from the hot side to the cold side. To determine the rate of heat flow per unit of surface area of the plate it is necessary to know: 1) the temperature difference, $\triangle t$, between the two sides of the plate, F; 2) the thermal conductivity, k, of the plate material, Btu per sq ft, hr, F per in. thickness; and 3) the thickness, l, of the plate in inches. The rate of heat flow is directly proportional to 1) and 2) and inversely proportional to 3).

Metals are good conductors and have high conductivities; materials such as asbestos are poor conductors with low conductivities and are suitable for insulation. Typical approximate values of thermal conductivities of various materials at room temperature are given in Table 1.

TABLE 1
Thermal Conductivity, k
(Btu/sq ft, hr, F/in. thickness)

Material	k°
Silver	2880
Copper	2640
Carbon steel	312
Alloy steel, 18% Cr-8% Ni	108
First-quality firebrick	4.5
Insulating firebrick	0.8
85% Magnesia block	0.5

°At room temperature.

Thermal conductivities of most pure metals decrease with increase in temperature, while conductivities of alloys may either increase or decrease. Thermal conductivities of various steels are shown in Fig. 1. Thermal conductivities of insulating materials generally increase with increasing temperature, as shown in Fig. 19, Chapter 14.

In the case of a simple plate heated on one side and cooled on the other, the distribution of temperature will be as shown in Fig. 2, heat flowing from the higher temperature to the lower. If the thermal conductivity of the material does not change with change in temperature, the temperature gradient will be a straight line; if the conductivity increases with increase in temperature, the temperature gradient will be convex; if the conductivity decreases with increase in temperature, the gradient will be concave. In most cases it is satisfactory to assume a constant conductivity at the average temperature of the material.

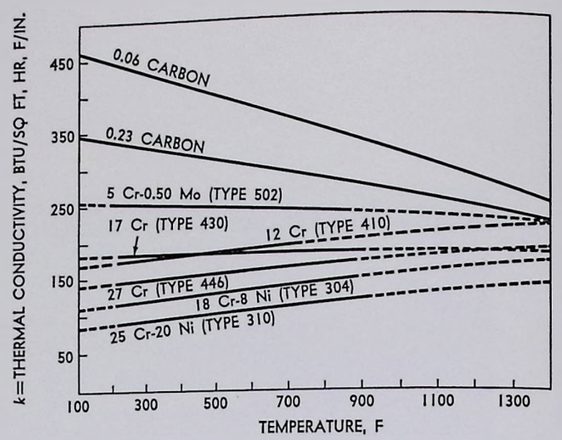
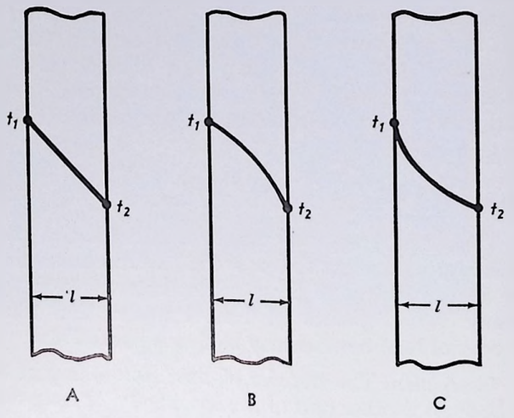


Fig. 1. *Thermal conductivity, k, of some commonly used steels*

A — k IS CONSTANT
B — k INCREASES WITH INCREASE IN TEMPERATURE
C — k DECREASES WITH INCREASE IN TEMPERATURE



$$q = \frac{kS\,(t_1 - t_2)}{l}$$

Fig. 2. *Diagrammatic temperature-thickness relationship for different thermal conductivities, k*

In the case of heat transfer by conduction only, the general equation (1) is expressed as:

$$q = U_d S \triangle t = \frac{S \triangle t}{R} = \frac{kS \triangle t}{l} \qquad (2)$$

where:

U_d = combined conduction conductance, Btu/sq ft, hr, F
k = thermal conductivity of the material, Btu/sq ft, hr, F/in.
l = length of heat flow path (thickness of material), in.
R = combined resistance of heat flow path as in equation (1) $= 1/U_d = R_1 + R_2 + R_3$, etc., $= l_1/k_1 + l_2/k_2 + l_3/k_3$, etc., for a series of resistances

In the simple example, Fig. 2, for a flat steel plate ¼ in. thick, 1 sq ft in area, and $t_1 - t_2 = \triangle t = 25$ F, the combined conductance, U_d, is the single conductance through the steel plate and is equal to the conductivity, k, of steel (312, from Table 1) divided by the thickness of the plate. Therefore in equation (2):

$$q = \frac{312 \times 1 \times 25}{0.25} = 31{,}200 \text{ Btu/hr}$$

Equation (1) is used for the composite example, Fig. 3, which may be the wall of a flue conveying hot gases. In this case U, the combined conductance, is not a single conductance but is the combination of incremental conductances in series through the fluid films and the various layers of materials, and R, the combined resistance, is equal to the sum of the incremental resistances, thus:

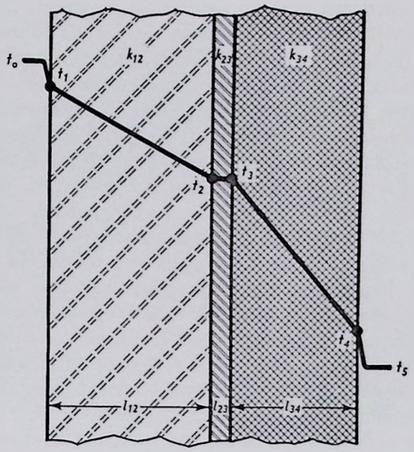
$$R = R_{01} + R_{12} + R_{23} + R_{34} + R_{45}$$

$$= R_{01} + \frac{l_{12}}{k_{12}} + \frac{l_{23}}{k_{23}} + \frac{l_{34}}{k_{34}} + R_{45}$$

For this example the assigned numerical values are:

$S = 600$ sq ft, wall surface
$t_o = 1080$ F, temp of hot gases
$t_5 = 80$ F, temp of room air
$\triangle t = t_o - t_5 = 1000$ F, temp difference
$k_{12} = 1.08$, for insulating firebrick
$k_{23} = 300$, for steel (trial value)
$k_{34} = 0.5$, for insulation
$l_{12}, l_{23}, l_{34} = 4$ in., ¼ in., and 3 in., respective thicknesses
$U_{01}, U_{45} = 5.0, 2.0$, respective film conductances
$R_{01}, R_{45} = 1/5, 1/2$, respective film resistances

The film conductances, U_{01} and U_{45}, depend upon a variety of factors, chiefly velocity or mass flow of the fluids giving up and receiving heat and the temperatures of the fluids (see below, also Chapter 11).



$$q = \frac{S\,(t_o - t_5)}{\frac{1}{U_{01}} + \frac{l_{12}}{k_{12}} + \frac{l_{23}}{k_{23}} + \frac{l_{34}}{k_{34}} + \frac{1}{U_{45}}}$$

Fig. 3. *Temperature distribution in composite wall (fluid films included)*

Using the above assigned values in equation (1) for the composite wall illustrated in Fig. 3:

$$q = \frac{600 \times 1000}{\frac{1}{5} + \frac{4}{1.08} + \frac{0.25}{300} + \frac{3}{0.5} + \frac{1}{2}}$$

$$= 57{,}600 \text{ Btu/hr}$$

$$\frac{q}{S} = \frac{57{,}600}{600} = 96 \text{ Btu/sq ft, hr}$$

Since the heat flow rate through each resistance is the same and in series, the face and interface temperatures may be determined:

In equation (1), $q = \frac{S \triangle t}{R}$ and $\triangle t = \frac{q}{S} \times R$

$t_o - t_1 = \frac{q}{S} \times R_{01}$; $1080 - t_1 = 96 \times \frac{1}{5}$

$t_1 = 1061$ F

$t_1 - t_2 = \frac{q}{S} \times \frac{l_{12}}{k_{12}}$; $1061 - t_2 = 96 \times \frac{4}{1.08}$

$t_2 = 705$ F

$t_2 - t_3 = \frac{q}{S} \times \frac{l_{23}}{k_{23}}$; $705 - t_3 = 96 \times \frac{0.25}{300}$

$t_3 = 705$ F

$t_3 - t_4 = \frac{q}{S} \times \frac{l_{34}}{k_{34}}$; $705 - t_4 = 96 \times \frac{3}{0.5}$

$t_4 = 129$ F

Equation (2) can be developed mathematically

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to take account of heat flow in all directions instead of only one and can be further developed to take account of heat flow where the temperatures not only vary from point to point but also vary with time. However, the fundamental principles outlined above form the basis of the whole theory of heat conduction, either in the steady state (temperatures do not vary with time, and heat flow is constant) or in the unsteady state (temperatures vary with time, and heat flow is variable). Heat conduction and electrical conduction are analogous and follow the same laws, temperature difference corresponding to voltage difference and heat flow rate to electrical current. This important analogy permits the use of electrical models and combinations of electrical circuits for the solution of heat conduction problems that are too complex for mathematical analysis.

The expression k/l in the equations above is characteristic of any area of the plate, and it is found useful to give this characteristic a special name, conductance, which has the dimensions Btu/sq ft, hr, F. The reciprocal of conductance is resistance, R = sq ft, hr, F/Btu = l/k. For any case in which heat flows through successive flat layers of material, the reciprocal of the over-all conductance equals the sum of the reciprocals of the individual conductances, and the total resistance equals the sum of the individual resistances. If the successive layers of material do not make good thermal contact with each other, there will be interface resistances from air or oxide films, etc. These resistances may be neglected in composite walls of insulating materials but become important and must be included in calculations if the resistances of the layers are small in comparison with the interface resistances.

When heat is conducted radially through a cylindrical wall, as in heat flow through the wall of a steam line from the inside to the outside of the pipe (Fig. 4), the flat-plate equation must be modified, since the heat-flow surface area, S, is no longer a constant but increases as the distance from the center of the pipe increases. If the outside surface area of the pipe is used for the value of S, then it can be shown that the thickness of the pipe wall, corresponding to length l in the flat-plate equation, must be replaced by the equivalent (thickness) length, l_e, given by the equation:

$$l_e = 0.5\, d_o \log_e (d_o/d_i) \qquad (3)$$

where:

l_e = equivalent length, in.
d_o = outside pipe diameter, in.
d_i = inside pipe diameter, in.

The equation for heat flow through a cylindrical wall thus becomes:

$$q = \frac{kS\,(t_1 - t_2)}{l_e} = \frac{2kS\,(t_1 - t_2)}{d_o \log_e (d_o/d_i)} \qquad (4)$$

If the heat flows through two successive cylindrical walls, as in an insulated steam pipe, and if the film conductances (see below under convection) U_{oi} of the steam and U_{si} of the air outside the insulation (Fig. 5) are included, the equation becomes:

$$q = \frac{S\,(t_o - t_s)}{\frac{d_i}{d_s}\frac{1}{U_{oi}} + \frac{d_s}{2k_{12}} \log_e \frac{d_2}{d_1} + \frac{d_s}{2k_{23}} \log_e \frac{d_3}{d_2} + \frac{1}{U_{si}}} \qquad (5)$$

where:

d_s = outside diameter of insulation, in.
S = outside surface area of insulation, sq ft

Many problems of heat transfer in the power field involve conduction through plane or cylindrical walls and can be treated by the methods outlined. For other geometrical shapes, such as thick-wall rectangles, ribbed construction, etc., special treatment is required.

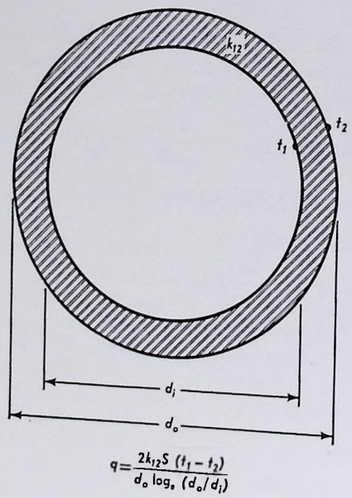


Fig. 4. Heat flow through cylindrical wall (see equation)

Conduction in Fluids

Conduction in fluids (liquids and gases) follows the same laws as conduction through solids. In engineering practice, heat transfer by conduction through fluids is small in comparison with convection. However, the thermal conductivities of fluids and solids enter into convection heat transfer. The thermal conductivity of water ranges from about 3.8 Btu/sq ft, hr, F/in. at 32 F to about 4.8 at 260 F. Most of the other nonmetallic liquids have conductivities between 0.60 and 1.80 Btu/sq ft, hr, F/in. The thermal conductivities of gases are independent of pressure* but increase with increasing temperature, the higher values being associated with the lighter and smaller molecules (see Table 2). The relatively high conductivity of hydrogen is utilized in hydrogen-cooled generators.

*At low pressures.

TABLE 2
Thermal Conductivity, k
(Btu/sq ft, hr, F/in. thickness)

Water and Steam Vapor

Temp, F	Sat. Liquid, k	Sat. Vapor, k
35	3.840	0.108
100	4.368	0.132
200	4.728	0.156
300	4.740	0.204
400	4.584	0.300
500	4.176	0.420
600	3.504	0.576
650	2.940	0.744
700	1.8	1.2
705.4	1.5	1.5

Gases

Temp, F	Air, k	CO_2, k	O_2, k	N_2, k	H_2, k
0	0.168	0.096	0.156	0.168	1.080
500	0.300	0.252	0.312	0.264	2.004
1000	0.408	0.384	0.444	0.336	2.724
1500	0.480	0.516	0.564	0.408	3.360
2000	0.564	0.624	0.672	0.468	3.924
2500	0.636	0.720	0.792	0.528	4.464
3000	0.696	0.804	0.912	—	5.004

Flue Gases

Temp, F	*Nat Gas, k	*Fuel Oil, k	†Coal, k
0	—	—	—
500	0.264	0.264	0.264
1000	0.360	0.348	0.348
1500	0.444	0.432	0.432
2000	0.528	0.516	0.516
2500	0.612	0.588	0.600

*For 115% total air.
†For 120% total air.

Unsteady-State Conduction

Thus far the discussion has been limited to steady-state conduction, where temperatures vary from point to point but do not change with time. Unsteady-state conduction is involved in heating or cooling processes where temperatures change with time, as in heating of billets, regenerative heaters, raising of pressure in a boiler, quenching of steel, and warming up and cooling down of steam lines and turbines. With the introduction of time as an additional variable, the problems of conduction become much more complicated. In this difficult field, electrical analogy models are particularly applicable.

All unsteady-state conduction involves heat storage. For instance, in heating up a furnace enough heat must be supplied to bring the walls to the operating

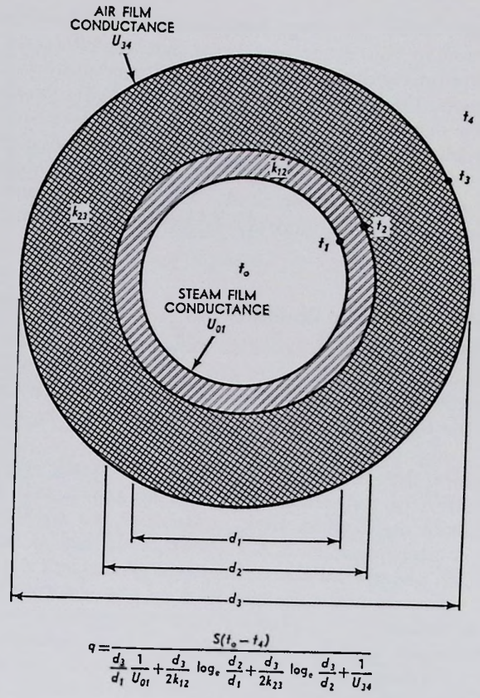


Fig. 5. Heat flow through composite cylindrical wall (pipe and insulation) with fluid films included (see equation)

temperature and also to make up for the steady-state losses of normal operation. In large power boilers that run for long periods of time, heat storage in the walls and boiler metal is an insignificant fraction of the total heat input. In small boilers with refractory settings that are operated only a part of each day or in heating furnaces that are frequently heated and cooled in batch-process work, heat storage in the walls during start-up may be a considerable proportion of the total heat input. Often in equipment of this type, the use of insulating firebrick of low heat-storage capacity instead of standard firebrick, even though the initial expense may be somewhat greater, is warranted by substantial savings in fuel cost and in the time required for start-up.

Unsteady-state conduction is important in the equalization of temperatures in boiler drums during pressure-raising and -reducing periods. When pressure is raised in a boiler, the water temperature increases. The inner surface of the steam drum is heated by contact with the water below the waterline and by the condensation of steam above the waterline. The temperatures of the inside and outside of the drum are raised by unsteady-state conduction. Heat flow takes time, however, and it is necessary to restrict the rate of pressure rise in thick

boiler drums to keep from exceeding allowable stresses in the drum metal, caused by differential expansion resulting from excessive temperature differences. During pressure-reducing periods the inside of the drum below the waterline is cooled by water. The top of the drum is cooled only by radiation to the water and by unsteady-state conduction through the metal walls of the drum. The rate of pressure reduction must therefore also be limited (see Chapter 21).

Heat Transfer by Radiation

The fractions of radiation reflected, transmitted, and absorbed by a surface are known respectively as the reflectivity, ρ, transmissivity, τ, and absorptivity, α, of the body. The sum of these fractions equals one:

$$\rho + \tau + \alpha = 1$$

Bodies that are good absorbers are equally good emitters of radiation, and it can be shown that at thermal equilibrium their emissivities are equal to their absorptivities. A blackbody is defined as one which absorbs all the radiant energy incident upon it, reflecting and transmitting none; the absorptivity and emissivity of a blackbody are each equal to one. In radiant heat transfer parlance, "black" signifies a body which neither reflects nor transmits radiant energy and has unit absorptivity and emissivity. It does not necessarily mean that the body appears black to the eye. Snow, for instance, absorbs only a few per cent of visible light falling upon it, but to the longer wavelengths (the bulk of thermal radiation) snow is almost a blackbody. A blackbody at a temperature of 2000 F will glow brightly, since part of its radiation is in the visible range. No actual bodies are completely black, but a hole through the wall of a large enclosure approximates blackbody conditions, since radiation entering the hole will undergo multiple reflections and absorptions inside the enclosure and will, so to speak, be unable to find its way out.

The radiation emitted by a blackbody depends upon its surface area and temperature and follows a relationship known as the Stefan-Boltzmann law:

$$q = \sigma S T^4 \tag{6}$$

where:

q = rate of heat flow, Btu/hr
σ = Stefan-Boltzmann constant, 1.71×10^{-9} Btu/sq ft, hr, T^4
S = surface area of body, sq ft
T = absolute temperature, R = F + 460

The above equation is based on the emissivity (equal to 1) of a blackbody and must be modified to account for the emissivity of an actual body, as follows:

$$q = \sigma \epsilon S T^4 \tag{7}$$

where ϵ, emissivity, is the ratio of the energies radiated, respectively, by an actual body and a blackbody at the same absolute temperature.

The radiation from a blackbody extends over the whole range of wavelengths, although the bulk of it is concentrated in a moderately narrow band. The wavelength at which the maximum radiation intensity occurs is inversely proportional to the absolute temperature of the body. Fortunately many commercial surfaces, particularly at high temperatures, have emissivities of 0.80 to 0.95 and behave very much like blackbodies. Typical emissivity values are noted in Table 3.

TABLE 3

°Normal Emissivities, ϵ, for Various Surfaces

Material	Emissivity, ϵ	Temp, F	Description
Aluminum	0.09	212	Commercial sheet
Aluminum oxide	0.63-0.42	530-930	
Aluminum paint	0.27-0.67	212	Varying age and Al content
Brass	0.22	120-660	Dull plate
Copper	0.16-0.13	1970-2330	Molten
Copper	0.023	242	Polished
Cuprous oxide	0.66-0.54	1470-2010	
Iron	0.21	392	Polished, cast
Iron	0.55-0.60	1650-1900	Smooth sheet
Iron	0.24	68	Fresh emeried
Iron oxide	0.85-0.89	930-2190	
Steel	0.79	390-1110	Oxidized at 1100 F
Steel	0.66	70	Rolled sheet
Steel	0.28	2910-3270	Molten
Steel (Cr-Ni)	0.44-0.36	420-914	"18-8" Rough, after heating
Steel (Cr-Ni)	0.90-0.97	420-980	"25-20" Oxidized in service
Brick, red	0.93	70	Rough
Brick, fireclay	0.75	1832	
Carbon, lampblack	0.945	100-700	0.003" or thicker
Water	0.95-0.963	32-212	

°From Hottel.

The emission of radiant energy described above depends solely upon the temperature and emissivity of the surface of the body and is independent of any other heat exchange, whether by conduction, radiation, or convection, which may be occurring at the same time.

When two blackbody surfaces are arranged so that all the radiant energy emitted by one falls upon the other (as, for example, two infinite parallel planes), the net rate of heat exchange between the hot surface 1 and the cold surface 2 is:

$$q_{12} = \sigma S (T_1^4 - T_2^4) \tag{8}$$

where:

q_{12} = Btu/hr
$\sigma = 1.71 \times 10^{-9}$ Btu/sq ft, hr, T^4
S = surface area of one plane, sq ft
T_1, T_2 = absolute temperatures of surfaces 1 and 2, R = F + 460

If the two blackbodies are arranged so that all the radiation emitted by one does not fall on the other, it is necessary to introduce an angle factor, F_{12} (less than one), into the equation, which then becomes:

$$q_{12} = F_{12} \sigma S_1 (T_1^4 - T_2^4) \tag{8a}$$

The angle factor depends upon the geometry of the situation and the body used to define S_1.

If the emissivity of a surface is less than one but is independent of wavelength, the surface is termed a nonselective radiator, or "gray" surface; if the emissivity depends upon wavelength, the surface is termed a selective radiator. Exact allowance for the departure of surfaces from black or ideal radiating characteristics is, in general, too complicated for engineering use. However, if the assumption that all surfaces are gray is permitted, a simpler treatment is possible. This involves introducing an emissivity factor, F_ϵ, which depends upon the geometry, the emissivities involved, and the surface area selected for use as S in the equations. Two special cases are of interest:

1) Where one body is completely surrounded by a much larger body (e.g., a steam pipe in a large room), the general equation for the net rate of heat exchange between one surface and another becomes:

$$q_{12} = \sigma \epsilon_1 S_1 (T_1^4 - T_2^4) \tag{9}$$

where:

q_{12} = Btu/hr
$\sigma = 1.71 \times 10^{-9}$ Btu/sq ft, hr, T^4
ϵ_1 = emissivity of smaller surface 1 = F_ϵ
S_1 = area of smaller surface 1, sq ft
T_1 = absolute temperature of smaller surface 1, R = F + 460
T_2 = absolute temperature of larger surface 2, R = F + 460

2) For the heat exchange between two small gray bodies at considerable distance from one another, the general equation for the net rate of heat exchange between one surface and another becomes:

$$q_{12} = \sigma \epsilon_1 \epsilon_2 (\omega/2\pi) S_1 (T_1^4 - T_2^4) \tag{10}$$

where:

q_{12} = Btu/hr
$\sigma = 1.71 \times 10^{-9}$ Btu/sq ft, hr, T^4
ϵ_1, ϵ_2 = emissivities of surfaces 1 and 2
$\epsilon_1 \epsilon_2$ = emissivity factor, F_ϵ
S_1 = area of surface 1, sq ft
ω = solid angle intercepted by body 2 as seen from body 1, steradians°
$\pi = 3.1416$
$\omega/2\pi$ = angle factor, F_{12}
T_1, T_2 = absolute temperatures of surfaces 1 and 2, R = F + 460

°2π steradians comprise a hemispherical solid angle about a point.

Since the solid angle intercepted by body 2, at a distance from body 1, is something less than 2π steradians, the factor $\omega/2\pi$ is less than unity, and the effective surface area of body 1 becomes $(\omega/2\pi)S_1$.

A complicated case of intersolid radiation is encountered in flames made luminous by particles, such as pulverized coal, averaging about 0.001 in. in diameter, or by soot formed by the thermal decomposition of hydrocarbons, having particle diameters of about 0.00001 inch. The treatment of radiation from luminous flames is beyond the scope of this chapter.

Radiation From and To Gases

Although many gases, such as oxygen and nitrogen, do not appreciably absorb or emit radiation, some others, such as water vapor, carbon dioxide, sulfur dioxide, and carbon monoxide, both absorb and emit. Water vapor and carbon dioxide are important because they are found in considerable quantities in the products of combustion of hydrocarbon fuels. These gases are selective radiators, which emit and absorb radiation only in certain wavelength bands that lie outside of the visible range. Their radiation, therefore, cannot be seen and is termed nonluminous gas radiation. Whereas solid radiation is a surface phenomenon, a gas both radiates and absorbs (within its absorption bands) at every point throughout the gas body. Furthermore, the emissivity of a gas changes with temperature, and the presence of one radiating gas has an effect on the radiating characteristics of another radiating gas with which it is mixed. The amount of radiant energy emitted by a radiating gaseous mixture depends upon the temperature of the mixture, the partial pressures of the gaseous radiating constituents, and the shape and dimensions of the gas body. The fraction of radiant energy emitted by the surroundings, which is absorbed by the gas, depends upon the temperature and surface of the surroundings as well as upon the foregoing.

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Heat Transfer by Convection

Heat transfer by convection between a fluid (gas or liquid) and a solid, such as a boiler tube, is expressed in the basic equation:

$$q_c = U_c S \Delta t \qquad (11)$$

where:

q_c = rate of heat flow by convection, Btu/hr
U_c = convection film conductance, Btu/sq ft, hr, F
S = heat transfer surface normal to heat flow, sq ft
$\Delta t = (t - t_s)$ or $(t_s - t)$, depending upon direction of transfer
t = fluid temperature, F
t_s = surface temperature, F

From this equation it is evident that heat flux density, the quantity of heat (Btu) transferred per sq ft, hr, depends upon the film conductance and the difference between the surface temperature and the fluid temperature. For the same difference in temperatures, heat flux densities are, therefore, proportional to the film conductances. In general, film conductance must be determined experimentally by painstaking test procedures involving precise measurements of surface temperatures and fluid temperatures. For the effect of film conductance, see example above of composite heat flow path (Fig. 3) and also Chapter 11.

Free Convection

The quantity of heat transfer by convection is vitally affected by the type of motion within a fluid. A fluid at rest, exposed to a heated surface, will be at a higher temperature adjacent to the surface than elsewhere. The difference in specific weights of the particles of the fluid because of this difference in temperature will cause the fluid to circulate, thus carrying heat from one place to another. Heat transfer in this manner is known as free or natural convection, as differentiated from forced convection, where outside mechanical force, change of state of the fluid, or stack induction is used to circulate the fluid. The relationships of the numerous variables affecting the free-convection heat transfer conductance for different media are fully discussed in texts on the subject. In the power field, however, air and flue gases at atmospheric pressure are the most important natural-convection heat transfer media, and in this case the following equation is applicable:

$$U_{fc} = C(\Delta t)^{1/3} \qquad (12)$$

where:

U_{fc} = free-convection film conductance, Btu/sq ft, hr, F
C = coefficient, characteristic of shape and position of heat transfer surface
Δt = difference in temperature between the surface and the air or gas, F

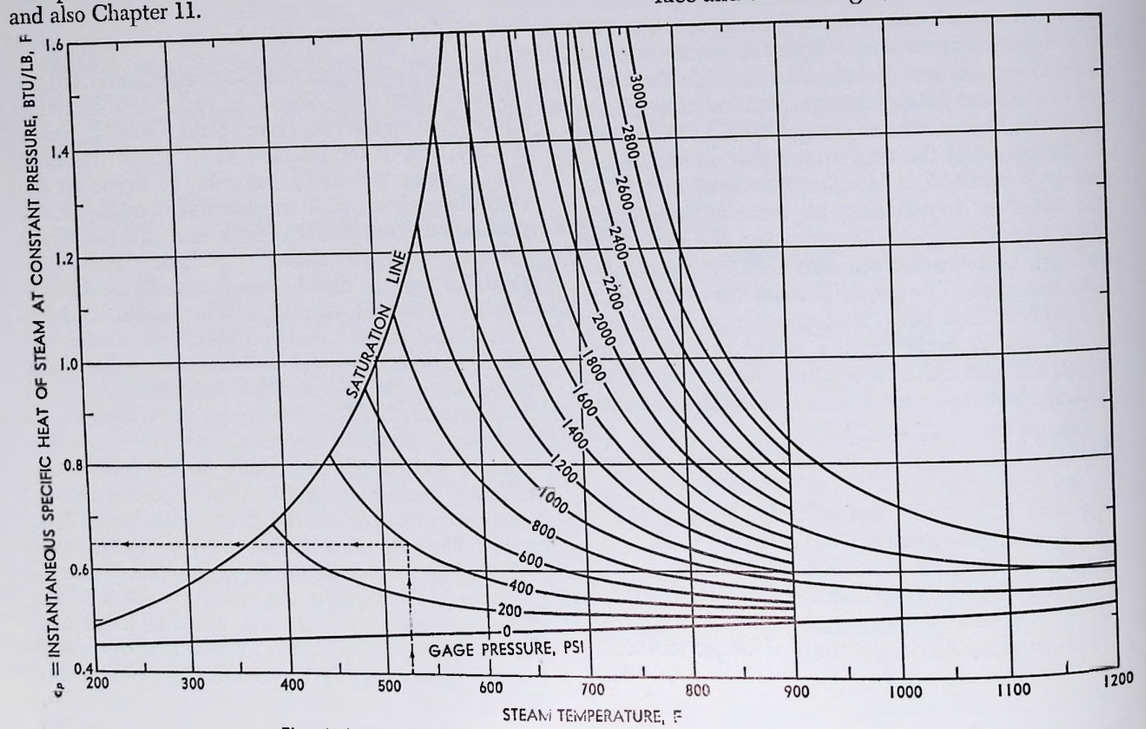


Fig. 6. Instantaneous specific heat of steam at constant pressure

Values of C for various surfaces at different attitudes are listed below:

Horizontal plates facing upward	0.22
Vertical plates more than 1 ft high	0.19
Vertical pipes more than 1 ft high	0.19
Horizontal pipes	0.18

The flow resulting from the boiling of a liquid at a heating surface should be differentiated from free convection. Whereas the free-convection film conductance for water is in the order of 25-50 Btu/sq ft, hr, F, the heat transfer conductance for boiling water may vary between very wide limits (see below and Chapters 21 and 29), depending upon the rate of heat input and other factors.

Forced Convection

Forced-convection heat transfer depends upon mechanically imposed fluid flow or equivalent. The mechanisms and types of fluid flow are described in Chapter 8. In subsections a) through e) which follow, radiation which may be occurring simultaneously is intentionally omitted.

a) Streamline Flow Inside Pipes or Tubes

In streamline or laminar flow, the elements or layers of the fluid flow parallel to the axis of the channel, with no appreciable lateral disturbance. For heating or cooling viscous liquids in the streamline-flow region inside horizontal or vertical tubes, the film conductance between fluid and surface, based on the arithmetic mean temperature difference, can be determined by the following empirical equation from McAdams:

$$U_{oi} = 1.86 \frac{k}{D_i} \left\{ \frac{\mu}{\mu_s} \right\}^{0.14} \left\{ \frac{GD_i}{\mu} \frac{c_p \mu}{k} \frac{D_i}{L} \right\}^{1/3} \qquad (13)$$

where:

U_{oi} = convection film conductance in long flow, Btu/sq ft, hr, F
D_i = inside tube diameter, ft
k = conductivity of fluid within tube, Btu/sq ft, hr, F/ft
G = mass velocity, or mass flow, of fluid within tube, lb/hr, sq ft of cross-sectional area
c_p = specific heat at constant pressure, Btu/lb, F (see Fig. 6 for steam)
L = heated length of straight tube, ft
μ = absolute fluid viscosity at bulk temperature, lb/ft, hr
μ_s = absolute fluid viscosity at surface temperature, lb/ft, hr

In the case of low-viscosity fluids, such as water and gases, a more complex equation is required to allow for the effects of natural convection at the heat transfer surface. This refinement is of little interest in industrial practice, since it is generally impractical to use water and gases in streamline flow.

b) Turbulent Flow

In turbulent flow, the fluid elements move radially as well as axially. The combination of radial and axial components of flow sets up an eddy motion, which increases the activity within the body of the fluid. Consequently, there is a considerable increase in the convection heat transfer as compared with streamline flow.

Studies of the velocity gradient across a fluid indicate that in turbulent flow the velocity at the heat transfer surface is zero. In the zone immediately adjacent to the surface, known as the sublaminar layer, the flow is streamline, and the heat leaving or approaching the surface travels by conduction. In the next zone, known as the buffer layer, where the motion is a mixture of streamline and turbulent flow, the heat is transferred partly by convection and partly by conduction. Heat is transferred mainly by convection in the bulk of the stream cross section, known as the turbulent zone, or turbulent core in the case of flow inside pipes or tubes.

The streamline flow in the sublaminar layer and the streamline component in the buffer layer act as a barrier to the convection transfer of heat to or from the surface. This barrier may be likened to a film, the thickness of which depends, to a certain extent, upon the velocity of the fluid. By increasing the velocity, a part of this film can be swept away, thus decreasing its thickness and reducing the resistance to heat transfer by conduction.

c) Reynolds Number and Others

It was only after the important variables were arranged in a number of so-called dimensionless groups that the data from the earlier research of various investigators in convection heat transfer could be plotted and compared in an orderly manner. In correlating and using forced-convection heat transfer information, the best method is to compare fluid flow, temperature, and heat transfer characteristics for geometrically similar systems. Geometrical similarity is a familiar concept, frequently encountered in the scale model, where size is changed but relative physical proportions remain constant.

A familiar example of dimensionless groups of factors is the Reynolds number, $\rho VD/\mu$, which is used as a criterion of fluid-flow similarity in pipes flowing full. This number represents the ratio of the inertial forces, F_I, to the viscous forces, F_μ, as shown below:

$$F_I = ma = \rho L^3 V^2 / L = \rho L^2 V^2$$
$$F_\mu = \mu AV/L = \mu L^2 V/L = \mu LV$$
$$F_I/F_\mu = \rho L^2 V^2 / \mu LV = \rho LV/\mu$$

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where:

ρ = density of fluid, lb (mass)/cu ft
V = mean velocity of fluid, ft/hr
μ = viscosity of fluid, lb (mass)/ft, hr
L = characteristic lineal dimension, ft

For fluid flowing full in closed conduits, the characteristic lineal dimension is the internal diameter for pipe flow or the equivalent diameter for conduits of noncircular cross section. It can be demonstrated mathematically that, for dynamically similar fluid motion in geometrically similar systems of different size, $\rho_1 V_1 D_1/\mu_1$ for one system is equal to $\rho_2 V_2 D_2/\mu_2$ for another system. The flow pattern in geometrically similar systems is therefore determined by the variables ρ, V, D, and μ.

In the derivation of the dimensionless expression $\rho VD/\mu$, it is assumed that the fluid is a continuous medium filling its conduit and that the gravitational and elastic forces are negligible compared with the inertial and viscous forces.

Where the velocity of the fluid is high and the viscosity is low, as in water flowing rapidly through a pipe (high Reynolds number), the inertial forces are predominant, and the flow will be turbulent.

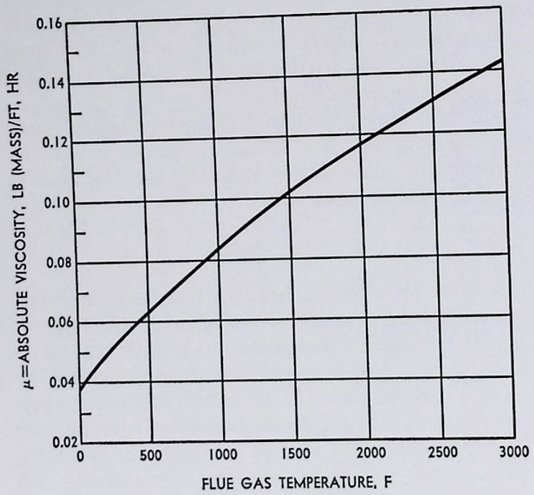


Fig. 7. Flue gas viscosity for various temperatures

Where the velocity is low and the viscosity is high, as in the flow of heavy oil in a pipe (low Reynolds number), the viscous forces predominate, and the tendency is for the flow to be laminar.

Another dimensionless group of factors known as the Prandtl number, $c_p\mu/k$, serves as a criterion of temperature gradient similarity. This group can be considered as specific heat, c_p, divided by the ratio of thermal conductivity to viscosity, k/μ. The numerator, c_p, is a measure of the temperature rise of a given mass of fluid for a given heat energy input. The denominator, k/μ, is a measure of the ease of heat transmission through the fluid, since high conductivity or low viscosity stimulates heat transmission.

The dimensionless group known as the Nusselt number, $U_o D/k$, is a criterion of heat transfer similarity and can be considered as a ratio of the size factor, D, to the thickness of the boundary layer of the fluid, expressed as k/U_o.

The dimensionless group known as the Stanton number, $U_o/\rho V c_p$, is the ratio of the conductance to the product of the mass flow and the specific heat at constant pressure, which is the ratio of the heat absorbed to that available for absorption. It is used in the following expression as a designating function.

The Nusselt number can be expressed as the product of the Stanton, Reynolds, and Prandtl numbers:

$$\frac{U_o D}{k} = \left\{\frac{U_o}{\rho V c_p}\right\}\left\{\frac{\rho VD}{\mu}\right\}\left\{\frac{c_p\mu}{k}\right\} \qquad (14)$$

where:

$\frac{U_o D}{k}$ = Nusselt number, dimensionless

$\frac{U_o}{\rho V c_p} = \frac{U_o}{G c_p}$ = Stanton number, dimensionless

$\frac{\rho VD}{\mu} = \frac{GD}{\mu}$ = Reynolds number, dimensionless

$\frac{c_p\mu}{k}$ = Prandtl number, dimensionless

d) Turbulent Flow Inside Pipes or Tubes

By applying these ratios, extensive research data on gases and liquids of low viscosity in turbulent flow through long tubes have been correlated to give the following equation:

$$\frac{U_{ct} D_i}{k} = 0.023 \left\{\frac{GD_i}{\mu}\right\}^{0.8}\left\{\frac{c_p\mu}{k}\right\}^{0.4}\left\{\frac{T_b}{T_f}\right\}^{0.8} \qquad (15)$$

$$U_{ct} = 0.023 \frac{G^{0.8}}{D_i^{0.2}} \frac{c_p^{0.4} k^{0.6}}{\mu^{0.4}} \frac{T_b^{0.8}}{T_f^{0.8}} \qquad (16)$$

where:

U_{ct} = convection film conductance in long flow, Btu/sq ft, hr, F
T_b = avg bulk absolute temp, R = F + 460
T_f = avg film absolute temp
$= \frac{\text{surface abs temp} + T_b}{2}$, R = F + 460

Equation (16) applies to both the heating and the cooling of fluids flowing inside clean conduits. For convenience in use, equation (16) is presented in the form:

$$U_{ct} = U'_{ct} F_{pp} F_T \qquad (17)$$

where:

U'_{ct} = basic convection conductance in long flow
$= 0.023\ G^{0.8}/D_i^{0.2}$, Btu/sq ft, hr, F (Fig. 31, Chapter 11)

F_{pp} = physical properties factor evaluated at t_f for $c_p^{0.4} k^{0.6}/\mu^{0.4}$ (Figs. 32 and 33, Chapter 11)
F_T = temperature factor = $(T_b/T_f)^{0.8}$ (Fig. 34, Chapter 11)
t_f = avg film temperature
$= \frac{\text{avg surface temp} + \text{avg bulk temp}}{2}$, F

Charts for evaluating the several factors for convection conductance in longitudinal flow appear in Chapter 11, as does also an empirical equation for determining the steam film conductance (see superheater example).

e) Turbulent Flow Outside Pipes or Tubes

In the steam boiler, the most important application of convection is in the transfer of heat from hot combustion gases to the various heat-absorbing tubular surfaces. Compared with the extensive research on heat transfer for fluids flowing inside tubes, little has been done to establish convection heat transfer coefficients for crossflow over tube banks. Perhaps the most authoritative and complete data on crossflow heat transfer are those obtained as part of a B&W research program. The following equation was developed from the correlation of these data:

$$\frac{U_{co} D_o}{k} = 0.287 \left\{\frac{G D_o}{\mu}\right\}^{0.61}\left\{\frac{c_p\mu}{k}\right\}^{0.33} F_a \qquad (18)$$

$$U_{co} = 0.287 \frac{G^{0.61}}{D_o^{0.39}} \frac{c_p^{0.33} k^{0.67}}{\mu^{0.28}} F_a \qquad (19)$$

where:

U_{co} = convection film conductance in crossflow, Btu/sq ft, hr, F
F_a = arrangement factor which corrects for difference in geometric configuration from base arrangement

Equation (19) applies to both the heating and the cooling of fluids flowing outside clean tubes in crossflow. For convenience in use, equation (19) is presented in the form:

$$U_{co} = U'_{co} F_{pp} F_a F_d \qquad (20)$$

where:

U'_{co} = basic convection conductance in crossflow
$= 0.287\ G^{0.61}/D_o^{0.39}$, Btu/sq ft, hr, F (Fig. 35, Chapter 11)
F_{pp} = physical properties factor evaluated at t_f for $c_p^{0.33} k^{0.67}/\mu^{0.28}$ (Figs. 36 and 37, Chapter 11)
F_a = arrangement factor (Fig. 38, Chapter 11)
F_d = depth factor (Fig. 8)
t_f = avg film temperature
$= \frac{\text{avg surface temp} + \text{avg bulk temp}}{2}$, F

Charts for evaluating U'_{co}, F_{pp}, and F_a will be found in Chapter 11.

The arrangement factor, F_a, depends upon the tube arrangement, the ratios of tube spacing to tube diameter, and the Reynolds number. Values for F_a are given in Fig. 38, Chapter 11, for various conditions. The mass flow factor, G, in equation (19) and in the Reynolds numbers for the curves in the figure just mentioned, is based on the minimum free area available for fluid flow, which area may occur either in the transverse or in the diagonal openings.

The value of the film conductance, U_{co}, in equation (19) applies to banks of tubes which are 10 or more rows deep in the direction of gas flow. Tests confirm the assumption that the heat transfer coefficient increases beyond the first row because of increased turbulence. However, in banks of ten or more rows of tubes, the effect of the relatively low conductance in the first row, because of undisturbed flow, is negligible. For undisturbed flow* approaching a bank of less than ten rows, the film conductance, U_{co}, must be multiplied by a correction factor, F_d, known as the depth factor, given in Fig. 8. Factor F_d should be taken as unity when the tube bank is preceded by a bend, screen, or damper.

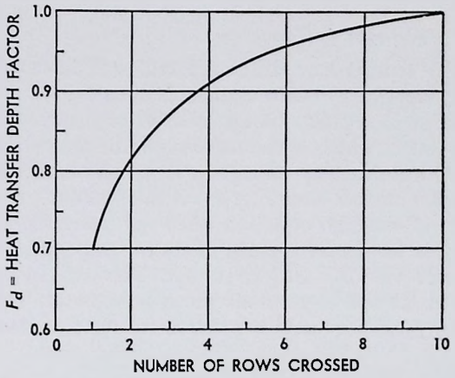


Fig. 8. Heat transfer depth factor for number of tube rows crossed in convection banks

Although equations (16) and (17) were developed for flow inside tubes, the same equations can also be developed by correlating the limited data available for flow of gases longitudinally over the outside of tubular surfaces except that an equivalent diameter must be substituted for the tube diameter. The equivalent diameter is defined as 4 times the area of the flow cross section divided by the contact perimeter. For flow along banks of circular tubes arranged in rectangular spacing, the equivalent diameter is:

$$D_e = \frac{4 \times (L_1 \times L_2 - 0.785\, D_o^2)}{\pi D_o} \qquad (21)$$

*Flow is considered undisturbed when it is straight and uninterrupted for at least 4 ft before entering a tube bank.

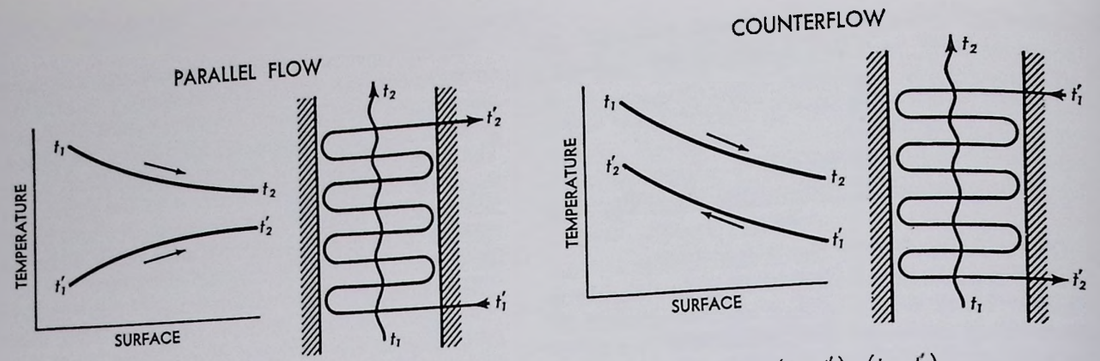


Fig. 9. *Temperature relationship (diagrammatic), hot and cold fluids, for parallel flow and counterflow, with corresponding formulas for logarithmic mean temperature difference*

where:

D_e = equivalent diameter, ft
D_o = tube outside diameter, ft
L_t and L_t are tube pitches, ft

f) Temperature Difference

The temperature difference indicated in equation (11) applies to that existing between a fluid and a solid at any point. In the transfer of heat between two fluids, which is the usual case, the temperatures of both fluids may change as they traverse the heat transfer surface (see Fig. 9). It then becomes necessary to modify equation (11) by an integrating process to obtain a mean effective temperature difference for the surface swept. This difference is known as the logarithmic mean temperature difference, $\triangle t_m$, a general expression for which follows:

$$^{\circ}\triangle t_m = \frac{\triangle t_1 - \triangle t_2}{\log_e \triangle t_1 / \triangle t_2} = \frac{\triangle t_1 - \triangle t_2}{2.3 \log_{10} \triangle t_1 / \triangle t_2} \qquad (22)$$

where:

$\triangle t_1$ = initial temp difference
$\triangle t_2$ = final temp difference

For practical design purposes, when the value of the ratio $\triangle t_1 / \triangle t_2$ in equation (22) is not over 1.5, the arithmetic mean of the temperature differences may be used as a reasonably close approximation.

The logarithmic mean temperature difference applies to the over-all heat transfer between two fluids when the specific heats and the local heat transfer coefficients are constant, that is, when these values are relatively unaffected by the temperature change along the surface. This is the condition in the majority of convection heat transfer applications in the generation of steam. When either or both the local heat transfer conductance and the specific heat vary along the heat transfer surface, special treatment is required.

[°] See Fig. 23, Chapter 11, for $\triangle t_m$ of specific arrangements, and Appendix, pages 7-A1 and 7-A2, for $\triangle t_m$ charts.

Combinations of Heat Transfer Mechanisms

In a practical heat exchanger, the transfer of heat usually depends upon the operation of a combination of at least two of the three fundamental mechanisms of heat transfer, conduction, convection, and radiation. For instance, in a tube wall separating two fluids, the convection conductances for each of the two fluid streams as well as the conduction through the tube wall must be evaluated. These conductances are combined into a single over-all conductance.

As noted above in the section on *Conduction*, there is an analogy between heat conduction and electrical conduction. By this analogy any resistance to heat flow may be treated as an equivalent resistance in an electrical circuit. The resistance to heat flow can be defined as the reciprocal of the conductance, or $R = 1/U$, where U is the unit for conductance, Btu/sq ft, hr, F.

a) Convection and Conduction Through Tube Wall

In Fig. 10 a tube wall separates two fluids with temperature t_o for the outside fluid and temperature t_i for the inside fluid. The over-all resistance is $R = R_{ab} + R_{bc} + R_{cd}$. Where D_o and D_i are respectively the outside and the inside diameters and k and l_c respectively the conductivity and the equivalent thickness of the tube wall, the conductances in Btu per sq ft of surface area per hr per F are: U_{co} through the outside film, k/l_c through the tube wall, and $U_{ci} \frac{D_i}{D_o}$ through the inside film, all referred to the outside surface. If the over-all heat conductance is U, then by substituting the reciprocals of the total and individual conductances for the total and individual resistances the expression becomes:

$$\frac{1}{U} = \frac{1}{U_{co}} + \frac{l_c}{k} + \frac{1}{U_{ci}} \frac{D_o}{D_i} \qquad (23)$$

For heat transfer between gases or superheated vapors separated by a tube wall, the resistances to heat flow in the films are generally many times greater than the resistance in the metal of the tube. In the evaluation of the over-all coefficient, the metal conductance may therefore be neglected without serious error. If the metal conductance is omitted and U_{co} and U_{ci} are based upon the same surface, equation (23) can be rearranged to read:

$$U = \frac{U_{co} \times U_{ci}}{U_{co} + U_{ci}} \qquad (24)$$

This is the general equation for the over-all conductance between two fluids, where the resistance to heat flow of the metal separating them is negligible. In this instance the combination of two convection conductances has been analyzed.

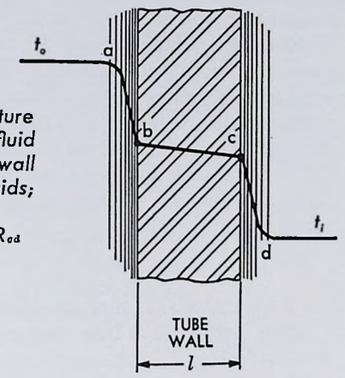


Fig. 10. *Temperature gradients through fluid films and tube wall separating two fluids; total resistance,* $R = R_{ab} + R_{bc} + R_{cd}$

b) Convection and Radiation in Tube Banks

Carbon dioxide and water vapor, which are found in considerable quantities in the products of combustion of hydrocarbon fuels, absorb and emit radiant energy or heat. In tube banks, therefore, heat is transferred by radiation as well as by convection. Since the resistance of circuits in parallel is the reciprocal of the sum of the reciprocals of the resistance of each circuit, heat transfer resistance, by analogy to electrical resistance, can be expressed as follows:

$$R = \frac{1}{\frac{1}{R_{rg}} + \frac{1}{R_{cg}}} + R_{tw} + R_{cs} \qquad (25)$$

where:

R_{rg} = resistance of gas film (the hotter fluid) to radiant heat flow
R_{cg} = resistance of gas film (the hotter fluid) to convection heat flow
R_{tw} = resistance of tube wall to conduction heat flow
R_{cs} = resistance of the other fluid film (the colder fluid) to convection heat flow

Neglecting the tube wall resistance, assuming that all conductances are based on the same surface, and substituting the reciprocals of the total and individual conductances for the total and individual resistances in the analogy to the electrical expression:

$$\frac{1}{U} = \frac{1}{U_{rg} + U_{cg}} + \frac{1}{U_{cs}}$$

and by rearranging:

$$U = \frac{(U_{rg} + U_{cg}) \times U_{cs}}{U_{rg} + U_{cg} + U_{cs}} \qquad (26)$$

where:

U_{rg} = intertube radiation conductance, Btu/sq ft, hr, F
U_{cg} = hotter fluid convection conductance, Btu/sq ft, hr, F
U_{cs} = colder fluid convection conductance, Btu/sq ft, hr, F

In equation (26), U represents the combined conductance for most boiler, economizer, superheater, and air heater heat transfer problems, where the resistance to heat flow through metal (such as tube walls) is small and may be neglected. This usually is so where the quantities of heat transferred and the resulting temperatures of the hotter and colder fluids are to be determined. However, where metal temperatures must be accurately established, conductance through the metal must be included.

The combined conductance, U, from equation (26) can now be used in the general equation for heat flow, $q = U S \triangle t_m$, which is identical with equation (1) with $\triangle t_m$ substituted for $\triangle t$ as the mean temperature difference.

An exact evaluation of intertube radiation is too cumbersome for engineering calculations. An approximate value can be obtained by applying the data for cavity radiation to the space bounded by a characteristic cluster of tubes (see Chapter 11).

c) Heat Transfer in Porous Materials

It is well known that porosity is an important factor in the effectiveness of materials as insulation in reducing heat losses. When so used in boiler work, porous materials are backed up by solid walls or casings, so that there is no appreciable flow through the pores of the material.

In porous insulating materials, heat flow is by conduction through the material itself and by the combination of conduction, convection, and radiation

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across the gas-filled voids in the material. The relative magnitudes of heat flow from these various mechanisms of heat transfer depend upon such variables as porosity of the material, the type of porosity (cellular or otherwise), the type of material, the composition and density of gas filling the voids, the temperature gradient through the material, and the absolute temperature of the material.

The analytical evaluation of the separate mechanisms of heat transfer in porous materials is extremely difficult. Some experimental work, however, has been done to establish an over-all conductivity (see Fig. 19, Chapter 14). Using the factor k = Btu/sq ft, hr, F/in. as over-all conductivity, the expression for the quantity of heat flowing through a porous material per unit of time then becomes similar to the expression for conduction of heat through a homogeneous substance. The effect on thermal conductivity of varying the pore size while maintaining a constant density of the material is shown by Fig. 11. The effect of other variables can be determined in a similar manner.

In high-temperature applications, the principal means of heat transfer across the voids is by radiation. Consequently for good insulation, the mean radiating path in the direction of heat flow should be as disrupted as possible. In low-temperature applications, heat flow by conduction and convection across the voids approaches the order of magnitude of heat flow by radiation. It is therefore important not only that the radiation path should be disrupted but also that the pores should cause the pools or pockets of fluid to be nearly stagnant, so that heat flow by free convection will also be restricted.

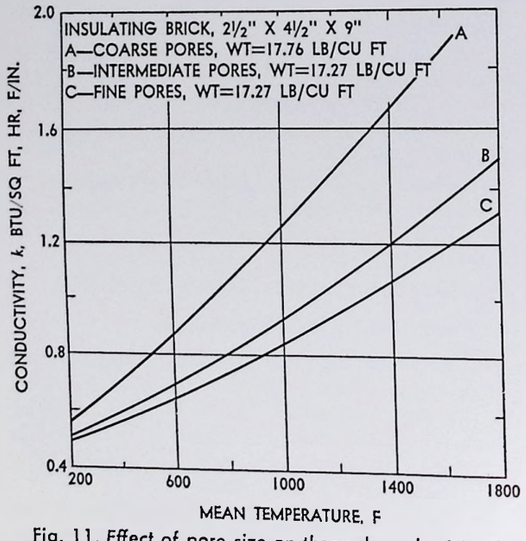


Fig. 11. *Effect of pore size on thermal conductivity, k, of a porous insulating brick*

An important recent development in the use of porous material is in what is called "transpiration" or "sweat" cooling. This extremely effective method of cooling is currently used for the protection of parts of gas turbines and rocket motors exposed to high gas temperatures, in the order of 3000 to 6000 F and even higher. The part to be cooled is constructed of a porous material consisting of metal particles having a high degree of structural interconnection. The particle interconnections in a porous metal part of this type are obtained by pressing powdered metal in a forming die, followed by heating ("sintering") in an inert atmosphere.

A cooling fluid, either liquid or gas, passes through the porous walls of the part to the surfaces exposed to heat. On leaving this exposed surface, the coolant passes through the adjacent laminar layer of gases in a direction opposite to the flow of heat by conduction through the layer. This opposite flow creates an effective barrier against convection heat transfer to the exposed surface. Heat reaching the surface by radiation or other means travels a short distance (1/32 to 1/16 in.) into the material by thermal conduction through the interconnected metal particles forming the part. The internal pore surface in this thin layer of material into which the heat penetrates is several times greater than the external surface exposed to the heat, and this very effective internal "extended" surface is cooled by the fluid (coolant) flowing through the material. The thermal conductance between the cooling fluid and the internal surface of the material is extremely high. For air at atmospheric pressure and 80 F as the coolant, this conductance will be in the order of 200 to 1000 Btu/sq ft, hr, F, depending upon pore size (inversely).

d) Heat Exchange

Porous material is also used effectively in various types of regenerative heat exchangers, where a fluid is made to flow through or over the material. A very high ratio of heat exchange surface to volume is possible by this means. Typical of well-known regenerative heat exchangers are: 1) the heating of combustion air for blast furnaces and open-hearth furnaces by passing the air through chambers of brick checkerwork which have been heated by furnace gases and 2) the pebble air heater (see Chapter 11) developed by B&W, where the air is heated by passing it through a heated mass of nonagglomerating solid particles in a moving bed.

A modern technique for regenerative heat exchange, utilizing what is called a "fluidized bed," has been developed in connection with hydrocarbon synthesis, coal gasification, and other chemical processes. The bed of finely pulverized nonagglomerating material is held in suspension (fluidized) by an upward stream of gas. The gas is evenly distributed by a perforated distributing plate at the base of the bed. The very effective transfer of heat to surfaces immersed in the bed results from the high ratio of heat exchange surface to volume of the suspended particles, the high rate of heat transfer between fluid (gas) and particles, and the rapid and thorough intermixing of particles in the fluidized bed.

In some applications of this process, heat is supplied by the combustion of fluidizing air and fuel within the bed itself, and the excess heat is removed by a tubular heat exchanger immersed in the bed. When combustion or other chemical reaction is made to take place in the fluidized bed, the fluidizing gas is not recirculated through the bed. However, almost all of the bed particles in the leaving gas are extracted by a cyclone separator and returned to the bed.

Other Forms of Heat Transfer

Condensation—Film Type

When a pure saturated vapor strikes a surface of lower temperature, the vapor condenses, and a film is formed on the surface. If it is assumed that the condensate film flows along the surface because of gravity alone and a condition of streamline flow exists throughout the film thickness, then heat transfer through this film is by conduction only.

Since the heat exchange is by conduction, the thickness of the condensate film has a direct effect on the quantity of heat transferred. The thickness of the film, in turn, depends upon the rate at which the condensate is removed. On a vertical surface, because of drainage, the thickness of the film at the bottom will be greater than at the top. The thickness of the film will increase as a plate is inclined from the vertical position.

An increase in the film temperature, which is the mean of the surface and vapor temperatures, decreases the film thickness, since the drainage velocity increases with the decrease in viscosity. The film thickness is affected by an appreciable velocity of the vapor because of the frictional drag between the vapor and the condensate. The preferable arrangement of the surface, therefore, is such that the vapor flows in the same direction (downward) as the condensate. As heat transfer increases with an increase in the temperature difference between the vapor and the surface, the thickness of the condensate film also increases. The surface conductance of heat transfer, therefore, decreases with an increase in temperature difference, which is an unusual relationship.

The theoretical equations for the conductance of heat transfer for film condensation involve the thermal conductivity, viscosity, and density of the condensate, the temperature difference between the vapor and the surface, and certain dimensions of the surface, such as height for vertical surfaces and diameter for horizontal tubes.

Dropwise Condensation

Steam will condense in drops, rather than as a film, on a surface contaminated with a substance which prevents the condensate from wetting the surface. Under these conditions a large part of the surface, not covered by an insulating film, is free for unobstructed heat transfer. This is known as dropwise condensation. The data of several investigators indicate that conductances for dropwise condensation are 4 to 8 times as high as the conductances for film condensation.

Mass Transfer and Diffusion

There are two related phenomena involved in some types of condensation. When a mixture of condensable vapor and a noncondensable gas is in contact with a surface that is below the dew point of the mixture, some condensation occurs, and a film of liquid is formed on the surface. A common example of this phenomenon is the condensation of water vapor on the outside of a glass of iced tea.

The mixture of vapor and noncondensable gas adjacent to the condensate film contains a lower proportion of vapor than the main body of the mixture, since some of the vapor has been removed as condensate. As vapor from the main body of the mixture diffuses through the vapor-lean layer, it is condensed on the cold surface. The rate of condensation is thus governed by the laws of gas diffusion, while the sensible heat transmission is controlled by the laws of heat conduction and convection. This mode of heat transfer is important in the design of cooling towers, humidifiers, etc., where mixtures of vapors and noncondensable gases are encountered.

Evaporation or Boiling

The boiling point of a liquid may be defined as the temperature at which its vapor pressure is equal to the total pressure above its free surface. Under special conditions the liquid may be heated until the vapor pressure is considerably higher than the total pressure, without the formation of bubbles as a visible indication of "boiling." When a liquid is heated to a temperature above its boiling point, it is said to be superheated.

A liquid must be superheated to start bubble formation (for temperatures below the "critical"), because more thermal energy is required to free an escaping molecule from a small concave surface than from a plane surface. This concept is based on the molecular action at the interface between a liquid and its vapor. Molecules of the liquid continually leave the surface and are projected into space. This space, however, is not empty but is occupied by rapidly moving vapor and gaseous molecules which

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are farther apart than the molecules escaping from the liquid. Some of the vapor molecules are continually entering the liquid. When the rate at which molecules escape from the liquid to the vapor equals the rate at which molecules from the vapor enter the liquid, a condition of equilibrium exists, and there is no net interchange of material between liquid and vapor phases. When heat is added to the liquid, its temperature and the corresponding kinetic energy of its molecules increase, with the result that molecules escape from the liquid surface at a faster rate than molecules from the vapor enter the liquid. This is the process known as *evaporation*.

Escaping molecules from a flat liquid surface collide only with the vapor and gaseous molecules above this surface, but when the liquid surface is in hollow or concave form, the escaping molecules occasionally collide with one another. Such collisions increase in frequency as the curvature of the surface increases, and they reach a maximum when the surface returns upon itself like the inside of a hollow sphere or bubble, where the opposing walls tend to cancel each other's efforts to eject molecules into space. It follows that a smaller number of the vapor and gaseous molecules are required within the bubble to prevent additional molecules from permanently escaping from the liquid. This means that the pressure within the bubble is lower than the pressure above a flat liquid surface or, conversely, that the superheat in the liquid must be higher around a bubble than at a flat liquid surface. If the diameter of the bubble is infinitely small, the superheat has to be infinitely high, so, theoretically, in an absolutely pure liquid a bubble cannot be started, and boiling below the liquid surface cannot be initiated.

Water can be evaporated without considerable superheating because of the presence in the liquid of small curved surfaces which tend to initiate bubble formation. This condition is caused in part by the roughness of the heating surface and by the presence of dissolved gases or small particles which act as nuclei for the formation of bubbles. Heat transfer from a heating surface to a vapor bubble may be divided into two parts: 1) from the heating surface to the boiling liquid and 2) from the liquid to the bubble.

At high temperature differences, the conductance reaches a maximum and then decreases with a further increase in temperature difference. This is because of the formation of a continuous vapor film at the surface, which insulates the surface from the liquid. The surface becomes so hot that the liquid no longer wets it, as in the case of a drop of water rolling on the cover of a hot stove.

The temperature difference between the surface and the liquid for maximum heat transfer varies but is usually in the order of 40 to 80 F. Maximum film conductance for boiling water at atmospheric pressure is approximately 10,000 Btu/sq ft, hr, F. Field tests of boilers, however, have indicated film conductances in the order of 20,000 Btu/sq ft, hr, F. The boiling-water film conductance is so high that this conductance is not a limiting factor and may be ignored in evaluating surface requirements.

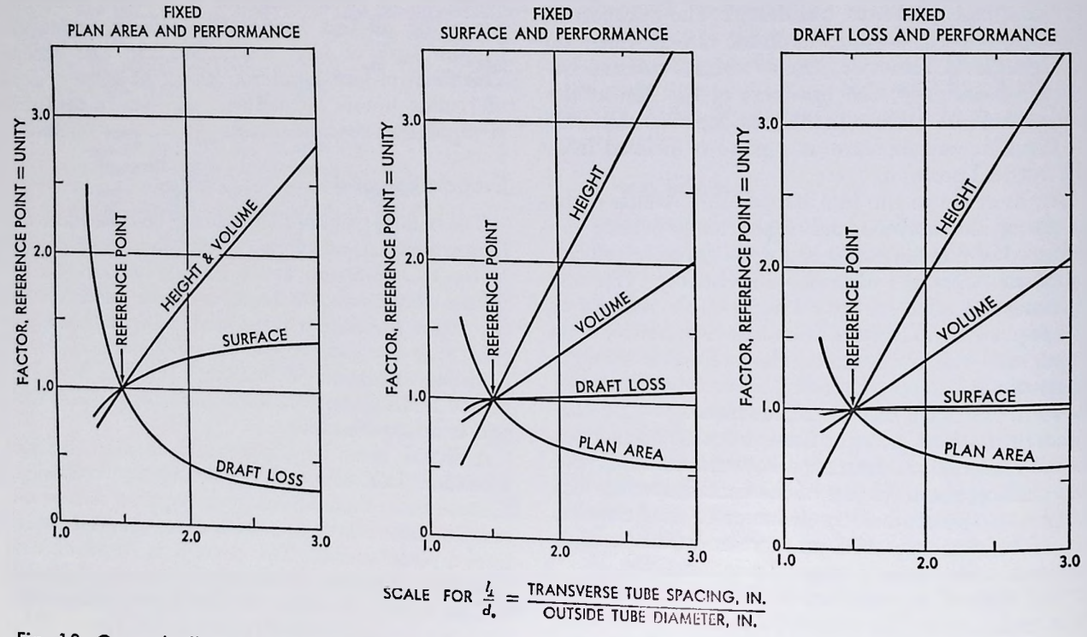


Fig. 12. General effect of convection tube arrangement on volume occupied, amount of surface, draft loss, and floor area for selected conditions fixed. Based on tubes in line and longitudinal spacing, $l_{ll} = 2 \times$ tube OD

It is apparent that with boiling liquids, forced circulation of the liquids or the gases is not necessary to obtain very high film conductances. There has been intensive research in connection with the possible use of this method of heat transfer in rocket- and jet-engine cooling and in removing heat from nuclear reactors.

Heat Transfer Applications

Convection Banks

In applying the theoretical relationships of heat transfer by convection to the design of practical convection banks in boilers, many other factors must be considered, which will usually require a compromise in selecting the optimum characteristics. Among these factors, the effects of which on freedom of design are covered in detail in Chapters 11 and 12, are:

1. Available space in a plant.
2. Type of fuel, firing rates, and feedwater conditions.
3. Maintenance and operating costs.
4. Allowable metal temperatures and drum- or header-wall thicknesses.
5. Manufacturing standards set up by boiler manufacturers for a range of tube sizes and arrangements based on operating experience.

a) Tube Spacing and Arrangement

The more important factors, other than heat absorption and resistance to gas flow, to be considered in establishing the optimum tube spacing and arrangement for a convection surface are slagging or fouling of surfaces, accessibility for cleaning, and space occupied. However, with these limitations, an optimum transverse spacing for in-line or staggered banks should be selected. A large longitudinal spacing (l_{ll}) in relation to the transverse spacing ($l_{\perp}$) is usually undesirable, since the length of flow path for the calculated surface may be excessive.

The effect on design of various ratios of transverse tube spacing to outside tube diameter is plotted in Fig. 12 for in-line tube banks and longitudinal spacing equal to two tube diameters. By entering the curves with calculated values for draft loss, heating surface, and space occupied based on an assumed tube diameter and transverse spacing, a pattern best suited to the requirements of the unit can be selected. Similar curves can be plotted for staggered arrangements with a fixed back pitch.

b) Tube Diameter

For turbulent flow, the heat transfer conductance is inversely proportional to some power of the tube diameter (see equations (16) and (19) above). The exponent for longitudinal flow is 0.20 and for cross-flow, 0.39. For most effective heat transfer, therefore, it is desirable to hold the tube diameter to a minimum. An optimum tube diameter for this purpose, however, may require an arrangement that is expensive to fabricate and difficult to install and maintain in operating condition, thus increasing the initial and maintenance costs. A compromise between heat transfer effectiveness and manufacturing, erection, and service limitations is, therefore, always necessary in the selection of tube diameter.

In oil-fired boilers of high rating for marine service, tubes of 1-in. OD are used in the boiler bank beyond the screen tubes. Because of the higher heat absorption rates, which necessitate an increase in the quantity of circulating water, the screen tubes and water-wall tubes are usually 1½-in. OD (see Chapter 11 for tube sizes in other applications).

c) Penetration of Radiation

A convection bank or banks of tubes bordering a furnace or a cavity act as blackbodies in absorbing radiant heat. Some of this heat, however, radiates through the spaces between the tubes of the first row and sometimes penetrates beyond the fourth row. The quantity of radiant heat penetration can be established by geometric or analytical methods. The effect of this penetration is especially important in establishing tube temperatures for superheaters located fairly close to a furnace or a high-temperature cavity (see Chapter 11).

d) Effect of Lanes

Lanes in tube banks, formed by the omission of a row of tubes, may decrease the heat absorption considerably, since, in effect, these passages act as bypasses for the hot gases through the banks. Although the over-all efficiency will decrease, the high mass flow and greater gas weight through the lanes will increase the absorption rates of the adjacent tubes and may develop critical tube temperatures in superheaters or steaming conditions in economizers. Whenever possible, therefore, lanes should be eliminated both within tube banks and between tube banks and walls. Lanes, however, are unavoidable under certain conditions. In installing superheaters, for instance, space must be allowed for additional surface to satisfy future specified increases in superheated steam temperature. A calculation for the lanes, taking into account the increased gas weight and mass flow, is therefore necessary.

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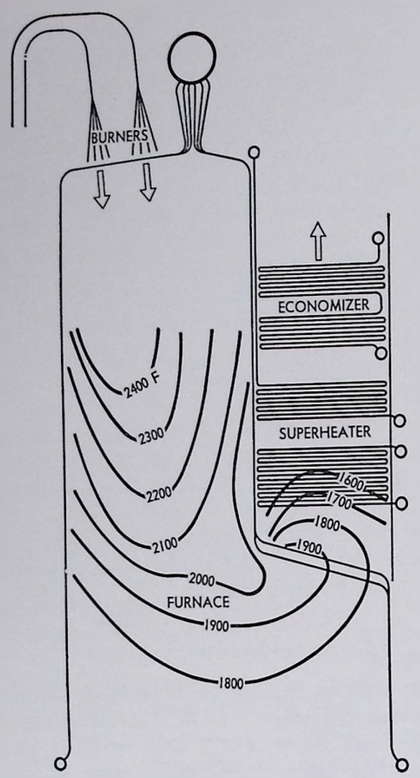


Fig. 13. *Typical temperature contours in the furnace of a down-fired Radiant boiler*

e) Heat Transfer to Water

Water Film Conductance. In boiler design practice, the film conductance for water in economizers is so much higher than the gas-side conductance that it is neglected in evaluating economizer surface.

Boiling-Water Conductance. In boiler practice the combined gas-side conductance (convection plus intertube radiation) seldom exceeds 30 Btu/sq ft, hr, F. In estimating the resistance to heat flow, the film conductance for boiling water is therefore neglected, since its reciprocal (the resistance to heat flow) is so very small compared with the gas-side resistance.

Effect of Oil or Scale. Scale deposits interpose a high resistance to the flow of heat, and the additional temperature drop required to maintain a given fluid temperature inside the tube as the thickness of the scale increases will necessitate a dangerously high metal temperature, which will eventually cause failure. With the high heat absorption rates to which furnace-row tubes of high-capacity boilers are exposed, it is therefore essential entirely to prevent the formation of scale (see Fig. 11, Chapter 29). The formation of scale and the deposit of oil, also a factor in resistance to the flow of heat, can be prevented by proper feedwater treatment and careful operation.

f) Heat Transfer to Steam

In the design of superheaters, the steam film constitutes a significant resistance to the flow of heat, and although this resistance is many times lower than the gas-side resistance, it cannot be neglected in computing the over-all resistance to heat flow or the heat transfer rate. It is particularly significant in estimating superheater tube temperatures, since the temperature of the inside wall is equal to the steam temperature plus the temperature drop through the steam film. If the steam film conductance is designated as U_s, the film temperature drop will be $q/S \div U_s$, where q/S and U_s are expressed in Btu/sq ft, hr and Btu/sq ft, hr, F, respectively, using the outside surface area of the tube as the base in each expression.

Because of the relative magnitude of the resistance to heat flow in the steam film and the elevated temperatures at which superheater tubes operate, it is imperative that scale deposits be prevented in superheater tubes. Even an extremely thin layer of scale forms an objectionable insulating barrier, which, together with the steam film, will be sufficient to cause overheating and ultimate failure of a tube.

Furnaces

An analytical solution of the problem of heat transfer in the furnace of a steam-generating unit is extremely complex, and it is impossible to calculate furnace outlet temperatures by theoretical methods alone. Nevertheless, the furnace outlet temperature must be predicted correctly since, in a large measure, this temperature determines the design of the remainder of the unit, particularly the superheater.

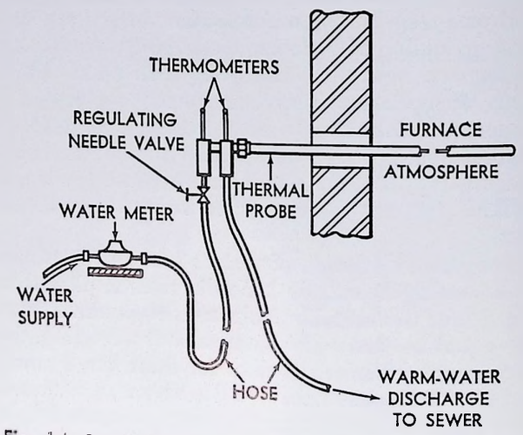


Fig. 14. *Special thermal probe or calorimeter (diagrammatic) sometimes used to establish the heat absorption rates of surfaces throughout a furnace*

An attempt will therefore be made to outline the complexities of the problem and its practical solution.

In the furnace of a steam-generating unit, all of the principal mechanisms of heat transfer take place simultaneously. These mechanisms are: intersolid radiation between solids in fuel beds or pulverized-coal particles and tubes and refractory materials; nonluminous gas radiation from the products of combustion; convection from the furnace gases to the furnace walls; and conduction through ash deposits on tubes.

Temperature varies throughout the furnace (see Fig. 13 for typical furnace-temperature contours). Fuel and air enter at relatively low temperatures, reach a high temperature during combustion, and cool again as the products of combustion give up heat to the furnace enclosure. Water-cooled surface is at a relatively low temperature, while deposits on tubes and refractory in the furnace walls are at high temperatures. All of these temperatures change with load, excess air, burner adjustment, and other operating conditions. A special water-cooled calorimeter, the thermal probe illustrated in Fig. 14 (as well as the high-velocity thermocouple, see Fig. 19, Chapter 6), is a useful tool for establishing the order of heat absorption in a furnace. Variations in heat absorption in a primary slag-tap furnace of a steam-generating unit are shown in Fig. 15.

Furnace geometry is complex. Furnaces vary in shape and size, in the location and spacing of burners, in the size of fuel beds, in the disposition and type of cooling surface, in the spacing of tubes in tube-and-refractory walls, and in the arrangement of arches and hoppers. Flame shape and length affect the geometry of radiation in the furnace.

Surface characteristics vary. The enclosing furnace walls may include any combination of fuel bed, refractory material, studded tubes, water-cooled cast-iron blocks, spaced tubes backed by refractory, close-spaced tubes, or tube banks. Emissivities of flames and various types of surfaces are different. The water-cooled surface may be covered with fluid slag or dry ash in any thickness, or it may be clean. Heat-absorbing surface in a furnace may, when clean, show absorption rates of 175,000 Btu/sq ft, hr, or even higher. When slag accumulations reach equilibrium, the absorption rate may be reduced to 50,000 Btu/sq ft, hr, or even lower.

Fuel variation is significant. Not only are there important differences inherent in the use of different fuels and in the manner in which they are burned, as in stoker, pulverized-coal, gas, oil, or waste-fuel firing, but there are also differences caused by the use of different types of the same fuel. Coal, for example, may be high volatile or low volatile, may have much or little ash, and the ash may fuse at high or low temperature, varying considerably or very little with the oxidizing properties of the furnace atmosphere.

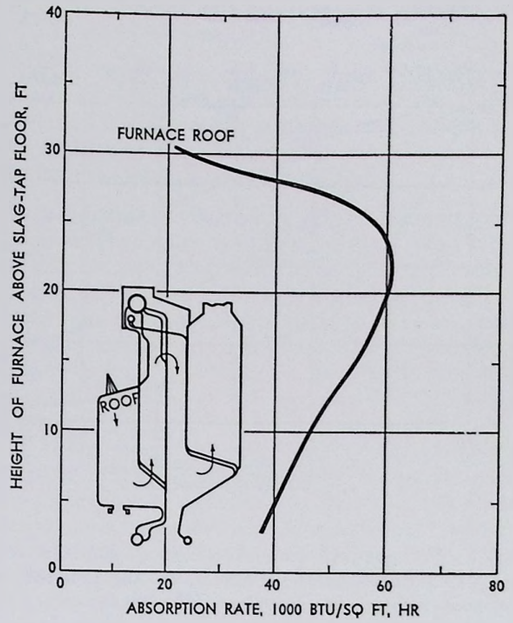


Fig. 15. *Typical heat absorption rates at different elevations in a furnace of type indicated, at maximum output*

Since furnace outlet temperatures cannot be calculated by theoretical heat transfer methods alone, it is necessary to use an empirical approach. Calculations are based on test results from similar types of units, supplemented by data accumulated from operating experience and by judgment based on a knowledge of heat transfer principles and the characteristics of fuels and slags. To acquire the necessary basic information on furnace outlet temperature, B&W pioneered in the development of the high-velocity thermocouple and conducted an extensive program of field tests for measuring gas temperatures in furnaces of operating units. A similar program has been undertaken by a special research committee of the ASME on furnace performance factors.

In setting up an empirical method of furnace temperature calculation, stoker, waste-fuel, pulverized-coal, Cyclone Furnace, natural gas, and oil firing must be considered separately. For simplification, the problem is divided into several classes, the number of variables is reduced by disregarding those of lesser importance, and other separate factors are combined into single variables, concentrating on the variables of greatest influence.

In the absorption of radiant heat, the effectiveness of tubes spaced other than touching and backed by refractory as compared with completely water-cooled surface (touching tubes or equivalent°) is shown in Fig. 16. The effectiveness of block-covered or stud-

°A wall of flat-stud tubes is considered completely water cooled.

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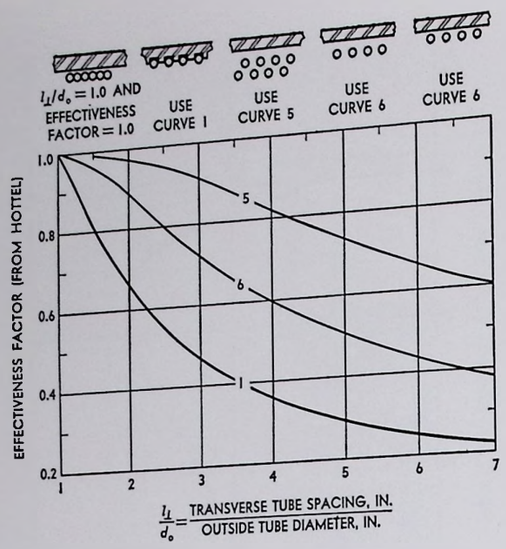


Fig. 16. *Furnace-wall area-effectiveness factor (1.0 for completely water-cooled surface). A reduced area (effective-cold surface) is determined from these curves for walls not completely water cooled ($l_1/d_o > 1$)*

ded-tube surface and its expected ash covering, as compared with completely water-cooled surface, can also be estimated. The entire furnace envelope can then be evaluated in terms of effective-cold surface.

The heat energy supplied by the fuel and by the preheated combustion air, corrected for minor items such as unburned combustible loss, may be combined into a single variable known as heat available (see examples, Chapter 4, Appendix, for calculated values of heat available). For correlating furnace outlet gas temperature and heat available, the Btu per sq ft of absorbing surface is a better basis than Btu per cu ft of furnace volume. Experience has shown that the outlet temperature of a given furnace burning a given fuel in suspension varies approximately as the logarithm of the heat available per hr per sq ft of effective-cold surface.

Assuming typical ratios between effective-cold surface and total furnace envelope, the approximate furnace outlet gas temperature will be of the order shown in Fig. 17. The curve shown for pulverized coal is for a dry-ash-removal furnace with fairly clean walls.

The furnace outlet temperatures and the heat absorption rates for heat available (heat input) rates of most pulverized-coal-fired furnaces lie within the shaded bands shown in Figs. 18 and 19, respectively. The limits indicated are intended to serve only as a general guide. The horizontal scale in each case is heat available in thousand Btu per hr per sq ft of water-cooled surface, excluding the floors of slag-tap furnaces. The bands for dry-ash and for slag-tap overlap between 100,000 and 150,000 heat available rates, but not with the same coal. Within this range, the coal used in the slag-tap furnace should have a relatively low ash-fluid temperature, and the coal used in the dry-ash furnace should have ash with a very high initial-deformation temperature. In the overlapping range, dry-ash and slag-tap each have about the same rate of heat absorption, as shown in Fig. 19, or the same "dirtiness" factor. Both bands may seem rather broad, but they cover a wide range of ash characteristics, considerable diversity in type of water-wall construction, and, in some cases, a difference in dirtiness as a result of difference in cleaning, load factor, etc.

The quantity of heat leaving the furnace can be calculated from the gas weight leaving the furnace, the composition of the gas, and the furnace outlet gas temperature, using gas enthalpy values given in Chapter 4 (Appendix). The quantity of heat absorbed in the furnace is the difference between the quantity of heat leaving the furnace and the quantity of heat supplied by the preheated combustion air and by the combustion of the fuel fired.

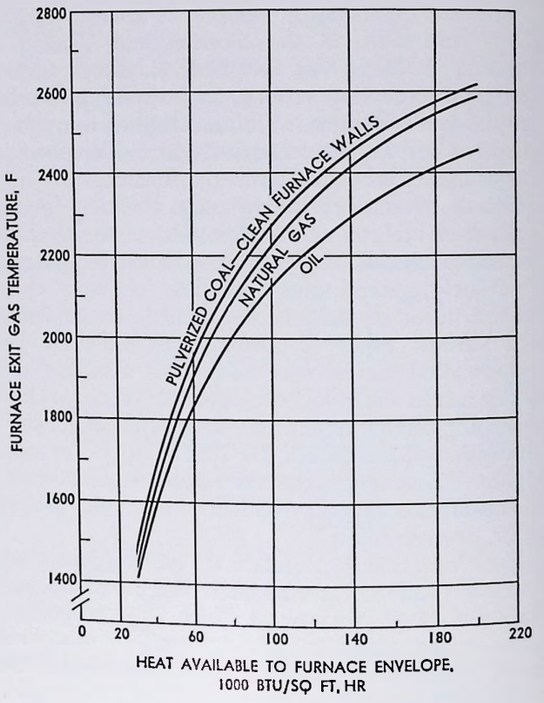


Fig. 17. *Approximate furnace outlet gas temperature for different fuels at different rates of heat available to water-cooled furnace surface*

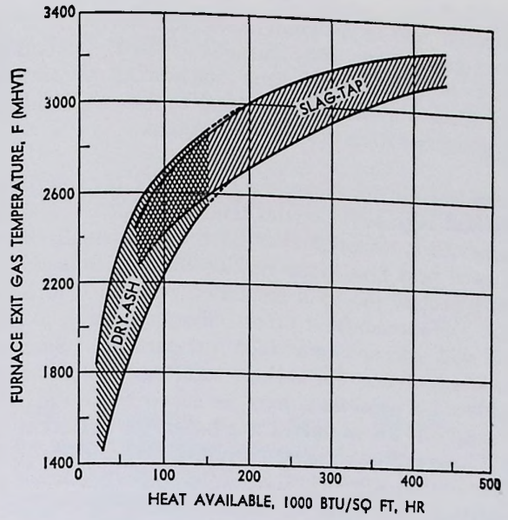


Fig. 18. *General range of furnace exit gas temperature for dry-ash and slag-tap pulverized-coal firing*

Cavities

In designing steam-generating units, it is necessary to provide cavities between tube banks for access, for soot blowers, or for the installation of additional surface as required. However, heat transfer from hot flue gas to heat-absorbing surface surrounding cavities does take place. The gas loses heat by radiation to the surroundings while passing through the cavity. Though the procedure for calculating the quantity of heat transferred in a particular case is too complex for treatment here, cavity radiation can be calculated by using methods applying to nonluminous radiation developed by authorities on heat transfer. It may be of interest, however, to discuss briefly some of the factors which affect heat transfer by nonluminous gas radiation in cavities.

Temperature Level. Radiation from nonluminous gases to the boundary surfaces and radiation to the gas by the surroundings increase approximately as the fourth powers of their respective absolute temperatures.

Composition of the Gas. In steam-generating units, the only normal constituents of flue gas which contribute importantly to nonluminous gas radiation are carbon dioxide and water vapor. The concentrations of these constituents depend upon the fuel burned and the excess air; the amount of radiation depends upon these concentrations.

Particles in the Gas. The particles carried by dust-laden gases receive heat from the gas by radiation, convection, and conduction and radiate by intersolid radiation to the surroundings.

Size of Cavity. The quantity of heat transferred per unit of time increases as the size of the cavity increases. Thick layers of gas radiate more vigorously than thin layers. The shape of the cavity also has a complicated effect upon heat transfer.

Receiving Surface. Refractory surface forming part of a cavity boundary will reach a fairly high temperature by convection and radiation from the flue gas and will reradiate to the gas and to the other walls of the enclosure. Reradiation from clean heat-absorbing surface is small unless the surface temperature is high, as it may be in superheaters and reheaters. Deposits of ash or slag on the tubes reduce heat absorption and increase reradiation.

In boiler design there are two effects of cavity radiation which are important. The temperature of flue gas will drop, sometimes as much as 100 F, in passing across a cavity. This is a significant fact in design calculations. Gas radiation also increases the heat absorption rates for the tubes forming the cavity boundaries. This effect is often important in high-temperature superheaters, where absorption rates strongly influence tube temperatures and the selection of alloys.

Insulation

The calculation of heat transfer through insulation follows the principles outlined for conduction through a composite wall. A number of practical matters must also be considered in the design of insulation. These will be briefly discussed. The reader is

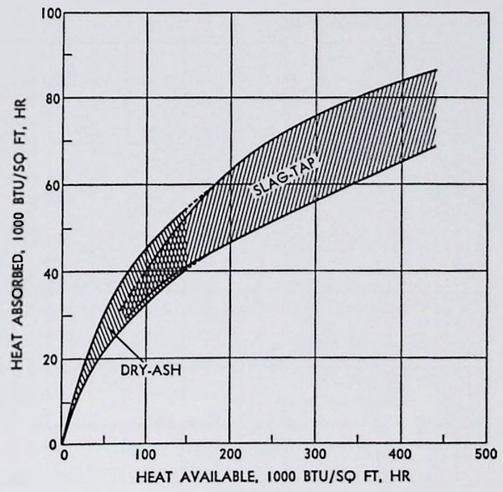


Fig. 19. *General range of furnace heat absorption in dry-ash and slag-tap pulverized-coal-fired furnaces*

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referred to Chapter 14 for detail information, including properties of insulating materials, the mechanics of application, and a sample calculation of heat flow through a tube-and-brick wall covered with insulation.

Hot-Face Temperature. In a boiler furnace with tube-to-tube walls, the hot-face temperature of the insulation may be taken as saturation temperature of the water in the tubes. If the inner face of the furnace wall is refractory, with or without cooling by spaced tubes, the hot-face temperature of the insulation must be calculated or estimated from a knowledge of conditions on the gas side of the furnace wall, based on radiation and convection heat transfer and empirical data. Fortunately, for a considerable error in the hot-face temperature of the insulation, the corresponding error in cold-surface temperature is much smaller, and the effect on total heat loss from the unit is not important, although it may be misleading in the selection of materials.

Heat Loss and Cold-Face Temperature. The heat loss to the surroundings and the cold-surface temperature decrease as the thickness of the insulation increases. A cold-face temperature of 130 to 160 F is usually considered satisfactory for a boiler room. When the insulation is thick, the change in cold-surface temperature for increase in thickness is small, whereas the cost of the insulation increases steadily with thickness (see Chapter 14). Standard commercial thicknesses of insulating materials should be used in the composite wall.

The detailed calculation of over-all heat loss by radiation and convection from the surfaces of a steam-generating unit (usually called "radiation loss") is tedious and time consuming. A simple approximate method is provided by the chart (Fig. 20) prepared from the ABMA original.

Ambient Air Conditions. Low ambient air temperature and high air velocities tend to reduce cold-face temperature, although they have only a small effect on total heat loss, since surface film resistance is a minor part of the total insulation resistance to heat flow. The proximity of other boilers radiating to the first will increase cold-face temperature. Surface film conductances for various wall temperatures and ambient air conditions may be taken from Fig. 20, Chapter 14. These curves are based on radiation to 80 F surroundings, using an emissivity of 0.95, plus natural-convection heat transfer to 80 F air at various velocities. The effect of surface resistance on heat loss through casings and on casing temperature is illustrated in Fig. 23, Chapter 14.

Temperature-Use Limits and Conductivities. Refractory or insulation material suitable for high-temperature applications is usually more expensive and less effective as insulation than low-temperature materials. It is therefore customary to use several layers of insulation, the lowest cost and most effective insulation in the cool zones and the higher cost material only where the temperature exceeds the permissible operating limits for low-temperature materials. Typical temperature-use limits and thermal conductivities are shown in Fig. 19, Chapter 14.

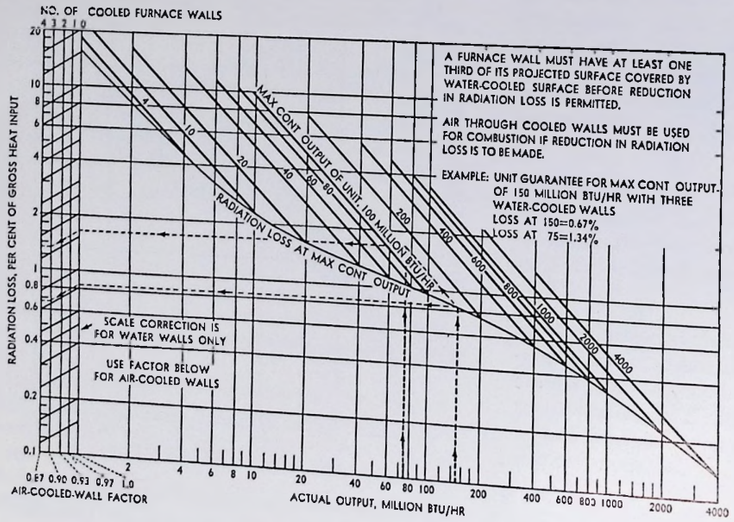


Fig. 20. *Radiation loss (ABMA) in per cent of gross heat input*

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FLUID DYNAMICS

IN the production and utilization of steam there are many problems involving the flow of fluids. The term *fluid* embraces substances in either the liquid or the gaseous state. A *liquid* is an essentially incompressible, homogeneous, fluid substance in such state as to permit the free movement of all of its contiguous molecules which have sufficient interattraction (cohesive forces) to prevent separation, thereby maintaining a definite volume but with freedom to conform to any shape and form a free surface between itself and any other immiscible fluid. A *gas* is an aeriform fluid, the independent molecules of which, because of low interattraction (cohesive forces), have unrestricted freedom of motion permitting indefinite expansion. A *vapor* is the gaseous state of a liquid or solid during the phase when certain physical properties resemble neither those of the liquid or solid from which it was formed nor those of the gas which it more closely simulates the higher its temperature. In contrast to a fluid, a *solid* is a substance in such state that the interattraction (cohesive forces) between contiguous molecules is sufficient to prevent any perceptible motion within its mass, which therefore maintains both definite volume and shape.

The broad subject of fluid mechanics must be culled for those elements which are of particular significance in the field of steam. These would include the flow of steam and water through pipes, fittings, valves, and tube bundles; the flow of air and gases through ducts, tube banks, fans, compressors, and turbines; the convection flow and circulation of water and steam; the convection flow of gases for draft effect; the flow of water and steam through nozzles, orifices, venturi tubes, pumps, and turbines; the throttling calorimeter; and the separation of liquids, vapors, and solid particles from flowing mixtures. The fluids-in-motion or fluid-dynamic aspects of these phenomena are more important in practice than are those of fluid statics.

Determination of the velocity, V, at which the fluid moves is a prime requisite in the study of the effects produced by fluid motion. To evaluate the quantity of flow, friction losses, viscosity forces, particle separation, heat absorption, and fluid flow paths, it is necessary to know the fluid velocity.

Energy Equation applied to Fluid Flow

Many basic theoretical relationships involving fluid flow stem from the fundamental *general energy equation* of thermodynamics:

$$V_1^2/2g + p_1v_1 + u_1 + Z_1 \pm W_k \pm {}_1Q_2 = V_2^2/2g + p_2v_2 + u_2 + Z_2 \qquad (1)$$

where:

V_1 and V_2 = fluid velocities @ stations 1 and 2, respectively, ft/sec
g = gravitational constant, 32.2 ft/sec^2
p_1 and p_2 = static pressures @ respective stations, lb/sq ft abs
v_1 and v_2 = specific volumes @ respective stations, cu ft/lb
u_1 and u_2 = internal energies of fluid @ respective stations, Btu/lb ÷ 778 = ft-lb/lb
Z_1 and Z_2 = elevations above datum @ respective stations, ft-lb/lb
W_k = work done on or by the fluid while it is in the device, ft-lb/lb
${}_1Q_2$ = heat added to or abstracted from the fluid while it is in the device, Btu/lb

Where there is no difference in elevation between the initial and subsequent stations 1 and 2 ($Z_1 = Z_2$), where the initial velocity of the fluid (V_1), often referred to as the velocity of approach, is zero or negligibly small, and where neither work nor heat is involved, equation (1) for the gain in kinetic energy becomes:

$$V_2^2/2g = (p_1v_1 + u_1) - (p_2v_2 + u_2) \qquad (2)$$

Since the terms represented by $(p_1v_1 + u_1)$ and $(p_2v_2 + u_2)$ are equal to the enthalpies, h_1 and h_2, of the fluid at the respective stations, equation (2) for the gain in kinetic energy becomes:

$$V_2^2/2g = h_1 - h_2 \qquad (3)$$

If enthalpy is measured in Btu per lb, as is generally the case, equation (3) for the gain in kinetic energy becomes:

$V_2^2/2g = (h_1 - h_2)778°$, ft-lb/lb = ft (4)

Solving equation (4) for velocity:

$V_2 = 223.9\sqrt{h_1 - h_2}$, ft/sec (5)

For steam as the fluid, the enthalpy drop may be determined from the Mollier chart (see Appendix, page 10-A6). Where a Mollier chart is not available and the perfect-gas laws are applicable, the velocity can be computed with the following equation for frictionless flow:

$$V_2 = 8.02\sqrt{\frac{\kappa}{\kappa-1}\,p_1v_1\left\{1-(p_2/p_1)^{\frac{\kappa-1}{\kappa}}\right\}}, \text{ ft/sec} \quad (6)$$

where:

$\kappa = c_p/c_v$, ratio of specific heats @ constant pressure and constant volume

Equations (1) through (5) apply to all fluids, equation (6) to fluids that follow the perfect-gas laws.

For an incompressible fluid ($v_1 = v_2$) in frictionless adiabatic flow, equation (1) for the gain in kinetic energy becomes:

$$(V_2^2 - V_1^2)/2g = (p_1 - p_2)v + (Z_1 - Z_2)$$
$$= \triangle h_p + \triangle h_z = H, \text{ ft-lb/lb} = \text{ft} \quad (7)$$

where:

$\triangle h_p$ = head from pv difference between stations 1 and 2, ft of fluid flowing

$\triangle h_z$ = head from difference in elevation between stations 1 and 2, ft of fluid flowing

$H = \triangle h_p + \triangle h_z$ = net head causing flow, ft

$v = v_1 = v_2$, specific volume of fluid, cu ft/lb

If the velocity of approach (initial velocity, V_1) is zero or negligibly small, equation (7) for the gain in kinetic energy becomes:

$V_2^2/2g = H$, head in ft of fluid flowing = velocity head, h_v (8)

Solving equation (8) for velocity:

$V_2 = 8.02\sqrt{H}$, ft/sec (9)

In establishing the head, H, in ft of fluid flowing, the following conversions are helpful:

$$H \text{ in ft of fluid} = \frac{\triangle p \text{ in psi} \times 144}{\text{sp wt of fluid}} \quad (10)$$

or using Fig. 21, Chapter 10 (for water):

$$H \text{ in ft of gas} = \frac{\text{head in in. of water}}{12} \times \frac{\text{sp wt of water}}{\text{sp wt of gas}} \quad (11)$$

For the conversion of manometer readings to head, H, in ft of fluid flowing:

$$H \text{ in ft of fluid} = \text{manometer reading in ft} \times \frac{\gamma_{mf} - \gamma_{sf}}{\gamma_{ff}} \quad (12)$$

where:

γ_{mf} = specific weight of manometer fluid

γ_{sf} = specific weight of superposed fluid

γ_{ff} = specific weight of fluid flowing

Where the superposed fluid is air or gas, the value of γ_{sf} is negligibly small and need not be considered.

From equation (9) it is evident that, for frictionless flow of a nonexpansive fluid with a negligible velocity of approach ($V_1 = 0$), the velocity of a free-spouting jet is determined solely by the pressure or head, H, in ft of fluid causing the flow and that the velocity of the jet is independent of the density of the fluid.

To determine the velocity for expansive fluids where the pressure drop is small, as in many metering nozzles or the flow of air in fan and draft systems, equation (5) or (6) cannot be used with reliability because of the difficulty in accurately evaluating the factors involved. In such cases, where the difference in specific volume of the fluid upstream and downstream is small, $(v_2 - v_1)/v_2 < 0.05$, the fluid may be considered practically nonexpansive and equation (9) used to determine the velocity.

The example below will serve to illustrate an application of equation (5) to a simple case of an expansive fluid in motion:

Example 1

To find the velocity of steam flowing from a header through a well-insulated horizontal pipe, using the Mollier chart to determine the enthalpy of the steam at stations 1 and 2 for the conditions given:

Station 1 in Header	
Velocity, fps	0
Pressure, psia	207
Temperature, F	468
Enthalpy, h_1, Btu/lb	1250
Station 2 in Pipe	
Pressure, psia	118
Temperature, F	380
Enthalpy, h_2, Btu/lb	1214

Substituting values of h_1 and h_2 in equation (5):

$V_2 = 223.9\sqrt{1250 - 1214} = 1343$ ft/sec

The value of h_2 reflects the effect of pipe friction on the steam, tending to raise the value of h_2 and thus decrease the value of V_2. Had the flow in the pipe to station 2 been assumed to encounter no friction, the observed hypothetical values of pressure and temperature for such an adiabatic flow might have been 120 psia and 358 F ($h_2 = 1200$ Btu/lb), giving a velocity of:

$V_2 = 223.9\sqrt{1250 - 1200} = 1583$ ft/sec

° Exact value = 778.26

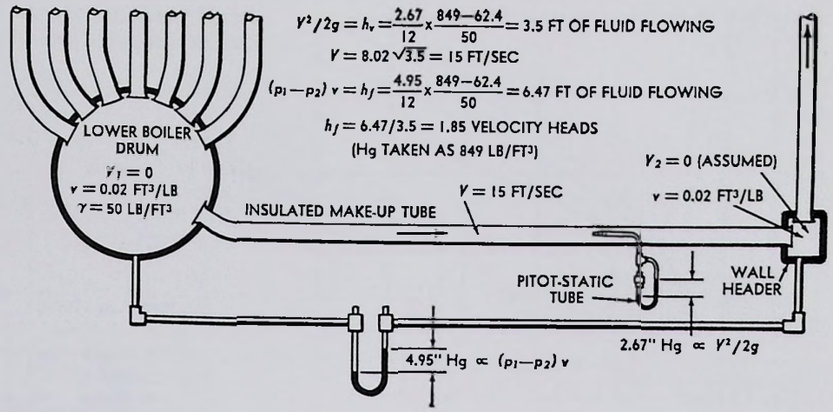


Fig. 1. *Determination of flow velocity and friction losses, in terms of velocity heads, in a simple boiler circuit*

Flow from well-formed nozzles and thin-plate orifices closely approaches the frictionless condition (constant entropy). Flow through rough pipes or labyrinthine channels, such as turbine shaft seals, is far removed from the frictionless condition.

Loss from Friction

To express the effect of friction on a nonexpansive fluid (or where the change in specific volume, v, is negligible) flowing between two points at the same elevation, equation (1) becomes:

$p_1v + V_1^2/2g = p_2v + V_2^2/2g + (u_2 - u_1) - {}_1Q_2$ (13)

where: $u_2 - u_1 - {}_1Q_2$ = a "loss," the result of friction

or $p_1v + V_1^2/2g = p_2v + V_2^2/2g + h_f$ (14)

and $h_f = (p_1 - p_2)v + (V_1^2 - V_2^2)/2g$ (15)

where: $h_f = u_2 - u_1 - {}_1Q_2$ = head loss from friction, ft of fluid flowing

If $V_1 = 0$, as in the isothermal flow of fluid from a large reservoir into a pipe, equation (15) becomes:

$h_f = (p_1 - p_2)v - V_2^2/2g$, ft of fluid flowing (16)

If $V_1 = V_2$, as in the isothermal flow of a fluid between two stations of a pipe of constant cross section, then:

$h_f = (p_1 - p_2)v$, ft of fluid flowing (17)

Since head loss from friction is a function of velocity squared, as given in equation (38), page 8-9, it is frequently expressed as a loss of so many velocity heads (head loss from friction ÷ the velocity head, $V^2/2g$, in the conduit).

The simple example in Fig. 1 will serve to demonstrate a typical application of equation (17) where the over-all friction loss, including entrance and exit losses, is to be determined for a portion of a flow circuit in which the elevation is the same, $Z_1 = Z_2$, and the change in specific volume is negligible.

The velocity head, $V^2/2g = h_v$, ft of fluid flowing, for the tube in the flow circuit is determined by the proper conversion of the reading of a manometer connected to a pitot-static tube suitably inserted in the stream of the moving fluid.

To find the friction loss, the pressure drop, $(p_1 - p_2)v$, is determined by a proper conversion of the reading of a U-tube manometer connected to the drum and the header to indicate the difference in static pressure between the two locations.

Volume and Weight Flow

The velocity of a free-spouting jet, as determined by the use of equation (5), (6), or (9), can be combined with the cross-sectional area of the flow path (conduit, nozzle, or aperture) to give the continuity equations for volume and weight flow:

$Q = AV$ (18)

$w = AV\gamma$ (19)

where:

Q = volume flow, cu ft/sec

A = cross-sectional flow area, sq ft

V = flow velocity, ft/sec

w = weight flow, lb/sec

γ = specific weight of fluid, lb/cu ft, the reciprocal of specific volume

Equations (9), (18), and (19) applied to cold water flowing through a nozzle of 1 sq in. area give the ideal flow velocities and quantities shown in Fig. 2. The flow velocity, the volume flow, and the weight flow all increase as the square root of the head.

With an expansive fluid, such as steam or air, the flow relationships are not as simple as for water. The results seem to be contrary to expectation. It is possible rationally to interpret the observed facts only after careful analysis and study. The complication stems from the changes in specific volume and temperature which accompany a drop in pressure and an increase in velocity. The flow relationships of an expansive fluid may be demonstrated by considering a convergent nozzle in which the initial pressure,

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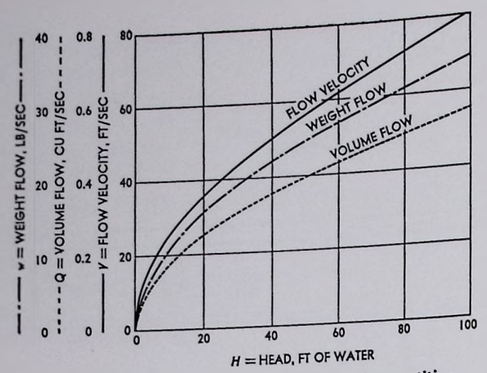


Fig. 2. *Ideal water flow velocities and quantities through a 1 sq in. (throat) nozzle*

p_1, is constant and the back pressure, p_2, is variable. In this case, for dry saturated steam where the initial pressure is fixed as 100 psia and the back pressure varies from 100 psia downward, the ideal weight flow is as shown in Fig. 3.

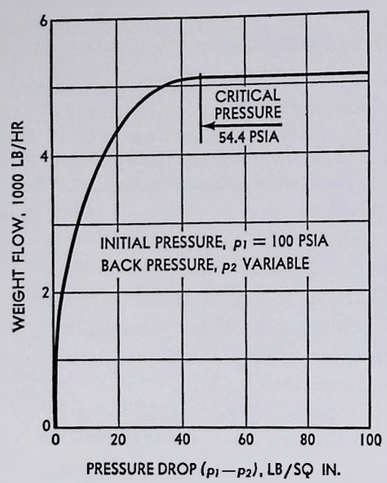


Fig. 3. *Ideal weight flow of dry saturated steam through a 1 sq in. (throat) nozzle*

The unique feature of this graph is the rapid increase in the weight flow as the pressure difference increases until it reaches a value of 45.6 psia. The weight flow does not increase for any further decrease in the back pressure, i.e., for any further increase in the pressure drop across the nozzle. This limitation is a critical condition characteristic of expansive fluids and is most conveniently measured in terms of a critical pressure, p_c. This critical pressure is usually expressed as a fraction of the initial pressure, p_1, and has a value between 0.5 and 0.6 for all gases and vapors. For fixed (perfect) gases it can be computed from the equation:

$$\text{Critical pressure ratio, } \frac{p_c}{p_1} = \left\{\frac{2}{\kappa+1}\right\}^{\frac{\kappa}{\kappa-1}} \qquad (20)$$

Values for κ and the critical pressure ratio are given in Table 1.

TABLE 1

Substance	κ	p_c/p_1
Air	1.40	0.528
Superheated steam		
$p_1 =$ 100 psia, $t_1 =$ 400 F	1.31	0.544
$p_1 =$ 500 psia, $t_1 =$ 900 F	1.28	0.549
$p_1 =$ 1000 psia, $t_1 =$ 1000 F $p_1 =$ 1500 psia, $t_1 =$ 1050 F	1.27	0.551
Dry saturated steam		
$p_1 =$ 100 psia	1.31	0.544
$p_1 =$ 500 psia	1.28	0.549
$p_1 =$ 1000 psia	1.26	0.553

The maximum weight flow is fixed by the acoustic velocity developed in the throat of the nozzle and the density of the fluid at the critical condition. This critical phenomenon is illustrated graphically in Fig. 4. If the initial pressure, p_1, is constant on the nozzle and the back pressure, p_2, is variable, the typical graph for weight flow is Fig. 4a like Fig. 3. If the back pressure, p_2, is constant and the initial pressure, p_1, is variable, the typical graph is Fig. 4b.

Where the initial pressure, p_1, is constant and the back pressure, p_2, is variable (Fig. 4a), it will be noted that, as the back pressure decreases, the weight flow increases rapidly, reaching a maximum when the back pressure is equal to the critical pressure, p_c, i.e., when p_2/p_1 is equal to the critical pressure ratio, p_c/p_1. Any further reduction in back pressure will not increase the flow rate, since the critical pressure, p_c, a constant for this condition, serves as a barrier and limits expansion in the nozzle to that pressure. The weight flow from this point to the lowest value of the back pressure is therefore constant.

Where the initial pressure, p_1, is variable and the back pressure, p_2, is constant (Fig. 4b), the critical pressure, p_c, increases as the initial pressure increases, since the critical pressure ratio, p_c/p_1, is a constant. Up to and including the point where the critical pressure, p_c, becomes equal to the constant back pressure, p_2, i.e., where $p_c/p_1 = p_2/p_1$, the velocity for calculating weight flow is based on complete expansion from the initial pressure, p_1, to the constant back pressure, p_2. From this point, as the initial pressure increases, the corresponding increase in the critical pressure limits the expansion from the initial pressure

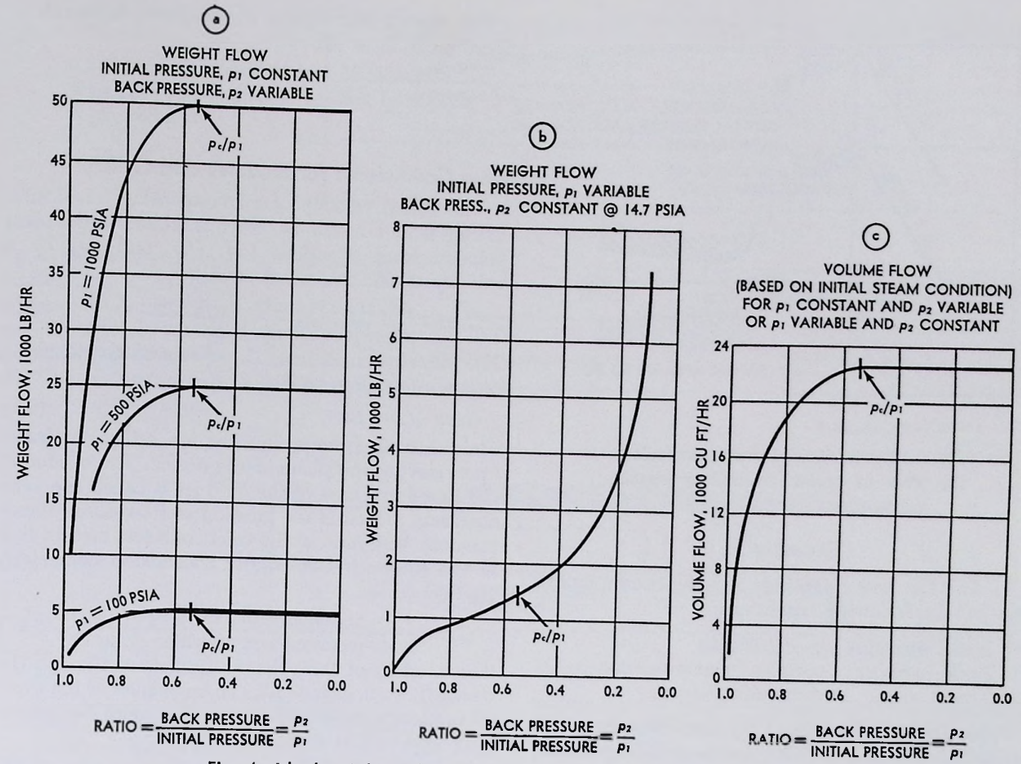


Fig. 4. *Ideal weight and volume flows of dry saturated steam through a 1 sq in. (throat) nozzle for various initial and back pressures*

to the critical pressure, and thus, while the weight flow continues to increase rapidly, the critical pressures serve as barriers to complete expansion to the back pressure.

It is interesting to note that the volume flow at the various ratios of p_2/p_1 (Fig. 4c) is constant for all initial pressures if the critical pressure ratio is rounded out at an average (see Table 1) for the gas concerned. The difference, using the particular values of p_c/p_1 for various values of p_1, is negligible. Also, there is no change in volume flow at various ratios of p_2/p_1 whether the initial pressure is constant and the back pressure is variable or the reverse. In other words, for the various conditions of weight flow shown, the volume flow does not change for the same size nozzle opening.

Because of the limitation on weight flow through nozzles with large pressure drops and since the acoustic velocity in a gaseous substance does not change materially with pressure but does with temperature, it is possible and convenient to develop simple rational equations for the approximate weight flow, w, in lb/sec through a 1 sq in. nozzle, as follows:

Dry saturated steam, Grashof's formula:

$$w = p_1^{0.97}/60 \qquad (21)$$

Superheated steam, modified Grashof formula:

$$w = p_1^{0.97}/60\ (1 + 0.00065\ \text{F superheat}) \qquad (22)$$

Wet steam, modified Grashof formula:

$$w = p_1^{0.97}/60\ \sqrt{\text{dryness fraction}} \qquad (23)$$

Air, with $p_1 > 2$ atmos and $p_2 \leqq 1$ atmos, Fliegner's formula:

$$w = p_1/1.9\ \sqrt{T_1} \qquad (24)$$

where:

p_1 = initial pressure, psia

T_1 = initial temp, deg R = F + 460

The velocity for determining the weight flow of steam at the critical condition may be found by using the Mollier chart (Appendix, page 10-A6) in conjunction with equation (5), as illustrated by Example 2 where dry saturated steam expands from 100 psia to atmosphere through a converging-diverging nozzle having a throat area of 1 sq in. A cross section of the nozzle and a trace of the flow process on the Mollier chart are shown by Fig. 5.

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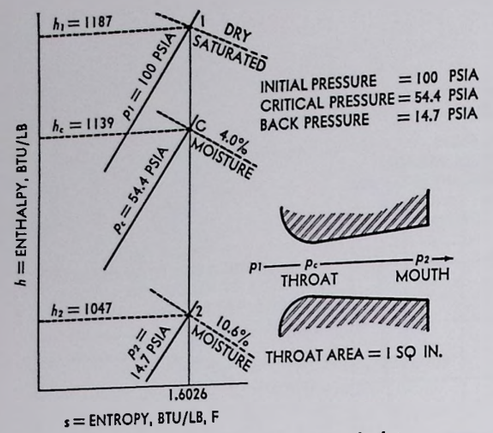


Fig. 5. Use of Mollier chart to calculate the flow of steam through a nozzle

Example 2

To find the flow quantities and the nozzle mouth area for the following conditions:

Initial pressure, $p_1 = 100$ psia
Back pressure, $p_2 = 14.7$ psia
Throat area, $A = 0.00695$ sq ft

From Table 1, the critical pressure ratio, p_c/p_1, for dry saturated steam is 0.544, so the critical pressure, p_c, for this example is 54.4 psia.

From the Mollier chart:

Initial enthalpy, $h_1 = 1187$ Btu/lb
Throat enthalpy, $h_c = 1139$ Btu/lb
Mouth enthalpy, $h_2 = 1047$ Btu/lb
all at constant entropy, $s = 1.6026$ Btu/lb, F
Moisture @ throat $= 4.0\%$
Sp vol @ throat $= 7.868\ (1.000 - 0.040)$
$= 7.553$ cu ft/lb
Moisture @ mouth $= 10.6\%$
Sp vol @ mouth $= 26.8\ (1.000 - 0.106)$
$= 23.96$ cu ft/lb
Throat velocity, $V_c = 223.9\sqrt{1187 - 1139}$
$= 1550$ ft/sec, by Eq (5)
Mouth velocity, $V_2 = 223.9\sqrt{1187 - 1047}$
$= 2649$ ft/sec
Vol flow @ throat $= 0.00695 \times 1550$
$= 10.77$ cu ft/sec, by Eq (18)
Wt flow @ throat $= 10.77/7.553$
$= 1.426$ lb/sec, by Eq (19)

Check by the simple rational equation (21):
Wt flow $= 100^{0.97}/60 = 87/60 = 1.45$ lb/sec

For the specific volume of the steam in mouth @ 23.96 cu ft/lb as above:

$1.426 \times 23.96 = A \times 2649$, by Eq (18)
Required area of mouth, $A = 0.0129$ sq ft
$= 1.858$ sq in.

Flow Coefficients for Nozzles and Orifices

The actual velocity of a free-spouting jet of fluid issuing from a nozzle or orifice is related to the ideal velocity from equations (5), (6), and (9) by a velocity coefficient, C_v, defined as:

$$C_v = \frac{\text{actual jet velocity, } V_a, \text{ ft/sec}}{\text{ideal jet velocity, } V_i, \text{ ft/sec}} \qquad (25)$$

The velocity coefficient, C_v, is related to the flow-resistance or drag coefficient, C_d, by:

$$C_v = \sqrt{1 - C_d} \qquad (26)$$

Fluid issuing from a square-edge orifice contracts upon leaving the plane of the orifice. The minimum cross-sectional area of the fluid jet is called the vena contracta, which is the location of jet-velocity measurement. The ratio of the vena contracta area to that of the orifice is termed the contraction coefficient, C_c, defined as:

$$C_c = \frac{\text{actual area of jet, } A_a, \text{ sq ft}}{\text{cross-sectional area of orifice, } A, \text{ sq ft}} \qquad (27)$$

The product of the velocity coefficient, C_v, and the contraction coefficient, C_c, can be defined as the over-all discharge coefficient, C_q (or C_{qo} where the velocity of approach is zero), expressed as:

$$C_q \text{ (or } C_{qo}) = C_v C_c \qquad (28)$$

and the actual flow = ideal flow $\times$ C_q (or C_{qo}), or:

$$Q = C_q A V \text{ or } C_{qo} A V \qquad (29)$$

Numerical values of C_q and C_{qo} are given in Fig. 6, derived from data published in ASME research publication *Fluid Meters* (1937). Values for over-all discharge coefficients are in general determined from actual test flow measurements and are related to the Reynolds number. Values of C_v and C_c are not easily determined by direct measurement. However, the square-edge thin-plate orifice, the bellmouthed streamlined flow nozzle, and the venturi nozzle all offer very little resistance to flow, and the value of C_v is usually between 0.98 and 0.99. The relatively low discharge coefficient of the orifice reflects a low contraction coefficient (about 0.62).

The two examples below show the use of the above data in calculating the flow through orifices and nozzles.

Example 3

Water under 100-ft head flows through a 1 sq in. aperture in (a) a square-edge orifice and (b) a bellmouthed flow nozzle. Velocity of approach is zero. To find the velocity and the volume of discharge in each case:

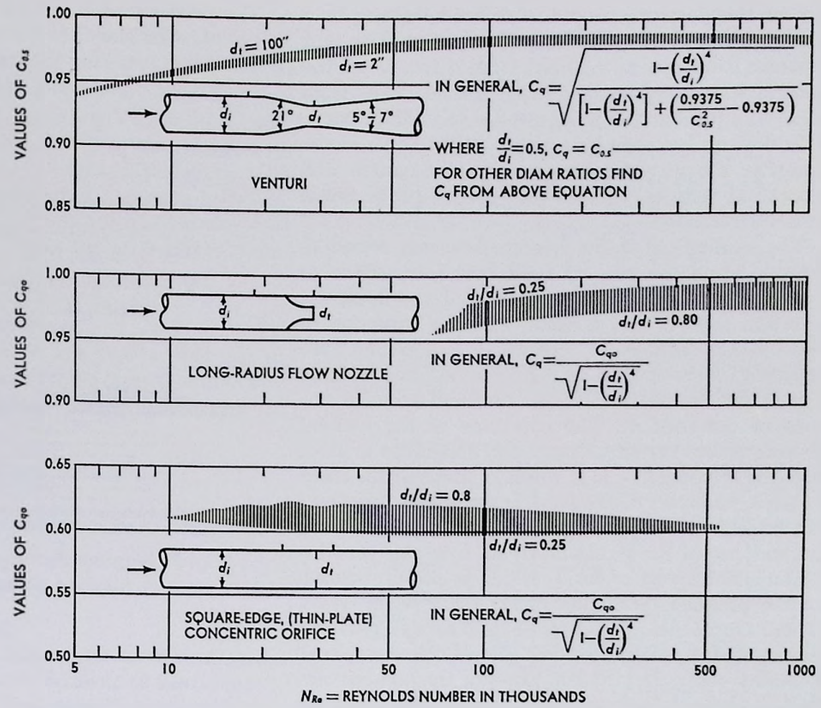


Fig. 6. Values of discharge coefficients, C_{qo} (velocity of approach not included) and C_q (velocity of approach included), for typical venturi, flow nozzle, and orifice in pipe

(a) Square-edge orifice

Ideal velocity, $V_i = 8.02\sqrt{100} = 80.2$ ft/sec, from Eq (9)
Discharge coefficient, $C_{qo} = 0.6$ (approx), from Fig. 6
Actual flow, $Q = 0.6 \times (1/144) \times 80.2$
$= 0.334$ cu ft/sec, from Eq (29)
Velocity coefficient, $C_v = 0.98$
Actual jet velocity, $V_a = 0.98 \times 80.2$
$= 78.6$ ft/sec

(b) Bellmouthed flow nozzle

Ideal velocity, $V_i = 8.02\sqrt{100} = 80.2$ ft/sec
Discharge coefficient, $C_{qo} = 0.98$ (approx), from Fig. 6
Actual flow, $Q = 0.98 \times (1/144) \times 80.2$
$= 0.547$ cu ft/sec
Velocity coefficient, $C_v = 0.99$
Actual jet velocity, $V_a = 0.99 \times 80.2$
$= 79.4$ ft/sec

If the orifice, nozzle, or venturi is installed in a pipe, the velocity of approach must be considered, and the over-all discharge coefficient, C_q, should be used instead of C_{qo}.

Example 4

Dry saturated steam at 100 psi is escaping from a safety valve where the diameter is 3 in. and the lift is ¾ in. The discharge coefficient, C_q, is 0.74 by test on this type of valve. To find the relieving capacity of the safety valve:

Area of the annulus $= 3 \times 3.14 \times \frac{3}{4} = 3.53$ sq in.
Ideal flow $= 3.53 \times 114.7^{0.97}/60 = 5.86$ lb/sec, from Eq (21)
Actual flow $= 0.74 \times 5.86 = 4.34$ lb/sec, from Eq (29)
$= 15{,}620$ lb/hr relieving capacity

Resistance to Flow in Pipes and Ducts

The flow of steam and water through tubes and pipes, the flow of air and gases through ducts and flues, the flow of fluids through valves and fittings, and the flow of fluids through and across tube banks are all problems of immediate concern to the boiler designer and the boiler user. These are all classified as flow through closed-conduit systems. The determination and correlation of rational data for application to theoretically sound equations for the use of engineers in the various fields of fluid dynamics are a major part of the experimental work conducted by hydraulic, aerodynamic, and many other research laboratories. The purpose of the following discussion is to present the experimental data that can be reliably and expeditiously used in the solution of boiler fluid-flow problems.

Perhaps the most important problem is the evaluation of the resistance to fluid motion, i.e., the friction

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loss or the pressure drop of the fluid flowing in a conduit. The distinction between friction loss and pressure drop can be realized from a review of the general energy equation (1) and the head loss from friction, equation (17). It can be seen that, for a fluid flowing between two points at the same elevation with a negligible change in specific volume and a constant velocity, the pressure drop is proportional to the friction loss.

The evaluation of the friction loss was found to depend upon the type of fluid flow (streamline or turbulent), the physical properties of the fluid, the size and form of the conduit, and the condition of the conduit surface. In streamline or laminar flow, adjacent layers of fluid slide over each other with no slip at the boundary between the fluid and the surface of the conduit. The condition of the surface, therefore, has no effect upon the resistance to flow, and the friction loss is a result of the viscous shear of adjacent layers of the fluid. In turbulent flow there is an irregular motion in the fluid, characterized by an eddying of the fluid and by an irregular shearing of adjacent layers of fluid, which is erratic because of the interchange of fluid particles between layers. There are no definite layers of fluid except, perhaps, at the surface of the conduit. This layer is so small that any change in the condition of the conduit surface, such as an increase in roughness, does affect the resistance to flow. The friction loss for turbulent flow is therefore the result of the random viscous shear between fluid layers, the more consistent shear near the surface of the conduit, and the impact of the fluid particles caused by eddy formation in the fluid.

The significance of viscous shear can be illustrated as in Fig. 7, where two flat plates are separated by a layer of fluid of thickness y. If the bottom plate is stationary and the top plate moves at a velocity V, the fluid layer in contact with the top plate, because of adhesion, moves with the same velocity as the plate, and the fluid layer in contact with the bottom plate, for the same reason, is stationary. At intermediate distances between the two plates, the velocity is proportional to the distance from the bottom plate. Because of internal fluid friction, each layer of fluid exerts a drag on the more rapidly moving layer above it, and the topmost layer exerts a drag on the moving plate. A force is therefore required to maintain the motion of the top plate. This force is equal to the drag, or frictional resistance, which can be expressed as:

$$F = \tau S \tag{30}$$

where:

F = force, lb
τ = shear stress at surface, lb/sq ft
S = plate or wall surface area at which the force acts, sq ft

The property of the fluid called viscosity is a measure of its resistance to the shearing of adjacent layers. The force opposing the movement of adjacent layers or of the top plate can, therefore, be expressed as:

$$F = \mu SV/y \tag{31}$$

where:

F = force, lb
μ = absolute viscosity, lb (force), hr/sq ft
S = plate surface area, sq ft
V = velocity, ft/hr
y = distance between stationary and moving plates, ft

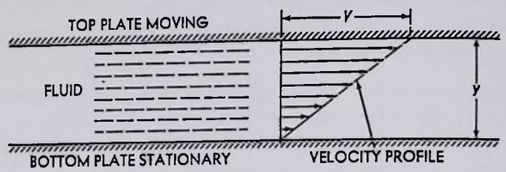


Fig. 7. *Diagram illustrating viscous drag reaction between adjacent fluid layers*

The above explanation of frictional resistance is applicable to a closed conduit, such as a pipe or duct, where the stationary surface or walls of the conduit can be considered to exert a drag or resistance on the fluid flowing inside the conduit. Experimental results have shown the resistance force can be expressed as:

$$F = f'S\gamma V^2/2g \tag{32}$$

where:

F = force, lb
f' = friction factor
S = surface area, sq ft
γ = specific weight of fluid, lb/cu ft
V = velocity, ft/sec
g = gravitational acceleration, ft/sec^2

A more useful form of equation (32) can be resolved by considering the components of S in a closed conduit of a constant cross section, that is:

$$S = PL$$

where:

S = surface area, sq ft
P = contact perimeter of conduit, ft
L = length of conduit, ft

and:

$$F = f'PL\gamma V^2/2g \tag{33}$$

As mentioned above, resistance to flow or friction loss can be synonymous with pressure drop, and in most cases it is better to express the loss as a pressure drop in either lb/sq ft or lb/sq in. For a conduit with a cross-sectional area, A, in sq ft, the pressure drop in lb/sq ft becomes:

$$\Delta p = f'PL\gamma V^2/2gA$$

or:

$$\Delta p = f'L\gamma V^2/2g(A/P) \tag{34}$$

For consistent evaluation of the pressure drop and the friction factor, it is necessary to establish some characteristic length term that is applicable to both circular and noncircular conduits. The length term satisfying this need is four times the mean hydraulic radius and is known as the equivalent diameter (see Chapter 7, page 7-11). The hydraulic radius of a conduit is defined as:

$$\frac{A}{P} = \frac{\text{cross-sectional flow area}}{\text{contact perimeter}} \tag{35}$$

In a circular conduit, such as a pipe or duct, having an internal diameter, D_i, in ft, the cross-sectional flow area is $\pi D_i^2/4$, the contact perimeter is πD_i, and the equivalent diameter, D_e, in ft is:

$$D_e = 4A/P = 4\,\frac{\pi D_i^2}{4}\,\frac{1}{\pi D_i} = D_i$$

Thus $A/P = D_i/4$, and equation (34) for a circular conduit becomes:

$$\Delta p = 4f'L\gamma V^2/2gD_i \tag{36}$$

or expressed in lb/sq in.:

$$\Delta p = 4f'L\gamma V^2/288gD_i \tag{37}$$

In some cases it may be convenient to express the friction loss as a head loss, h_f, in ft of fluid flowing. This is obtained from equation (36) divided by the specific weight, as:

$$h_f = 4f'L\gamma V^2/2gD_i\gamma$$

or:

$$h_f = 4f'LV^2/2gD_i \tag{38}$$

Equations (36), (37), and (38) can be applied to noncircular conduits by substituting D_e for D_i.

The several equations given above for determining fluid friction loss are variously attributed to Chezy, Darcy, or Fanning. Careful attention is required in the selection of the proper numerical value for the friction factor, f'. It has been found experimentally that the most rational base for evaluating the friction factor is by the use of the Reynolds number, N_{Re}, criterion for defining the flow pattern, and the ϵ/d_e function for defining the relative roughness of the conduit surface. Experimental data on friction factors using this basis were correlated by Moody, as shown in Figs. 8 and 9. The Moody friction factor, f, incorporates the "4" of the previous equations and is, therefore, four times the Fanning friction factor, f', in equations (36), (37), and (38). These equations and the quantities used for determining the friction factor can be rearranged for application to boiler calculations by using the following symbols and units:

Δp = pressure drop, psi
f = friction factor = $4f'$
L = developed length of flow path (incl bends), ft
v = specific volume, cu ft/lb
G = mass flow = w/A, lb/sq ft, hr (see page 11-22)
d_e = equiv diam of conduit, in.
V = fluid velocity, ft/hr
g = gravitational acceleration, ft/hr^2
μ = absolute viscosity, lb (mass)/ft, hr
N_{Re} = Reynolds number = $d_eG/12\mu$
ϵ/d_e = relative roughness of surface
ϵ = avg height of roughness projections, in.

When these definitions are substituted in equation (37), the equation for pressure drop becomes:

$$\Delta p = f\,Lv\,(G \times 10^{-5})^2/d_e, \text{ lb/sq in.} \tag{39}$$

The friction factor, f, for various conduits is obtained from Fig. 9, using the relative roughness value,

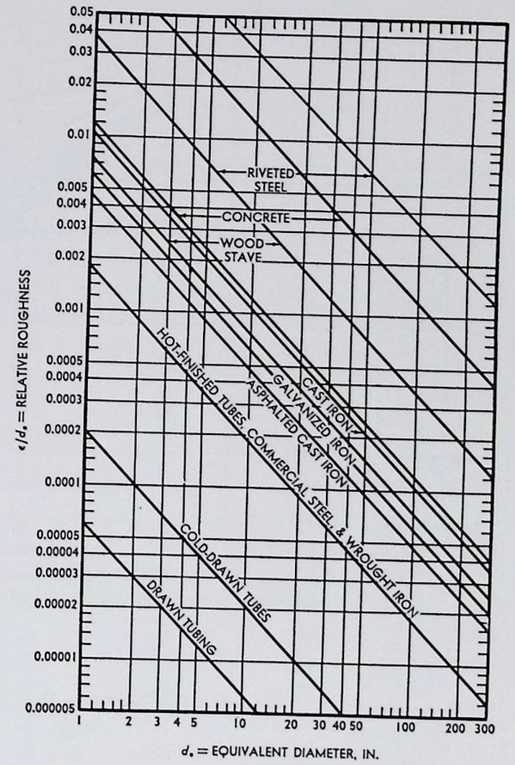


Fig. 8. *Relative roughness of various conduit surfaces*

ϵ/d_e, from Fig. 8. From a practical standpoint, the increase in rugosity with age should be considered when designing or establishing the power required to overcome the maximum probable losses.

Realistic values of flow velocity, based on experience, for an assortment of piping and duct systems are given in Table 2. The specific weights and volumes of liquids and gases in Tables 3 and 4, respec-

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tively, and the steam tables in the Appendix, page 10-A1 ff, are useful in calculating mass flows.

A component of the Reynolds number, N_{Re}, is the viscosity of the fluid flowing (see page 7-9). Some numerical values of absolute viscosity for selected liquids, gases, and vapors are given in Figs. 10, 11, and 12, respectively. The independent variable in all of these plots is temperature, and it will be noted that there is a very material change in viscosity with change in temperature. For liquids (Fig. 10) the

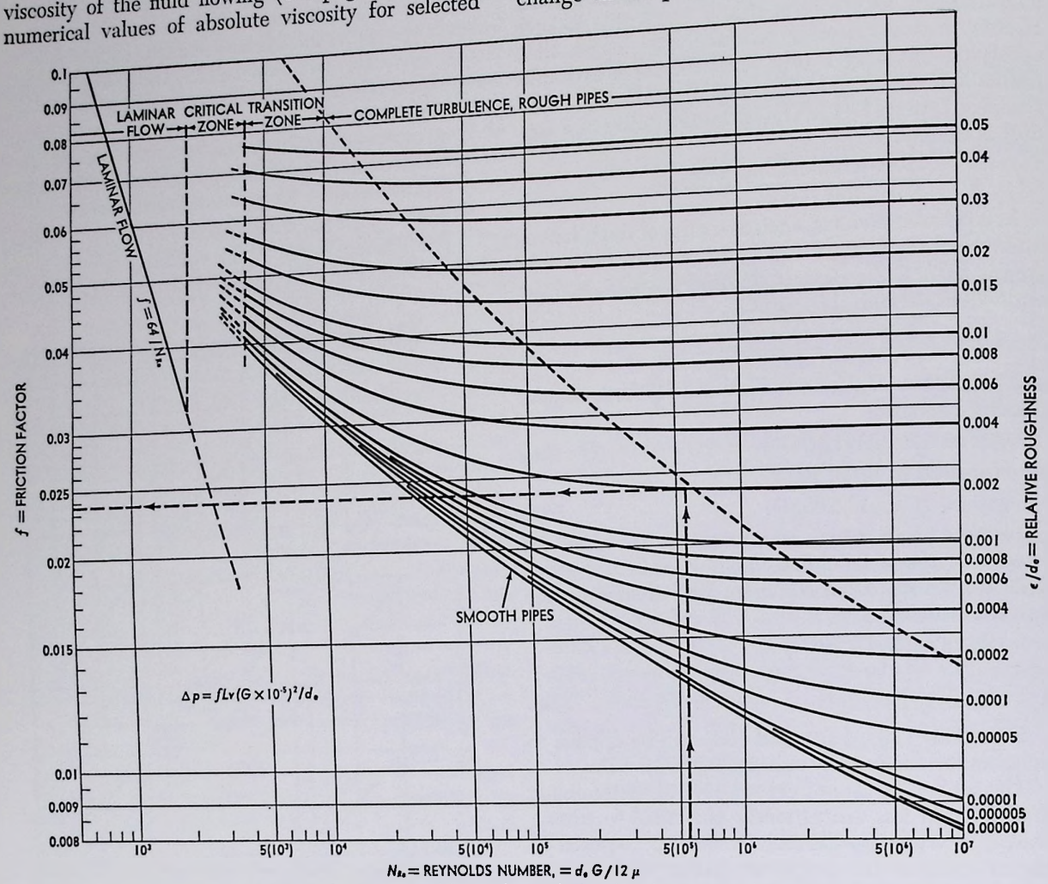


Fig. 9. *Friction-factor Reynolds-number relationship for use in determining pressure drop of fluids flowing through closed conduits (pipes and ducts)*

TABLE 2

Range of Allowable Velocities in Duct and Piping Systems

Nature of Service	Velocity, ft/min	Nature of Service	Velocity, ft/min
Steam lines:		Stacks and chimneys	2,000-5,000
High pressure	8,000-12,000	Boiler gas passes°	3,000-6,000
Low pressure	12,000-15,000	Coal-and-air lines, pulverized coal	3,000-4,500
Vacuum	20,000-40,000	Air heater, gas and air sides	1,000-5,000
Superheater tubes, steam	2,000-5,000	Ventilating ducts	1,000-3,000
Compressed-air lines	1,500-2,000	Register grills	300-600
Natural-gas lines (large transcontinent)	1,000-1,500	Water lines	500-750
Forced-draft ducts	1,500-3,600	Economizer tubes, water	150-300
Forced-draft ducts, entrance to burners	1,500-2,000	Crude-oil lines (6″ to 30″)	60-360
Induced-draft flues and breeching	2,000-3,500	Boiler circulation	70-700

° See also Tables 3 and 4, Chapter 18.

TABLE 3

Physical Properties of Liquids
(at 14.7 psia)

Liquid	Temp, F	Sp Wt, lb/cu ft	Sp Vol, cu ft/lb	Sp Ht, c_p
Water	70	62.4	0.0160	1.000
	212	59.9	0.0167	1.000
Automotive oil	70			
SAE 10		55-57	0.0182-0.0175	0.435
SAE 50		57-59	0.0175-0.0170	0.425
Mercury	70	846	0.0018	0.033
Fuel oil, #6	70	60-65	0.0167-0.0154	0.40
	180	60-65	0.0167-0.0154	0.46
Kerosene	70	50-51	0.0200-0.0196	0.47

TABLE 4

Physical Properties of Gases
(at 14.7 psia)

Gas	Temp, F	Sp Wt, lb/cu ft	Sp Vol, cu ft/lb	Inst Sp Ht, c_p	Inst Sp Ht, c_v	κ, c_p/c_v
Air	70	0.0751	13.32	0.241	0.172	1.40
	200	0.0603	16.58	0.242	0.173	1.40
	500	0.0415	24.10	0.248	0.180	1.38
	1000	0.0273	36.63	0.265	0.197	1.34
CO_2	70	0.1148	8.71	0.202	0.155	1.30
	200	0.0922	10.85	0.216	0.170	1.27
	500	0.0634	15.77	0.247	0.202	1.22
	1000	0.0417	23.98	0.280	0.235	1.19
H_2	70	0.0052	191.2	3.440	2.440	1.41
	200	0.0042	238.0	3.480	2.490	1.40
	500	0.0029	345.0	3.500	2.515	1.39
	1000	0.0019	526.3	3.540	2.560	1.38
Flue gas°	70	0.0836	11.96	0.253	0.187	1.35
	200	0.0776	12.88	0.255	0.189	1.35
	500	0.0428	23.38	0.265	0.199	1.33
	1000	0.0282	35.48	0.283	0.217	1.30
CH_4	70	0.0416	24.05	0.530	0.406	1.30
	200	0.0334	29.95	0.575	0.451	1.27
	500	0.0230	43.50	0.720	0.596	1.21
	1000	0.0151	66.22	0.960	0.836	1.15

° From coal, 120% total air. Flue gas molecular weight taken as 30.

fluid viscosity decreases substantially as the temperature is raised. On the other hand, for gases and vapors (Figs. 11 and 12) there is an increase in viscosity as the fluid temperature increases. These opposite effects are the consequence of cohesion and diffusivity in fluids. As the temperature of the liquid increases, the cohesive forces are lowered more than the diffusivity is increased, and consequently the viscosity decreases. For gases and vapors cohesive forces are relatively insignificant, and the viscosity

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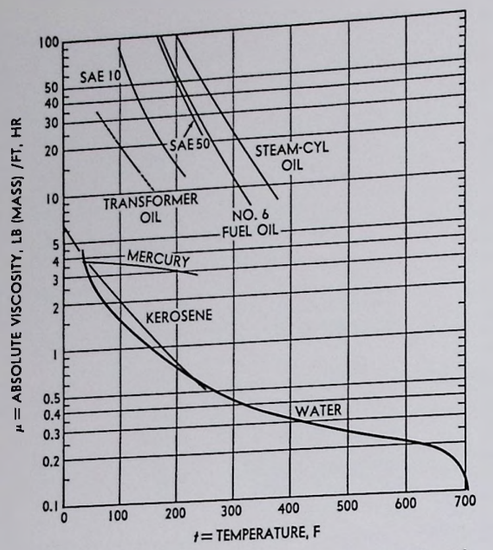


Fig. 10. *Absolute viscosities of some common liquids*

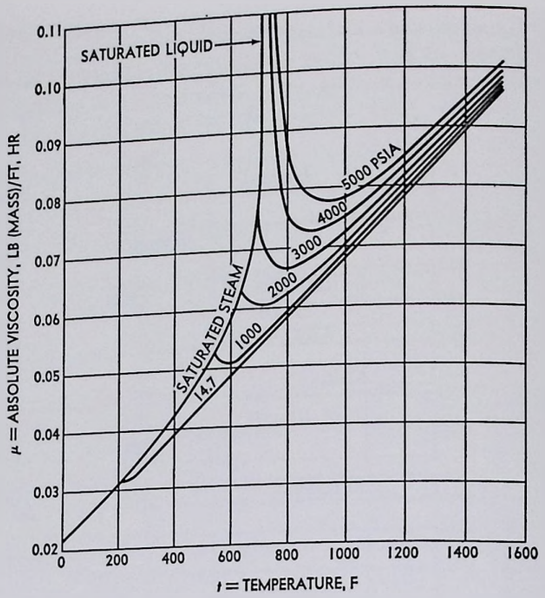


Fig. 12. *Absolute viscosity of saturated and superheated steam*

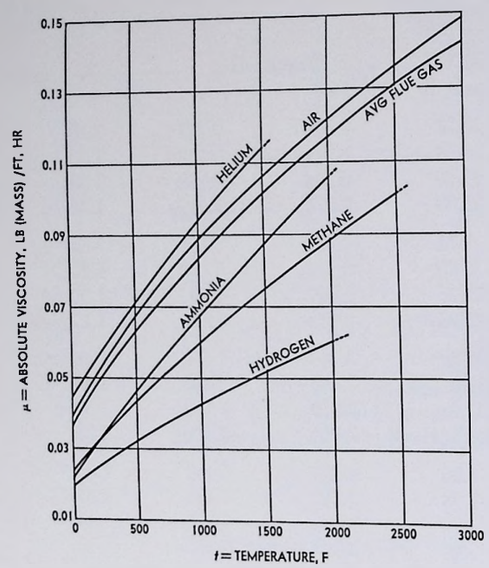


Fig. 11. *Absolute viscosities of some common gases at atmospheric pressure*

increases with rising temperature because of the increased diffusivity. That there is also some change in viscosity with pressure is indicated by Fig. 12 for steam. The relationship between various units of viscosity is given in Table 5.

Resistance to Flow in Valves and Fittings

Pipe lines and duct systems contain many valves and fittings to make a working entity of the assembly. Unless the lines are used to transport fluids over long distances, as in the distribution of process steam at a factory or the cross-country transmission of oil or gas, the straight runs of pipe or duct are relatively short. Usually the water, steam, air, and gas lines in a power plant have relatively short runs of straight pipe and many valves and fittings. Consequently the resistance to flow offered by the valves and fittings is a large, if not major, part of the total resistance. The methods outlined above are helpful in the exact evaluation of fluid pressure for straight-run pipes and ducts. The data available for estimating the resistance to flow in valves and fittings are much less exact, since basic difficulties practically preclude the correlation of experimental data on a rational basis. These losses are furnished by the manufacturer, expressed either as velocity heads or as equivalent lengths of straight pipe. The latter are calculated by equation (39), the former by equation (40) below, which is equation (8) converted to psi units:

$$\Delta p = Nv(G \times 10^{-5})^2/12, \text{ lb/sq in.} \qquad (40)$$

where:

N = number of velocity heads
v = sp vol of fluid, cu ft/lb
$G = w/A$ = mass flow, lb/sq ft, hr

The range in pressure drop through an assortment of commercial fittings is given in Table 6 in equivalent velocity heads. As noted, pressure drop through fittings may also be expressed as the loss in equivalent lengths of straight pipe.

TABLE 5

Relationship between various Units of Viscosity

Absolute (Dynamic) Viscosity Values — Metric Units: Centipoise, 0.01 poise	Metric Units: Poise, g(mass)/cm, sec	English Units: lb(mass)/ft, sec	English Units: lb(mass)/ft, hr	English Units: lb(force), sec/sq ft	Kinematic Viscosity Values — Metric Units: Centistoke, 0.01 stoke	Metric Units: Stoke, sq cm/sec	English Units: sq ft/sec
1.0	0.01	0.000672	2.42	0.0000209	$62.4/\gamma$	$0.624/\gamma$	$0.000672/\gamma$
100	**1.0**	0.0672	242	0.00209	$6{,}240/\gamma$	$62.4/\gamma$	$0.0672/\gamma$
1,488	14.88	**1.0**	3,600	0.0311	$92{,}900/\gamma$	$929/\gamma$	$1/\gamma$
0.413	0.00413	0.000278	**1.0**	0.0000086	$25.8/\gamma$	$0.258/\gamma$	$0.000278/\gamma$
47,900	479	32.2	115,900	**1.0**	$2{,}990{,}000/\gamma$	$29{,}900/\gamma$	$32.2/\gamma$
0.016γ	0.00016γ	0.000011γ	0.0396γ	0.00000034γ	**1.0**	0.01	0.0000108
1.6γ	0.016γ	0.0011γ	3.96γ	0.000034γ	100	**1.0**	0.00108
$1{,}490\gamma$	14.9γ	1.02γ	$3{,}680\gamma$	0.032γ	92,900	929	**1.0**

γ = specific weight, lb/cu ft

TABLE 6

Resistance to Flow of Fluids through Commercial Fittings*

Fitting	Loss in Velocity Heads
Ell, 90° standard sweep	0.3-0.7
Ell, 90° long sweep	0.2-0.5
Tee, flow through run	0.15-0.5
Tee, flow through 90° branch	0.6-1.6
Return bend, close	0.6-1.7
Gate valve, open	0.1-0.2
Check valve, open	2.0-10.0
Globe valve, open	5.0-15.0
Angle valve, 90° open	3.0-7.0
Boiler nonreturn valve, open	1.0-3.0

*See Fig. 16 for loss in velocity heads for flow of fluids through pipe bends. See also "Losses in Pipe and Fittings," by R. J. S. Pigott, *ASME Transactions*, vol 79, 1957, pp 1767-1783.

The following examples show the use of the above data in estimating the pressure drop in a steam pipeline and in a forced-draft duct system:

Example 5

A steam pipe system of 100-ft developed length, one nonreturn valve, seven standard-sweep elbows, and one angle valve must deliver 10,000 lb/hr of dry saturated steam @ 100 psi. To find:

1. Nearest commercial nominal pipe size, in.
2. The over-all pressure drop, lb/sq in.

Assume: $V = 12{,}000$ ft/min $= 200$ ft/sec (Table 2)
$\Delta p = 20$ lb/sq in.
Then: $v = 3.587$ cu ft/lb (Appendix, page 10-A2) for avg $p = 125$ psia

$Q = 10{,}000 \times 3.587/3600 = 9.96$ cu ft/sec
$A = 9.96/200 = 0.0498$ sq ft $= 7.18$ sq in.
$d_i = 3.003$ in.

Nearest nominal pipe size, 3 in., where $d_i = 3.068$ in.
$= 0.256$ ft (see Table 7, Schedule No. 40)
$A = 0.0515$ sq ft
$V = 9.96/0.0515 = 193.5$ ft/sec

Use equation (39) for pressure drop in straight pipe:

$\Delta p = fLv(G \times 10^{-5})^2/d_e$, lb/sq in.
$G = 10{,}000/0.0515 = 194{,}300$ lb/sq ft, hr
$d_e = 3.068$ in.
$\mu = 0.0405$ lb (mass)/ft, hr (from Fig. 12 for sat. steam @ 125 psia)

$$N_{Re} = \frac{dG}{12\mu} = \frac{3.068 \times 194{,}300}{12 \times 0.0405} = 1{,}227{,}000$$

$\epsilon/d_e = 0.00059$, for $d_e = 3.068$ and commercial steel (Fig. 8)
$f = 0.0177$, for $\epsilon/d_e = 0.00059$ and $N_{Re} = 1{,}227{,}000$ (Fig. 9)

For Δp in 100 ft of straight pipe:

$\Delta p = 0.0177$, for $\epsilon/d_e = 0.00059$ and $N_{Re} = 7.81$ lb/sq in.

Use equation (40) for loss in bends and fittings:

$\Delta p = Nv(G \times 10^{-5})^2/12$, lb/sq in.

Loss/nonreturn valve $= 2.0$ vel heads (Table 6)
Loss/elbow $= 0.5$ vel head (Table 6)
Loss/angle valve $= 5.0$ vel heads (Table 6)

$N = 2.0 + (7 \times 0.5) + 5.0 = 10.5$
$\Delta p = 10.5 \times 3.587 \times 1.943^2/12$
$= 11.85$ lb/sq in.

Over-all pressure drop for pipe system
$= 7.81 + 11.85 = 19.66$ lb/sq in.

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Example 6

A forced-draft duct (depth limited to 2 ft) of 100-ft developed length and four 90° bends with square outside corners and 2-ft radius inner corners (see Fig. 18b) must deliver 300 F air at a rate of 60,000 cfm. To find:

1. Dimensions of duct system.
2. The over-all draft loss, in. of water.

Assume: $V = 3600$ ft/min $= 60$ ft/sec (Table 2)

Then: $Q = 60{,}000/60 = 1000$ cu ft/sec

$A = 1000/60 = 16.7$ sq ft

Dimensions of duct system: 2 ft × 8.35 ft

$$d_e = 4 \times \frac{16.7 \times 144}{(2 \times 8.35 + 2 \times 2)\,12} = 38.7 \text{ in.}$$

Use equation (39) for pressure drop in straight duct:

$\triangle p = fLv(G \times 10^{-6})^2/d_e$ lb/sq in.

v for 70 F air $= 13.32$ cu ft/lb (Table 4)

v for 300 F air $= 13.32 \times 760/530$
$= 19.1$ cu ft/lb

$$G = \frac{60{,}000 \times 60}{19.1 \times 16.7} = 11{,}270 \text{ lb/sq ft, hr}$$

$d_e = 38.7$ in.

$\mu = 0.057$ lb (mass)/ft, hr (Fig. 11)

$$N_{Re} = \frac{dG}{12\mu} = \frac{38.7 \times 11{,}270}{12 \times 0.057} = 637{,}600$$

$\epsilon/d_e = 0.000046$, for $d_e = 38.7$
and commercial steel (Fig. 8)

$f = 0.0132$, for $\epsilon/d_e = 0.000046$
and $N_{Re} = 637{,}600$ (Fig. 9)

For $\triangle p$ in 100 ft of straight duct:

$\triangle p = 0.0132 \times 100 \times 19.1 \times 0.1127^2/38.7$
$= 0.00828$ lb/sq in. $= 0.23$ in. of water

TABLE 7

Nominal Wall Thickness of Welded and Seamless Steel Pipe, Inches

Nominal Pipe Size, in.	Outside Diam, in.	Schedule Number 10	20	30	40	60	80	100	120	140	160
⅛	0.405	—	—	—	**0.068**	—	**0.095**	—	—	—	—
¼	0.540	—	—	—	**0.088**	—	**0.119**	—	—	—	—
⅜	0.675	—	—	—	**0.091**	—	**0.126**	—	—	—	—
½	0.840	—	—	—	**0.109**	—	**0.147**	—	—	—	0.187
¾	1.050	—	—	—	**0.113**	—	**0.154**	—	—	—	0.218
1	1.315	—	—	—	**0.133**	—	**0.179**	—	—	—	0.250
1¼	1.660	—	—	—	**0.140**	—	**0.191**	—	—	—	0.250
1½	1.900	—	—	—	**0.145**	—	**0.200**	—	—	—	0.281
2	2.375	—	—	—	**0.154**	—	**0.218**	—	—	—	0.343
2½	2.875	—	—	—	**0.203**	—	**0.276**	—	—	—	0.375
3	3.5	—	—	—	**0.216**	—	**0.300**	—	—	—	0.438
3½	4.0	—	—	—	**0.226**	—	**0.318**	—	—	—	—
4	4.5	—	—	—	**0.237**	—	**0.337**	—	0.438	—	0.531
5	5.563	—	—	—	**0.258**	—	**0.375**	—	0.500	—	0.625
6	6.625	—	—	—	**0.280**	—	**0.432**	—	0.562	—	0.718
8	8.625	—	0.250	**0.277**	**0.322**	0.406	**0.500**	0.593	0.718	0.812	0.906
10	10.75	—	0.250	**0.307**	**0.365**	**0.500**	0.593	0.718	0.843	1.000	1.125
12	12.75	—	0.250	**0.330**	0.406	0.562	0.687	0.843	1.000	1.125	1.312
14	14.0	0.250	0.312	0.375	0.438	0.593	0.750	0.937	1.093	1.250	1.406
16	16.0	0.250	0.312	0.375	0.500	0.656	0.843	1.031	1.218	1.438	1.593
18	18.0	0.250	0.312	0.438	0.562	0.750	0.937	1.156	1.375	1.562	1.781
20	20.0	0.250	0.375	0.500	0.593	0.812	1.031	1.281	1.500	1.750	1.968
24	24.0	0.250	0.375	0.562	0.687	0.968	1.218	1.531	1.812	2.062	2.343
30	30.0	0.312	0.500	0.625	—	—	—	—	—	—	—

NOTE: Values in bold face in Schedules 30 and 40 are identical with the old "standard-weight" pipe; those in Schedules 60 and 80 are identical with the old "extra-strong" pipe.
Adapted from ASME *Boiler and Pressure Vessel Code*, section II, *Material Specifications*.

Use equation (40) for loss in bends:

$\triangle p = Nv(G \times 10^{-6})^2/12$, lb/sq in.
Width of duct, $b = 8.35$ ft
Depth of duct, $D = 2$ ft
Aspect ratio, $b/D = 8.35/2 = 4.17$
Curve ratio, $r_i/r_o = 2/4 = 0.5$
Loss/90° bend $= 0.18$ vel head (Fig. 18b)
$N = 4 \times 0.18 = 0.72$
$\triangle p = 0.72 \times 19.1 \times 0.1127^2/12$
$= 0.0146$ lb/sq in. $= 0.41$ in. of water

Over-all draft loss for duct system:

$0.23 + 0.41 = 0.64$ in. of water

Fluid Flow in Boilers and Tube Banks

The above data are usually adequate for estimating fluid-dynamic performance values in nozzles, orifices, piping, and duct systems. However, for estimating fluid-flow performance values in a boiler on the gas side, more exact and more elaborate data are generally desirable. While it is not within the scope of this chapter to provide an exhaustive review, the following data and equations will indicate the nature of the problem (see also Chapter 11). The problem is complicated by the presence of not only viscous or skin-friction losses but also losses from shock and impact caused by changes in the direction of flow and changes in the cross-sectional area available for flow.

The shock or impact losses are represented by the loss of energy from the formation of eddies and swirls when fluid particles of different velocities come into contact. Such losses can be conveniently expressed in terms of the velocity head of the flowing fluid. Another method in common use is to consider these losses as equivalent to the viscous loss in the flow through a certain length of straight conduit. The latter method is not as accurate as the former since the coefficient of friction for straight conduit resistance varies with the surface roughness and the Reynolds number and is inversely proportional to the fifth power of the diameter, whereas impact losses are inversely proportional to the fourth power of the diameter.

Effect of Contraction and Enlargement

The simplest sectional change in a conduit is where the boundaries converge. Converging boundaries have a tendency to steady the flow, and there is no shock loss when the pressure energy is converted to kinetic energy if the flow is constricted by a gradually converging section terminating tangentially in the smaller downstream section. If the included angle of the converging boundaries is about 30° or less and the terminal junction is smooth, the loss of energy is a result of the skin friction only and is approximately 0.05 velocity head based on the smaller downstream section. Identifying the upstream and downstream sections by subscripts 1 and 2, respectively, and applying equation (1) where $\triangle u = 0$, $\triangle Z = 0$, $v_1 = 1/\gamma_1$, and $v_2 = 1/\gamma_2$, the total energy equation becomes:

$$\frac{p_1}{\gamma_1} + \frac{V_1^2}{2g} = \frac{p_2}{\gamma_2} + \frac{V_2^2}{2g} + K_c\frac{V_2^2}{2g} \qquad (43)$$

where:

$K_c = 0.05$ = contraction loss factor

$K_cV_2^2/2g$ = energy loss from contraction, ft of fluid flowing

The total energy equation (43) is also applicable to a sudden contraction of a conduit section. For this case, however, the contraction loss factor, K_c, accounts for the energy loss resulting from the orifice effect of a sudden contraction of the fluid stream followed by its expansion from the vena contracta to the downstream conduit area. Values of K_c obtained by experiment are plotted in Fig. 13 for various values of α, the ratio of the smaller area to the larger.

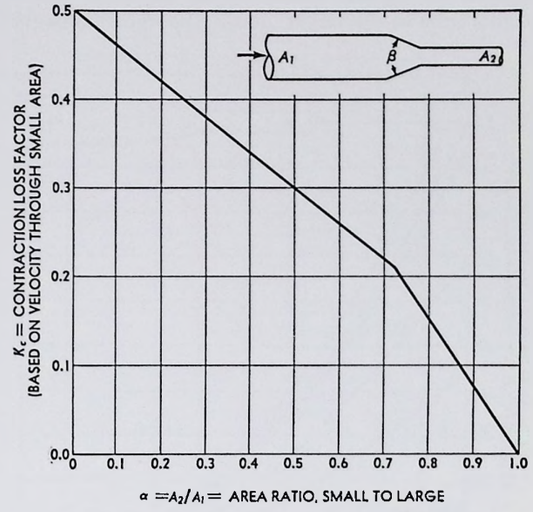


Fig. 13. *Contraction loss factor for* $\beta > 30°$
$K_c = 0.05$ *for* $\beta \leqq 30°$

When there is a sudden enlargement of the conduit section, the rapidly moving particles impinging on the slowly moving particles cause an eddy formation in the downstream section 2, and the energy lost is $(V_1 - V_2)^2/2g$. Since $A_1V_1 = A_2V_2$ or $V_2 = V_1A_1/A_2$, the energy loss can also be expressed as:

$$(1 - A_1/A_2)^2V_1^2/2g \qquad (44)$$

and the total energy equation becomes:

$$\frac{p_1}{\gamma_1} + \frac{V_1^2}{2g} = \frac{p_2}{\gamma_2} + \frac{V_2^2}{2g} + (1 - A_1/A_2)^2\frac{V_1^2}{2g} \qquad (45)$$

For sudden enlargement it is evident from the

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above that the downstream static pressure, p_2/γ_2, will always be larger than the upstream static pressure, p_1/γ_1, ahead of the enlargement if $V_2^2/2g + (1 - A_1/A_2)^2 V_1^2/2g$ is less than $V_1^2/2g$. Where $V_2 = 0$, as in submerged discharge into a large tank, it is also evident that the downstream static pressure, p_2/γ_2, will be equal to the upstream static pressure, p_1/γ_1, and that the energy lost is $V_1^2/2g$.

When there is gradual enlargement of the conduit section, the fluid has a tendency to leave the walls, eddies are formed, and energy is lost, the amount depending upon the size of the included angle. The energy lost can be expressed as $K(1 - A_1/A_2)^2 V_1^2/2g$ or $K_e V_1^2/2g$, and the total energy equation becomes:

$$\frac{p_1}{\gamma_1} + \frac{V_1^2}{2g} = \frac{p_2}{\gamma_2} + \frac{V_2^2}{2g} + K_e \frac{V_1^2}{2g} \qquad (46)$$

where:

K_e = enlargement loss factor

$K_e V_1^2/2g$ = energy loss from enlargement, ft of fluid flowing

Values of K_e obtained by experiment are plotted in Fig. 14 for various values of α, the ratio of the smaller area to the larger, and β, the included angle.

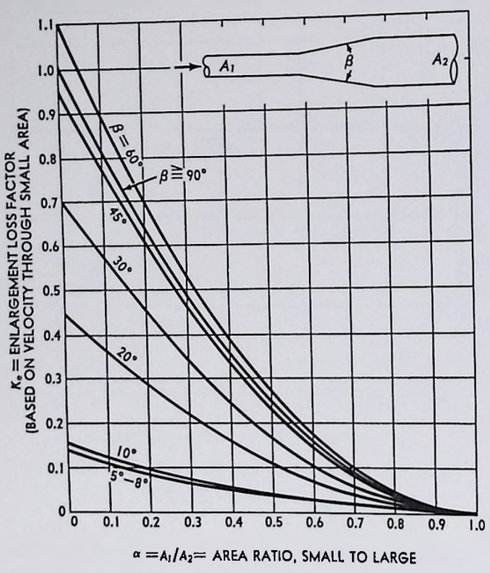


Fig. 14. *Enlargement loss factor for various included angles*

For gradual enlargement it is evident from the above that the downstream static pressure, p_2/γ_2, will always be larger than the upstream static pressure, p_1/γ_1, ahead of the enlargement if $V_1^2/2g + K_e V_1^2/2g$ is less than $V_1^2/2g$.

The differences in static pressure caused by sudden and gradual changes in section are shown graphically in Fig. 15 in terms of the velocity head at the smaller area plotted against the section area ratios.

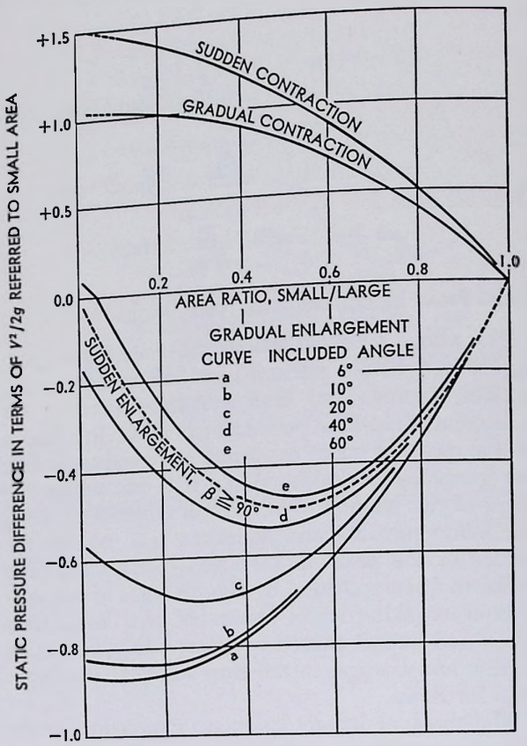


Fig. 15. *Static pressure difference resulting from sudden and gradual changes in section*

Flow through Bends

Bends in a pipeline or duct system result in losses of energy caused by friction or viscous shear of the fluid along the wall of the bend or within the fluid itself and by the impact or shock produced by the change in direction. The complex flow pattern in the bend does not permit an accurate evaluation of these losses analytically. Accordingly, it has been found more convenient to evaluate the losses empirically. Since the axial length of the bend is normally included in the straight friction loss of the pipeline or duct system, it has also been found more convenient to use experimental bend-loss values from which the frictional component has been deducted.

The energy losses for bends in round pipe expressed in terms of velocity heads in excess of straight-pipe friction vary slightly with Reynolds numbers below 150,000. For Reynolds numbers above this value they are reasonably constant and depend solely upon the dimensionless ratio, r/d, defined as the ratio of the centerline radius of the bend to the internal diameter of the pipe. For commercial pipe, the combined effect of radius ratio and angle of bend in terms of velocity heads, is shown in Fig. 16.

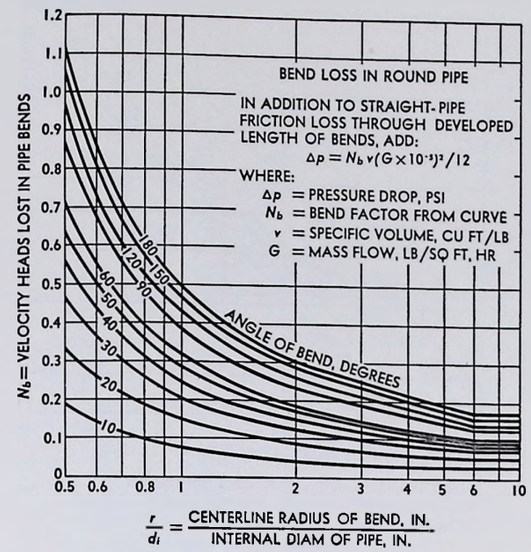


Fig. 16. *Bend loss for round pipe, in terms of velocity heads*

Flow through Coils

A convenient method for calculating the pressure drop in flow through coils is to apply a factor to the drop through an equivalent length of straight pipe. This factor depends upon the type of flow, laminar or turbulent, and the radius of the coil. The type of flow and the factors for laminar and turbulent flow can be determined from the curves and formulas of Fig. 17.

Flow through Rectangular Ducts

The loss of energy caused by a change of direction in a rectangular duct system is similar to that for cylindrical pipe. However, an additional factor, the shape of the duct in relation to the direction of bend, must be taken into account. This is known as the aspect ratio, defined as the ratio of the width to the depth of the duct. The width of the duct is always considered to be the dimension of the sides which curve into and away from the direction of flow (see Fig. 18). The bend loss for the same radius ratio will decrease as the aspect ratio increases. This is because of the smaller proportionate influence of the secondary flow on the stream as a whole. The combined effect of radius and aspect ratios on 90° duct bends is given in terms of velocity heads in Fig. 18.

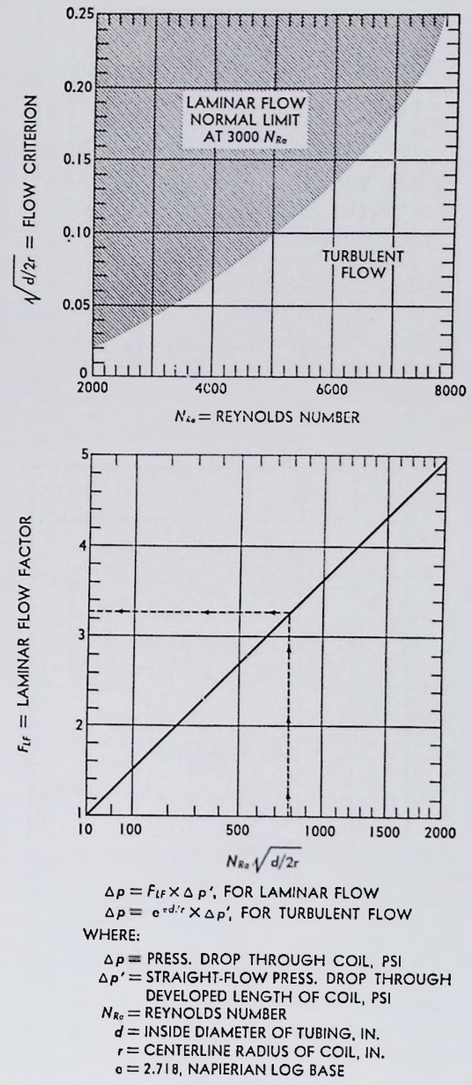
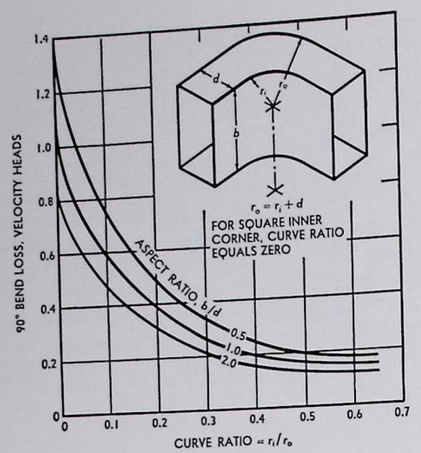


Fig. 17. *Pressure drop for laminar flow and for turbulent flow through coils*

The losses shown here are empirical and are average values of test results which indicated that, for the given range of aspect ratios, b/d, the losses were relatively independent of the Reynolds number. Outside this range the variation with Reynolds number was found to be erratic. It is therefore recommended that values for $b/d = 0.5$ be used for aspect ratios less than one half and values for $b/d = 2.0$ be used for ratios greater than two.

Losses for bends other than 90° can be taken as proportional to the angle of bend. It is apparent that bend losses in rectangular ducts can be reduced by

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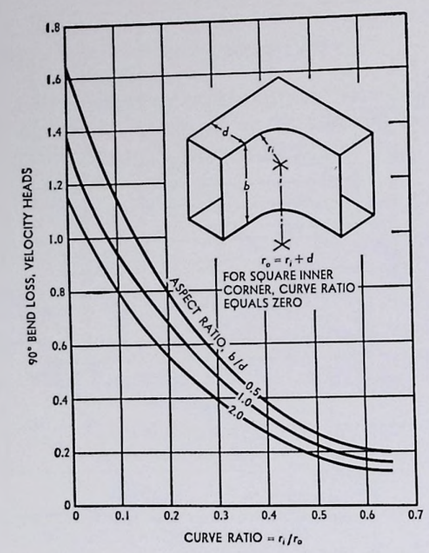


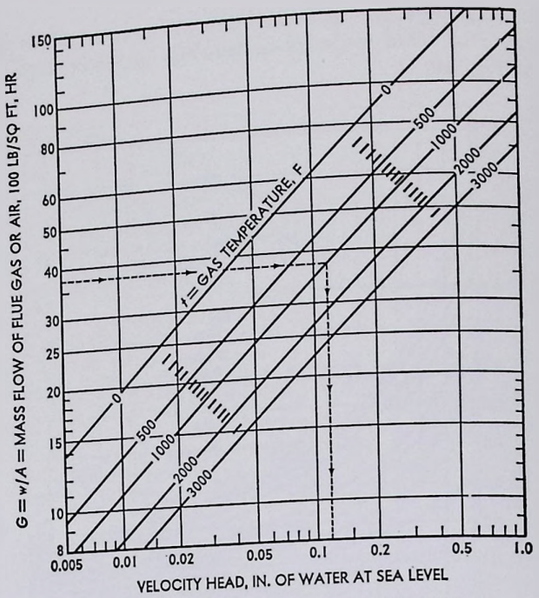
Fig. 18. *Loss for 90° bends in rectangular ducts*

the use of splitters, which increase both the aspect ratio and the average radius ratio.

A convenient chart for the calculation of draft loss resulting from impact losses in duct systems conveying flue gas or air is Fig. 19. Knowing the mass flow and temperature, a base velocity head in inches of water at sea level can be obtained. This is subject to an altitude correction if such applies.

Flow of Gases over Tube Banks

Resistance to the transverse flow of gases over in-line tube banks is an example of more or less extreme sectional changes. When the tubes are staggered, both sectional and directional changes affect the resistance. Experimental results and the analytical



VELOCITY-HEAD VALUES ARE FOR AIR OR NORMAL FLUE GAS WHICH HAS A SPECIFIC VOLUME OF 25.2 CU FT/LB AT 1000 F ABSOLUTE AND 30 IN. Hg.

Fig. 19. *Mass-flow velocity-head relationship for flue gas or air*

conclusions of an extended research by B&W, using the orifice formula as a basis, indicate that three principal variables affect resistance to the transverse flow of gases over tube banks. The primary variable is the number of major restrictions. The second variable is the so-called friction factor, relatable to the Reynolds number (based on the tube diameter) and the tube pitch and pattern (in-line or staggered). The third variable is the depth factor, applicable to banks of a depth less than 10 rows, as shown by Fig. 20.

The product of the friction factor and the number of major restrictions is, in effect, the summation of the velocity-head losses in the flow of fluids through apparatus when the major resistance is caused by impact losses. This fact can be applied in establishing procedures for the calculation of draft loss through units of a fixed type or design. In units of this type, the ratio of the length of flow path to the equivalent diameter of any longitudinal flow section remains unchanged, as does the ratio of heating surface to free-flow area. In calculating the draft loss through units of this type, the procedure is resolved into the three factors: temperature, type, and loading. The physical properties of the fluid, such as density and viscosity, vary with temperature and are accounted for by the temperature factor. The type factor accounts for the geometric pattern of the design. The loading factor accounts for the rate

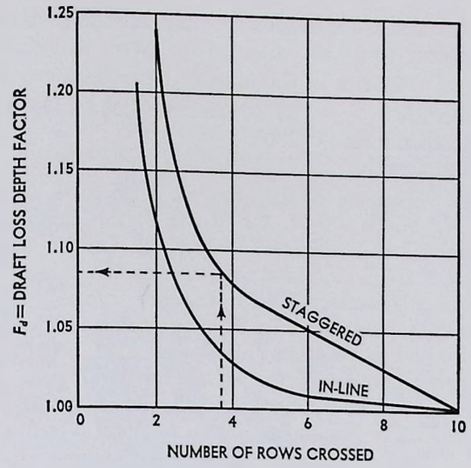


Fig. 20. *Draft loss depth factor for number of tube rows crossed in convection banks*

of driving and is proportional to the gas weight per sq ft of surface area raised to a power. The product of these three factors is the base draft loss subject to the applicable stack effect and correction for altitude (see Chapter 5). A further correction is required when the gas or air outlet is above or below the inlet. This correction is additive if the hot gases are forced to flow downward against their natural tendency to rise in relation to the cooler ambient air, and it is subtractive when the hot gas outlet is above the hotter gas inlet. Examples of draft loss calculations for simple tube-bank arrangements appear in Chapter 11.

Flow through Stacks or Chimneys

The flow of gases through stacks or chimneys is established by the natural-draft effect of the stack and/or the mechanical draft produced by a fan. The resistance to this flow or the loss in energy between the bottom and the top of the stack is a result of the friction and stack exit losses. The algebraic treatment of these losses and examples of actual application are given in Chapter 5.

Boiler Circulation

The flow in a natural-circulation boiler is established by the difference in specific weights of the water in downflowing tubes and the water or water-steam mixtures in the upflowing tubes and by the restriction to flow resulting from friction and shock losses. The algebraic treatment of the above forces and energy losses affecting natural circulation and examples of actual application are given in Chapter 9.

Throttling Calorimeter

The throttling calorimeter is used for the determination of the enthalpy and the quality of wet steam. As shown in Fig. 21, steam taken from the header in a sampling tube passes through a valve where it is throttled to a lower pressure. The kinetic energy of the high-velocity steam is all returned as reheat in the fluid when it is brought to rest in the low-pressure downstream calorimeter chamber. The expansion, thermodynamically and fluid dynamically, is therefore at "restored" enthalpy, i.e., the same initially and finally, even though not constant throughout. This was first recognized by Joule in his porous-plug experiments. The phenomenon can be defined in mathematical terms by reference to the general energy equation in the reduced form of equation (1) applicable to fluid flow. Since there is no change, over-all, in the elevational head, Z, and velocity, V (see above), equation (1) can be reduced to the form:

$$p_1v_1 + u_1 = p_2v_2 + u_2 \qquad (47)$$

and since enthalpy, $h = pv + u$, Btu/lb

$$h_1 = h_2 \qquad (48)$$

This can be verified by equation (26), Chapter 10, since no work is done.

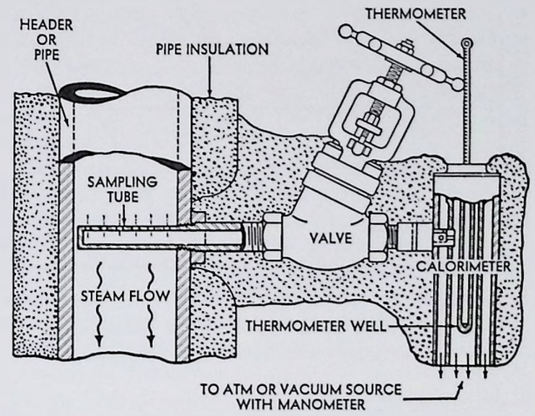


Fig. 21. *Throttling calorimeter showing sampling tube installed in steam header or pipe*

In the operation of the throttling calorimeter, the temperature and pressure are taken in the low-pressure chamber. The readings can be used only if there is superheat at this point. Use of the calorimeter can be demonstrated by the following examples in finding the quality of steam:

Example 7

Steam in Header

Pressure	100 psi (114.7 psia)
Temperature	338 F

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The steam tables show that these readings apply to saturated water, wet steam, and dry saturated steam. There is, however, a vast difference in the quality (0 to 100% dryness) and in the enthalpy, h, (309 to 1189 Btu/lb) implicit in these readings, and they give no information on whether the fluid in the header is water, steam, or a mixture. By expanding a sample of the steam from the header through the needle valve of a throttling calorimeter, the readings obtained were:

Steam in Calorimeter Chamber

Pressure	14.7 psia
Temperature	262 F
Enthalpy, h_1	1175 Btu/lb (steam tables)

Since the saturation temperature at 14.7 psia is 212 F, the steam in the calorimeter chamber has 50 degrees of superheat. On the Mollier chart, if the reading for the steam in the calorimeter chamber, represented by point 2 in Fig. 22, is projected horizontally to the left at constant enthalpy of 1175 Btu/lb in accordance with equation (48), the moisture of the steam at 114.7 psia (the pressure in the header) is 1.6% and the quality is 98.4% at point 1.

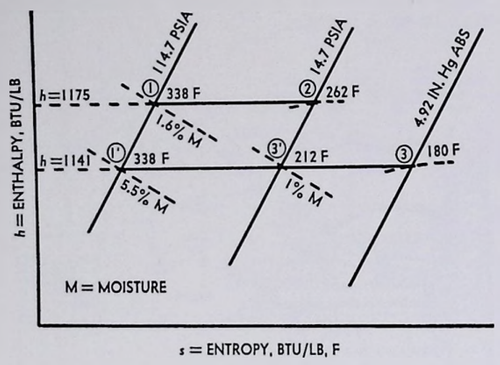


Fig. 22. *Trace of Mollier chart for determining steam quality with throttling calorimeter*

If no superheat appears in the calorimeter chamber at atmospheric pressure, it is often possible to extend the range by connecting the chamber to a vacuum source and thus operate at negative pressure. The following example will serve to demonstrate this condition:

Example 8

Steam in Header

Pressure	100 psi (114.7 psia)
Temperature	338 F

Steam in Calorimeter Chamber

Pressure	14.7 psia
Temperature	212 F

If the calorimeter chamber is connected to a vacuum source:

Steam in Calorimeter Chamber

Pressure	25 in. vacuum (4.92 in. Hg abs)
Temperature	180 F
Enthalpy, h_1	1141 Btu/lb (steam tables)

Since saturation temperature at a pressure of 4.92 in. Hg abs is 133 F, the steam in the calorimeter chamber has 47 degrees of superheat. On the Mollier chart, if the reading for the steam in the calorimeter chamber (connected to a vacuum source), represented by point 3 in Fig. 22, is projected horizontally to the left at constant enthalpy of 1141 Btu/lb, the moisture content of the steam for 114.7 psia (the pressure in the header) is 5.5% (94.5% quality) at point 1'. The reading projected to point 3' will give 212 F with 1% moisture in the calorimeter chamber when operated at atmospheric pressure (14.7 psia).

To measure steam quality beyond the range of the throttling calorimeter, a device, such as a separating calorimeter, must be used in which the liquid is physically separated from the vapor and measured—a trying operation, at best.

Entrainment by Fluid Flow

Collecting or transporting solid particles or a second fluid by the flow of a primary fluid at high velocity is known as entrainment. This is usually accomplished by jets using a small quantity of high-pressure fluid to carry along large quantities of another fluid or of solid particles. The pressure energy of the high-pressure fluid is converted into kinetic energy by nozzles, with a consequent reduction of pressure. The material to be transported is drawn in at the low-pressure zone, where it meets the high-velocity jet and is carried along by it. Usually the jet is followed by a parallel throat section, to equalize the velocity profile, before the fluid with its entrained substance enters a diverging section in which kinetic energy is partially reconverted into pressure energy. The chief loss of energy is caused by impact or shock.

Entrainment is a very useful means of performing certain functions. The injector is a jet pump that uses steam as the high-pressure fluid to entrain low-pressure water for delivery against a back pressure higher than the pressure of the actuating fluid. The ejector, similar to the injector, is designed to entrain gases, liquids, or even mixtures of solids and liquids for delivery against a pressure less than the pressure of the primary fluid. In a water-jet aspirator, water is used to entrain air to obtain a fair vacuum. In the Bunsen burner, a jet of gas issuing from a small orifice entrains air for combustion. In a gasoline carburetor, the air entrains liquid fuel to provide a flammable mixture. Steam-jet blowers are sometimes used in the base of stacks of small natural-draft boiler plants to increase the draft for short peak loads.

In several instances, entrainment is a source of trouble in the operation of steam boilers. Particles of ash, entrained by the products of combustion, when deposited on heating surfaces reduce thermal conductance, when passing through fans erode the blades, and when discharged into the atmosphere are sometimes a public nuisance. Moisture, carrying solids either in suspension or in solution, is entrained in the steam, and these solids may be carried through to the turbine and deposited on the blades, increasing the back pressure and decreasing the turbine efficiency. In downcomers or supply tubes, steam bubbles are entrained in the water when the drag on the bubbles is greater than the buoyant force, reducing the specific weight in the pumping column of natural-circulation boilers.

Separation

In the boiler field there are two clearly recognizable separation problems concerned with the removal of solid or liquid particles from a flowing gas or vapor stream: the elimination of dust and fly-ash from the products of combustion and the separation of water or moisture from steam vapor as it leaves the boiler drum. In the case of dust and fly-ash, several different principles of separation are employed.

1. Entrainment in another fluid, as with washing by water sprays.
2. Filtration, in which the dust-laden gases are passed through fiber filters where the small openings serve to catch the solid particles.
3. Ionization, in which the dust particles are given an electrostatic charge and withdrawn from the gas stream by the attraction of a charged electrode.
4. Inertia, which utilizes an abrupt change in the direction of the flow path.
5. Gravitation, in which there is a sudden reduction in the flow velocity.
6. Centrifugal force, as with the whirling action of a cyclone separator.

Some of the principles of separation listed above are essentially fluid dynamic and are applicable to the separation of both solid particles and liquid drops. For an understanding of these principles it is necessary to recognize that 1) the inertia of the solid particle or liquid drop is a function of its volume (for a given density) and 2) the resistance to flow of the particle or drop through a gaseous or vapor medium is a function of the exterior surface of the particle or drop. With a spherical configuration for the particle or drop, this means that its inertia is proportional to the cube of its diameter while the flow resistance is proportional to the square of its diameter. By Newton's law of motion, a particle or drop entrained in a flowing stream will continue to move in a straight line unless acted upon by an external force. This continuance of motion in a straight line is a function of the inertia. Opposing the motion is the fluid resistance. If separation is then attempted by fluid dynamic means in which the direction of the flowing stream is altered, the solid or liquid particles will, by virtue of this inertia, tend to continue moving in their original direction. They will not follow the altered direction of the general gas stream. The particles will move across that stream, but energy will be dissipated by fluid friction, and ultimately the particles will conform to the pattern of the gas stream. The greater the particle mass and size, the greater will be the inertia effects and the consequent effectiveness of the separation of the particle from the general gas stream.

The inertia principle is employed in a wide variety of devices, of which the B&W cyclone steam separator, illustrated in Chapters 9 and 29, is one of the most prominent. Here the bulk of the physically entrained water is removed from the steam in a series of cyclones that communicate with a girth compartment into which the steam-water mixture from the risers is discharged. From the cyclone steam separators the steam passes through scrubber baffles for final drying. The scrubber baffles are made of a number of corrugated plates, closely nested so that a large surface is available for collecting moisture as the steam, at relatively low velocity, continually changes direction through the tortuous passage between the baffles.

Eliminating the discharge of steam from the risers below the free surface of the water and thorough removal of water from the steam in the cyclone steam separators insure the densest fluid to the downcomers and, consequently, a maximum differential head for circulation. The high quality of the steam delivered, after passing through the cyclone steam separators and scrubber baffles, is indicated by a solids carry-over of only 0.3 to 0.6 part per million. This freedom from scale-forming material permits long periods of continuous trouble-free operation of both superheater and turbine.

These fluid dynamic principles are used also in the separation of dust, fly-ash, and cinders from flue gases. Because of the inertia and fluid-resistance aspects considered above, it is found that mechanical dust separators are more effective 1) when dealing with large and dense solid particles rather than with small and flocculent solids, 2) when operating with higher rather than lower fluid velocities, 3) when running at full load rather than at part load, and 4) when operating with heavy rather than light dust loadings.

It should be noted that these results are quite the

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opposite from experience with electrostatic dust precipitators, which are more effective for light and small particles, low velocities, and partial loads.

The performance of typical flue gas dust separators as applied to boilers is shown by Fig. 23. For most complete dust elimination, one of the best methods is to use mechanical and electrostatic precipitators in series. The changes in efficiency with load for the two components are largely offset, and a high sustained over-all efficiency is maintained for the entire range of load, equivalent to the sum of the two component efficiencies minus one hundredth of their product.

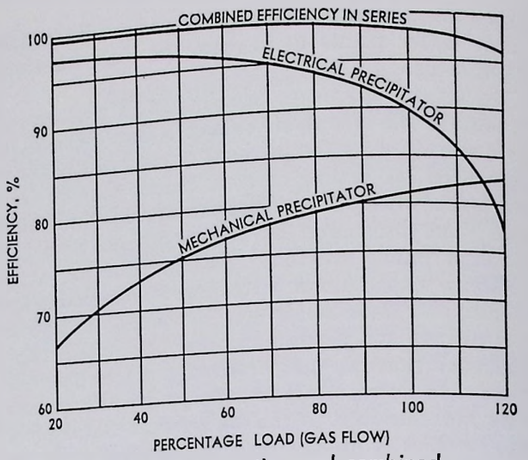


Fig. 23. *Comparative and combined performance of flue gas dust collectors*

Despite all the efforts made to separate fly-ash and cinders from flue gas, additional precautions are necessary to reduce the nuisance of dust to a satisfactory minimum. Basically, any small particles (from 100 to 30 microns, sometimes extended to as low as 3 microns) discharged from a stack will settle in still air, according to Stokes. The results of applying this and similar rules to various particle sizes and the applicable formulas for rate of settling are given in Fig. 24. High stacks or high issuing velocities (up to about 100 fps) contribute greatly to the satisfactory dissemination of dust particles. The dust deposit rate per sq ft is reduced in proportion to the cube of the stack height. Wind velocity upsets these calculations and by its turbulent effects causes the deposition of dust particles in close proximity to the stack. Deflecting and baffle effects of adjacent structures or ground contours also introduce great turbulence.

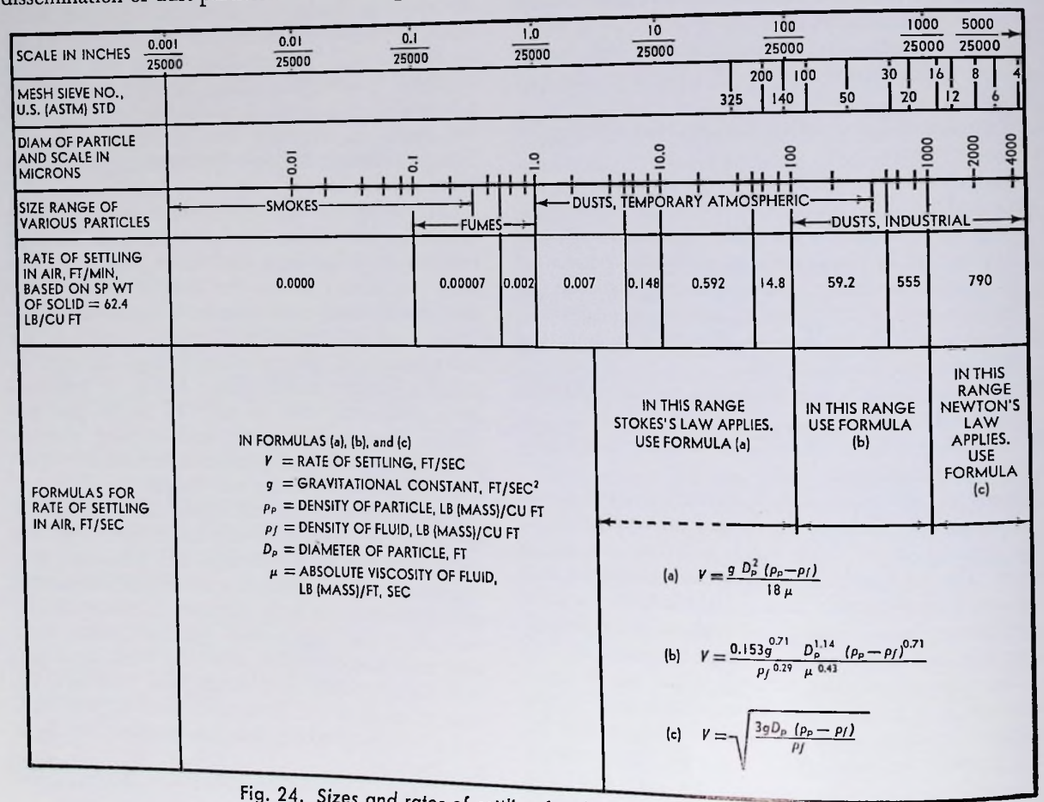


Fig. 24. Sizes and rates of settling for dusts and other particles

9

STEAM GENERATION AND BOILER CIRCULATION

THE process of boiling water to make steam is a familiar phenomenon. This is usually done at pressures below the critical (3206.2 psia), but modern practice has introduced supercritical operations.

The terms *boiling, saturation, liquid, vapor, steam, mixture, heat of vaporization,* and *superheat,* all relevant to the process of steam generation, define the physical states or properties of the working substance. Thermodynamically, boiling may be considered a special case of adding heat to the working substance in a constant-pressure constant-temperature process.

On the other hand, when the working substance, water, is compressed to and above the critical pressure, 3206.2 psia, and the temperature is increased to and above 705.4 F, the above terms no longer have their former significance. As the pressure and temperature of water approach the critical, the value of the heat of vaporization (h_{fg} of the steam tables) rapidly decreases and finally at the critical condition (3206.2 psia and 705.4 F) becomes zero, and the term is meaningless.

Heat added to the working substance at or above the critical pressure produces an increase in temperature. As a result there are changes in temperature-dependent properties, such as specific heat, viscosity, thermal conductivity, diffusion, and specific volume. With the exception of the last, the relationship is one of decreasing values with increasing temperature over the lower temperature range and increasing values with increasing temperature over the higher temperature range. There are no discontinuities in this relationship.

When heat is added at or above the critical pressure, the term *fluid* may be used to describe the working substance in its continuous passage from the cold to the hot end of the circuit. Unlike the practice in the subcritical region, attempts to define the state of the working substance at a particular point in the circuit are generally unsatisfactory and unnecessary.

In the heat-power field the operating pressures of natural-circulation boilers have increased over the years to above 2800 psi. Since 1945 many experimental operations have been conducted in the supercritical region. Today the supercritical pressures are accepted standards in large-scale commercial practice.

Much of the analysis which follows is concerned with the phenomena of boiling as encountered in natural-circulation boilers. Experience has demonstrated that in some instances, economic advantages accrue from the use of the forced-circulation (Universal-Pressure type) boiler in large sizes and for the higher subcritical pressures.

Types of Boiling

The change from liquid to vapor occurs both at a solid-liquid interface, as at the inside tube surface of a water-tube boiler, and at a liquid-vapor interface, as with water surrounding a steam bubble. It has been observed that two distinct types of boiling, known as nucleate boiling and film boiling, occur at a solid-liquid interface. Nucleate boiling is characterized by the formation and release of steam bubbles at the interface with the water still wetting the surface, while in film boiling this surface is covered by a film of steam. To transfer heat from the surface to the fluid, a temperature gradient ($\triangle t$) is necessary. The magnitude of this gradient depends mainly upon whether nucleate or film boiling is taking place and, to a lesser extent, upon the pressure, the surface material, and the condition of the surface.

Except for the once-through type, boiler designs for subcritical pressures must stay within nucleate boiling conditions, where the water still wets the tube surfaces and the $\triangle t$ values are low. For a given pressure under the condition of nucleate boiling, incremental increases in $\triangle t$ produce quite uniform increases in conductance, Btu/sq ft, hr, F. Beyond this condition there is a sharp decrease in conductance for further increase in $\triangle t$. This limit marks the transition from nucleate to partial nucleate partial steam-film boiling, and the tube wall is no longer continuously wetted.

The fluid velocity through vertical heat-absorbing

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boiler tubes covers a relatively wide range (1 to 10 fps). With horizontal or sloping tubes, mass flow and velocity must be sufficient to prevent separation in order to maintain the wetted condition. Entrance velocities to circuits of this nature are conventionally in the upper half of the above range, depending upon heat absorption rate, direction of heat application, and degree of slope.

Types of Circulation

An adequate flow of water and water-steam mixture, necessary to the proper performance of all the circuits of a steam-generating unit, may be produced naturally by the action of the force of gravity, mechanically by means of pumps, or by a combination of the two.

In natural circulation the force of gravity available to produce flow comes from the difference between the densities (lb/cu ft) of the fluids in the downcomer (downflow) and riser (upflow) circuits. Ideally, the fluid in the downcomer is water at or slightly below saturation temperature and undiluted by steam bubbles (see *Steam Separation and Purity* below). Heat-absorbing upflow circuits at saturation temperature convey to the boiler drum a water-steam mixture of less density than that of the water in the downflow circuits, thus establishing the force available for circulation.

Forced Circulation

In boiler units designed for forced circulation at subcritical pressures, the flow in the various circuits is produced by pumps. There are two general types:

1. In the "once-through" forced-circulation type, water from the feed supply is pumped to the inlet end or ends of the heat-absorbing circuits. Evaporation, or change of state, gradually takes place along the length of the circuit, and, when evaporation is complete, further progress through the heated circuits results in superheating the vapor. Conventionally this type of forced circulation requires no steam-and-water drum.

A modification of the "once-through" type evaporates to partial dryness (90% quality), and the water is removed in a separator.

2. In the "recirculating" forced-circulation type, the water supplied to the circuits by a separate circulating pump is considererably in excess of the steam produced. A steam-and-water drum is required (as in a natural-circulation boiler), in which steam separation takes place. The separated water together with feedwater from the feed pump is returned through downcomer circuits to the circulating pump for another "round trip."

In the recirculating type of forced circulation there is a net thermal loss for the boiler unit because of the separate circulating pump. While practically all of the energy put into the pump reappears in the pumped water as added enthalpy, the energy came originally from the fuel pile at a conversion-to-useful-energy factor of less than 1.0. If an electric motor is used to drive the pump, the net energy loss referred to the fuel pile is in the order of double the energy supplied to the pump motor (based on a plant thermal efficiency of 33%).

Natural Circulation

In any given natural-circulation system, the circulation will increase with increased heat input (and increased steam output) until a maximum value is reached, after which further increase in heat absorption will result in a decrease in flow. The general form of the curve is shown in Fig. 1. Two opposing forces are present: The increase in flow results from the increase in the difference of the densities of the respective fluids in downcomers and risers caused by the increase in heat absorption. However, at the same time the friction and impact losses (see Chapter 8) in both downcomers and risers are increasing. When the rate of increase in these losses, caused primarily by the increase in specific volume in the riser circuits, becomes greater than the gain from increase in the density differences, the flow rate will begin to drop. A proper objective, therefore, is always to design all the circuits to operate in the region of the rising part of the curve.

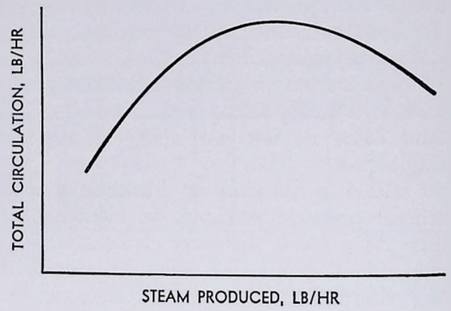


Fig. 1. *Typical relationship between circulation in a boiler circuit (at a given pressure) and amount of steam produced (scale arbitrary)*

When the design is limited to the rising portion of the circulation curve, a natural-circulation boiler tends to be self-compensating for the numerous variations in heat-absorption conditions usual in an operating unit. Some of these are: sudden overloads, change in heat-absorbing-surface cleanliness, nonuniform fuel bed or burner conditions, and even the inability to forecast precisely the actual conditions over the operating lifetime.

No similar compensating effect is inherent in a forced-circulation unit operating at subcritical pressures, since a large part of the total resistance of the riser circuits is caused by flow-distribution devices required at the circuit inlets. Under these conditions an increase in heat absorption to an individual circuit or group of circuits changes the flow rate only a little because of the disproportionately large resistance of the distributors.

The method of producing flow in boiler circuits, whether natural or mechanical, has virtually no bearing on the effectiveness of heat-absorbing surface with nucleate boiling. The prime requirement is that the inside surface should be wetted at all times by the water in a water-steam mixture of suitable quality for the given conditions (see above and Fig. 2). Having met this fundamental requirement, the water-film resistance to heat flow is negligibly small (see Chapter 11), and the over-all heat conductance depends upon gas-film and tube-wall resistances. Any departure from this practice requires special consideration of the forced-convection steam-film heat transfer coefficient and its relation to permissible metal temperatures. In other words, boiler heat-absorbing surface in the furnace or convection portion of the unit will absorb substantially the same amount of heat per sq ft regardless of whether the circulating flow is produced by natural methods or by pumps.

Forced or Natural Circulation

Under certain conditions forced circulation can be usefully applied to steam generation. Mechanical means to move the fluid within the circuits are employed for boilers designed to operate above or near the critical pressure (3206.2 psia). There are instances, also, in the process and waste-heat fields where temperature control or consolidating heat pick-up from widely separated points can be economically effected by circuits activated by pumps. The conditions where forced circulation can be applied to advantage are quite specific.

Experience has shown that natural circulation by difference in density is ample for boiler pressures even above 2800 psi. When the difference between the densities of water and steam becomes too small, a pump will assure positive circulation. The forced-circulation principle, however, is equally operable in both the supercritical and subcritical pressure ranges. The selection of the identifying name "Universal-Pressure" boiler reflects the broad applicability of the forced-circulation principle. Its choice, as opposed to the retention of natural circulation in the subcritical range, is essentially determined by the economics of the installation.

The differential in specific weights of steam and water for the range of 14.7 to 3206.2 psia is given in Fig. 3. It will be noted that a substantial differential persists well up toward the critical pressure. As long as the maximum effectiveness of this differential is maintained by the efficient separation of the steam from the water in the circulating system, as with the use of the cyclone steam separator (see below), mechanical aid to circulation is not essential.

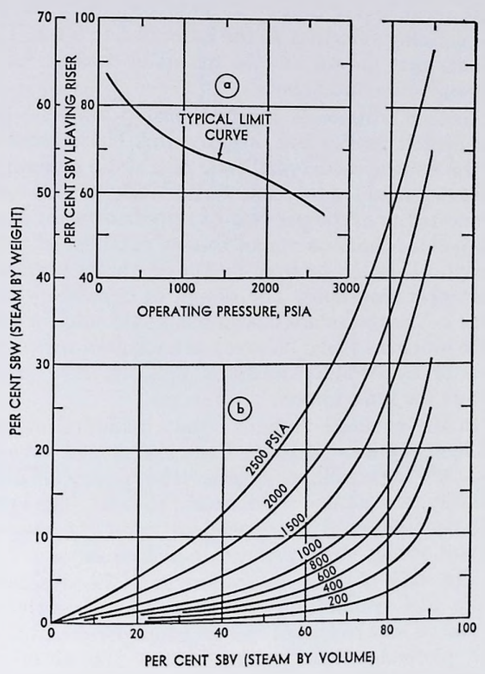


Fig. 2. *Diagram a: Recommended maximum per cent SBV leaving riser at various pressures. Diagram b: Per cent SBW corresponding to per cent SBV for various pressures*

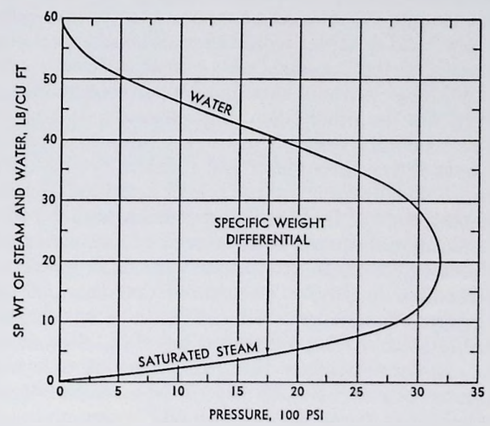


Fig. 3. *Specific weights of steam and water at saturated steam temperature for pressures atmospheric to critical*

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Circulation Design Criteria

In the rational design of boiler circulating systems, the primary requisite is that nucleate boiling is maintained for all anticipated operating conditions. This insures the continued wetting of the inside surface by the water phase of the two-phase mixture. Accumulated data on limiting values and design criteria are derived from operating experience and laboratory investigation. Furthermore, it is assumed that the fluid supplied to the boiler will be of suitable purity after proper treatment. (See *Treatment of Water for Steam Boilers*, Chapter 21.)

In the actual design, the riser circuits are normally fixed by arrangement. They form the enclosure and provide the surface required for heat absorption. They have dimensional limitations as a result of location, permissible burner clearances, and gas velocities. The structural arrangement of the downcomer circuits usually is not as rigidly prescribed as that of the risers. The establishment of flow areas in the various parts of the system, particularly in the downcomer (supply) circuits, is the end result of the design calculations.

One design criterion is the permissible per cent steam by volume (%SBV), or the corresponding per cent steam by weight (%SBW), throughout the fluid flow path. This limit is a function of many variables, including pressure, heat flux, and mass flow. For each pressure and heat flux there is a maximum permissible quality (%SBW), which is dependent upon mass flow. This is the so-called "burnout point," or departure from nucleate boiling. Designs which would result in operation at qualities in excess of this limit could be subject to tube failures as a result of metal temperatures above material limitations. Moreover, for a constant heat flux and mass flow, the permissible steam by volume (%SBV) decreases with increased pressure.

For low-pressure designs, below 1200 psi, it is possible economically to design to a maximum quality limit leaving the riser as a function of pressure. Satisfactory local qualities would then obtain in the lower portions of the furnace, where heat fluxes are normally higher, without sacrificing a suitable margin of safety. On the other hand, high-pressure designs require a detailed analysis of the probable distribution of heat fluxes throughout the furnace during operation. These fluxes are a function of heat input, burner location, type of fuel fired, and possible maloperation or operational upsets. It therefore becomes necessary to investigate these various factors which affect the permissible quality, to be assured that the final circulatory arrangement will result in a satisfactory quality (%SBW) not only at the top of the riser circuit but also incrementally along its length. A unit with burners located near the top of the riser circuit has high heat fluxes in the high-quality region and must be designed to quality limits different from those of a unit with burners located near the bottom of the riser circuit in the low-quality region. The limit curve of Fig. 2a is typical; explicit values would be dependent upon local conditions of heat flux and mass flow.

Laboratory investigations verified by field tests have shown that the permissible quality for a given heat flux, mass flow, and pressure can be increased by the use of turbulence promoters, which aid in wetting the inside tube surface. This increase has been achieved through the use of internally ribbed tubes.

The quality of the mixture is not, however, the only criterion of proper conditions within the tubes or flow paths. Minimum allowable internal velocities are also important criteria. These values vary widely with the heat absorption rate, quality, manner of heat application, and arrangement of tubes in the circuit. Sloping tubes in particular, receiving heat on either half or full circumference, require adequate velocities in order to prevent separation of the two-phase fluid, steam-blanketing, and eventual tube failure. Adequate velocities are also necessary for sloping tubes receiving heat only from below, to minimize the possibility of solids deposition from the water and the consequent overheating and ultimate failure of the tube metal. Slightly inclined tubes receiving heat only from above, as in a furnace floor, or around the full circumference, as in a screen, require fluid velocities in the order of 5 to 10 fps entering the heat absorption zone. On the other hand, substantially vertical riser circuits, regardless of heat application, require relatively low entering velocities, in the order of 1 to 5 fps. The criteria that govern involve too many factors for a thorough description here.

Another criterion is the relationship between the anticipated density and pressure drop characteristics of the flowing media and those of a static column of saturated liquid of the same vertical height. This ratio is referred to as the per cent of saturated-water head (%SWH). Minimum values can be established from analytical studies or from measured values obtained from operating units. The design of circulatory systems below the established limits could lead to possible problems (tube failures) as a result of recirculation of steam-water mixtures between flow paths within the same circuit.

In the finalized design of the circulation system all limits must be satisfied. Each circuit will have its own limiting condition, whether it be quality (%SBW), velocity, or saturated-water head (%SWH). The limiting condition will be dependent upon the over-all circuit arrangement, pressure, and heat flux.

The rate of heat absorption must be evaluated from fuel combustion data and from knowledge, based on experience, of the probable performance of the particular arrangement of the heat-absorbing surface. In practice, the heat input pattern is rarely uniform along the length of the heat-absorbing path, and integrated values of the instantaneous density-

height relationship are required to permit precise determination of mean fluid specific weights.

Analyses of downcomer circuits as compared with riser circuits are less involved. The specific weight of the fluid in the downcomer (preferably all in the liquid phase) is readily determined, since it is at a temperature related to those of the feedwater and saturation. Steam bubbles in the downcomer circuit seriously affect the head available for natural circulation (see *Steam Separation and Purity* below).

In the simple circulation-calculation example to follow, the application of all refinements is beyond the scope of the present treatment. It must be emphasized that the possible arrangements of boiler circuits are innumerable, with parallel and series circuits inevitably present. To establish a circulation balance in a complete system is an involved problem, and, while the principles of equations (1), (2), (3), and (4) given below apply, the procedure for a thorough solution requires consideration of other factors peculiar to two-phase flow.

Principles of Natural Circulation

The elementary circuit, shown by Fig. 4, and the basic relationships noted may be used to demonstrate the principles of natural circulation, simple in conception but somewhat tedious in application.

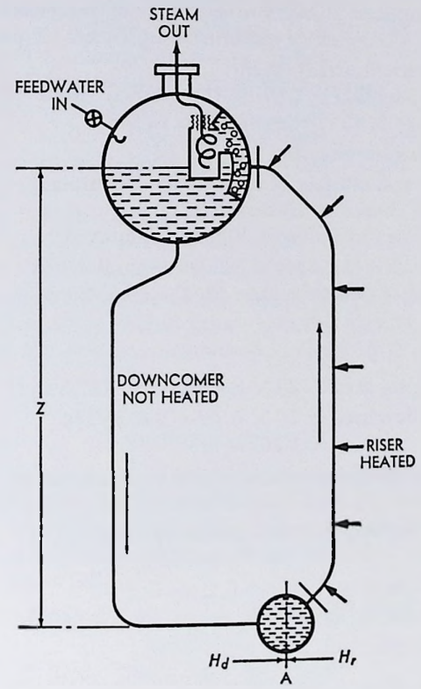


Fig. 4. *Hypothetical simple circuit (diagrammatic) including primary steam separator in drum*

For stabilized flow (system in equilibrium), the weight flow in the downcomer must equal the weight flow in the riser. Also, the net pressure at A, Fig. 4, of the fluid in the downcomer must be balanced by the net pressure of the fluid in the riser; that is, the net head, H_d, in the downcomer must equal the net head, H_r, in the riser.

Referring to Fig. 4:

Z = difference in elevation, ft

γ_d = mean sp wt of downcomer fluid, lb/cu ft

γ_{ex} = sp wt of mixture leaving riser, lb/cu ft

γ_r = mean sp wt of riser fluid, lb/cu ft

$Z\gamma_d/62.4$ = downcomer gravity head, ft of std water

$Z\gamma_r/62.4$ = riser gravity head, ft of std water

$h_{\Sigma d}$ = sum of downcomer losses (friction, bend, entrance, exit, change of section), ft of std water

$h_{\Sigma r}$ = sum of riser losses (friction, bend, entrance, exit, change of section, baffle, separator, acceleration), ft of std water

$$H_d = H_r \qquad (1)$$

$H_d = Z\gamma_d/62.4 - h_{\Sigma d}$ = downcomer net head, ft of std water (2)

$H_r = Z\gamma_r/62.4 + h_{\Sigma r}$ = riser net head, ft of std water (3)

$Z(\gamma_d - \gamma_r)/62.4 = h_{\Sigma d} + h_{\Sigma r}$, from equations (1), (2), and (3) (4)

The term "std water" refers to water at sp wt = 62.4 lb/cu ft.

From equation (4) it is evident that the difference between the gravity heads of downcomer and riser is equal to the sum of the circulation losses in the system. The latter, as noted above, include friction losses along the straight flow paths, and bend, entrance, exit, change-of-section, baffle, steam-separator, and acceleration losses.

In general, therefore, the losses in the downcomer portion of the circuit are:

$$h_{\Sigma d} = h_f + h_b + h_{en} + h_{ex} + h_k$$

and similarly for the riser circuit,

$$h_{\Sigma r} = h_f + h_b + h_{en} + h_{ex} + h_k + h_s + h_{ac}$$

The h increments represent the various circulation losses in the downcomer and riser portions of the circuit as enumerated above. The flow path friction, h_f, may be expressed by a form of the Moody equation (see page 8-9). The bend loss, h_b, entrance loss, h_{en}, exit loss, h_{ex}, change-of-section loss, h_k, and the steam-separator loss, h_s, being shock losses (see Chapter 8), are best expressed in "velocity heads." The acceleration loss, h_{ac}, is peculiar to the case of adding heat to the working substance and is reversible in the sense that it becomes a gain on extracting

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heat. It is a result of the change in momentum in the system, and the energy loss is equal to the difference between the velocity heads at the beginning and the end of the heated section of the flow stream. All incremental losses are converted to ft head of std water.

In general:

h_f, downcomer $= 2.31\, fL\,(G \times 10^{-5})^2/d_e\gamma_s$, ft of std water (5)

h_f, riser $= 2.31 fL\,(G \times 10^{-5})^2/d_e\gamma_r$, ft of std water (6)

$h_b = N_b\,(G \times 10^{-5})^2/5.2\gamma$, ft of std water (7)

h_{en}, h_{ex}, and $h_k = N_{ek}\,(G \times 10^{-5})^2/5.2\gamma$, ft of std water (8)

h_s = value from separator manufacturer, ft of std water

$h_{ac} = (1/\gamma_s - 1/\gamma_f)\,(G \times 10^{-5})^2/5.2$ (9)

where:

γ_s = sp wt of mixture or steam leaving heated section

γ_f = sp wt of mixture or water entering heated section

In the above equations (5), (6), (7), (8), and (9):

f = friction factor, from Fig. 9, Chapter 8

L = developed length of flow path (incl bends), ft

d_e = equiv diam (usually ID of tube), in.

γ = fluid sp wt at region considered, lb/cu ft

N_b = number of velocity heads lost in bends, from Fig. 16, Chapter 8

N_{ek} = number of velocity heads lost in entrance (N_{en}), exit (N_{ex}), and change of section (N_k); 0.5 and 1.0 are commonly used for N_{en} and N_{ex}, respectively; N_k is obtained from Fig. 13 (as K_c) and Fig. 14 (as K_c), Chapter 8

G = mass flow, lb/sq ft, hr

For two-phase flow, as is usual in the risers of steam boilers, a good approximation of the friction loss at relatively low steam qualities (where the steam phase is submerged in the water phase) can be made as follows:

1. Calculate the Reynolds number, using the total mass flow (steam + water) and the viscosity of the water.
2. Obtain the corresponding friction factor, f, from Fig. 9, Chapter 8.
3. Multiply the f thus obtained by 1.5, to account for the increased level of secondary-flow losses associated with two-phase flow.
4. Use the result in place of f of equation (6).
5. Obtain the average sp wt, γ_r, as the reciprocal of the average of the specific volumes entering ($1/\gamma_f$) and leaving ($1/\gamma_s$), assuming uniform steam generation along the length of the tube.

The following additional definitions and relationships are helpful in circulation analyses:

$$\gamma_{mix} = \frac{\%SBV \times \gamma_g}{\%SBW}, \text{ lb/cu ft} \quad (10)$$

$$\gamma_{mix} = \gamma_f - \frac{\%SBV\,(\gamma_f - \gamma_g)}{100}, \text{ lb/cu ft} \quad (11)$$

where:

%SBV = per cent steam by volume

%SBW = per cent steam by weight

γ_f = sp wt of saturated water, lb/cu ft

γ_g = sp wt of saturated steam, lb/cu ft

γ_{mix} = sp wt of mixture, lb/cu ft

Example

A simple example using the above relationships for an hypothetical natural-circulation circuit, similar to the one shown by Fig. 4, follows:

Conditions

Operating pressure = 600 psia

Feedwater temperature = 486.2 F

γ_f = 1/sp vol = 49.8 lb/cu ft, from steam tables

γ_g = 1/sp vol = 1.3 lb/cu ft, from steam tables

h_{fg} = 731.6 Btu/lb water, heat added, from steam tables

Z = 23 ft, difference in elevation

L_r = 25 ft, length of riser circuit

L_d = 28 ft, length of downcomer circuit

Heat input = 70,000 Btu/hr, sq ft of projected riser area (uniform along L_r for simplicity)

Riser circuit arrangement:

30, 3-in.-OD × 2.67-in.-ID vertical tubes (touching) 3, 45° bends @ 18-in. radius

Downcomer circuit:

Size and number of tubes to be determined

3, 90° bends @ 18-in. radius

Bend loss factors, from Fig. 16, Chapter 8

h_s = 1.10 ft of std water, steam separator loss (from manufacturer for the conditions)

Procedure

Riser, proj area = $30 \times 3 \times 25/12 = 187.5$ sq ft

Riser, flow area = $30 \times 0.7854(2.67)^2/144$
= 1.167 sq ft

Steam, total produced = $70{,}000 \times 187.5/731.6$
= 17,950 lb/hr

%SBV leaving riser, maximum allowable = 73, from Fig. 2

%SBW leaving riser = 6.6, from Fig. 2

Lb steam/lb mixture, leaving riser = 0.066 lb

Circulation ratio = 1/0.066 = 15.1 lb mixture/lb steam produced

Flow in circuit, total = $17{,}950 \times 15.1$
= 271,000 lb/hr

$\gamma_s = \gamma_f$ = 49.8 lb/cu ft, mean sp wt in downcomer and at entrance to riser

$\gamma_s = \gamma_{mix} = 73 \times 1.3/6.6 = 14.38$ lb/cu ft, sp wt @ riser exit, from Eq (10)

$\gamma_r = 2/(1/49.8 + 1/14.38) = 22.33$ lb/cu ft, mean sp wt in riser*

Riser Flow Losses

G, mass flow, riser = 271,000/1.167
= 232,000 lb/sq ft, hr

μ, abs viscosity of sat. water @ 486.2 F = 0.255 lb (mass)/ft, hr, from Fig. 10, Chapter 8

N_{Re}, Reynolds No. = $232{,}000 \times 2.67/(12 \times 0.255)$
= 202,400

f, friction factor, riser = 0.019 for hot-finished tubes, from Figs. 8 & 9, Chapter 8

h_f, riser $= \dfrac{2.31 \times 1.5 \times 0.019 \times 25 \times (2.32)^2}{2.67 \times 22.33}$
= 0.149 ft of std water, from Eq (6)

N_b, bend factor for 45° bends, 2.67-in. ID, 18-in. radius = 0.10, from Fig. 16, Chapter 8

h_b, riser, lower = $1 \times 0.1(2.32)^2/(5.2 \times 49.8)$
= 0.002 ft of std water, from Eq (7)

h_b, riser, upper = $2 \times 0.1(2.32)^2/(5.2 \times 14.38)$
= 0.014 ft of std water, from Eq (7)

h_{en}, riser = $0.5(2.32)^2/(5.2 \times 49.8)$
= 0.010 ft of std water, from Eq (8)

h_{ex}, riser = $1.0(2.32)^2/(5.2 \times 14.38)$
= 0.072 ft of std water, from Eq (8)

h_k, riser (no change in section) = 0.0

h_s = 1.10 ft of std water, from manufacturer

h_{ac}, riser = $(1/14.38 - 1/49.8)\,(2.32)^2/5.2$
= 0.051 ft of std water, from Eq (9)

$h_{\Sigma r}$, sum of riser head losses = 0.149 + 0.002 + 0.014 + 0.010 + 0.072 + 1.10 + 0.051
= 1.40 ft of std water

$23(49.8 - 22.33)/62.4 = h_{\Sigma d} + 1.40$, from Eq (4)

$h_{\Sigma d} = 10.15 - 1.40 = 8.75$ ft of std water

To effect a balance in the circuit, the flow area of the downcomer should be so selected that $h_{\Sigma d}$, the sum of the downcomer losses, equals 8.75 ft of std water.

Downcomer Flow Losses

The downcomer tube size is selected by trial and error. For this example, preliminary trials are omitted. Three 3-in.-OD × 2.67-in.-ID tubes will produce a balance, and the evaluation of the individual downcomer losses follows:

Downcomer, flow area = $3 \times 0.7854(2.67)^2/144$
= 0.1167 sq ft

G, mass flow, downcomer = 271,000/0.1167
= 2,320,000 lb/sq ft, hr

μ, abs viscosity of sat. water @ 486.2 F = 0.255 lb (mass)/ft, hr, from Fig. 10, Chapter 8

N_{Re}, Reynolds No. = $2{,}320{,}000 \times 2.67/(12 \times 0.255)$
= 2,024,000

f, friction factor, downcomer = 0.018 for hot-finished tubes, from Figs. 8 & 9, Chapter 8

h_f, downcomer $= \dfrac{2.31 \times 0.018 \times 28(23.2)^2}{2.67 \times 49.8}$
= 4.72 ft of std water

N_b, bend factor for 90° bends, 2.67-in. ID, 18-in. radius = 0.12, from Fig. 16, Chapter 8

h_b, downcomer = $3 \times 0.12(23.2)^2/(5.2 \times 49.8)$
= 0.75 ft of std water, from Eq (7)

$h_{en} + h_{ex}$, downcomer = $1.5(23.2)^2/(5.2 \times 49.8)$
= 3.13 ft of std water, from Eq (8)

h_k, downcomer (no change in section) = 0.0

$h_{\Sigma d}$, sum of downcomer head losses = 4.72 + 0.75 + 3.13 = 8.60 ft of std water

As calculated, the total of the downcomer flow losses, $h_{\Sigma d}$ = 8.60 ft of std water, is in substantial agreement with the value 8.75 ft of std water, as established by equation (4). The sum of the calculated downcomer losses should not exceed the losses as derived by the equation, since, if it does, the allowable %SBV, one of the criteria imposed, will be exceeded.

Multiple Circuits in Parallel

Parallel riser circuits are frequently used in boiler arrangements. Detail examples of such arrangements are not within the scope of this chapter, but the general approach is as follows:

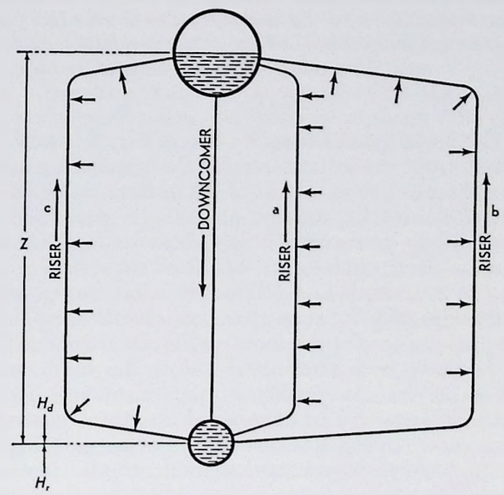


Fig. 5. *Hypothetical multiple circuit (diagrammatic) with three heated riser components*

* An approximate method; an integrated value should be used for more accurate calculations.

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An hypothetical multiple circuit with three heated riser components (c, b, and a) and a downcomer is illustrated diagrammatically by Fig. 5. In the diagram, Fig. 6, the flow in each of these riser components and the total flow in all are evaluated horizontally, while the downcomer losses in terms of $h_{\Sigma d}$ in equation (4) are evaluated vertically.

PROCEDURE

FOR HEAT INPUT PERTINENT TO EACH RISER

1. CALCULATE POSITION OF POINTS ⊙ AT MAXIMUM ALLOWABLE VALUE FOR PER CENT STEAM BY VOLUME.
2. CALCULATE POSITION OF POINTS ⊗ FOR VALUES GREATER THAN THE MAX ALLOWABLE PER CENT STEAM BY VOLUME.
3. CALCULATE POSITION OF POINTS X FOR VALUES LESS THAN THE MAX ALLOWABLE PER CENT STEAM BY VOLUME.
4. DRAW TRACES THROUGH POINTS ⊙-⊗-X AND FOR THE TOTAL FLOW (ADDITIVE).
5. DRAW LINE m-n THROUGH THE LOWEST POINT ⊙ ; THE INTERCEPTS DETERMINE THE FLOW IN EACH RISER AND THE TOTAL FLOW.

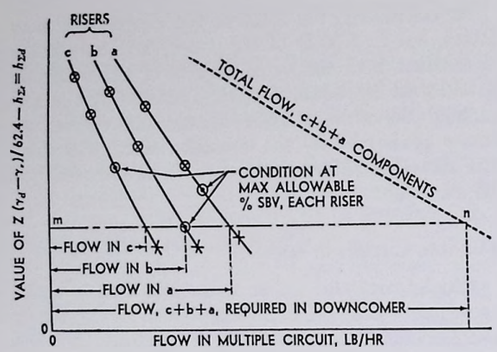


Fig. 6. *Method of establishing a circulation balance for a multiple parallel circuit (see Fig. 5). The flow in each riser component and the allowable downcomer loss (at m intercept) are determined for a specified heat input to each riser, using the % SBV criterion*

The initial point for each curve in Fig. 6 is established from the criteria set for the particular riser component, such as quality of the mixture and allowable %SBV leaving the heat-absorbing region. (Such a point was determined in the above example for a 30-tube riser, where $h_{\Sigma d} = 4.60$ ft of std water for a flow of 271,000 lb/hr.) Additional values for $h_{\Sigma d}$ are then calculated for each riser component, using the original value of heat input or lb steam produced and values both above and below the maximum allowable for the quality of the mixture. Curves through the points so determined are of the general form shown in Fig. 6. The lowest point for the group, designating the allowable criterion assigned to the particular riser component, thus establishes the position of a horizontal line, m-n, as indicated. The intercept of this line with the curve for each riser defines the flow in that particular component. The sum of all the riser component flows establishes, as at n, the total flow in the downcomer, for which the flow area and loss in head are to be determined. The intercept of the horizontal line with the ordinate representing the $h_{\Sigma d}$ values fixes the losses permissible in the downcomer for a circulation balance.

It is clear from Fig. 6 that once the position of the horizontal line is fixed (at the lowest value of $h_{\Sigma d}$) for maximum allowable %SBV (when used as a criterion) in the controlling riser component, the %SBV and the flow in the other riser components are both precisely set by the intercepts.

Steam Separation and Purity

In the usual form of steam boiler operating at pressures below the critical, the steam drum is important functionally. It is the vessel in which the saturated steam is separated from the steam-water mixture and into which feedwater is introduced. Boiler-water treatment by chemicals may take place here. Any necessary blowdown for reduction of boiler-water concentration is usually from the drum. Further, it is from the drum that the water separated from the mixture, together with feedwater, must be recirculated, preferably in all-liquid phase, to the heat-absorbing surfaces.

In a modern boiler drum, the separation of steam from the mixture delivered by the riser portions of the circuit usually takes place in two steps. The primary separation removes nearly all the water from the mixture, so that, in effect, no steam is recirculated through the circuits. The steam may still contain contaminants which must be removed or reduced in amount before the steam is sufficiently pure for use in high-pressure turbines. This step is called secondary separation or "steam scrubbing." Both steps are usually accomplished in one steam drum.

Part of the contamination of the steam is caused by dissolved solids contained in tiny droplets that may remain after primary separation. The rest of the contamination appears to be silica, either in solution in vapor-phase steam or in vaporized form. This type of contaminant cannot be mechanically removed by primary separation, and washing or scrubbing is necessary for its dilution or complete removal.

Other instances of steam contamination are possible. Gross impurities in the steam may be caused by periods of abnormally high water level from operational upsets, during which the separating equipment is submerged, allowing the water to be carried over in gulps. This action may be called "priming." Another type of gross carry-over may occur if the boiler water produces excessive foam in the drum. This action, known as "foaming" and also as "priming," may be severe enough to render

the separating devices ineffective. Foaming and priming are comparatively rare occurrences in the modern boiler equipped with proper water-level regulation and control of boiler-water quality by chemical methods.

Factors Affecting Steam Separation

Separation of the steam from the steam-water mixture discharged into the drum from the riser circuits is related to both design and operating factors, which may be listed as follows:

Design Factors

1. Operating pressure.
2. Rate of steam generation.
3. Rate of water circulation (the circulation ratio).
4. Arrangement of downcomer and riser circuits in the drum.
5. Size of the steam drum.

Operating Factors

1. Boiler-water analysis.
2. Type of steam load.
3. Water level carried.

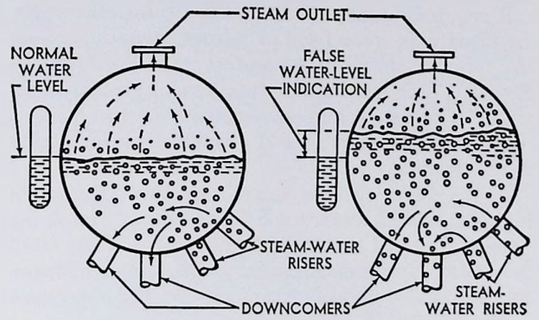


Fig. 7. *Effect of rate of steam generation on steam separation in an unbaffled boiler drum*

In steam drums without separation devices, where separation is by gravity only, the manner in which some of the above items affect separation is indicated in simplified form in Figs. 7 and 8.

For a low rate of steam generation (up to about 3 ft/sec velocity of steam leaving the water surface) there is sufficient time for the steam bubbles to separate from the mixture by gravity without being drawn into the downcomers and without carrying entrained water droplets into the steam outlet (see Fig. 7a). However, for this same arrangement at a higher rate of steam generation (Fig. 7b) the time is insufficient to attain either of these desirable results. Moreover, the dense upward traffic of steam bubbles in the mixture may cause a false water level, as indicated.

The effect of the location of the riser circuits in relation to the water level is illustrated in diagrams a and b, Fig. 8. Neither arrangement is likely to yield desirable results in a drum where gravity alone is used for separation.

It is apparent from the above that separation by the action of gravity alone is possible if the velocity

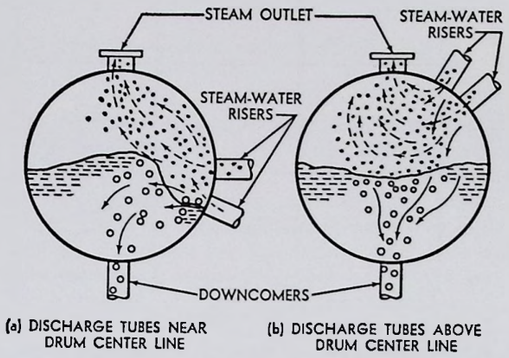


Fig. 8. *Effect of location of discharge from risers on steam separation in an unbaffled boiler drum*

of either the mixture or the steam bubbles within the mixture is sufficiently low, but the arrangement may be uneconomical. For gravity separation in a single drum, the steam generated per ft length of drum must be kept extremely low. A single drum under these conditions is generally uneconomical except for small low-duty boilers. Using multiple drums of reasonable size in series, as in Fig. 10a, somewhat higher steam outputs per ft length of drum are possible with gravity separation.

Operating pressure has an effect on the natural tendency of steam and water to separate. The relationship between pressure and the differential in the specific weights of water and steam (Fig. 3) is indicative of this effect. In the separation of steam from water, the limiting velocity of a water particle conveyed in steam and the force of gravity both vary directly with the differential in the specific weights of the water and the steam (Stokes's law, see Chapter 8). Hence, as the sp wt differential diminishes with increase in operating pressure, so does the force of gravity available for separation. The effect of increasing operating pressure (above 300 psi for comparison) on steam flow per unit of flow area, on steam velocity above the drum water level, and on the force of gravity for separation is typically shown in Fig. 9. It will be noted that the percentage drop in gravity-separating force closely follows the drop in limiting velocity.

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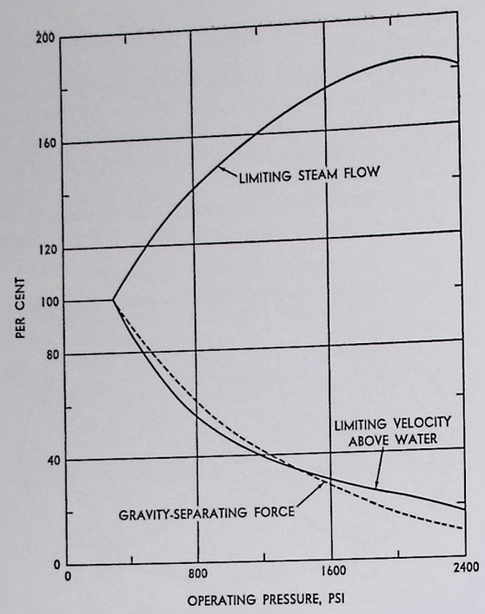


Fig. 9. *Typical effect of increase in operating pressure (above 300 psi for comparison) on 1) limiting steam flow per unit of flow area, 2) limiting velocity above water, and 3) gravity-separating force—steam and water*

As steam pressure increases, sp wt of steam increases, and the size of steam bubbles in the mixture decreases. Consequently, the velocity of the mixture leaving the riser circuits is reduced, which justifies a higher duty per ft of drum length, for a given size of drum, or the use of a smaller drum.

Boiler-Water Characteristics

Steam contamination (solid particles in the superheated steam) comes from the boiler water largely in the carry-over of water droplets. Therefore, in general, as boiler-water concentration increases, steam contamination may be expected to increase. The tendency to foam formation also increases with boiler water high in dissolved solids, but a strict correlation of this tendency with boiler-water analyses is not yet available despite extensive research.

The need for extreme purity of steam for use in high-pressure turbines has prompted the development of highly satisfactory devices for separating steam from water in a boiler drum. Consequently, steam contamination has been steadily reduced. Troublesome deposits on turbine blades may occur with surprisingly low (0.6 ppm) total solids contamination in steam. In the 500 to 900 psi range these deposits are usually water soluble and can be removed by periodic washing. In the 1000 to 1500 psi range, however, silica deposits predominate, and these deposits are not easily removed by water-washing. With operating pressures of 2000 psi and above, insoluble deposits appear to diminish. In some instances soluble deposits do occur, which may be controlled by water-washing at intervals.

With a rapid increase in boiler load, steam pressure usually starts to decrease as the firing rate is increased. The volume of steam throughout the boiler is increased, and the resulting "swell" raises the water level in the drum. The amount of the rise in water level depends upon the rate of load change and its magnitude. Where rapid load changes are anticipated, the steam drums should be large enough to prevent carry-over from this action.

When there is a foam layer in the steam drum, the steam space available for gravity steam separation is decreased, and separation is adversely affected. Priming may occur. Judicious lowering of the normal water level may relieve the tendency to moisture carry-over.

Mechanical Steam Separators for Drums

As noted above, gravity steam separation alone is generally unsatisfactory for boilers in the usual sizes and required duties. Most steam drums, therefore, are fitted with some form of primary separator. Simple types of primary separators are illustrated in Fig. 10. These facilitate or supplement gravity separation. The extent and arrangement of the various baffles and deflectors should always allow for access to the drums.

In the B&W cyclone steam separator, shown in Figs. 11 and 12, centrifugal force (many times the force of gravity) is used to separate the steam from the water. Cyclones, essentially cylindrical in form, and corrugated scrubbers are the basic components of this type of separator.

The cyclones are arranged internally along the length of the drum, and the steam-water mixture is admitted tangentially. The water forms a layer against the cylinder walls, and the steam (of less density) moves to the core of the cylinder and then upward. The water flows downward in the cylinder and is discharged through an annulus at the bottom, below the drum water level. Thus, with the water returning from drum storage to the downcomers virtually free of steam bubbles, maximum net head is available for producing flow in the circuits, which is the important factor in the successful use of natural circulation, as previously noted. The steam moving upward from the cylinder passes through a small corrugated scrubber at the top of the cyclone (see Figs. 11 and 12) for additional separation. Under many conditions of operation no further refinement in separation is required, although the cyclone separator is considered only as a primary separator.

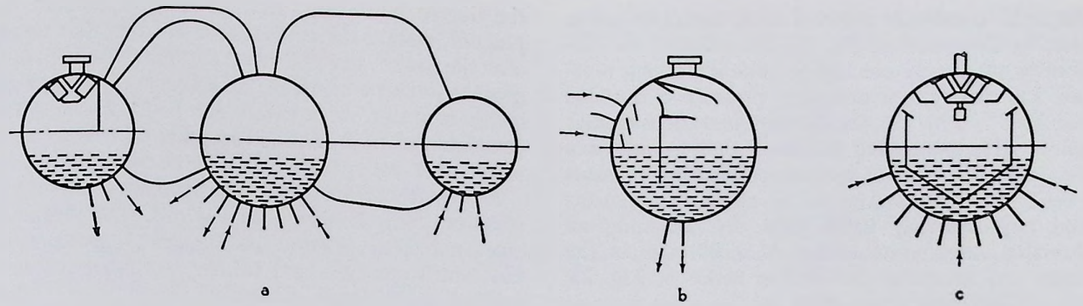


Fig. 10. *Simple types of primary steam separators in boiler drums: a) deflector baffle, b) another type of deflector baffle, and c) compartment baffle*

When wide load fluctuations and variations in water analyses are expected, large corrugated scrubbers may be installed at the top of the drum (also shown in Figs. 11 and 12) to provide nearly perfect steam separation. These scrubbers may be termed secondary separators. They provide a large surface which intercepts water particles as the steam flows sinuously between the closely fitted plates. Steam velocity through the corrugated-plate assembly is very low, and thus re-entrainment of water is avoided. The collected water is drained from the bottom of the scrubber assembly to the water below.

Single or double rows of cyclone separators are installed in boiler drums, with ample room for access. For smaller boilers (FH and Two-Drum Stirling) at lower pressures (100 psi), the separation of clean steam (see below) by single and double rows of cyclone separators is in the order of 4000 and 6000 lb, respectively, per hr per ft of drum length and, at pressures around 1050 psi, increases to 9000 and 15,000 lb, respectively, per hr per ft of drum length. For large utility boilers (RB) at pressures as high as 2500 psi, separation by single and double rows of cyclone separators is in the order of 15,000 and 30,000 lb, respectively, per hr per ft of drum length.

The combination of cyclone separators and scrubbers described above provides the means for obtaining, under a wide variation in operating conditions, steam purity in the order of less than 1.0 ppm solids content, a result that is commercially quite acceptable. However, further refinement in steam purification is required to remove boiler-water salts, such as silica, which are entrained in the steam by vaporization or solution mechanism. Washing the steam with condensate or feedwater of acceptable purity may be used for this purpose.

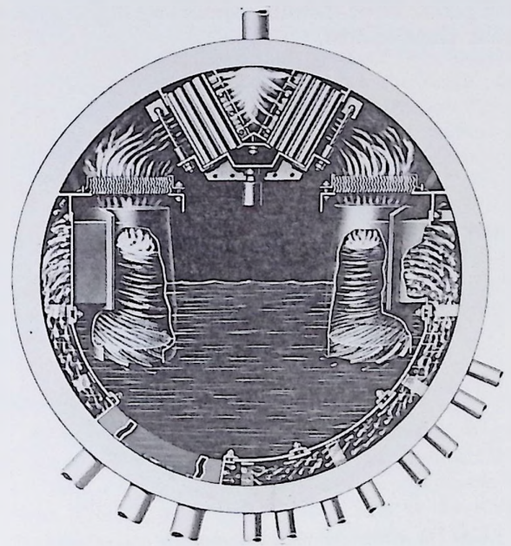


Fig. 11. *Double-row arrangement of cyclone-type primary steam separators, with scrubber elements at top of drum for secondary separation*

Steam-Washing

For any given operating condition, silica is distributed between boiler water and steam in a def-

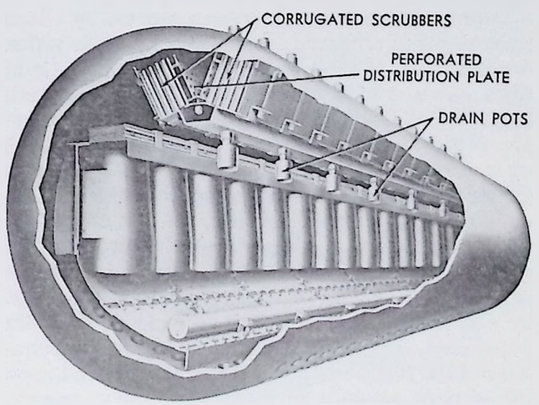


Fig. 12. *Single-row arrangement of cyclone steam separators with scrubbers*

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inite ratio, commonly referred to as the distribution ratio. As illustrated in Fig. 13, the value of the distribution ratio increases rapidly with increasing pressure. For example, if operating pressure is doubled from 1300 to 2600 psi, the distribution ratio increases tenfold. Experience has indicated that troublesome silica deposits will not form in condensing turbines if the silica concentration in the steam does not exceed approximately 0.025 ppm. By assuming an allowable silica concentration of 0.025 ppm in the steam and using the distribution ratios of Fig. 13, it is evident that above 2000 psi the silica content in the boiler water must be kept well below 1 ppm if turbine fouling is to be avoided. Since it is often impractical to maintain boiler-water concentrations of silica sufficiently low to prevent turbine fouling, other measures, such as steam-washing, must be used to control this type of steam contamination.

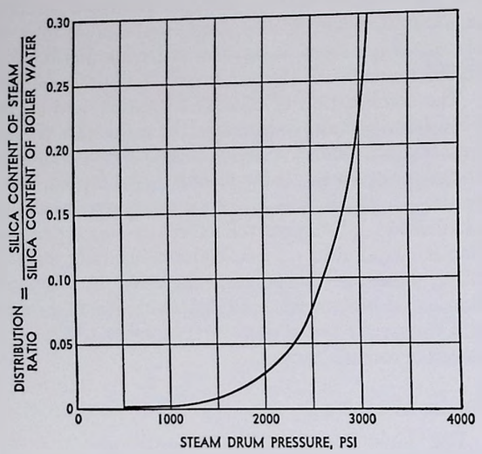


Fig. 13. *Effect of pressure on silica distribution ratio*

Steam-washing is an absorption process by which silica vapor is absorbed from steam by wash water. A net transfer of silica takes place when the ratio of silica concentration in steam to that in the wash water differs from the distribution ratio. For example, if pure steam is in contact with water of high silica content, silica will transfer from the water to the steam. This action will continue until the distribution ratio has been established. In steam-washing, silica-laden steam is brought into intimate contact with relatively pure wash water, such as condensate or feedwater, and silica is absorbed from the steam by the wash water. The actual amount of silica transferred, and therefore the final silica concentrations of the steam and wash water, will depend upon the relative amounts of steam and water present, the degree of mixing, and the time of contact. In practice, neither the contact area between the steam and the water nor the contact time is sufficiently great to permit a complete interchange of silica from steam to water. As a result, the ratio of the silica concentration of the washed steam to that of the wash water does not quite reach theoretical equilibrium.

A typical steam-drum arrangement employing steam-washing is shown in Fig. 14. The drum is equipped with primary mechanical separators of the centrifugal type and primary scrubbers. Steam leaving the primary separators flows to a steam washer arranged in the top of the steam drum. The washer consists of a rectangular column approximately the length of the steam drum. Steam passes vertically upward through a perforated plate, a pack of stainless steel wire mesh, a second perforated plate, and finally a corrugated scrubber element. Wash water enters the drum through a nozzle and flows downward through the washer, counterflow to the steam. The steam velocity through the tray perforations maintains, above each tray, a layer of wash water which is kept in violent agitation by the steam. The wire mesh provides a large surface area for achieving intimate contact between the steam and the wash water.

In many installations where it is impractical to maintain the silica content of the boiler water low enough to prevent turbine deposits, the use of steam washers will eliminate these deposits. In other cases where it is now possible to prevent turbine deposits by large blowdown rates with resultant low silica content in the boiler water, the use of steam washers will permit lower blowdown rates and higher boiler-water silica content.

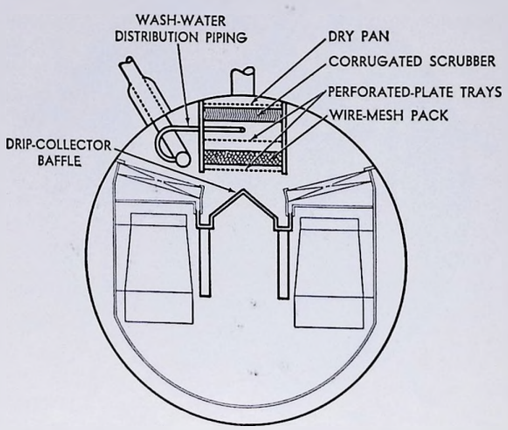


Fig. 14. *Arrangement of steam-drum internals for washing silica-laden steam*

10

STEAM AND OTHER ENERGY CYCLES

STEAM generated in power boilers is used in prime movers for the development of mechanical work, as a source of heat for various processes, and for a combination of these purposes. The central station public utility and the steamship are examples of its use for the development of mechanical work. The heating system of a building exemplifies its use as process heat. A steam plant in a paper mill or in a chemical factory is an example of its use for a combination of the two.

Steam is particularly favored for many heating operations because vapor pressure is an ideal thermostat. This fact and steam's high latent heat, moderate density, and reasonable pressure make it the prevalent economical process fluid. The steam tables (Appendix, page 10-A1 ff) give some of the physical properties of steam. The first two columns of Tables 1 and 2 give the vapor-tension relationship for water and exactly define the vapor pressure that must be established for equilibrium between the liquid and vapor phases for any specified temperature. Steam may be heated beyond this saturation condition to any desired higher value of temperature. Properties for this superheated condition are given in Table 3.

The steam tables also contain data on specific volume, sensible heat, and latent heat, which are essential for determination of the suitability of steam for heat purposes. There are also other data, the significance of which is developed below in a consideration of the thermodynamic aspects of the problem.

Laws of Thermodynamics

Thermodynamics is concerned with the mechanics of heat and especially with the conversion of heat into work. There are two basic principles involved, which have become firmly established in the scientific field as the first and second laws of thermodynamics. These principles can be variously stated, and the reader should consult any standard book on thermodynamics for a complete statement.

According to the law of the conservation of energy, or the first law of thermodynamics, energy can be neither created nor destroyed. Thus, in any device or system into which energy is put and out of which some energy is taken, this principle means that an exact balance must prevail between the positive and negative quantities. There can be no excess or deficiency. Energy may be converted within the device from one form to another, such as heat to work, but there must be complete conservation of the aggregate. This law can be expressed in symbols, as in equation (1), and diagrammatically, as in Fig. 1. To account for the energy flow, in at 1 and out at 2 in Fig. 1, which may represent a boiler, a turbine, a compressor, or any thermodynamic system, assuming for convenience that 1 lb of working fluid moves through the device, an equation can be written, as follows:

$$V_1^2/2g + p_1v_1 + u_1 + Z_1 \pm W_k \pm {}_1Q_2 = V_2^2/2g + p_2v_2 + u_2 + Z_2 \qquad (1)$$

This is the *general energy equation,* in which:

V = the flow velocity through the intake and discharge pipes, ft/sec

g = gravitational acceleration, ft/sec^2

$V^2/2g$ = the kinetic energy, ft-lb/lb

p = the static pressure under which the fluid exists at the inlet and outlet pipes, lb/sq ft abs

v = the specific volume of the fluid, cu ft/lb

pv = the flow work, ft-lb/lb

u = the internal, or intrinsic, energy of the fluid, Btu/lb

Z = the elevational head (potential energy), ft-lb/lb

W_k = work done on or by the fluid while it is in the device, ft-lb/lb

${}_1Q_2$ = heat added to or abstracted from the fluid while it is in the device, Btu/lb

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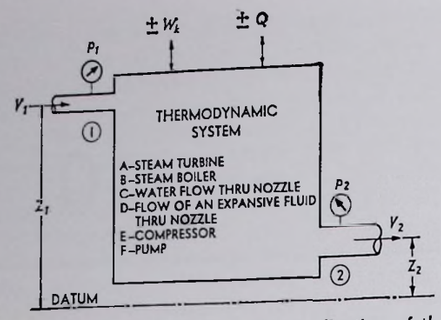


Fig. 1. *Diagram to illustrate the application of the general energy equation to thermodynamic systems*

In using the general energy equation (1), it is essential that all terms be expressed in a consistent system of units. It is not customary to measure all the energy items in the same units, hence the diversity in equation (1). Thus, elevation head, Z, is measured in ft; the kinetic energy, $V^2/2g$, the flow work, pv, and the work done, W_k, are usually all measured in ft-lb. On the other hand, the intrinsic energy, u, and the heat added, ${}_1Q_2$, are generally measured in Btu. Joule's experiments based on the law of the conservation of energy give the mechanical equivalent of heat. The heat unit in use today is the international Btu. This is defined rigorously in terms of the electrical unit, the kilowatthour, so the exact conversion is given by the relation:

$$1 \text{ Btu} = 778.26 \text{ ft-lb} \qquad (2)$$

This conversion factor (usually taken as 778 in engineering work) will serve to give the necessary consistency in the use of equation (1), which may be rewritten as:

$$\frac{V_1^2}{2g \times 778} + \frac{p_1v_1}{778} + u_1 + \frac{Z_1}{778} \pm \frac{W_k}{778} \pm {}_1Q_2 = \frac{V_2^2}{2g \times 778} + \frac{p_2v_2}{778} + u_2 + \frac{Z_2}{778} \qquad (3)$$

The terms pv and u, representing, respectively, the flow work and the intrinsic energy of a fluid, can be treated conveniently as a group. They are then usually expressed as a single symbol, h, called enthalpy, thus:

$$h = pv/778 + u, \text{ Btu/lb} \qquad (4)$$

Enthalpy values for water and steam are given in the steam tables (Appendix, page 10-Al ff).

The general energy equation can be applied to specific equipment examples (see Fig. 1) as follows, in each case considering the performance to be for a theoretically perfect machine.

(A) Steam Turbine

The change in elevational head, $\triangle Z$, the change in flow velocity, $\triangle V$, from throttle, 1, to exhaust, 2, and the heat transferred to the surroundings, ${}_1Q_2$, are all zero. Hence, energy equation (3) reduces to:

$$p_1v_1/778 + u_1 = p_2v_2/778 + u_2 + W_k/778$$

or $$h_1 = h_2 + W_k/778$$

or $$W_k/778 = h_1 - h_2 \qquad (5)$$

From equation (5) it is evident that the work done, $W_k/778$, for an ideal steam turbine is equal to the difference between the enthalpy, h_1, of the steam entering and the enthalpy, h_2, of the steam leaving.

(B) Steam Boiler

The change in elevational head, $\triangle Z$, the change in flow velocity, $\triangle V$, from feed introduction, 1, to steam take-off, 2, and the work done, W_k, are all zero. The energy equation (3) therefore reduces to:

$$p_1v_1/778 + u_1 + {}_1Q_2 = p_2v_2/778 + u_2$$

or $$h_1 + {}_1Q_2 = h_2$$

or $${}_1Q_2 = h_2 - h_1 \qquad (6)$$

From equation (6) it is evident that the heat added, ${}_1Q_2$, for the conversion of water into steam in a boiler is equal to the difference between the enthalpy, h_2, of the steam leaving the boiler and the enthalpy, h_1, of the feedwater entering the boiler.

(C) Water Flow through a Nozzle

For water flowing through a nozzle, the change in elevational head, $\triangle Z$, the change in specific volume, $\triangle v$, the change in intrinsic energy, $\triangle u$, the work done, W_k, and the heat added, ${}_1Q_2$, are all zero. Energy equation (3) therefore reduces to:

$$V_1^2/(2g \times 778) + p_1v/778 = V_2^2/(2g \times 778) + p_2v/778$$

or $$V_2^2/2g - V_1^2/2g = (p_1 - p_2)v \qquad (7)$$

The increase in kinetic energy, or the acceleration, of the fluid is given in equation (7). If the velocity of approach, V_1, is zero, then equation (7) becomes:

$$V_2^2/2g = (p_1 - p_2)v$$

or $$V_2 = \sqrt{2g(p_1 - p_2)v} \qquad (8)$$

But $(p_1 - p_2)$ is the difference between the static pressures upstream and downstream, which, when multiplied by the specific volume, v, gives the static head, H:

$$(p_1 - p_2)v = H \qquad (9)$$

where H is measured in ft-lb/lb, or ft. The velocity of a free-spouting jet is then by substitution in equation (8):

$$V_2 = \sqrt{2gH} = 8.02\sqrt{H}, \text{ ft/sec} \qquad (10)$$

(D) Flow of an Expansive Fluid through a Nozzle

If steam, air, or any other expansive fluid is substituted for the flow conditions given under (C), the changes in the specific volume, $\triangle v$, and in the intrinsic energy, $\triangle u$, are no longer zero, and energy equation (3) becomes:

$$\frac{V_1^2}{2g \times 778} + \frac{p_1v_1}{778} + u_1 = \frac{V_2^2}{2g \times 778} + \frac{p_2v_2}{778} + u_2$$

or $$V_1^2/(2g \times 778) + h_1 = V_2^2/(2g \times 778) + h_2$$

or $$V_2^2/(2g \times 778) - V_1^2/(2g \times 778) = h_1 - h_2 \qquad (11)$$

If the velocity of approach, V_1, is zero, then:

$$V_2^2/(2g \times 778) = h_1 - h_2$$

or $$V_2 = \sqrt{2g \times 778(h_1 - h_2)} \qquad (12)$$

From equation (12) it is evident that the velocity of a free-spouting jet of an expansive fluid, such as steam or air, is a function of the enthalpies (h_1 and h_2) of the fluid entering and leaving the nozzle. This equation may be reduced to:

$$V_2 = 223.9\sqrt{h_1 - h_2}, \text{ ft/sec} \qquad (13)$$

(E) Compressor

If an expansive fluid moves through a well-insulated compressor, then the change in elevational head, $\triangle Z$, the change in velocity, $\triangle V$, from inlet, 1, to discharge, 2, and the heat transferred to the surroundings, ${}_1Q_2$, are all zero, and energy equation (3) reduces to:

$$p_1v_1/778 + u_1 + W_k/778 = p_2v_2/778 + u_2$$

or $$h_1 + W_k/778 = h_2$$

or $$W_k/778 = h_2 - h_1 \qquad (14)$$

From equation (14) it is evident that the work done, $W_k/778$, by the ideal compressor is equal to the difference between the enthalpies of the fluid leaving and entering the unit (h_2 and h_1).

(F) Pump

When a nonexpansive fluid, such as water, is substituted in (E), then the changes in specific volume, $\triangle v$, and in intrinsic energy, $\triangle u$, become zero, and energy equation (3) reduces to:

$$p_1v/778 + W_k/778 = p_2v/778$$

or $$W_k = (p_2 - p_1)v \qquad (15)$$

but $(p_2 - p_1)\,v = H$, the static head in ft of fluid flowing, and equation (15) becomes:

$$W_k = H \qquad (16)$$

Equation (16) is a rigorous demonstration of the principle that, in pumping a truly noncompressible fluid, all the energy added as work goes ideally to raising the static head. There is no change in temperature.

Second Law of Thermodynamics

The examples (A) to (F), noted above, represent some of the basic manipulations which can be made in applying the general energy equation to specific apparatus. The great significance of the property of enthalpy is amply demonstrated by the examples cited. All of these manipulations are based on the first law of thermodynamics. They account for all energy added or abstracted, and the conclusions satisfy the requirement for an exact balance (thermal) between the energy supplied to a system and the energy rejected from it.

The general energy equation, however, has one very serious shortcoming—it tells practically nothing of the possible extent of conversion of one form of energy into another in the many kinds of apparatus found in practice. Thus, if 100 heat units were added to a system, how much could be converted into work? According to the first law, not more than the 100 units could be converted, but nothing is said about how much less than the 100 units would be converted. The basic purpose of the second law of thermodynamics is rigorously to define the extent of possible conversion of heat into work. This concept is very broad, very tantalizing, and yet most useful in all fuel-burning power plants, whether steam, internal combustion, gas turbine, or nuclear.

Therefore, to define the extent of the possible conversion of heat energy into work, the first law of thermodynamics and the general energy equation must be supplemented by the second law. By using the concept of entropy, it is possible to represent heat added (+) or heat rejected (−) as areas and so demonstrate the extent of this possible conversion. For ideal processes this property (entropy) is defined methematically (assuming unit mass of the substance) as:

$$ds = dQ/T \qquad (17)$$

where:

ds = entropy change, Btu/lb, F
dQ = heat added, Btu/lb
T = absolute temperature at which heat is added, R = F + 460

Entropy can be more generally defined as the property which measures that portion of the heat added which cannot be converted into work no matter how nearly perfect the operation may be. Or expressed in another way, it is the measure of unavailability.

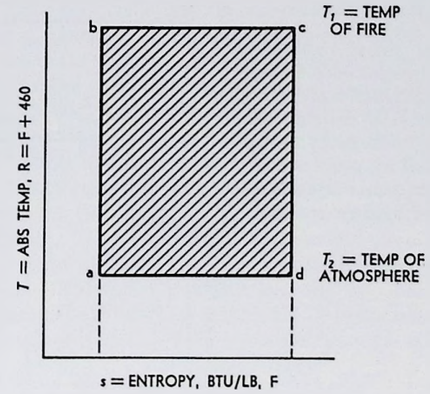


Fig. 2. *Carnot cycle, temperature-entropy diagram*

The limitations imposed by the second law are perhaps best illustrated by considering the temperature-entropy (T-s) diagram, Fig. 2, where absolute

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temperature is plotted vertically, entropy horizontally, and areas represent heats added (+) or rejected (−). Consider any two temperatures such as T_1, which might be the temperature of a fire (the heat source), and T_2, the temperature of the heat sink, which for practical purposes is limited to a minimum value set by atmospheric temperature. Then by Carnot's principle, the maximum area which can be constructed on the T-s diagram between these two temperatures is the rectangle as shown. The net crosshatched area represents the ideal work of this cycle. The area under T_1 represents the heat added from the source, and the area under T_2 represents the heat rejected to the sink. The thermal efficiency of any ideal cycle is defined as:

$$\text{Thermal efficiency} = \frac{\text{Work done}}{\text{Heat added}} \qquad (18)$$

By the first law this can be written as:

$$\text{Thermal eff} = \frac{\text{Heat added} - \text{Heat rejected}}{\text{Heat added}} \qquad (19)$$

For the Carnot cycle this is resolved to the form:

$$\text{Thermal efficiency} = \frac{T_1 - T_2}{T_1} \qquad (20)$$

The value of efficiency given by equation (20) is a rigorous maximum limit imposed by the second law of thermodynamics. No cycle can have a better efficiency between specified temperature limits than the value given for the Carnot cycle. Other cycles may equal it, but none can exceed it.

Because the atmosphere is the usual thermodynamic sink and its temperature is of the order of 500 deg R, it is evident from equation (20) that the conversion efficiency of heat into work is substantially less than 100%. By what extent it is less is measured by the heat which must be thrown away as heat. This is the unavailable portion which cannot be converted into work. The entropy, s_a or s_d in Fig. 2, is the practical measure of this unavailability.

In the cycles of real thermal power plants there is practically only one source of heat (T_1)—the fuel, and one sink (T_2)—the atmosphere, to which the residue must be discarded. As an example, these temperatures can be taken as 3000 deg R and 500 deg R, respectively. Any ideal heat engine working between these temperatures will then have a limiting thermal efficiency as set by the Carnot cycle. From equation (20) this is:

$$\text{Thermal efficiency} = \frac{3000 - 500}{3000} = 0.83 \qquad (21)$$

No power plant actually even approaches this value. The best steam plants and internal combustion engines give thermal efficiencies between 0.35 and 0.40 over-all. These are the best values obtained in practice, and the average is much lower.

Using the temperatures noted in equation (21), a Carnot cycle, where a fixed gas like air is the working fluid, is plotted on both the p-V_L and T-s planes, as shown in diagrams (A), Fig. 3. In this cycle there are two isothermal and two isentropic processes. This cycle has never been attained for any real engine using a fixed gas as the working fluid. Isothermal compression and expansion cannot be realized in practice. The cycle gives a very low mean effective pressure, which is defined as:

$$\text{MEP} = \frac{\text{Work of cycle}}{\text{Displacement}} \qquad (22)$$

The Otto, Diesel, and Brayton cycles variously overcome the practical limitations of the Carnot gas cycle and are represented, respectively, in modern practice by the gasoline engine, the oil engine, and the gas turbine. These cycles are shown in diagrams (B), (C), and (D), Fig. 3, for the same temperature limits set in the specification of the Carnot cycle.

The ratio of compression (V_{La}/V_{Lb}, p_b/p_a, or T_b/T_a) is the sole element, thermodynamically, which influences the thermal efficiency of the Otto (B) and the Brayton (D) cycles. With the Diesel cycle, diagram (C), Fig. 3, the length of the combustion line (phase b-c) must be included as a second variable.

Calculated thermal efficiencies of the Otto, Brayton, and Diesel cycles for the cold-air standard are shown in Fig. 4 and are derived from the first two of the following equations:

Thermal efficiency, Otto and Brayton cycles =

$$1 - T_a/T_b = 1 - (V_{Lb}/V_{La})^{\kappa-1} \qquad (23)$$

Thermal efficiency, Diesel cycle =

$$1 - (V_{Lb}/V_{La})^{\kappa-1}\left\{\frac{(V_{Lc}/V_{Lb})^{\kappa} - 1}{\kappa(V_{Lc}/V_{Lb} - 1)}\right\} \qquad (24)$$

Thermal efficiency, Carnot cycle (as noted above)

$$= (T_b - T_a)/T_b = 1 - T_a/T_b$$

In the above equations, V_L and T are the volumes and absolute temperatures, respectively; the subscripts are as given in Fig. 3; and $\kappa = c_p/c_v$, the ratio of specific heats for the gas. In each of the T-s diagrams of (B), (C), and (D), Fig. 3, the Carnot cycle has been sketched in to show the magnitude of the deviations and losses of each of the other gas cycles with respect to the Carnot cycle.

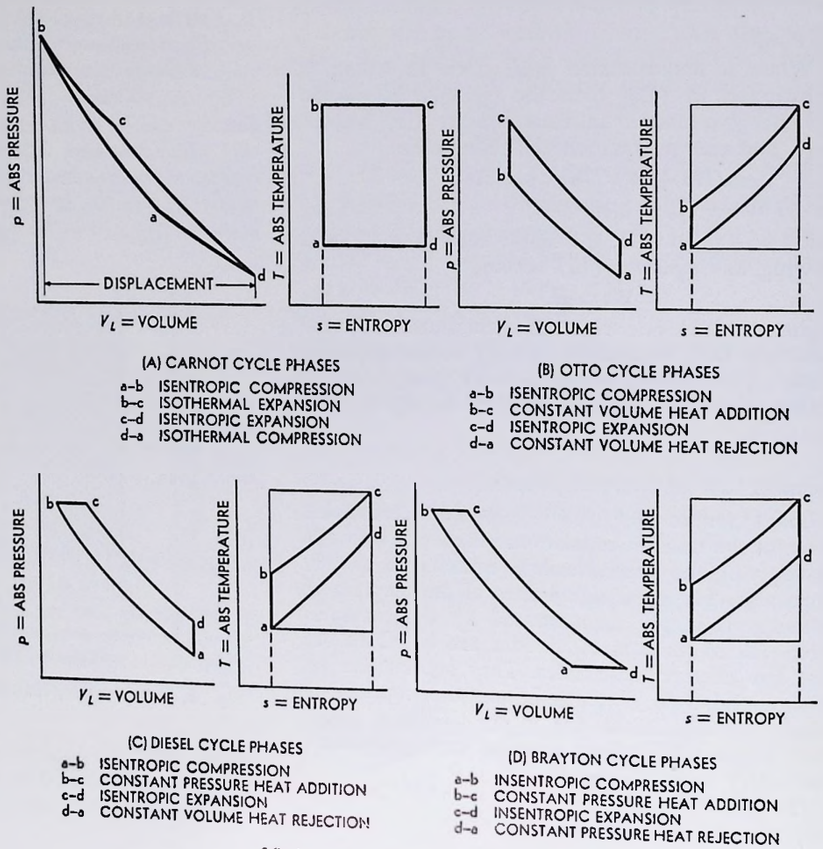


Fig. 3. *Pressure-volume and temperature-entropy diagrams for Carnot, Otto, Diesel, and Brayton gas cycles. Work of cycle represented by area a-b-c-d. MEP = work of cycle divided by displacement*

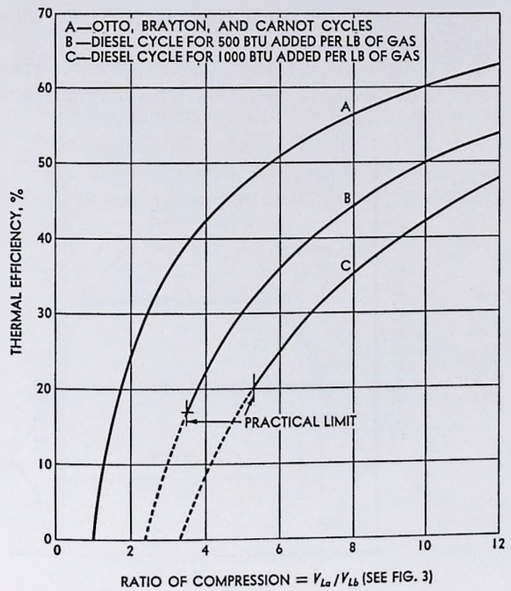


Fig. 4. *Thermal efficiencies of ideal gas cycles as a function of the compression ratio*

The Carnot Steam Cycle

Despite the lower working-fluid temperatures, steam systems are competitive in thermal efficiencies with internal combustion systems. This is so because in steam systems a fluid that changes state (evaporation and condensation) is substituted for the isothermal phases of the fixed-gas Carnot cycle. Thus, in accordance with the vapor-tension relation, if steam, in the wet region between water and saturated vapor, is the working fluid, constant pressure is the same as constant temperature. Such a situation is represented in the p-V_L and T-s diagrams of Fig. 5, with complete isentropic expansion and compression connecting the throttle (T_1) and the exhaust (T_2) conditions. This situation is approached in uniflow-steam-engine plants. Thus, for a saturated steam temperature of 540 F (1000 R) and for the same sink temperature of 500 R as before, the thermal efficiency of the Carnot steam cycle by equation (20) is:

$$\text{Thermal efficiency} = \frac{1000 - 500}{1000} = 0.5 \qquad (25)$$

This value is substantially lower than under the conditions previously established for direct-fired operations, equation (21), as 0.83 for the Carnot gas cycle. However, the value of 0.5 is entirely comparable with the ideal efficiencies for the Otto, Brayton,

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and Diesel cycles, as given in Fig. 4. Basically, therefore, the steam cycle remains competitive.

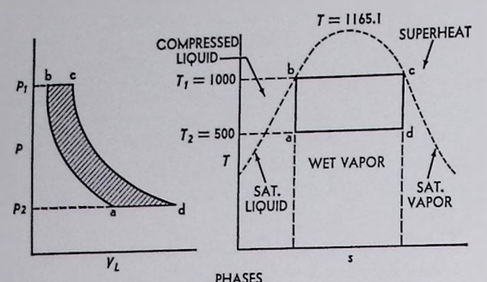


Fig. 5. *Carnot cycle with steam as working fluid*

The Rankine Cycle

The Carnot steam cycle, Fig. 5, is not realistic of modern steam plant practice because compression (phase a-b) is nonexistent in steam turbines. It also omits superheat, reheat, and regeneration. The Rankine cycle more closely defines practice with a steam turbine and is equally applicable to wet, dry, and superheated steam. This cycle is shown schematically in Fig. 6 where, in the p-V_L diagram, the two constant-pressure phases of admission (4-1) and exhaust (2-3) are connected by an isentropic-expansion phase (1-2). The work of this cycle, shown in the designated area of Fig. 6, is most conveniently evaluated by thermal methods, using the properties of steam from the steam tables (Appendix, page 10-A1 ff), the Mollier chart for steam (Appendix, page 10-A6), and the general energy equation (3). It was demonstrated in the derivation of equation (5) that the work of the ideal prime mover, diagram (A), Fig. 6, is found exactly as:

W_k/778, prime mover $= h_1 - h_2$, Btu/lb (26)

where h_1 and h_2 are the initial and final enthalpies of the steam entering and leaving the prime mover. The isentropic relationship between h_1 and h_2, most conveniently manipulated by using a vertical line on the Mollier chart, is shown in diagram (B), Fig. 6. The water rate, or steam rate, (WR) of the prime mover follows from the value of work, equation (26), and the mechanical equivalents of 3412.75 Btu/kwhr and 2545.1 Btu/hp-hr (usually taken as 3413 and 2545, respectively, in engineering work), thus:

$$\text{WR} = 3413 \times 778/W_k, \text{ lb/kwhr} \quad (27)$$

or

$$\text{WR} = 2545 \times 778/W_k, \text{ lb/hp-hr} \quad (28)$$

Equations (26), (27), and (28) are most useful in the calculation of work and water rate, but these values must be related to the heat supplied (Q added) in order to determine thermal performance. This goes far beyond the prime mover and requires the inclusion of accessory equipment in the steam plant. The arrangement of equipment in a rudimentary condensing steam power plant is shown schematically by Fig. 7. In addition to the prime mover there are the boiler, the condenser, and the feed pump. The functional elements of boiler, prime mover, condenser, and feed pump are essential for a workable steam plant that complies with the basic thermodynamic cycle.

The T-s diagram (C), Fig. 6, again serves to show the nature of the entire operation. This diagram must

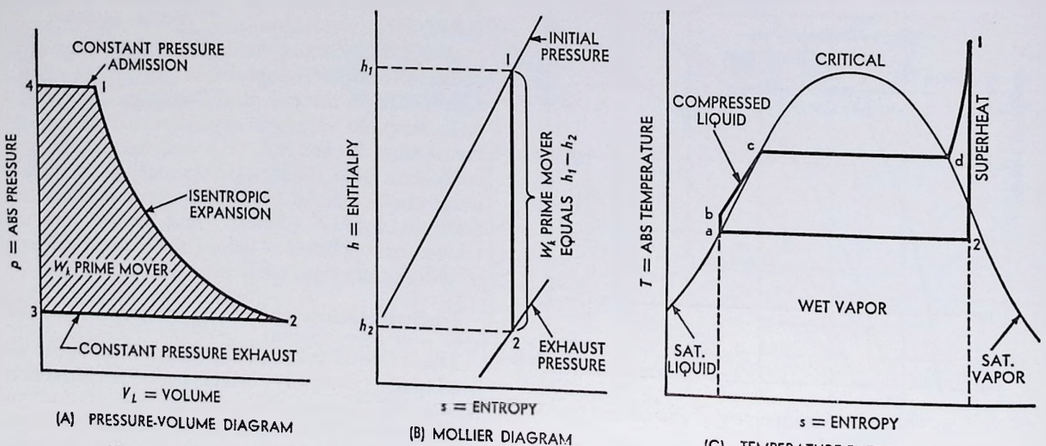


Fig. 6. *Pressure-volume, Mollier, and temperature-entropy diagrams for ideal Rankine cycle*

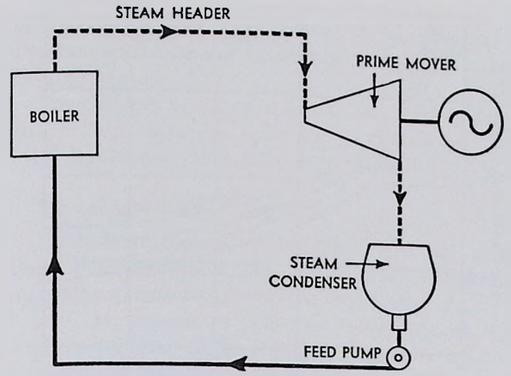


Fig. 7. *Diagram of rudimentary steam power plant*

contain, in graphical form, the properties for liquid, wet vapor, and superheat, as taken from the steam tables. The Rankine cycle can then be superimposed, as shown, where the heat added (Q added) is the area under the irregular line b-c-d-1; the heat rejected (Q rejected to the condenser) is the area under the line 2-a; the isentropic expansion in the prime mover is the vertical line 1-2; and the isentropic compression in the feed pump is the vertical line a-b. Usually the compression phase is so small as to be negligible, and the points a and b can, therefore, be considered as the single point, saturated liquid at the exhaust pressure. The compressed-liquid phase need only be recognized for pressures higher than 1000 psi. The feed-pump p-V_L diagram, or indicator card, is substantially a rectangle, as shown in Fig. 8. The net work of the Rankine cycle for the complete power plant of Fig. 7 is then:

Net work of cycle =

Prime-mover work — Feed-pump work (29)

Net work $= h_1 - h_2 - W_{FP}/778$ (30)

The heat supplied (Q added) to make steam is the difference between the enthalpy, h_1, of the steam leaving the boiler and the enthalpy, h_b, of the feed entering the boiler, diagram (C), Fig. 6. The value h_b is for compressed liquid and is most conveniently found by using the saturated liquid value, h_a (called h_f hereafter), from the steam tables and adding to it the feed-pump work, W_{FP}, or:

Q added $= h_1 - h_b$ (31)

Q added $= h_1 - (h_a + W_{FP}/778)$ (32)

Q added $= h_1 - h_f - W_{FP}/778$ (33)

The thermal efficiency of the Rankine cycle is then from equations (18), (30), and (33):

Thermal efficiency, Rankine cycle =

$$\frac{h_1 - h_2 - W_{FP}/778}{h_1 - h_f - W_{FP}/778} \quad (34)$$

As explained above, if steam pressures are not higher than 1000 psi, equation (34) may be simplified to the approximate form:

Approx thermal efficiency, Rankine cycle =

$$\frac{h_1 - h_2}{h_1 - h_f} \quad (35)$$

The heat rate (HR), which is the heat supplied per unit of power output, is generally preferred to thermal efficiency because it is more directly applicable to fuel performance. It is defined as:

HR = 3413/Thermal efficiency, Btu/kwhr (36)

and on the horsepower basis:

HR = 2545/Thermal efficiency, Btu/hp-hr (37)

These several equations for the Rankine cycle are equally applicable to steam that is wet, dry, or superheated. The influence of steam quality (wet, dry, and superheated) is evident from the several Rankine cycle diagrams drawn in Fig. 9. It will be noted that the shape of the Rankine cycle diagram is greatly

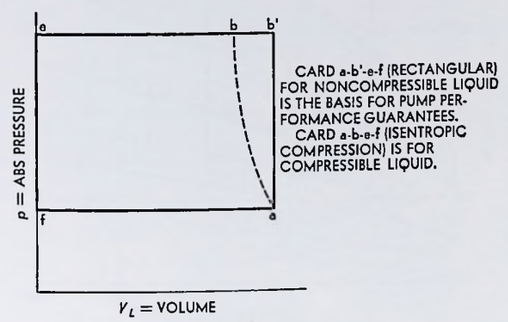


Fig. 8. *Indicator card or p-V_L diagram for ideal feed pump*

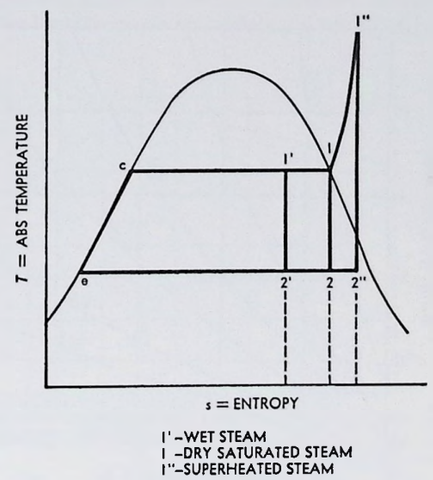
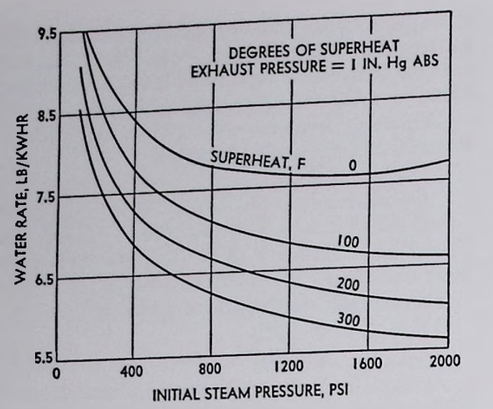
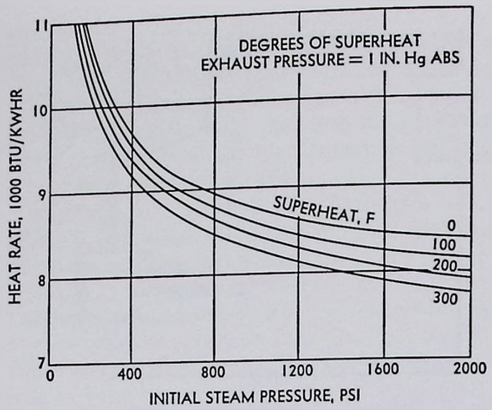
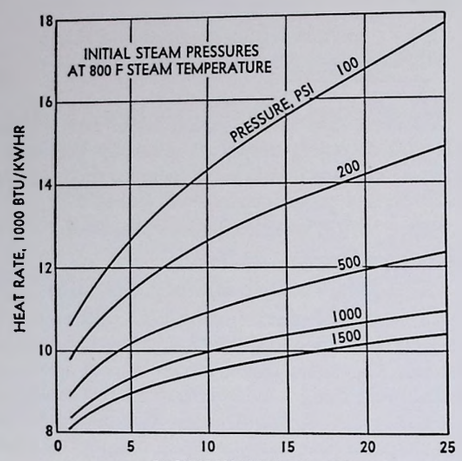


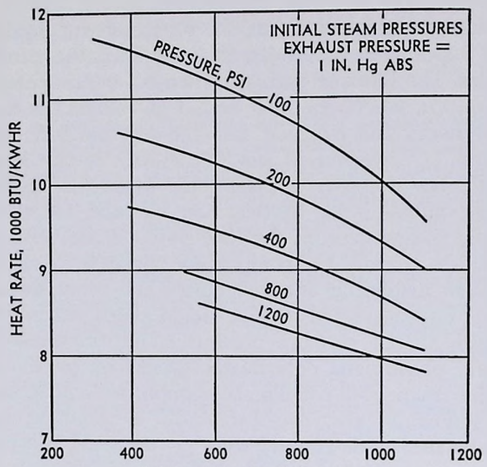
Fig. 9. *Temperature-entropy diagrams of Rankine cycle, using wet, dry saturated, and superheated steam*

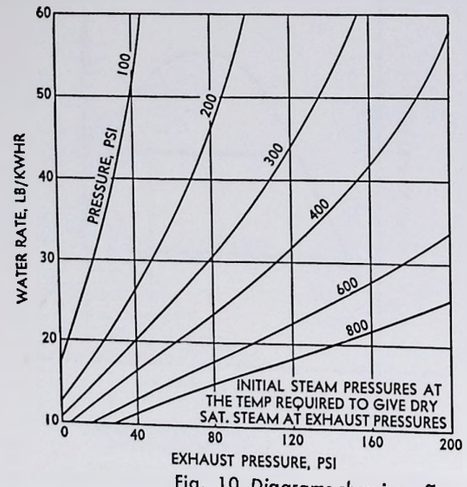
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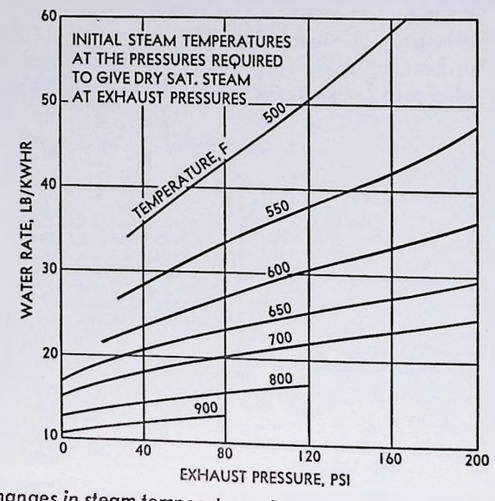


Fig. 10. *Diagrams showing effect of changes in steam temperature, steam pressure, and exhaust pressure on ideal Rankine cycle performance*

altered by shifting point 1 from its position for dry steam to position 1′ for wet steam or 1″ for superheated steam.

The effect of changes in superheat, steam pressure, and exhaust pressure upon the performance of the ideal Rankine cycle is shown graphically by Fig. 10.

The Reheat (Resuperheat) Cycle

The reheat cycle is a modification of the simple Rankine cycle, which is limited in steam temperature by practical considerations to about 1000 F. As shown in Fig. 11, expansion proceeds from 1 to an intermediate pressure 2; the steam is then reheated to an elevated temperature 3, approximately equal to the value at 1, and expansion proceeds isentropically from 3 to the exhaust pressure 4. The work of the complete cycle is found by use of the Mollier chart and is the sum of two Rankine cycle values, equation (26), or:

$W_k/778$, prime mover =
$(h_1 - h_2) + (h_3 - h_4)$, Btu/lb (38)

$W_k/778$, net work of reheat cycle =
$(h_1 - h_2) + (h_3 - h_4) - W_{FP}/778$, Btu/lb (39)

The heat added (Q added) is the sum of the two components, primary and reheat, or:

$$Q = (h_1 - h_b) + (h_3 - h_2), \text{ Btu/lb} \quad (40)$$

$$Q = (h_1 - h_f) + (h_3 - h_2) - W_{FP}/778, \text{ Btu/lb} \quad (41)$$

The thermal efficiency and heat rate are then, respectively:

Thermal efficiency, reheat cycle =

$$\frac{(h_1 - h_2) + (h_3 - h_4) - W_{FP}/778}{(h_1 - h_f) + (h_3 - h_2) - W_{FP}/778} \quad (42)$$

Heat rate, reheat cycle =
3413/Thermal efficiency, Btu/kwhr (43)

Some of the implications of the influence of re-

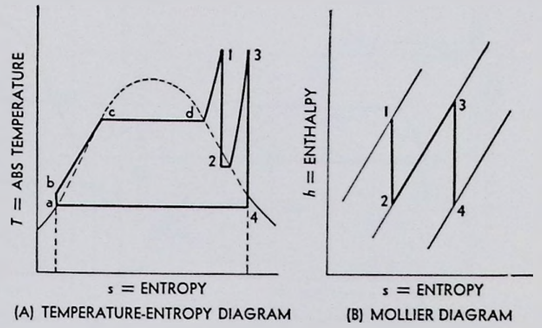


Fig. 11. *Illustrating effect of resuperheat on steam cycle*

superheat pressure on the thermal performance of reheat cycles are shown in Fig. 12. In fact, resuperheating gains are so vitally affected by reheat pressure that it is entirely possible to show a loss in heat rate compared with the simple Rankine cycle, as indicated in the right-hand diagram of Fig. 12. Further thermodynamic gains accrue from the use of multiple reheat, as with the modern supercritical-pressure power plants where two stages are used.

The Regenerative Cycle

No practical steam plant operates without a feedwater heater. For every 10 F rise in feed temperature, there is an approximate reduction of 1% in the heat that must be added to make steam. However, this heating must be accomplished by utilizing some otherwise waste-heat source, or, in the language of thermodynamics, it must be done regeneratively. Thus a feedwater heater serves regeneratively to raise the temperature of the feed above the saturated-liquid condition for the back pressure, as contained in the Rankine cycle specification (see Fig. 6). Much

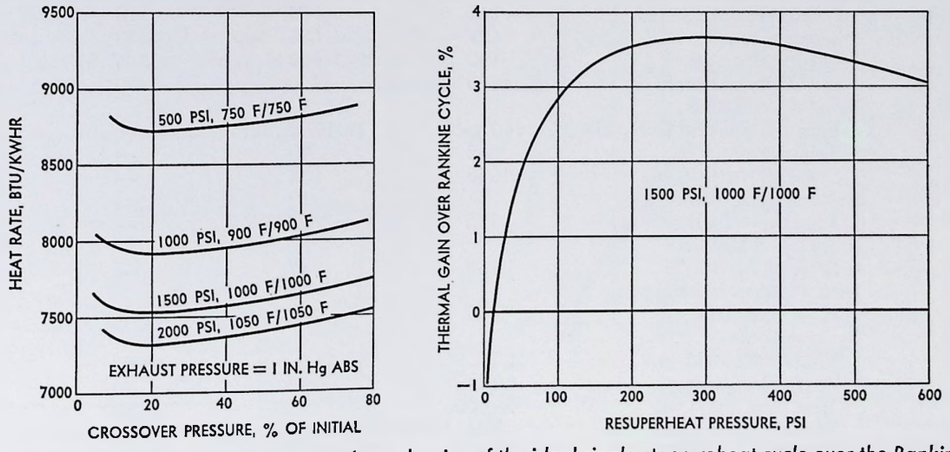


Fig. 12. *Influence of resuperheat pressure on thermal gains of the ideal single-stage reheat cycle over the Rankine cycle*

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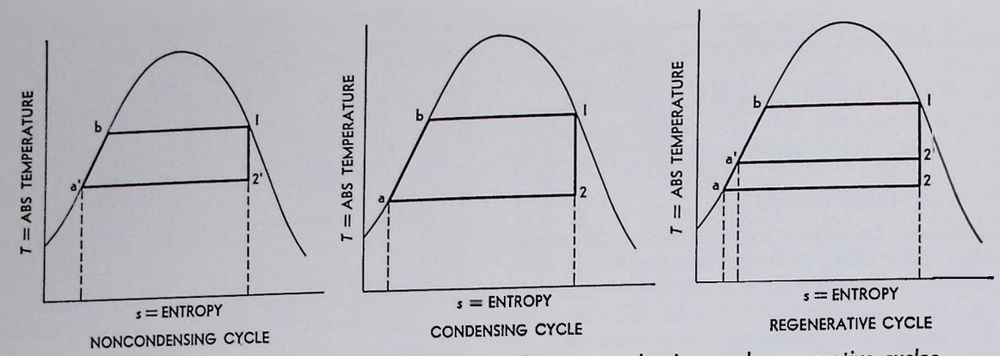


Fig. 13. *Temperature-entropy diagrams for noncondensing, condensing, and regenerative cycles*

could be written about the theoretical calculations involving regenerative feed heating. Real cycles, however, are so divergent from theory that only a brief treatment of the theoretical aspects is given here, followed by elaboration of the practical aspects of actual cycles for power plants.

Consider two steam cycles, each supplied with steam at the same pressure, say 115 psia dry and saturated, $h_1 = 1190$ Btu/lb, one exhausting to the atmosphere at 14.7 psia and the other exhausting to a condenser in which a vacuum of 1 in. mercury absolute is maintained. The evaluation of the performance of these two cycles (Fig. 13), as given in Table 1, can be made by applying the basic equations (27), substituting $(h_1 - h_2) \times 778$ for W_k, (35), and (36).

Regenerative principles can be demonstrated by combining these cycles, as shown in Figs. 13 and 14 (performance given in Table 1). If the exhaust from the noncondensing cycle is used regeneratively to heat the feedwater coming from the hot well of the condenser, then in Fig. 13 it is necessary that the area under a-a′, the heat per lb required to raise the temperature of the condensate multiplied by the

TABLE 1

Performance of Three Steam Cycles Compared

See Figs. 13 and 14

Cycle	1 Non-condensing	2 Condensing	3 Combining 1 & 2
Initial Conditions			
Pressure, psia	115	115	115
Temperature, F	338	338	338
Quality, % dryness	100	100	100
Enthalpy, h_1, Btu/lb	1190	1190	1190
Exhaust Conditions			
Pressure, psia and in. Hg abs	14.7 psia	1.0 in. Hg abs	—
Temperature, F	212	79	—
Enthalpy, h_2, Btu/lb	1040	855	—
Rankine Cycle Work			
$W_k = h_1 - h_2$, Btu/lb	150	335	—
Enthalpy of Liquid at Exhaust, h_f, Btu/lb	180	47	—
Heat added to make Steam $h_1 - h_f$, Btu/lb	1010	1143	—
Rankine Cycle			
Water rate, lb/kwhr	22.8	10.2	11.0
Thermal efficiency	0.148	0.293	0.307
Heat rate, Btu/kwhr	23,000	11,630	11,113
Power taken out, Btu/kwhr	3,413	3,413	3,413
Heat rejected, Btu/kwhr	19,587	8,217	7,700

weight of the condensate, must equal the area under 2′-a′, the heat available per lb in the noncondensing cycle exhaust multiplied by the weight of the noncondensing exhaust used. This equality can be established only by adjusting the weights of steam supplied to the condensing and noncondensing cycles, respectively. To heat 1 lb of condensate, only a fraction of a lb of steam is required. For the example given in Table 1, the balance is effected with $(212 - 79) \times 10.2/860 = 1.577$ lb of 14.7 psia exhaust steam to heat 10.2 lb of condensate. The aggregate generation is then, for steam flow rates of 10.2 and 1.577 lb/hr, as follows:

Condensing cycle,	$10.2/10.2 = 1.000$ kw
Noncondensing cycle,	$1.577/22.8 = 0.069$ kw
Total	1.069 kw

The combined performance of the two cycles, or of the regenerative cycle of Fig. 14, is then:

Total throttle flow $= 10.2 + 1.577 = 11.777$ lb/hr
Water rate (WR), throttle $= 11.777/1.069$
$= 11.00$ lb/kwhr
Water rate (WR), condenser $= 10.2/1.069$
$= 9.55$ lb/kwhr
Heat rate (HR), throttle $= 11.00\ (1190 - 180)$
$= 11{,}113$ Btu/kwhr
Heat rate (HR), condenser $= 9.55\ (855 - 47)$
$= 7{,}700$ Btu/kwhr
Power taken out $= 3{,}413$ Btu/kwhr

Comparing these values with the corresponding values in Table 1 for the condensing cycle, it is evident that the throttle WR is greater (11.0 vs 10.2) and the condenser WR is less (9.55 vs 10.2) for the regenerative cycle. Despite this water rate situation, the throttle heat rate, 11,113 Btu/kwhr (and therefore the thermal efficiency) is best of all for the regenerative cycle. This superior thermal performance results from the reduction, through regeneration, of the heat rejected to the condenser to the minimum value of 7700 Btu/kwhr.

This example, confined to one feed temperature and one stage of heating, serves to demonstrate the principle of regenerative feedwater heating. Different

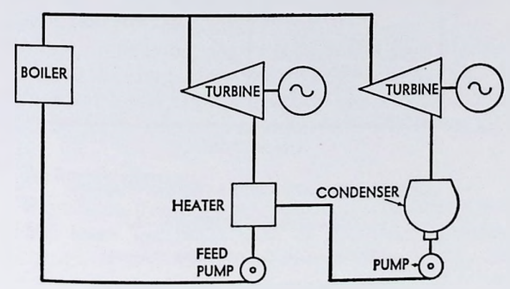
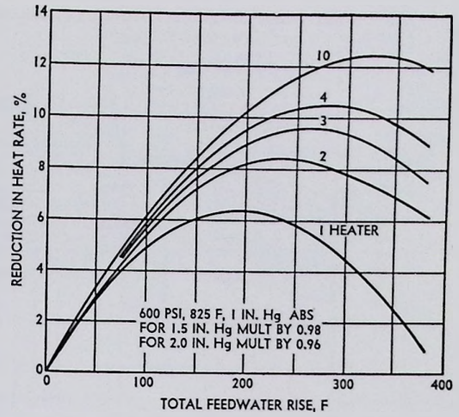
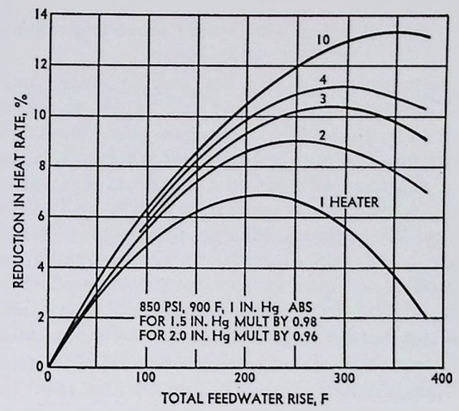


Fig. 14. *Single-stage regenerative cycle—a combination of the noncondensing and condensing cycles*





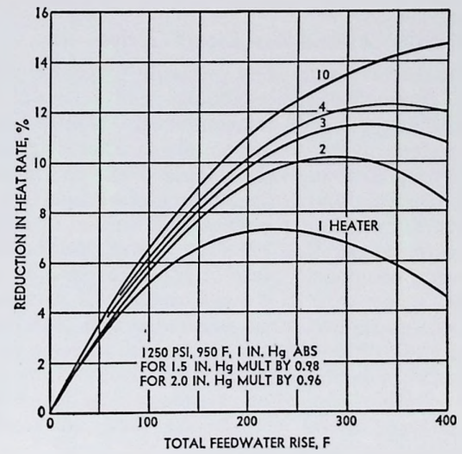


Fig. 15. *Gain in thermal efficiency of ideal regenerative cycle over Rankine cycle, showing the influence of feed temperature and number of stages of feedwater heating*

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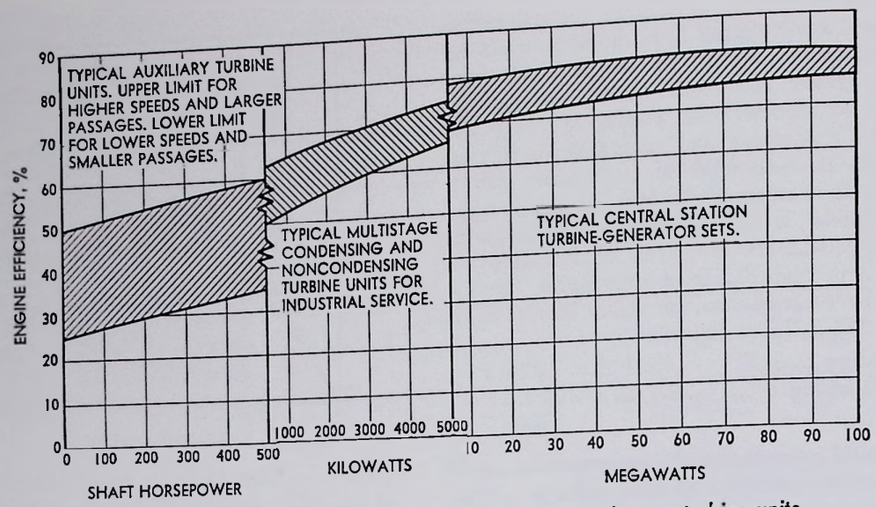


Fig. 16. *Representative engine efficiencies for some selected steam turbine units*

feed temperatures (from hot well to boiler saturation) and various numbers of stages of heating (from zero to infinity) will alter the thermal improvement. The effect of feed temperature and number of stages of heating on the gain in thermal efficiency, with ideal regenerative cycles, is indicated in Fig. 15. The multistage regenerative heating cycles shown are all based on the thermodynamic desirability of equal temperature rises in all stages. There is also the basic implication that maximum work has been obtained from a lb of steam before that steam is used for feed heating. In other words, no thermodynamic gain is possible by using live steam to heat feedwater, because steam so used does no work.

Heat Balances and Real Steam Cycles

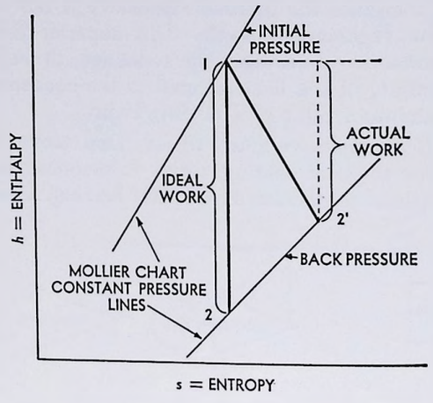
The discussion above is primarily concerned with ideal thermodynamic conditions and is useful in establishing the solid foundation on which to build the performance of real steam plants. Flow diagrams, cycle diagrams, and heat-balance diagrams are useful in showing schematically the performance of real plants. They include numerical data on size, quantity, pressure, temperature, enthalpy, power, and the like, for each component piece of apparatus used in the plant. Being real, they must contain all necessary allowances for the actual performance of each component. The heat-balance data can be no more exact than the understanding, interpretation, and correctness of the equipment-performance data which are used. Some useful performance data, not treated elsewhere, are given below.

Steam Turbines

Engine efficiency. The actual work obtained per lb of steam in real turbines is less than the theoretical Rankine cycle value from equations (5) and (26). Engine efficiency, a measure of this reduction in value, is defined as:

$$\text{Engine efficiency} = \frac{\text{Actual work}}{\text{Rankine cycle work}} \qquad (44)$$

Representative values of efficiency for different classes and sizes of turbines are given in Fig. 16. Marine turbines, because of larger clearances, are usually less efficient than shore turbines by 2-5%. Engine inefficiency is manifest in raising point 2 on the Mollier diagrams, Fig. 6 (B) and Fig. 17, to new and larger values of enthalpy and entropy at point 2′. The



TURBINE LOSSES APPEARING AS REHEAT $= h_{2'} - h_2$

ACTUAL TURBINE WORK $= h_1 - h_{2'}$

IDEAL RANKINE CYCLE WORK $= h_1 - h_2$

ENGINE EFFICIENCY $= (h_1 - h_{2'})/(h_1 - h_2)$

Fig. 17. *Condition curve, or state line, of steam turbine*

losses which occur in a turbine appear as reheat in the exhaust. The result when plotted on the Mollier diagram (Fig. 17) is a curved or inclined line, as shown, called the condition curve, or state line.

The values of engine efficiency at part load can be estimated from the percentage Willans lines of Fig. 18 or other more exact data. Throttle losses are

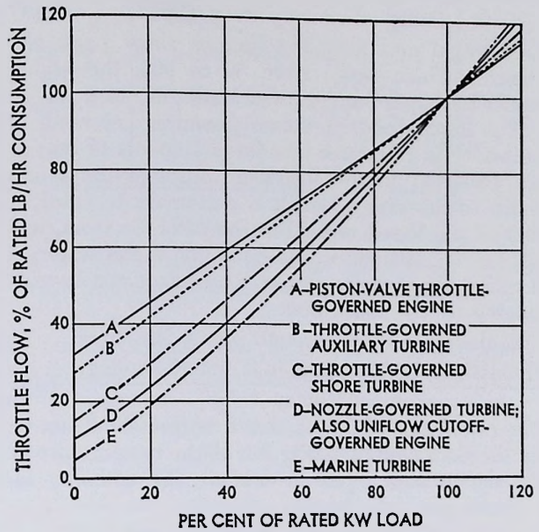


Fig. 18. *Willans lines for some selected steam engines and turbines*

less than 2% on the pressure-drop basis, and gland leakages on well-maintained turbines seldom exceed 0.5-1% of full-load throttle flow for each positive-pressure external gland. The pressure at any stage in a turbine is a function of the flow beyond that stage, so the absolute pressure is almost directly proportional to the weight flow. Thus, if the pressure is known for any stage at a particular flow, the pressure for any other flow can be estimated by direct proportionality. The enthalpy, in turn, can be read from the appropriate state line, or condition curve (Fig. 17). The exhaust from a turbine contains residual, unabsorbed, kinetic energy. There is thus an exhaust, or leaving, loss which varies with the flow and the blading size, shown typically in Fig. 19. Generator losses, mechanical and electrical, are about 1-2% for large units and seldom exceed 5% in any case.

Steam Condensers

Representative condenser performance can be computed from the basic data of the Heat Exchange Institute, New York. Water velocities seldom exceed 7 fps in practice, and a cleanliness factor of 0.7-0.9 is usually applied to the data. Circulating-water requirements are limited generally to a temperature

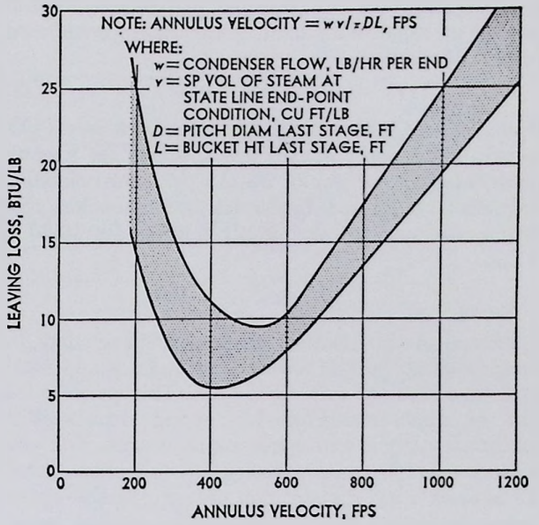


Fig. 19. *Typical steam turbine leaving-loss range*

rise at full load of 5-10 F for single-pass and 10-20 F for two-pass condensers. Steam-jet air ejectors require 300-600 lb of steam per hr per element, and the heat is usually recovered in the condensate.

Feedwater Heaters

Heat transfer rates are given by the data of Fig. 20. Selection of the suitable size heater usually resolves itself into an allowable terminal temperature difference (TTD) of less than 8 F at full load and

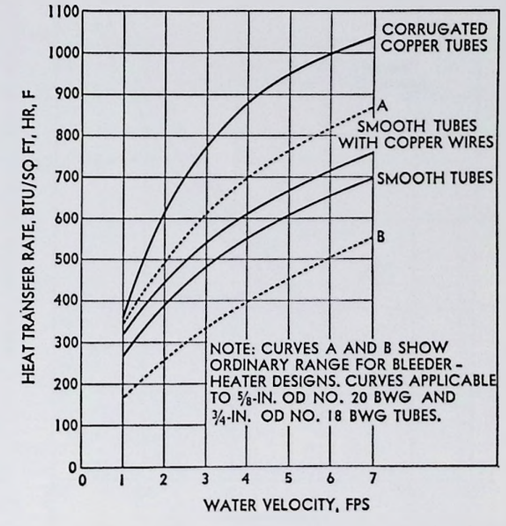


Fig. 20. *Some selected heat transfer rates for feedwater heaters*

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pressure drops of less than 25 psi. If the TTD is desired at some other load, it can be approximated by the relation:

$$\text{TTD} \propto (\text{Water flow rate})^2 \qquad (45)$$

Open, contact, and deaerating heaters give zero TTD. Deaerating heaters usually are specified for a minimum temperature rise of 25-30 F to assure effective degasification of the feedwater. Drain coolers and heat exchangers are frequently selected for 10-20 F TTD.

Pumps

Except in the smallest sizes, when they may be reciprocating, pumps are of centrifugal or, in some rare instances, of axial-flow type. This permits the use of direct-connected high-speed drivers with minimum weight and space requirements. The customary drive is the constant-speed induction motor. Economics may dictate the use of variable-speed motor, hydraulic coupling, magnetic coupling, steam turbine, or internal combustion engine. Steam auxiliary drive, while giving good reliability, always poses the problem of absorbing the heat of the exhaust regeneratively at all loads.

The performance of pumps is based on handling cold water (85 F max), which is considered to be truly noncompressible (see Fig. 8). Thus, the theoretic water horsepower (whp) is given by the following equation:

$$\text{whp} = \text{gpm} \times \text{head in ft} \times \text{sp gr}/3960 \qquad (46)$$

$$\text{or} \quad \text{whp} = \text{lb per min} \times \text{head in ft}/33{,}000 \qquad (47)$$

The efficiency of a pump is defined as the ratio of the theoretic water horsepower, whp, to the shaft horsepower input, shp, or:

$$\text{Pump efficiency} = \text{whp}/\text{shp} \qquad (48)$$

Centrifugal and axial-flow pumps show peak efficiencies which range from 50 to 90%, the higher values for larger pumps, lower heads, or fewer stages.

The impact of high steam pressures has made it necessary to recognize that large amounts of energy are imparted to the feedwater through the pump shaft. In this connection it is customary to consider that all the losses, as well as the theoretic work, will appear as a change in the enthalpy of the water as it passes through the pump. This effect can be calculated by the expression:

$$\text{Enthalpy increase through pump, Btu/lb} = \frac{\text{Head developed by pump in ft of fluid flowing}}{\text{Pump efficiency} \times 778} \qquad (49)$$

The calculation of the head in ft of fluid is facilitated by the data plotted in Fig. 21. If the temperature of the compressed liquid is desired, the enthalpy increase from equation (49) can be added to the saturated-liquid value from the steam tables, and the resulting compressed-liquid enthalpy can be used to get the liquid temperature.

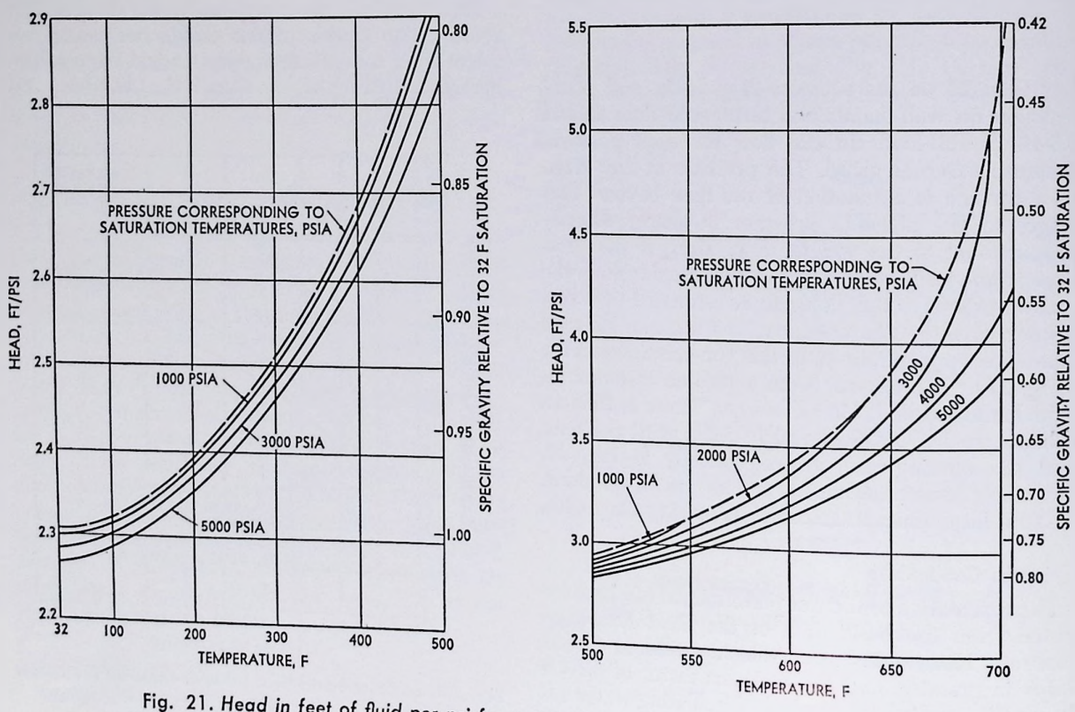


Fig. 21. *Head in feet of fluid per psi for water at various temperatures and pressures*

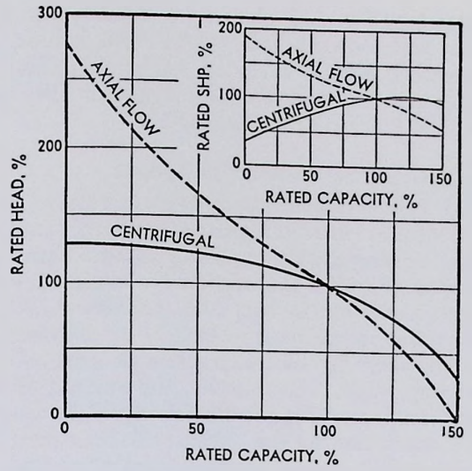


Fig. 22. *Characteristic curves of selected centrifugal and axial-flow pumps—percentage basis*

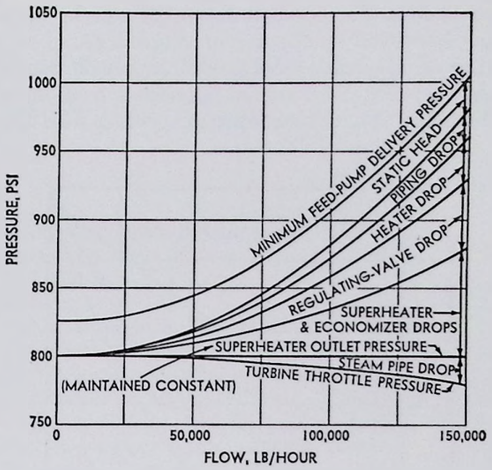


Fig. 23. *Typical flow-resistance curves for a boiler-feed-pump system*

Centrifugal and axial-flow pump performance is best shown through the medium of characteristic curves. These are generally plotted with capacity (gpm) horizontally and head, shaft horsepower, and efficiency vertically, all at constant speed. These characteristics are exact in their definition of the performance of fluid-acceleration machinery. If performance characteristics at other speeds are sought, they can be evaluated by applying the following basic laws at a given point on the efficiency curve, or:

$$\text{Capacity, gpm, varies as the rpm} \qquad (50)$$

$$\text{Head, ft, varies as the (rpm)}^2 \qquad (51)$$

$$\text{Power, shp, varies as the (rpm)}^3 \qquad (52)$$

The curves can be plotted on a percentage basis, as in Fig. 22, where the rating (100%) is defined as the performance at the point of maximum efficiency. These percentage curves apply substantially to all pumps of an homologous series at all usual speeds.

In the specification of pump head and capacity requirements, a system-resistance curve should be plotted, as in Fig. 23. This should include allowances for all such items as elevational head between pump and delivery point; pressure drop through piping, heaters, economizers, and superheaters; and margins for the operation of flow regulators and safety valves. Pump head characteristics must be matched against the system-resistance curve for proper selection.

Fans

Fans, fluid dynamically, are like pumps in their behavior. Representative characteristic curves, on a percentage basis, are shown in Fig. 24 for different

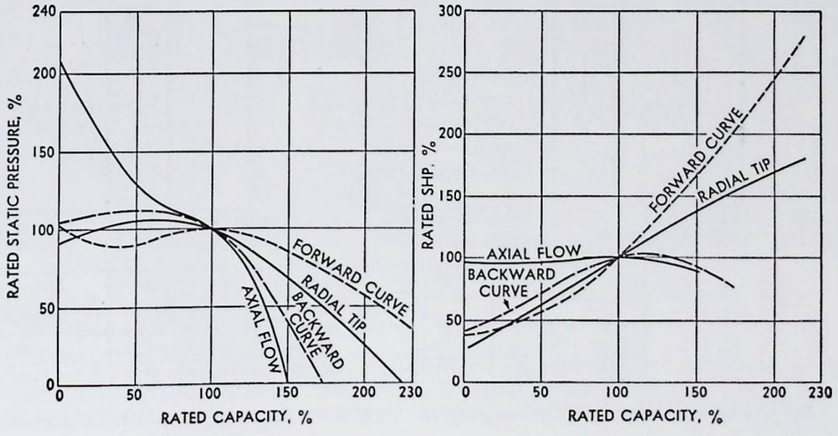


Fig. 24. *Selected centrifugal and axial-flow fan-characteristic curves—percentage basis*

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types of fans. Part-load performance can be appreciably improved by the use of variable speed or by the use of adjustable inlet guide vanes with constant speed (see Fig. 25). In the specification of fan requirements, the system-resistance curves like those

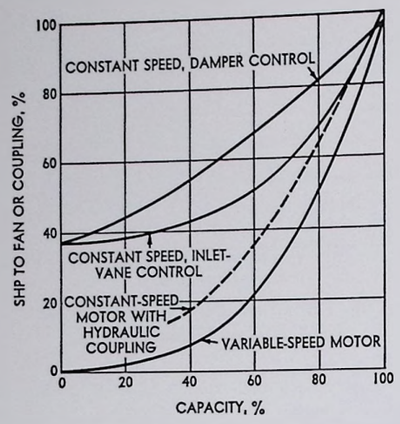
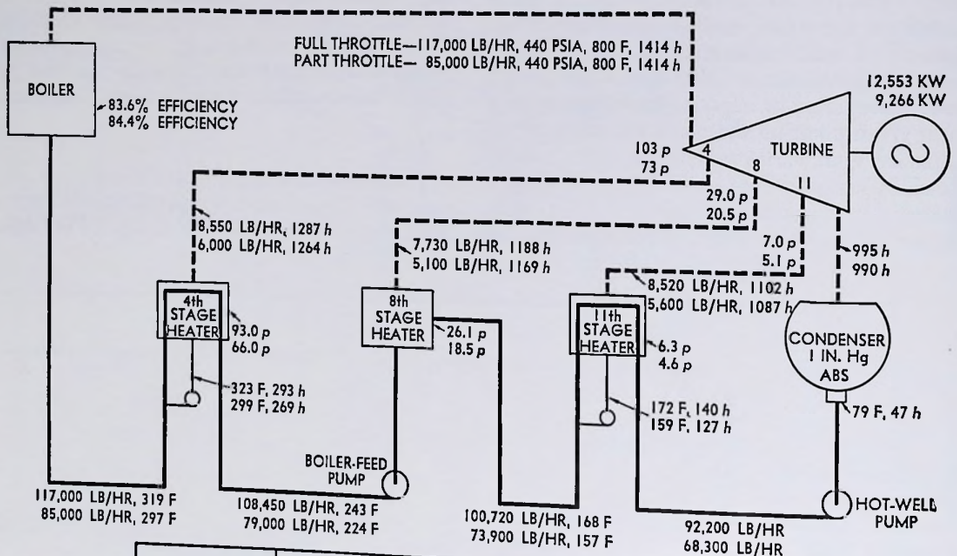


Fig. 25. *Power comparison for a selected centrifugal fan at constant speed, variable speed, and vane control*

for pumps (Fig. 23) should be constructed. For forced- and induced-draft service with boilers, allowances should be included for unusual fuel or firing conditions, worn blades, plugged or dirty passages, and leakage (see Chapter 5).

Circulating-Water Reclamation Systems

It has been found necessary, in an increasing number of steam plants, to reclaim circulating water by the use of spray ponds or cooling towers. These installations all utilize the evaporative effect for cooling purposes. Thus the heat contained in the exhaust of the prime mover must ultimately be delivered to the atmosphere by the evaporation of some of the circulating water. Theoretically, the amount of the evaporation loss will approximately equal the weight of the steam flow to the exhaust. To this must be added the drift, which can vary from 10 to 100% of the theoretic loss, depending upon equipment type and wind conditions. Cooling towers can generally be selected to give a 10-20 F range in the cooling of the water and to approach to within 10-15 F of the wet-bulb temperature. The summer wet-bulb temperature will be crucial in the selection, size, and performance of evaporative reclaiming equipment.



CASE	GROSS GENERATION, KW	NET PLANT SEND-OUT, KW	HEAT SUPPLIED IN FUEL, BTU/HR	OVER-ALL PLANT HEAT RATE, BTU/KWHR
FULL THROTTLE	12,553	12,050	165.7 X 10^6	13,750
PART THROTTLE	9,266	8,849	122.9 X 10^6	13,900

Fig. 26. *Heat-balance diagram for 10,000-kw steam power plant. Calculations shown in Table 2*

TABLE 2

Heat-Balance Calculations for Cycle shown in Fig. 26

10,000-kw Condensing Unit

	Full Throttle	Part Throttle
Turbine		
Steam flow, lb/hr	117,000	85,000
Pressure, psia	440	440
Total temperature, F	800	800
Condenser		
Steam flow, lb/hr	92,200	68,300
Pressure, in. Hg abs	1	1
Hot-well temperature, F	79	79
Hot-well enthalpy, Btu/lb	47	47
11th-Stage Extraction Heater		
°Water flow to heater, lb/hr	92,200	68,300
Turbine stage press., psia	7.0	5.1
Press. drop to heater, psi	0.7	0.5
Heater shell press., psia	6.3	4.6
Heater sat. temp, F	172	159
Terminal temp diff, F	4	2
Water temp out, F	168	157
Water temp in, F	79	79
Water temp rise, F	89	78
Heat required, M Btu/hr	8,206	5,320
Extraction enthalpy, Btu/lb	1,102	1,087
Drip enthalpy, Btu/lb	140	127
Heat avail. in extraction, Btu/lb	962	960
Extraction required, lb/hr	8,520	5,600
8th-Stage Extraction Heater		
Water flow to heater, lb/hr	100,720	73,900
Turbine stage press., psia	29.0	20.5
Press. drop to heater, psi	2.9	2.0
Heater shell press., psia	26.1	18.5
Heater sat. temp, F	243	224
Terminal temp diff, F	0	0
Water temp out, F	243	224
Water enthalpy out, Btu/lb	211	192
Water enthalpy in, Btu/lb	136	125
Enthalpy rise of water, Btu/lb	75	67
Heat required, M Btu/hr	7,555	4,960
Extraction enthalpy, Btu/lb	1,188	1,169
Drip enthalpy, Btu/lb	211	192
Heat avail. in extraction, Btu/lb	977	977
Extraction required, lb/hr	7,730	5,100
4th-Stage Extraction Heater		
Water flow to heater, lb/hr	108,450	79,000
Turbine stage press., psia	103	73
Press. drop to heater, psi	10	7
Heater shell press., psia	93	66
4th-Stage Extraction Heater (cont'd)		
Heater sat. temp, F	323	299
Terminal temp diff, F	4	2
Water temp out, F	319	297
Water enthalpy out, Btu/lb	289	267
Water enthalpy in, Btu/lb	211	192
Enthalpy rise of water, Btu/lb	78	75
Heat required, M Btu/hr	8,300	5,925
Extraction enthalpy, Btu/lb	1,287	1,264
Drip enthalpy, Btu/lb	293	269
Heat avail. in extraction, Btu/lb	994	995
Extraction required, lb/hr	8,350	6,000
Boiler feed, lb/hr	117,000	85,000
Turbine Generator		
Enthalpy at throttle, Btu/lb	1,414	1,414
Work up to 4th stage, Btu/lb	127	150
Work up to 8th stage, Btu/lb	226	245
Work up to 11th stage, Btu/lb	312	327
Expansion-line end point, Btu/lb	979	981
Leaving loss, Btu/lb	16	9
Exhaust enthalpy, Btu/lb	995	990
Work up to exhaust, Btu/lb	419	424
Internal Generation up to:		
4th stage, kw	318	264
8th stage, kw	513	366
11th stage, kw	780	536
Exhaust, kw	11,340	8,485
Total generation, kw	12,951	9,651
Generator losses, mechanical plus electrical, kw	398	385
Gross load, gen terminals, kw	12,553	9,266
Boiler		
Enthalpy of steam, Btu/lb	1,414	1,414
Enthalpy of feed, Btu/lb	289	267
Heat added to steam, Btu/lb	1,125	1,147
Heat added to steam, 10^6 Btu/hr	131.6	97.5
Boiler efficiency, %	83.6	84.4
Heat in fuel to attain above efficiency, 10^6 Btu/hr	157.4	115.5
Plant Performance		
Auxiliary power use, %	4.0	4.5
Auxiliary power use, kw	503	417
Net plant send-out, kw	12,050	8,849
†Plant realization ratio	0.95	0.94
Plant heat supplied, 10^6 Btu/hr	165.7	122.9
Over-all plant heat rate, Btu/kwhr	13,750	13,900

° The water flow from the condenser to the last-stage extraction feedwater heater can be approximated by a trial estimate of the total extraction required, which can be computed from its simple relation to the total rise in temperature from hot well to boiler feed. Trial values can then be adjusted as necessary to achieve an over-all balance.

† An over-all ratio to cover all losses not included in the heat balance, such as soot blowing, blowdown, make-up, radiation, gland leakage, steam-driven auxiliaries, and banking.

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Make-up Requirements

In condensing plants, the water required for make-up seldom exceeds a maximum value of 1 to 2% of full-load steam output. This make-up must be pure. Chemical treatment, demineralization, compression distillation, or an evaporator will give the degree of purity required. To conserve the heat needed for distillation, the evaporator is included in the heat balance for the plant cycle.

Auxiliary Power

The power used for driving the auxiliaries of a condensing steam plant lies between 4 and 8% of the gross generation at full load. For part load, the auxiliary power may be estimated by using a straight-line relationship in which, at no load, it is approximately half the requirement at full load.

Realization Ratio

Since no heat-balance calculation can include allowance for all losses that occur in a real plant, such as soot blowing, blowdown, make-up, gland leakage, steam-driven auxiliaries, and banking, it is necessary to apply an over-all realization ratio to the computed figures. This ratio may range from 0.90 to 0.98.

Heat-Balance Calculations

To show the applicability of the above data, a typical heat balance is worked out for the following conditions in a 10,000-kw condensing unit:

Steam supplied at 440 psi, 800 F, 1414 Btu/lb

Exhaust at 1 in. Hg abs

Three stages of feedwater heating with equipment arranged as shown in Fig. 26

Maximum throttle flow = 17,000 lb/hr

The calculations are for full-throttle and part-throttle flows and proceed step by step, as noted in Table 2.

A simplified heat-balance diagram for a supercritical double-reheat steam power plant is shown in Fig. 27. Results for these and other selected steam power plants of various sizes are given in Fig. 28.

Correction Factors and Comparative Data

In the interpretation and application of heat-balance data, it is often necessary to estimate the effects of changes in conditions on thermal performance of the plant. Thus, proposed alterations in steam pressure, steam temperature, feed temperature, number of stages of feed heating, or vacuum call for an estimate of the change in performance. Data useful for such estimates are presented in Fig. 29. Care should be exercised, however, in employing these generalized data, since relative values are more reliable than absolute values and this matter cannot be

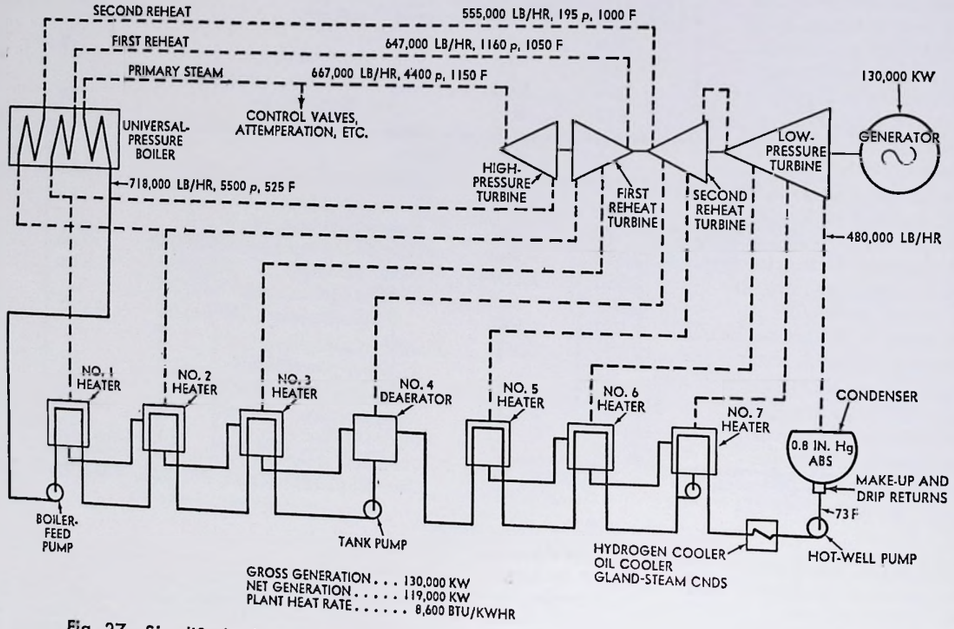


Fig. 27. *Simplified schematic heat-balance diagram for a supercritical double-reheat cycle*

PLANT	A	B	C	D	E	F	G	H	I
CYCLE TYPE	REGENERATIVE	REGENERATIVE	REGENERATIVE	REGENERATIVE REHEAT	REGENERATIVE	REGENERATIVE	REGENERATIVE REHEAT	REGENERATIVE REHEAT	REGENERATIVE
PLANT TYPE	SINGLE UNIT	TOPPING	SINGLE UNIT	CROSS COMPOUND	TOPPING	SINGLE UNIT	SINGLE UNIT	CROSS COMPOUND	DOUBLE REHEAT TANDEM COMP
NET PLANT OUTPUT, KW	11,700	62,000	22,000	53,000	150,000	103,000	120,000	148,000	117,000
THROTTLE PRESS., PSIA	440	1,240	865	1,215	1,250	1,275	1,465	2,015	4,400
THROTTLE TEMP, F	800	900	900	725	935	925	1,000	1,050	1,150
REHEAT TEMP, F	—	—	—	725	—	—	1,000	1,000	1050/1000
CONDENSER PRESS., IN. Hg ABS	1.0	1.0	1.5	1.0	1.0	VARIES	1.2	VARIES	VARIES
CIRCULATING-WATER TEMP, F	VARIES	VARIES	VARIES	VARIES	VARIES	66	VARIES	67	34
BOILER EFFICIENCY, %	83.4	86.6	87.6	86.5	87.5	87.9	90.0	90.1	89.3
MAKE-UP, %	2.0	2.0	2.0	1.25	3.0	2.25	2.0	1.5	0.15

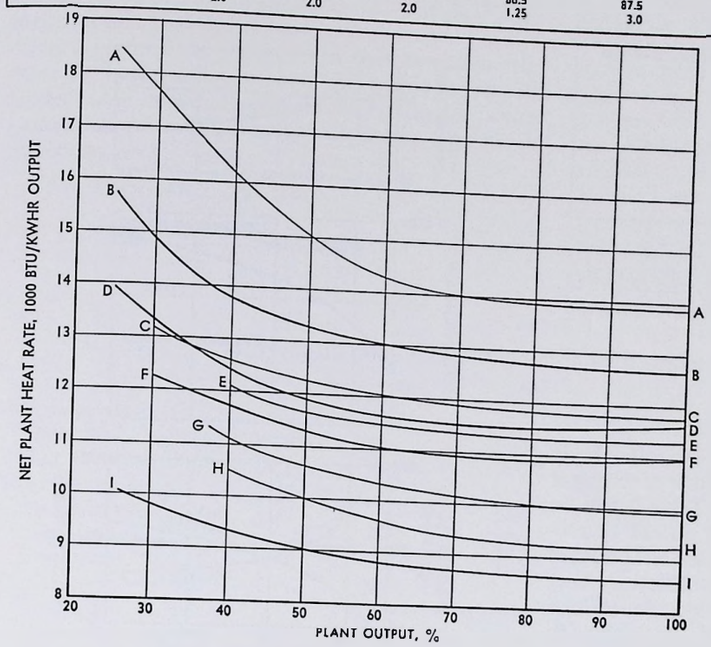


Fig. 28. *Heat-rate curves for selected condensing steam power plants*

TABLE 3

Approximate Correction Factors
For Condensing Steam Plants

Initial Conditions					
Pressure, psi	400	600	850	1250	2000
Temperature, F	800	825	900	950	1050
Reheat temp, F	—	—	—	—	1000
Corrections for:					
Pressure, Btu/psi	3.3	2.3	0.7	0.55	0.3
Initial temp, Btu/F	4.3	3.8	3.0	3.0	1.3
Reheat temp, Btu/F	—	—	—	—	1.2
Vacuum, Btu/in. Hg	320	220	200	200	200
Feed temp, Btu/F	1.5°	1.4°	1.1°	0.8°	1.5†

° Change in boiler efficiency is not included.

† Change in boiler efficiency is included.

Note: Corrections represent the change in net plant heat rate (Btu per net kwhr) per unit change in variable.

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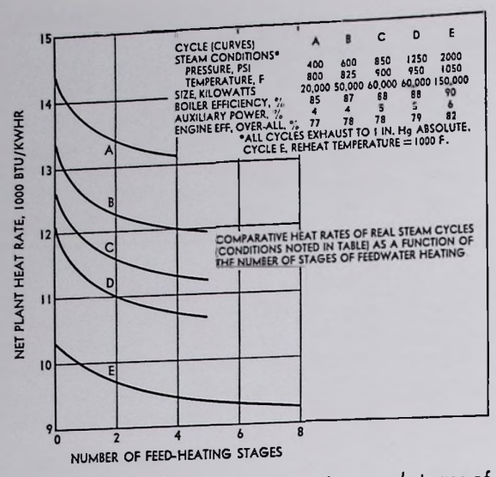
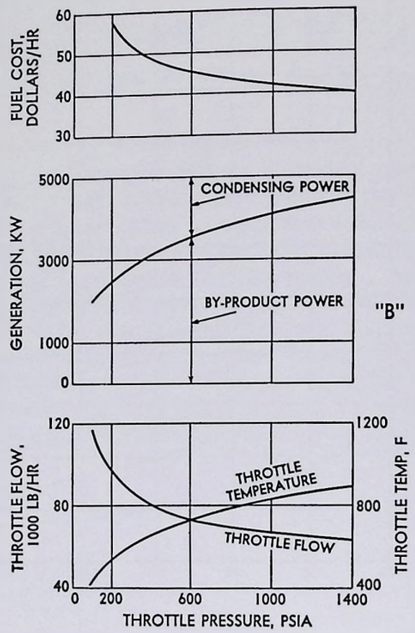


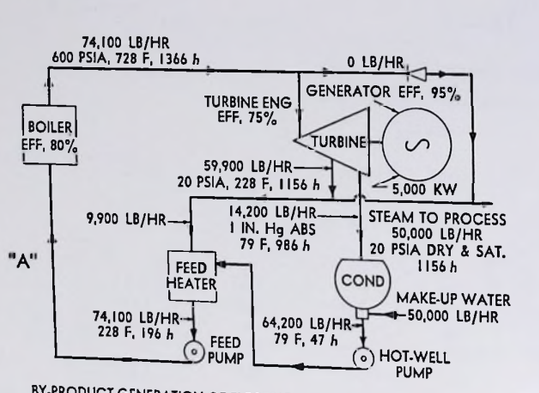
Fig. 29. *Effects of pressure, temperature, and stages of feedwater heating on performance of some selected steam cycles*

reliably simplified. If a rough over-all estimate is desired, particularly to cover the case of changes within an existing plant, the data in Table 3 will be of assistance. In using these data, it should be recognized that some items, such as feed-temperature change, may have a compensating or offsetting effect in the performance of the boiler versus the turbine room. The data should accordingly be used with caution.

Industrial and By-Product Steam Plants

Industry has numerous demands not only for electric power but also for heat at various temperature levels. If the desired temperature lies within the range of 200-500 F, steam is the ideal working fluid. The thermostatic effect of the vapor-tension curve, coupled with the high latent heat and the density of the fluid, makes steam especially attractive for





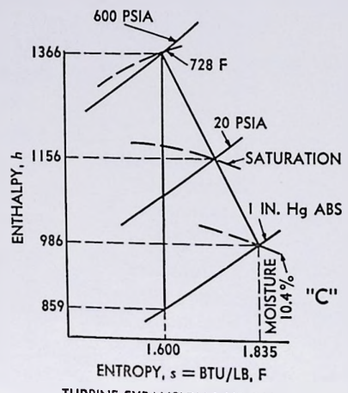


Fig. 30. *Diagrams "A" and "C" show the conditions for the heat-balance calculations, Table 4. The effects of changes in throttle pressure and temperature on by-product power and fuel costs are indicated in diagram "B"*

these heating purposes. Furthermore, if the power and process-heat loads can be coordinated, it is entirely practical to generate steam in a boiler at an elevated pressure, pass it through turbines or engines, and then exhaust or extract the steam at a proper pressure level for use in the process. This is how the multipurpose power plant of industry operates, where the heat for power and the process heat are developed as mutual by-products. If a complete balance can be effected, the heat loss represented in the exhaust of the condensing steam plant is eliminated, since the heat in the exhaust is chargeable to process. Under these circumstances, the heat rates for power generation are reduced practically to the irreducible minimum, or:

By-product-power heat rate =

$$\frac{3413 \times 100}{A \times (1 - B) \times C}, \text{ Btu/kwhr} \qquad (53)$$

where:

A = generator efficiency, %
B = auxiliary power, %
C = boiler efficiency, %

The values of the power heat rate under these conditions are of the order of 4200-5000 Btu/kwhr. This is by-product power, and the very low heat rate of the operation gives the steam cycle a tremendous advantage over any other form of heat-power cycle or system for industrial application. Selection of the by-product cycle hinges about the economic aspects of the solution. The heat balance, Table 4, based on the conditions illustrated in diagrams "A" and "C" of Fig. 30, shows the pertinent features of the basic thermal calculations. In this example, 64% of the power generated is a by-product of the process steam delivered. Consequently, the net plant heat rate chargeable to power is only 8800 Btu/kwhr.

TABLE 4

Heat-Balance Calculations for By-Product Power Cycle

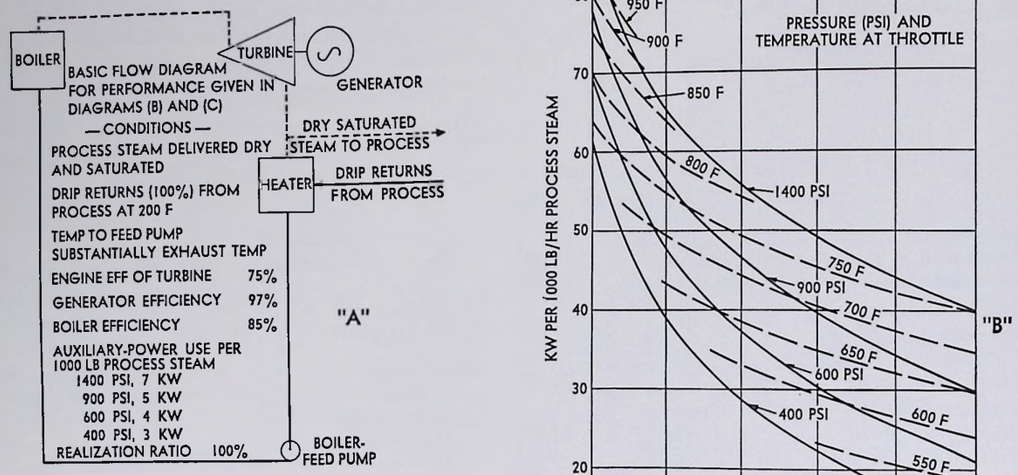
See "A" and "C," Fig. 30

Requirements	
Gross load at generator terminals, kw	5,000
Process steam, lb/hr @ 20 psia, dry & sat.	50,000
Throttle Conditions	
Pressure, psia	600
Temperature, F	728
Steam flow, lb/hr	74,100
Condenser	
Steam flow, lb/hr	14,200
Pressure, in. Hg abs	1
Hot-well temperature, F	79
Hot-well enthalpy, Btu/lb	47
Make-up water flow, lb/hr	50,000
Water flow out, lb/hr	64,200
Feed Heater	
Heater pressure, psia	20
Heater saturation temperature, F	228
Enthalpy, water in, Btu/lb	47
Enthalpy, water out, Btu/lb	196
Heat added to water, Btu/lb	149
Heat added to water, 10^6 Btu/hr	9.5
Bleed enthalpy, Btu/lb	1,156
Heat available from bleed, Btu/lb	960
Extraction required, lb/hr	9,900
Boiler feed, lb/hr	74,100
Turbine Generator	
Enthalpy at throttle, Btu/lb	1,366
Enthalpy at bleed point, Btu/lb	1,156
Enthalpy drop to bleed point, Btu/lb	210
Extraction, lb/hr	59,900
Internal generation by extracted steam, kw	3,690
Enthalpy at exhaust, Btu/lb	986
Enthalpy drop to exhaust, Btu/lb	380
Condenser flow, lb/hr	14,200
Internal generation by condenser flow, kw	1,580
Total internal generation, kw	5,270
Generator efficiency, %	95
Electrical and mechanical losses, kw	270
Gross load, generator terminals, kw	5,000
Throttle water rate, lb/kwhr	14.8
Condenser water rate, lb/kwhr	2.84
Boiler	
Enthalpy at superheater outlet, Btu/lb	1,366
Enthalpy at feed, Btu/lb	196
Heat added to steam, Btu/lb	1,170
Heat added to steam, 10^6 Btu/hr	86.7
Boiler efficiency, %	80
Heat supplied in fuel, 10^6 Btu/hr	108
Plant Performance	
Auxiliary power, %	7
Auxiliary power, kw	350
Net plant electric send-out, kw	4,650
Plant realization ratio	0.93
Net plant heat supplied, 10^6 Btu/hr	114
Fuel price, cents/10^6 Btu	40
Fuel cost/hr	$45.60
Allocation of Heat Charges	
Steam flow, lb/hr	50,000
Enthalpy of steam, Btu/lb	1,156
Enthalpy of make-up, Btu/lb	47
Heat to process, Btu/lb	1,109
Heat to process, 10^6 Btu/hr	55.4
Heat to power (balance), 10^6 Btu/hr	31.3
Fraction chargeable to process	0.64
Fraction chargeable to power	0.36
Fuel cost, process steam, per hr	$29.20
Fuel cost, power, per hr	$16.40
Fuel cost, process steam, cents/M lb	58.4
Fuel cost, power, mills/net kwhr	3.53
Net plant heat rate chargeable to power, Btu/net kwhr	8,800

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By-Product Power Estimates

The example noted and the data of Fig. 30 indicate some of the elements of variation in by-product power. The ratio of power (kw) to process steam (lb/hr) is vital in the determination of a suitable boiler pressure if all power is to be generated as a by-product of the process heat. To facilitate estimates in this connection, the curves of Fig. 31 have been prepared. These curves are useful within the limitations of engine efficiency, generator efficiency, water return temperature, and auxiliary power values that were used in the calculations. Together with such theoretical data as those of Fig. 10, they can be conveniently used in the preparation of approximate performance.



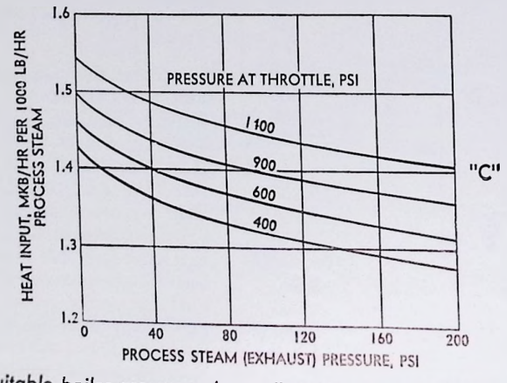


Fig. 31. For estimating suitable boiler pressure where all power generated is a by-product of process heat

11

BOILERS, ECONOMIZERS, AND AIR HEATERS

In a natural-circulation unit, boiler heating surface may be defined as those parts of shells, drums, or tubes which are a part of the boiler circulatory system and which are in contact with the hot gases on one side and water or a mixture of water and steam on the other side. By convention, the economizer is not considered a part of the boiler heating surface. In some types of forced-circulation boilers, the amount of water-wetted heating surface in a given unit is not constant but varies with operating procedure, and an exact designation is not practical. For other types of forced-circulation boilers, the definition given above for natural-circulation boilers will apply. Heat from the hot gases flows through the confining walls of these pressure parts to the water on the opposite side. The heat so transmitted, after raising the temperature of the water in the boiler to the boiling point, produces steam at a greater or lesser rate, depending upon the rate of heat input. Boilers may be broadly classified in three types: (a) shell, (b) fire-tube, and (c) water-tube.

Shell-Type Boiler

The shell boiler may be defined simply as a closed vessel containing water, with a portion of the shell exposed to a source of heat, such as a fire burning under it. However, it is now used chiefly in such modern adaptations as the electric boiler and the accumulator. The water in the former is heated by electrodes and in the latter by steam from an outside source. In neither case is the shell exposed to the source of heat. In the early shell boilers externally fired, the vessel was given a cylindrical form in order to withstand the steam pressure and to provide heating surface along the length of the bottom of the shell. The steam capacity and efficiency of this simple shell-type boiler set over a fire were both very low because of the relatively small area of boiler heating surface. Only the lower part of the shell and one end of the cylinder absorbed heat from the gases, chiefly by radiation, with very little heat transfer by convection.

Fire-Tube Boiler

The heating surface of the shell-type boiler was increased by passing the hot gases from the fire through the inside of tubes located within the shell. This was the beginning of the type known as the fire-tube boiler. The inner surfaces of the tubes absorbed heat from the gases by convection and radiation. Because heat is absorbed by the tubes as well as by the shell, the efficiency of the fire-tube is much higher than that of the simple shell-type boiler. This made it possible to reduce fuel consumption and to increase capacities for the same over-all dimensions of the boiler. Under favorable conditions of fuel and at a rating not exceeding 100%, the efficiency is of the order of 70%. Since the fire-tube boiler is not well suited for superheat, the better economy with superheated steam is not available.

Many arrangements of the fire-tube boiler have been developed. Tubes have been placed in horizontal, vertical, or inclined positions, with one or more gas passes. The simplest and usual arrangement is the horizontal-return-tubular boiler shown in Fig. 1. In this boiler the furnace and grates are located directly below the front end of the shell. The gases pass over the bridge wall and horizontally along the bottom of the shell to the rear of the boiler and then return through horizontal tubes to the front of the boiler and stack—hence the name horizontal-return-tubular (HRT) boiler.

In a fire-tube boiler known as the firebox class, the firebox or furnace as well as the fire tubes is located within the shell. A well-known example is the Scotch marine boiler (see Fig. 2). Combustion takes place within one or more cylindrical water-cooled chambers, with the gases leaving the chambers toward the rear and returning through the fire

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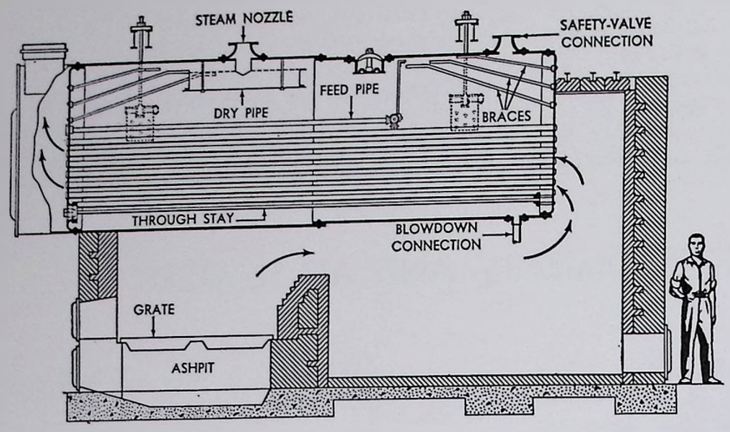


Fig. 1. *Early horizontal-return-tubular boiler set for hand firing*

tubes to the stack at the front of the boiler. Compared with the horizontal-return-tubular boiler, the firebox type is self-contained and compact. The Scotch marine boiler, however, is limited in steam output since the diameter seldom exceeds 14 ft. A greater diameter tends to require an excessive plate thickness. Within this limiting diameter of shell, the maximum grate surface that can be installed (with a four-furnace arrangement) is of the order of 100 sq ft. The steam output is accordingly limited to approximately 25,000 lb of steam per hr for the larger sizes.

Many types of fire-tube boilers are in use, but, as a class, the fire-tube boiler is confined to relatively small capacities and low steam pressures. It is, however, extensively used to supply steam for space heating and is still used to some extent in the railroad locomotive in steam outputs to 4000 bhp or more.

A fire-tube type of boiler known as the gas-tube boiler, developed by B&W for utilizing the heat in waste gases, has been in use for many years. This boiler is described and illustrated in Chapter 19.

Water-Tube Boiler

As the requirements for steam capacity and pressure increased, the shell diameter of the fire-tube boiler became very large, and it was necessary to use thick plates to withstand the high stresses. Furthermore, these early fire-tube boilers were subjected to stresses of high and indeterminate magnitude because of the differences in temperature in the various parts. The combined effect of the known pressure stresses and the indeterminate thermal stresses, aggravated by scale deposits from hard water, caused many boiler explosions. The water-tube boiler then being developed had the advantage, in this respect, of lower stresses from pressure and temperature differentials, since its components were relatively much smaller in diameter and were

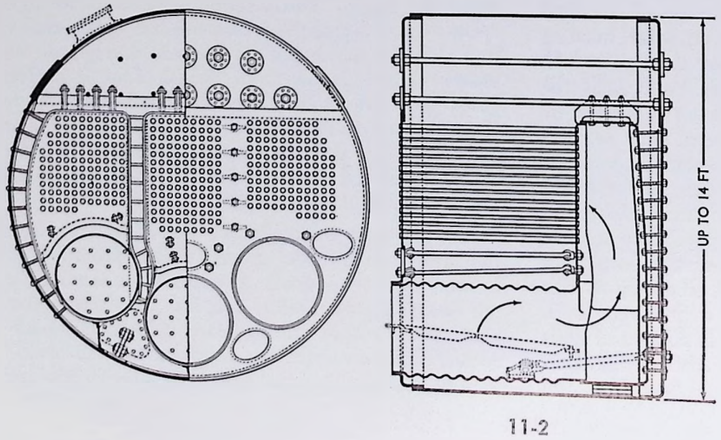


Fig. 2. *Scotch marine boiler, fitted for coal firing*

arranged to accommodate the expansions caused by differences in temperature. It was apparent, because of the lower stresses inherent in its design, that the water-tube boiler was much more suitable for larger capacities and higher pressures while still maintaining a high margin of safety.

In principle, the water-tube boiler is the reverse of the fire-tube boiler. The hot gases are in contact with the outside surface of the tubes, and the boiler water and steam are in contact with the inside surface of the tubes. The term "water-tube" is, therefore, universally recognized as descriptive of this type of boiler.

Straight-Tube Boiler

All B&W straight-tube boilers (see Figs. 3 and 4) are made up of sections of staggered tubes, inclined at an angle of 15° and expanded at the ends into vertical headers. The area of the heating surface is varied by changing the length of the tubes, the number of tubes high, and the number of sections wide. Each section is made up of a downtake header supplying water to the tubes and an uptake header discharging water and steam from the tubes. Sections may be single deck, with a single downtake and a single uptake header, or double or triple deck using more than one downtake and more than one uptake header connected by expanded nipples. The lower ends of the downtake headers are connected by nipples into a small (7¾-in. sq or 10¾-in. round) header, known as a mud drum. The mud drum, as its name implies, serves as a collecting chamber for the sediment, solids, or mud in the boiler water. This

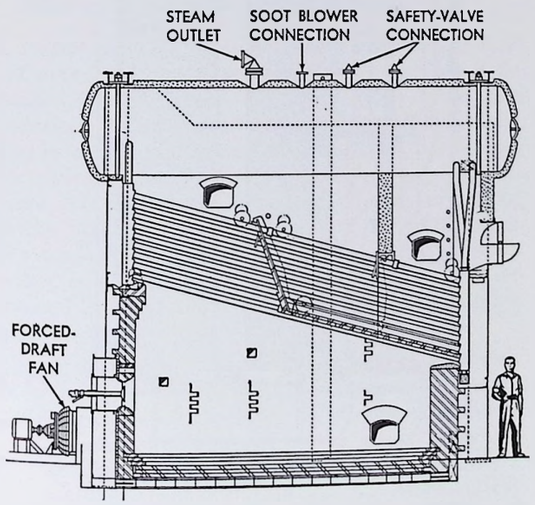


Fig. 3. *Long-drum boiler (B&W) installed in Brazilian rubber-processing plant. Design pressure 250 psi; steam output 18,000 lb/hr*

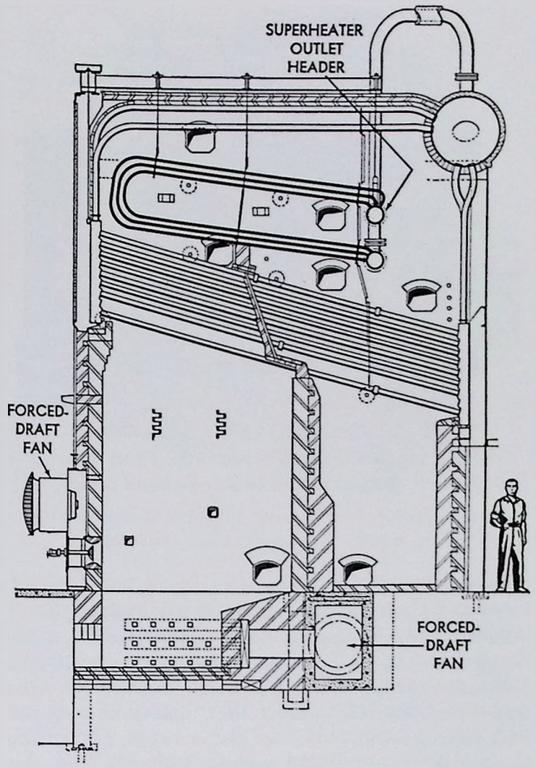


Fig. 4. *Cross-drum boiler (B&W) fired with fuel oil and cottonseed hulls. Design pressure 400 psi; steam temperature 790 F; maximum steam output 16,500 lb/hr*

header is provided with a blowoff connection which also serves as a boiler drain.

The generating tubes are usually 4-in. or 3½-in. OD spaced on 7-in., 7¾-in., 8¼-in., or 8¾-in. centers horizontally, depending upon the design pressure. As the design pressure increases, the header thickness increases, and consequently the horizontal tube spacing increases. Tube rows are spaced vertically on 6-in. centers (or at 12 in. for some lower rows where a slag screen is desired).

In the headers, opposite each tube end, there is a handhole of sufficient size to permit removal or renewal of the tube. These handholes can also be used for turbining and cleaning the inside of the tubes.

Vertical or slightly inclined gas baffles are arranged for two or three gas passes across the tubes. The arrangement of gas baffling depends upon many factors, including acceptable draft loss, size and arrangement of boiler heating surface, and position of gas outlet. Sometimes a straight-tube boiler is designed without baffles, and the gases make a single pass across the tubes.

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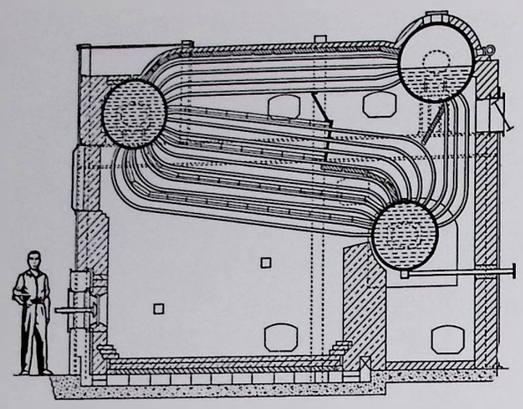


Fig. 5. *Type H Stirling boiler (B&W). Particularly suitable for low headroom. Fired with fuel oil, gas, and coal (stoker or hand firing)*

In the long-drum boiler, usually of one bank of tubes and three gas passes, the steam drum is placed longitudinally in relation to the boiler tubes, as indicated in Fig. 3. The steam and water circulating tubes are connected to the steam drum in circumferential rows. This limits the number of sections that can be connected to a single drum. Obviously, the maximum number of sections wide per drum depends upon the drum diameter. A 48-in.-diameter steam drum can accommodate as many as 14 sections in width. For wider long-drum boilers more than one steam drum must be used. Because of the limitations in the amount of heating surface with one steam drum, the long-drum boiler is economically suitable only for small units and low pressures which require relatively large steam-release areas. Heating surface ranges from about 1000 to 10,000 sq ft per boiler. Capacities range from 5000 to 80,000 lb of steam per hr per boiler. Design pressures range from 160 to 325 psi. Comparatively few long-drum boilers are being built today.

As indicated in Fig. 4, the steam drum of a cross-drum boiler is placed across or at right angles to the tube bank, and there may be one or more banks of boiler tubes with one or more gas passes across the tubes. The steam and water circulating tubes are connected to the steam drum in longitudinal rows. In contrast to the long-drum boiler, the single drum of the cross-drum boiler is not limited in the number of sections, since the length of the drum can be increased to suit the increase in width required for additional sections. The cross-drum boiler has a wide field of application. Boiler heating surfaces range from about 1000 to 25,000 sq ft. Capacities range from 5000 lb of steam per hr upward. The largest boiler unit of this type has a capacity of 525,000 lb of steam per hr. Design pressures range from 160 to 1450 psi. This design provides flexibility in the amount and disposition of heating surfaces to suit specific requirements of capacity, efficiency, draft loss, over-all dimensions, position of gas outlet, and space for superheater, reheater, and gas by-pass.

As mentioned above in reference to long-drum boilers, very few cross-drum straight-tube boilers are being built today, the preference being for water-tube boilers of the bent-tube designs.

Bent-Tube Boiler—General Classification

Many important modern designs of boilers, such as the type H Stirling, the Two-Drum Stirling, the Integral-Furnace, the Open-Pass, and the Radiant, are included in this general classification. The main elements of these water-tube boilers (also many subordinate designs) are essentially drums or drums and headers connected by bent tubes. Furthermore, the water-cooled furnace, of bent tubes suitably connected to the circulatory system to form a furnace enclosure, is an integral part of the boiler unit. The principal designs referred to are illustrated in Figs. 5 through 12.

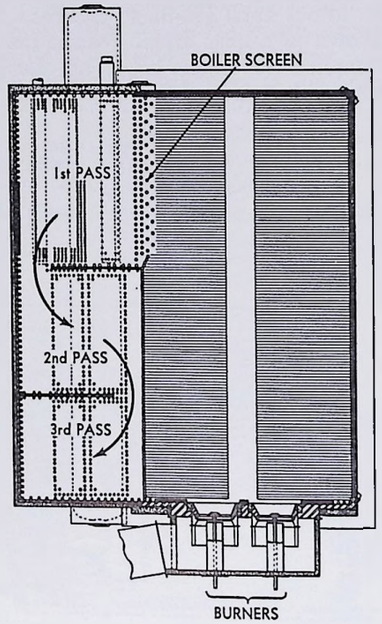


Fig. 6. *Plan section through generating banks of type FH Integral-Furnace boiler (B&W) illustrated in Fig. 6, Chapter 21*

For the best and simplest connection of tubes to drums, the tubes must enter the shell radially, hence all or almost all of the tubes so connected are bent. A common classification of bent-tube boilers is by the number of drums per boiler: two-, three-, and four-drum boilers. Gas baffles are arranged in many different patterns with gas flowing across and along the tubes in one or more passes.

Type H Stirling Boiler (B&W). One of the types of bent-tube three-drum boilers for low headroom, widely used in smaller sizes, is the H-type boiler illustrated in Fig. 5. Two drums are completely filled with boiler water, and the third drum, with feed inlet and steam outlet connections, is a steam-and-water drum. Boiler water circulates from the steam drum down to the lower drum, up through the inclined bank of generating tubes to the intermediate drum, and horizontally back to the steam drum, where the steam and water are separated. The steam passes through the steam outlet, and the separated water mixed with the incoming feed continues to circulate through the boiler. The hot gases make three passes over the boiler heating surface.

Boiler tubes, 3¼-in. OD spaced on 6-in. centers, are arranged vertically over each other to constitute a section. The horizontal or side spacing of sections is alternately on 5¼-in. and 6¾-in. centers to permit tube renewal at any point within the tube bank without disturbing adjacent tubes.

The size and capacity of the H-type boiler are varied by changing the length of the tubes and the number of sections wide.

Integral-Furnace Boilers

In the earlier stages of development, boilers were set over brick or refractory furnaces, and all of the heat-absorbing surface was in the boiler. As the units increased in size, furnace temperatures increased, and brick maintenance became excessive, particularly for pulverized-coal firing. The higher gas temperatures also resulted in increased slagging or fouling of the boiler surface. To overcome these difficulties, the furnaces were first partly and later completely water cooled. Furnace water-cooling not only decreased furnace maintenance and boiler slagging but also generated steam.

At first, furnace water-cooling, designed separately from the boiler, was applied to existing boiler designs with the water circulation more or less independent of the boiler circulation. However, boiler water was always circulated in the furnace circuits. Later, the furnace water-cooled surface and the boiler surface were designed together, so that each was an integral part of the other. This principle was applied to an extended series of boiler units of various arrangements and outputs.

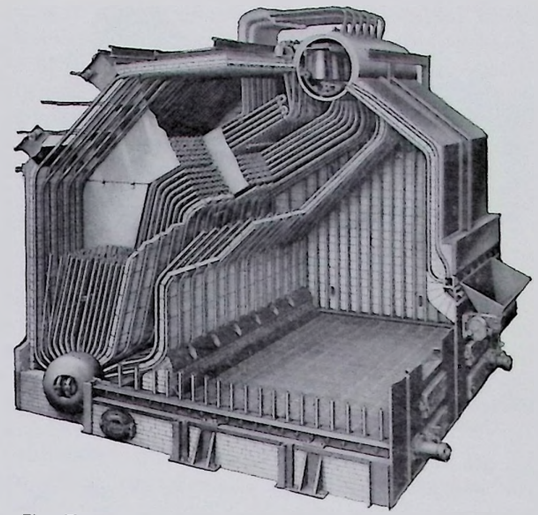


Fig. 7. *Type FF Integral-Furnace boiler (B&W) fitted for spreader-stoker firing with dump grates*

Type F Integral-Furnace Boiler (B&W). One of the earlier designs of the Integral-Furnace boiler was the F-type, which is essentially a two-drum boiler with a distinctive type of gas flow. The furnace gases enter one side of the boiler, flow through the full height, and make three passes horizontally across the boiler surface before leaving at the opposite side (similar to the FH boiler, see Fig. 6). This is a most efficient type of gas flow, since the gases are constrained to flow across all the tubes with no flow along the tubes. This desirable crossflow is obtained by using simple baffles placed between the tube sections. The interior of the steam drum is so arranged that the boiler water circulates down to the lower drum through a section of boiler tubes in the third or last gas pass of the boiler. The water from the lower drum then circulates up through the remainder of the boiler tubes and the tubes in the furnace water walls. The F-type boiler has been superseded by other types with improved features.

Type FH Integral-Furnace Boiler (B&W). This boiler (see Fig. 6, also Fig. 6, Chapter 21) is essentially the same as the F boiler; the main differences are that the boiler tubes are vertical rather than inclined and the drums are directly in line one above the other. Support is simplified, particularly for the larger units, since the boiler can be supported from either the bottom or the top. The outside walls of the FH boiler are completely water cooled with refractory backing supported on the water-cooled tubes.

Type FF Integral-Furnace Boiler (B&W). This boiler (Fig. 7) was developed and standardized for smaller capacities, 10,000 to 50,000 lb of steam per hr, and for pressures up to 600 psi. It is particularly suitable where headroom is limited and is designed for fuel

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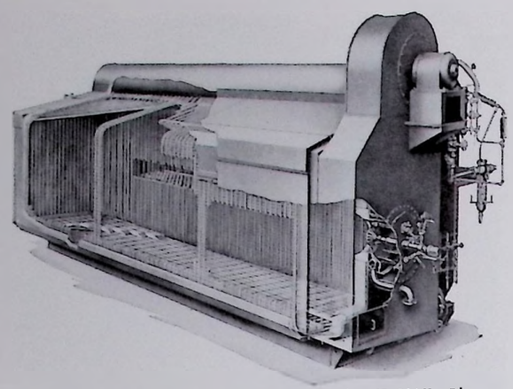


Fig. 8. *Type FM Integral-Furnace boiler (B&W). Shop-assembled unit, complete and ready to operate*

oil, gas, and stoker firing. The boiler-water circulation and gas flow through the boiler are similar to those in the F-type boiler. An additional water-cooled wall in the furnace of the FF boiler provides an "open pass" through which the gases flow after leaving the furnace.

Type FM Integral-Furnace Boiler (B&W). This shop-assembled unit, Fig. 8, is shipped complete with support steel, casing, forced-draft fan (unmounted in largest sizes), firing equipment, and controls, ready for operation when water, fuel, and electrical connections have been made. Only a stub stack is required. It is built for outputs from 8000 to 60,000 lb steam per hr and for steam pressures to 1050 psi. It can be fired with fuel oil, gas, or a combination of the two. A superheater may be fitted at the gas-entrance end.

Only a forced-draft fan is required, as the casing is airtight (welded) and the combustion gases are under pressure. Size of the unit is varied principally by changes in setting depth and drum length. Two combinations of width and height facilitate standardization of parts and assembly.

Gas baffles are so arranged that the gas temperature drop ($\triangle t$) and gas pressure drop ($\triangle p$) from gas inlet to outlet are about the same at a given rate of steam output, such as maximum load, for each size of unit. As the depth of the tube bank increases (from burner to rear) with increase in boiler size, the four baffles installed are progressively shortened until, in the largest size, the gases make but one pass through the bank from rear to front with little or no baffle diversion.

To assure proper steam quality, drum internals are provided in the larger higher-capacity boilers. The type and arrangement of steam separator depend upon the pressure and temperature as well as the capacity.

For still other types of Integral-Furnace boilers, see Chapter 23.

Stirling Four-Drum Boiler (B&W). This well-known bent-tube four-drum boiler with gas baffles arranged variously for one, three, or four passes has persisted without much change since its inception in the early 1890's (see Fig. 9). Only the lower drum is completely filled with water. The three top drums are steam-and-water drums. The feed inlet and steam outlet are connected to the rear steam drum. Boiler water circulates down through the rear bank of tubes to the lower drum and through upflow tubes to the center and front drums. A large part of the steam is separated from the water in the two front drums, and the separation is completed in the rear drum. Steam and water flow between the upper drums is through steam and water circulators connecting the drums.

The Stirling Four-Drum boiler has many different arrangements of gas baffling. The selected arrangement of baffling will depend upon the acceptable draft loss, efficiency, superheat, type of fuel, and position of gas outlet. A common arrangement is the three-gas-pass baffling shown in Fig. 9.

Boiler tubes are 3¼-in. OD and are arranged in sections which are spaced alternately on 5¼-in. and

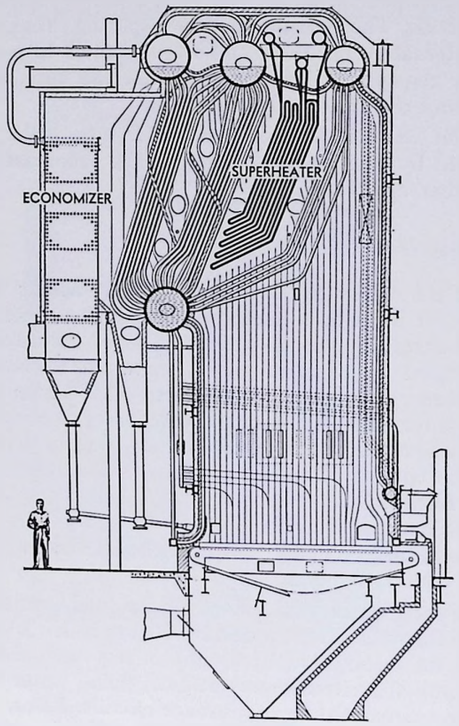


Fig. 9. *Stirling Four-Drum boiler (B&W) for traveling-grate spreader-stoker firing with gas as auxiliary fuel. Design pressure 725 psi; steam temperature 825 F; maximum continuous steam output 125,000 lb/hr*

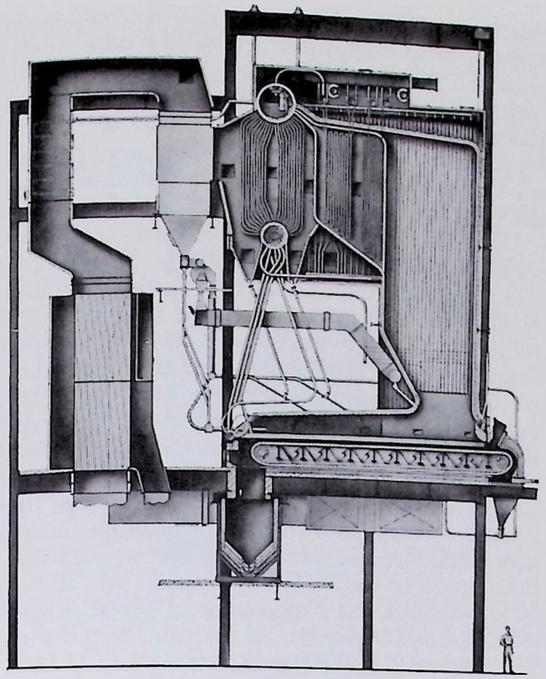


Fig. 10. *Stirling Two-Drum boiler (B&W) with single gas pass; 220,000 lb steam/hr, arranged for chain-grate-stoker firing*

6¾-in. centers to permit tube renewal without disturbing adjacent tubes. To increase gas velocities along the tubes, the front-to-back spacing of the tubes in the section is about 4 in. on centers.

The size and capacity of these boilers are varied by changing the length of the tubes and the number of sections wide. The size varies over a considerable range. This type of boiler is particularly well suited for rapidly fluctuating loads and where water conditions may not be maintained at high quality.

Although there are many Stirling Four-Drum boilers in operation, this design has been superseded, for economic and other reasons, by the simpler Stirling Two-Drum boiler.

Stirling Two-Drum Boiler (B&W). The simple arrangement possible for the connecting tubes, with one upper steam-and-water drum directly over one lower drum, led to the development of a series of designs known as the Stirling Two-Drum boiler. The most important unit of the series, generally superseding earlier designs, is the single-gas-pass arrangement, illustrated in Fig. 10, where the tubes in the generating banks, of 2½-in. OD swaged to 2 in. entering the drums, are sidespaced on $3\frac{9}{16}$-in. centers and the front-to-back spacing is 4 in. on centers.

Superheaters usually are fitted for steam temperatures to 1000 F or even more. As indicated in the illustration, a single row of 3-in. OD tubes on 18-in.-center sidespacing forms a boiler screen ahead of the superheater. Dry steam (from cyclone separators) enters the superheater through the saturated-steam connection. For most fuel applications, the furnace is completely water cooled and is an integral part of the boiler unit as a whole. With swaged tubes giving an increased ligament between tube seats, an economical drum-sheet thickness is possible for a gas mass flow based on a given arrangement and spacing in a tube bank. Tubes swaged at the ends may be cleaned internally by chemical means or by mechanical turbines.

The heating surface and output of the Two-Drum boiler may be varied by changing the length of tubes and the number of sections wide.

High-Pressure and High-Temperature Boilers

Because of factors noted in Chapter 1 and particularly the advent of pulverized-coal firing, the adoption of furnace water-cooling inevitably resulted in the integration of furnace, boiler, superheater, and economizer as a complete unit. The substantial results of such integration in units of moderate steam output, such as Integral-Furnace, Two-Drum, and Four-Drum Stirling boilers, have been outlined above.

In the rapid development of power-plant economy, the single-boiler single-turbine combination is now favored for the central station and where electric power is the end product of heat transformation. There is an incentive to use very large electrical generators, since the heat rate, unit investment, and labor costs decrease as the size increases. In the design of large boiler units for this application, the important factors are: (a) high steam pressure, (b) high steam temperature, (c) bleed feedwater heating, and frequently (d) reheat. High steam pressure means high saturation temperature and low temperature difference. High steam temperature means high initial temperature and, usually, reheating to high temperature for re-use of the steam. Bleed feedwater heating lowers the mean temperature difference and increases the gas temperature leaving an economizer. An air heater is then used, not only to lower the final gas temperature but to provide hot air for combustion. These factors and, above all, the economic need for continuity of operation to realize an optimum return on the large investment involved have combined to produce a "boiler unit" unique in many respects from earlier concepts. It is therefore apparent that the principle of the integrated boiler unit is firmly established for very large boilers as well as for boilers of smaller outputs.

Thus, as steam pressures increased, the steam temperatures also increased, which required proportionally more superheating surface and less boiler

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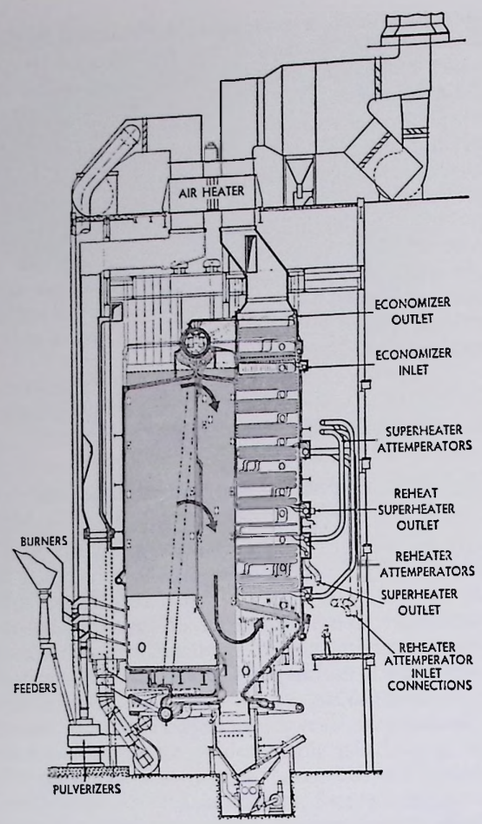


Fig. 11. *Large Open-Pass boiler (B&W), pulverized-coal fired; slag-tap furnace*

surface. When pressures exceed 1000 psi, it is usually more economical to replace whatever boiler surface is still required with economizer surface. All the steam is then generated in the furnace, water-cooled wall enclosures of superheater and economizer, boiler screens, platens, division walls, and in some cases the outlet end of a steaming economizer. The B&W Open-Pass (OP) and Radiant (RB) boilers, illustrated in Figs. 11 and 12, are typical high-pressure and high-temperature units. Radiant boilers have been built to generate up to a peak of 2,400,000 lb of steam per hr which, in this particular example with steam reheat, required a fuel input at the burners of 3,135,000,000 Btu per hr. Of this input, 2,840,000,000 Btu per hr were absorbed in the boiler unit and were delivered (with a boiler efficiency of 90.5%) to the turbine.

Slag-Tap Boiler Unit. The liquid-ash-removal (slag-tap) boiler unit illustrated in Fig. 11 has an output of 930,000 lb steam per hr at continuous rating, with steam pressure 2170 psi, final steam temperature 1050 F, and reheat steam temperature 1000 F. In this instance of an integrated unit, not only is the furnace completely water cooled by tube-to-tube construction but even the boundary walls of the superheater and economizer are water cooled, although not with touching tubes. This unit is top supported from a single elevation, permitting downward expansion in all its components. In a unit of this type, wall tubes usually are 3-in. OD, superheater tubes are 2½-in. and 2¾-in. OD, economizer tubes are 2¾-in. OD, and reheater tubes are 2½-in. OD.

In the particular unit illustrated, there is very little so-called boiler surface. The small bank of staggered boiler convection tubes directly below the superheater is the sole remaining vestige of boiler surface as formerly defined. Known as an "Open-Pass" boiler, this type derives its name from the manner of gas flow, where the gas from the furnace flows through open "thoroughfares" or "passes" (one such pass in the unit illustrated) before entering the convection passes. The lower portions of the furnace walls on all four sides are full-studded, which is customary practice for B&W slag-tap units. At full load the gases enter the superheater at about 2250 F (MHVT, see Chapter 6). At the point of gas entry the sidespacing of the superheater sections is 12 in. on centers. At the gas exit end of the superheater the tube sidespacing is 4 in. on centers. Regulation of steam temperature is by spray attemperation.

Dry-Ash Boiler Unit. The dry-ash hopper-bottom boiler unit illustrated in Fig. 12 has an output of 575,000 lb of steam per hr at continuous rating, with steam pressure 1675 psi (design), final steam temperature 1000 F, and reheat steam temperature 1000 F. This type is known as the "Radiant" boiler and, like the Open-Pass boiler, is almost devoid of boiler convection surface. The only remnants of so-called boiler surface are the single row of tubes (omitted in later designs, see Chapter 23) before the pendent superheater and the single staggered row following the reheater. The entire furnace and pressure-part enclosures are constructed of 3-in. tubes, touching in the furnace and slightly separated in the superheater and economizer vestibules. Superheater tubes are 2½-in. and 2¾-in. OD. Reheater and economizer tubes are 2½-in. OD.

The entire unit is supported from a single elevation overhead, and all components expand together. Elements of the horizontal superheater surface drain, but elements of the pendent superheater surface are nondraining, and at start-up the superheater is boiled to dryness (see Chapter 21). A certain degree of steam-temperature control is possible with burners at two elevations, as shown in Fig. 12. At low loads, for instance, the steam temperature can be raised by firing the top set of burners rather than the bottom set of burners. The over-all (final) temperature control, however, is by spray attemperation.

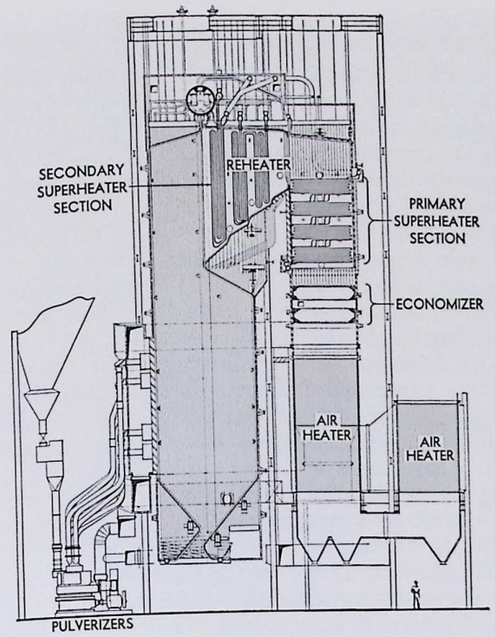


Fig. 12. *Large Radiant boiler (B&W), pulverized-coal fired; dry-ash removal; short air heater section at cold air inlet*

At full load the gases enter the superheater at about 2100 F (MHVT). At the point of gas entry the sidespacing of the superheater tubes is 12 in. on centers. At the gas exit end of the superheater the sidespacing of the tubes is 3½ in. on centers.

For units designed to operate at either subcritical or supercritical pressures, see the Universal-Pressure (UP) boilers in Chapters 1 and 23.

Boiler Supports

Boiler and furnace wall tubes are usually supported by the drums or headers to which they are connected. Some boilers are bottom supported from the lower drums and headers, while others are suspended from the upper drums. The FH boiler is typical of a bottom-supported boiler. The boiler load is taken on saddles placed under the lower drum and under the lower headers of furnace side, front, and rear walls. The Stirling boiler is typical of a top-supported unit, with the upper drums seated on saddles at the ends of the drums. Stirling boilers may also be supported by U-type hangers around the drums carried to steel above. Two-Drum and high-capacity boilers such as the Open-Pass and Radiant are almost always top supported through U-type hangers at the ends of the steam drum and by rod hangers fastened to upper headers and tubes.

In the design of proper supports the following considerations are important:

1. The tubes must be so arranged that they will not be subject to excessive bending-moment stresses in carrying the weight of the tubes, drums, parts, and contained water. When the unit is bottom supported, the tubes must satisfy column requirements.
2. Holding strength of the tube seats must not be exceeded.
3. Provision must be made to take care of expansion of the pressure parts. For a top-supported unit, the hanger rods must be designed to swing at the proper angle, and they must be long enough to take the movement without excessive stresses in either the rods or the pressure parts. Bottom-supported boilers should be anchored only at one point, guided along one line, and allowed to expand freely in all other directions. To reduce the frictional forces and resultant stresses in the pressure parts, roller saddles or mountings are desirable for bottom-supported heavy loads.

Boiler Horsepower and Rating

In the early days the "size" of a boiler was denoted by boiler horsepower. One rated or nominal boiler horsepower was defined as equivalent to 10 sq ft of boiler heating surface. A "developed" boiler horsepower was arbitrarily set as equal to the evaporation of 30 lb of water per hr from an initial temperature of 100 F to steam at 70 psi pressure. When the term "equivalent evaporation from and at 212 F" became popular, the old value of 30 lb at 100 F and 70 psi for a developed boiler horsepower was converted to its equivalent in heat capacity of 34.5 lb of water per hr from and at 212 F. Still later, when steam flow began to be recognized as a better measure of size (and metering equipment became available), an equivalent heat unit of 33,480 Btu per hr was used for conversion purposes. The relations between rated boiler horsepower, developed boiler horsepower, boiler heating surface (sq ft), per cent rating, and steam flow are given in the following formulas:

$$\text{blr-hp} = S/10$$

$$\text{blr-hp}_d = \text{blr-hp} \times R_b/100$$

$$\text{blr-hp}_d = \frac{w(h - h_w)}{33{,}480}$$

$$R_b = \frac{w(h - h_w) \times 100}{33{,}480 \times \text{blr-hp}}$$

$$R_b = \frac{w(h - h_w) \times 100}{3348 \times S}$$

$$w = \frac{R_b \times 3348 \times S}{(h - h_w) \times 100}$$

where:

blr-hp = rated boiler horsepower
blr-hp$_d$ = developed boiler horsepower

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S = boiler heating surface, sq ft
R_b = boiler rating, per cent
$w(h-h_w)$ = total heat absorbed in steam, Btu/hr
h = enthalpy of steam at boiler outlet or at superheater outlet when unit has superheater, Btu/lb
h_w = enthalpy of water at boiler inlet or at economizer inlet when unit has economizer, Btu/lb
w = steam flow, lb/hr

When the major part or all of the heat-absorbing surface is in the boiler with little or no furnace, superheater, and economizer surface, the terms "boiler horsepower" and "boiler rating" are quite suitable and represent not only some measure of capacity but also some measure of boiler duty. Obviously, a boiler operating at 300% rating is operating at a higher duty than one operating at 200% rating. The term "boiler rating" is still in use today in the small-boiler field, and it indicates some measure of duty where furnace-cooling and superheater surfaces are absent or relatively small compared with the boiler convection surface.

Most boiler units now, as noted above, whether small or large, have a relatively large area of heating surfaces other than boiler convection surface, so that a rating based solely on boiler heating surface has very little or no meaning. The terms "boiler horsepower" and "boiler rating" are, therefore, not properly applied to these boilers.

In all types of boilers and particularly in the medium- and larger-capacity units, the output is now almost invariably given in lb per hr of steam flow at maximum continuous load. Partial loads are usually designated in percentages of full load, such as 75% of load or three-quarters load. Peak or overloads may be designated in percentages of full load, such as 110% load.

The amount of heat-absorbing surface in all parts of the boiler unit is of prime importance in indicating size and capacity. However, many other important factors, as noted below, must be considered in evaluating a modern boiler unit.

Boiler Design

The design of boiler heating surface as such must necessarily be governed to a great extent by the design of other heating surfaces and of the unit as a whole. The selection of the type of unit or the design of a new unit for a given service depends upon many factors. Some of the more important factors are:

1. Steam flow, pressure, and temperature.
2. Reheat steam flow, pressure, and temperature, entering and leaving.
3. Feedwater temperature and conditions.
4. Load characteristics and type of service.
5. Fuel and firing.
6. Draft.
7. Efficiency and economics.

In modern practice the amount of boiler surface ahead of the superheater may vary greatly. In some instances, as in the unit illustrated in Fig. 12, boiler surface is reduced to a single row of tubes, widely spaced to form a slag screen. In other units there is considerable boiler surface arranged in several rows of tubes, the first one or two rows widely spaced to form a slag screen. The boiler surface ahead of the superheater serves to reduce the gas temperature entering the superheater, and, by so doing and by shielding the superheater from the direct radiant heat of the furnace, it protects the superheater from excessive metal temperatures and from fouling and slagging of the surfaces when firing coal or oil fuels with slagging characteristics (see Chapters 2, 3, and 18). The amount of boiler surface, including the slag screen, ahead of the superheater depends upon the desired gas temperature entering the superheater. The screen is usually in two or more rows of tubes.

The gas temperature leaving the furnace or entering the boiler depends mainly upon the furnace size and the extent of furnace-wall cooling surface. Since the cost of furnace-wall cooling surface is relatively higher than that of boiler surface, the furnace size and surface are limited to the requirements for the proper burning of the fuel and for the sufficient lowering of the gas temperature entering the boiler screen to avoid excessive slagging. The gas temperature entering the superheater must be high enough to give the superheat desired with a reasonable amount of superheater surface and the use of economical materials. The superheater surface must also be arranged so that it can be kept clean to absorb heat as intended (see Chapter 12).

Boiler tubes ahead of the superheater may be inclined as in the Long-Drum, Cross-Drum, H-type, and Stirling Four-Drum boilers or vertical as in the FH, Stirling Two-Drum, and Radiant boilers. The tendency toward slag adherence is less with vertical or nearly vertical tubes, and many operators prefer this construction where slagging is liable to occur.

Slag screens are arranged in many different tube patterns. It is essential to space the tubes far enough apart, particularly perpendicular to the gas flow. The clear space between tubes should be large enough so that any slag that adheres to the tubes cannot build out and bridge or plug the gas passage. Both staggered and in-line arrangements of tubes, Fig. 13, are used. In the staggered arrangement not only the spacing perpendicular to the gas flow but also the backspacing must be made large enough to prevent plugging. In the in-line arrangement the backspacing is not so important. When the backspacing is small (tubes nearly touching) compared with the sidespacing in the in-line arrangement, the tube sections

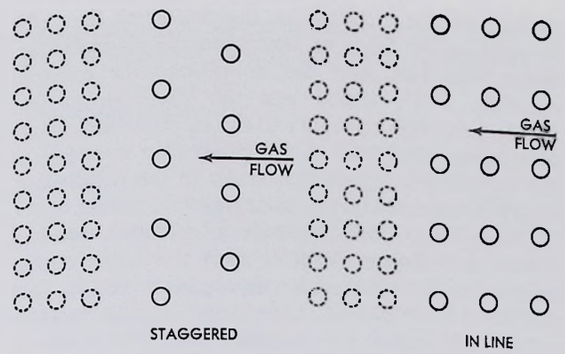


Fig. 13. *Diagrammatic arrangement of slag screens ahead of tube banks*

are sometimes called platens. Platens spaced the proper distance apart are effective in dropping the gas temperature through a slagging-temperature zone. Clear distance between surfaces should be in the order of 9 to 12 in., or even greater in some instances.

Design of boiler surface after the superheater will depend upon the particular type of unit selected, desired gas temperature drop, and acceptable draft loss through the boiler surface. Typical arrangements of boiler surface for various types of boilers have been illustrated. The object in the design of boiler heating surface—and this is true of other so-called convection heating surfaces of a unit, such as superheater, economizer, and air heater—is to establish the combination of tube diameter, tube spacings, length of tubes, number of tubes wide and deep, and gas baffling that will give the desired gas temperature drop with an acceptable draft loss. In other words, the design should provide an optimum combination of heating surface and gas mass flow to give the desired results.

For a given temperature, weight, and specific heat of a gas entering any component of a boiler, sufficient surface must be specified to drop this temperature to a desired value. To lower the gas temperature it is necessary to provide both surface (at lower temperature than that of the gas) and pressure drop (to cause flow). The amounts of surface and pressure drop required depend upon the gas temperature drop desired. The two requirements are interrelated. If one is reduced, the other must be increased. It is, of course, impractical to pursue this procedure beyond obvious limits.

For a given gas mass flow (weight of gas, lb/hr ÷ area of passage, sq ft) or a given gas velocity, a considerably higher gas film conductance, heat absorption, and draft loss result when the gases flow at right angles to the tubes (crossflow) than when they flow parallel to the tubes (longitudinal flow). Gas turns between tube banks generally add draft loss with little or no benefit to heat absorption and should be designed for easy flow.

Economizers—Early Types and Development

In a steam-generating unit, the economizer absorbs heat from the flue gas and adds it as sensible heat to the feedwater before the water enters the boiler. In this way the gases are still further cooled, and a better economy is effected, hence the term "economizer."

The design and use of economizers followed naturally the development of boilers. Among the various designs applied during the nineteenth century, the straight-tube boiler and economizer of the B&W type and the bent-tube boiler and economizer of the Stirling type survived as the most important prototypes for the modern economizer. The earlier designs were based on the requirements for straight-tube boilers operating on natural draft and usually called for large diameter tubes, widely spaced to reduce resistance to gas flow.

Cast iron, commonly used for boilers operating at pressures up to 250 psi, proved to be an economical material for economizers because of its inherent resistance to corrosion, both internal and external.

When steel was first available, it was successfully used in boilers but could not be used in economizers because of the corrosive effect of free oxygen. In the boiler, most of the dissolved oxygen in the feedwater was driven off in the steam drum with the steam, and the corrosive effect on the boiler tubes was therefore reduced. In the economizer, as the temperature of the feedwater rose, the oxygen was gradually driven out of solution to attack the inside of the tubes. Also, at saturation temperature, the temperature of the water in the boiler was above the dew point of the gases, while in the economizer, the feedwater, with a temperature often less than 100 F, cooled the surrounding flue gases so much that moisture condensed on the outside of the tubes. This moisture, with some of the sulfur dioxide and sulfur trioxide from the flue gas in solution, caused corrosion. The moisture also formed a bond for the collection of ash on the tubes, which greatly reduced heat transfer and even restricted the gas passages. Cleaning apparatus had to be used to remove this deposit. Early economizers of cast iron were equipped with mechanical scrapers. Some cast-iron economizers of this general type are still in use with low-pressure boilers.

Following these early designs, the economizer was gradually improved in design and performance, keeping step with other developments in the generation of steam. Steel replaced other metals in boilers and engines, so that steam pressures could safely be raised. It became economical carefully to treat the

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feedwater and to remove the oxygen by deaeration. Development of the steam turbine with stage bleeding increased the feedwater temperature to the economizer. The additional pressure drop through the unit, as a result of increased surface, led to the improvement of fans and the development of better soot blowers for surface cleaning.

The steel-tube economizer has now replaced the cast-iron design. Deaeration of feedwater has reduced internal corrosion. High feedwater temperatures have reduced condensation and, consequently, the corrosion and plugging on the outside of the tubes. The higher drafts now available make it possible to use the most desirable tube diameters and tube spacings for economical heat transfer and cleaning by steam or air.

One of the earliest straight-tube steel economizers of B&W design was built to ASME standards in 1915 and, like the boiler, was of the sinuous-header type, with three passes for gas flow to improve its efficiency. In 1916 a similar type economizer was built for use with a Stirling boiler.

A successful two-drum bent-tube type of economizer was developed early in 1917. A modern version of this type applied to a Radiant boiler is illustrated in Fig. 14. While an integral-type bent-tube economizer with separate lower drum and with the upper ends of the tubes connected to the rear top drum of a Stirling boiler is occasionally built, the two-drum type is preferred (if drums are to be used) as it is less limited in size, location, and shape. In either case, the tube diameters and spacing can be the same as for the boiler, or they can be modified to improve heat absorption, draft loss, water velocities, and the conditions for external cleaning. Water is fed into the lower drum and flows up through the tubes to the upper drum. Gases flow either along the tubes, preferably entering at the top and flowing down counter to the water flow, or across the tubes in single or multiple passes.

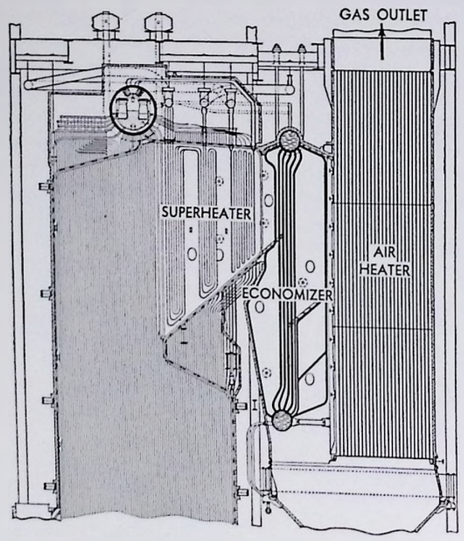


Fig. 14. *Two-drum bent-tube economizer arranged with three gas passes*

Before 1920, economizers were built with tubes of the same or larger diameter than the boiler tubes. One of the first economizers to use tubes of 2-in. OD was built early in 1920. Compared with larger-tube units, the small-tube economizer requires less space, is easier to manufacture, uses less steel for tubes and casing, and is particularly suitable for high pressures.

In the early 1920's, most 2-in.-tube economizers were made with straight horizontal tubes rolled into horizontal square headers. Handhole fittings were used in the headers opposite the ends of the tubes so that every tube could be inspected and mechanically cleaned internally. Water fed into the bottom header flowed up through the tubes from header to header and then from the top header into the boiler. Gas entered the top of the economizer and flowed down across the staggered tubes, leaving at the bottom. This type of economizer was used with the first 650 psi boiler units in 1922, and with the pulverized-coal-fired high-pressure units developed in 1924. (See Figs. 12, 13, and 15, Chapter 1.)

An improved design of the horizontal steel-tube economizer, frequently used when it was necessary to clean the inside of the tubes mechanically, was developed during the 1920's. At one end the tube bends were formed by bending the 2-in. OD tubes through 180 degrees. At the other end special flanged and bolted return-bend fittings were used.

Tube Sizes and Spacing

The use of inhibited acid to clean internal surfaces (see Chapter 21) has materially simplified the design of economizers. The tubes can be continuous from inlet to outlet headers with terminals rolled or welded. The tubes can be made any length and diameter, with 2-in. and 2½-in. OD in most cases. The sidespacing and backspacing can be arranged for good external cleaning, absorption of heat, and draft loss. Clear lanes of 1 in. and under should be used only for "clean" fuels. Lanes usually not less than 1½ in. and as much as 2 in. are generally used for fuels liable to cause gas-side deposits. For cleaning, backspacing is less critical and is usually about 3 in. on centers for 2-in. OD tubes and 3½ in. on centers for 2½-in. OD tubes.

All high-pressure and most intermediate- and low-pressure economizers are now designed with horizontal, draining, continuous tubes. In most high-capacity high-pressure units the economizers are located inside the boiler setting (see Fig. 4, Chapter 21).

Steaming Economizer

There is sometimes an advantage in producing part of the steam in an economizer rather than in the boiler surface. So far as the amount of water that can be evaporated in the steaming type of economizer is concerned, there appears to be no operating limit in size. However, in practice it is customary to limit the evaporation to steam in the economizer to a maximum of about 20% of the feed at full boiler output and, of course, less as the load decreases. Since 1924, steaming economizers have been installed in a considerable number of boiler units. Because of the practical difficulties in treating a high percentage of the feedwater to a condition suitable for steaming economizers, they cannot be used to good advantage in boiler units where a high feed make-up is required.

Economizer Size

Economizer surface can be justified only where it can absorb heat at less cost than other types of surface. The temperature of the feedwater entering the economizer is always less than the saturation temperature in the boiler. Since the economizer surface is at a temperature lower than that of the boiler surface, the heat remaining in the gases can be absorbed at a greater rate and the gases can be cooled to a lower temperature with economizer surface than with additional boiler surface. Therefore, in a steam-generating unit, there is usually a gas temperature level at which it is economical to stop absorbing heat through surfaces at saturation temperature and to start absorbing heat in an economizer.

Where air preheaters are not used, the gas temperature from the economizer is set by making an economical balance between the cost of economizer surface and the saving in the cost of fuel. Where air preheaters are used, the gas temperature leaving the economizer is determined by the requirement that the temperature of the gas entering the air preheater should be sufficiently high to heat the air for adequate fuel drying or for good combustion performance.

Sufficient surface is installed in the economizer to absorb enough heat economically to give the desired gas exit temperatures. The amount of surface will vary with the temperature of the incoming feedwater. The feedwater should always be deaerated, with the deaerator at or above 212 F to minimize stray air leaks in the system. For high-pressure units, the temperature of the feed to the economizer is determined by the economical bleed points on the turbine and will rise as the pressure increases, with consequent decrease in size of economizer, since the amount of heat to be absorbed is less.

Location, Arrangement, and Cleaning

Location of the economizer will vary with the over-all design of the boiler unit. Where air heaters are not installed, as in many small boilers of so-called standard type, economizers are located in the gas duct between the boiler gas exit and the fan or stack. It is always preferable that the gas from the boiler should flow down across the economizer tubes and that the water should enter at the bottom and flow up through the tubes. The counterflow design will reduce the required amount of surface and draft loss to a minimum. It will eliminate unstable water flow, give the most uniform gas distribution, and make possible the proper use of a "steaming economizer" wherein some steam is actually generated in a portion of the water-exit end.

In many high-pressure units, location of the economizer in an upward flowing gas stream is fixed by the layout. In such cases small economizers can be installed for all-parallel flow (see Fig. 4, Chapter 21). Large economizers require special consideration and should be installed with counterflow for the water-inlet section and with parallel flow for the water-outlet section (see Fig. 15).

Since the air heater, when installed, is the final heat trap, the economizer is generally located ahead of the air heater in the gas stream. In some cases where very low exit gas temperatures and high air temperatures are desired, it may be necessary to divide the economizer and the air heater and place the cooler section of the economizer between the air heater sections.

When ash-bearing fuels are to be fired and tenacious ash deposits are liable to form on the tubes, the economizer should be located directly over a hopper so that water-soluble constituents of the deposits can be dissolved and the deposits washed off (during out of service periods) and disposed of.

In addition to washing as may be necessary, the external surface of economizers is cleaned with the same type of equipment used to clean boiler surfaces. Economizers are designed with tube spacing and

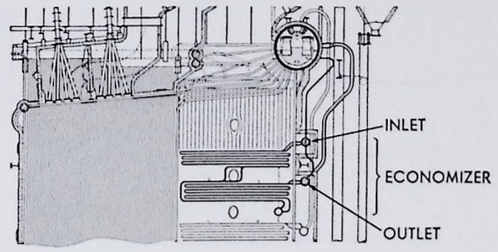


Fig. 15. *Continuous-tube economizer arranged for part counterflow and part parallel flow*

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depths of banks best suited for external cleaning by either steam or air blowers.

External Corrosion

External corrosion of economizers may occur when the water vapor in the flue gas condenses on the surface of the tubes, and corrosion is accelerated when this happens in the presence of the products of combustion of sulfur. The rate of corrosion increases as the metal temperature is reduced. As the amount of sulfur in the fuel increases, the dew point increases and so does the potential rate of corrosion. Pulverized-coal ash seems to act as an inhibitor in retarding the rate of corrosion. Experience indicates that the rate of corrosion can be reduced to safe limits by keeping the temperature of the feedwater to the economizer above certain minimum values. Minimum safe values of tube-metal temperatures for varying percentages of sulfur and for several types of firing are indicated by the curves in Fig. 16. When sulfur is burned under a boiler, as, for instance, in one commercial process in the production of sulfuric acid, the dew point is so high that all steel surfaces must be at or above 450 F to assure a reasonable life. Such a boiler would not be fitted with an economizer.

Internal Corrosion

Economizers are also subject to internal corrosion from dissolved oxygen and low hydroxyl ion concentration (a low pH). Oxygen corrosion can be eliminated by deaeration to zero oxygen (see Chapter 21). Steel in the economizer is attacked faster by pure water, which has equal hydrogen and hydroxyl ion concentrations (pH = 7), than by water which has a higher hydroxyl ion concentration. It is therefore necessary to maintain a pH value of between 8 and 9 for the water passing through the economizer. A successful means sometimes used to maintain the desired pH value of the feedwater is to recirculate some boiler water to the inlet end of the economizer.

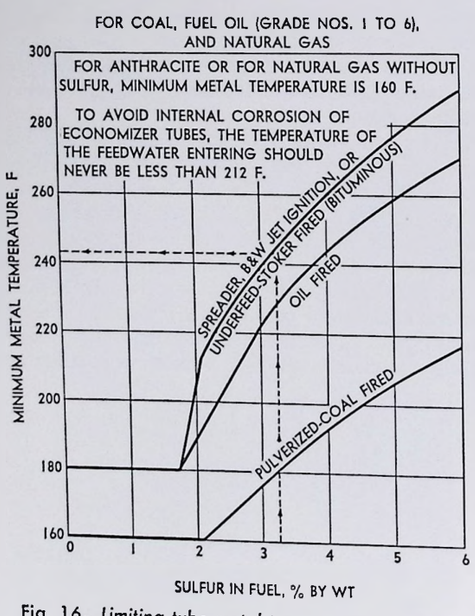


Fig. 16. *Limiting tube-metal temperatures to avoid external corrosion in economizers or air heaters when burning fuels containing sulfur*

Air Heaters — Early Types and Development

The air heater in a steam-generating unit reclaims some heat (which would otherwise be lost) from the flue gas or occasionally from some other source and adds that heat to the air required for the combustion of the fuel. Normally, air-cooled furnace walls or floors, which are sometimes used as a source of heat to improve combustion conditions, are not included in the term "air heater."

As in the case of economizers, air heaters originated in Europe before they were used in the United States. Early patent files reveal the invention of air heaters long before the industry had developed to the stage where they could be used economically.

The first air heater to be installed in a commercial boiler unit in the United States was built in 1922. It was made of flat parallel steel plates which formed alternate gas and air passages. It was necessary that the passages be completely separate one from the other, since any air leakage into the gas stream not only reduced the efficiency of the air heater but increased the load on the forced- and induced-draft fans. Plate air heaters are still in use but are not as suitable as tubular or regenerative heaters for the higher gas and air pressures which are now common practice.

The first tubular air heater developed by B&W was placed in service early in 1923. Demand for air heaters developed quickly. A total of 50 air heaters was built by B&W during the two-year period of 1923 and 1924. Several factors influenced this demand. Before 1922, the highest pressures commonly used in power generation ranged from 350 to 400 psi. In 1922, units of 650 psi were developed to generate more kilowatthours per pound of steam. This in turn increased the demand for more pounds of steam per pound of fuel. The use of air heaters helped to achieve this end.

Before the early 1920's, practically all coal used in the production of steam was burned on stokers or grates. Furnace walls were generally all refractory or, in a few instances, partly water cooled. Hot air supplied to the fire improved stoker performance

by raising the temperature level in the combustion zone. While this elevated temperature aided in the combustion of fuel, it also raised the temperature of all stoker parts and furnace surfaces, thus tending to increase the cost of their maintenance. So much, however, was gained in combustion efficiency by reclaiming some of the heat of the flue gas for heating the combustion air that boiler units were soon designed for operation with hot air without excessive maintenance costs. The development of water-cooled furnace walls that could withstand high combustion temperatures contributed greatly to the effective use of air heaters in boiler units.

In the early and middle 1920's, there was a tremendous increase in the development and use of pulverized-coal firing in central stations. For this type of firing, hot air was ideal for drying, transporting, and burning the coal in suspension. The demand for air heaters was thus well established before the end of 1924.

Advantages of Preheated Air

Preheated air is always required or justified with certain types of steam-generating units. Practically all pulverized-coal-fired units require hot air for drying the fuel, and the larger stoker-fired units burning bituminous coal operate more efficiently with preheated air up to temperatures suitable for stoker parts. Preheated air, however, is not essential for the smaller stoker-fired units. In deciding on the use of an air heater, as with most other component equipment in all fields, much depends upon the conditions and requirements of the service.

Whether or not an air heater should be installed, and if so, how large the heater should be, will depend upon economic and engineering factors. The major economic factors are: the original cost of the air heater, the operating cost for fuel and fan power, and the maintenance cost. The principal engineering factors are: the available space, the characteristics of the fuel used, and the desired temperatures for the preheated air and the exit gas.

With the common fuels (coal, oil, and gas) and identical furnace conditions, the efficiency of a boiler unit as a whole will increase by about 2.5% for 100 F drop in exit gas temperature from the unit. In terms of an air heater, this corresponds to an increase of approximately 2% for each 100 F rise in air temperature. An air heater providing combustion air at temperatures ranging upward from 300 F will often effect savings in fuel ranging from 5 to 10%, as indicated by the curve in Fig. 17. While temperatures above the maximum in the graph (600 F) have been and are used in boiler practice, it may be noted that the difficulties of arrangement in the boiler unit and in obtaining economical materials suitable for the temperature become increasingly troublesome if this limit is exceeded.

Use of preheated air will speed up combustion at all loads, improve combustion at low loads, and increase efficiency. In a new installation, the use of an air heater will frequently permit a reduction in the size of the boiler. Some saving will likewise be possible in the size and cost of the firing equipment, without impairment of the boiler output.

Classification of Air Heaters

Air heaters may be classified according to the principle of their operation as (a) recuperative and (b) regenerative. Heat is transferred to the air from the flue gases of the boiler or from other sources. Among successful air heaters operating on the recuperative principle, there are tubular and plate types, with the heat provided by 1) the flue gases of the boiler, 2) steam in coils, or 3) separately fired furnaces. Among air heaters operating on the regenerative principle are the rotary regenerative (Ljungstrom) and the diphenyl, with heat provided by the flue gases, and the pebble type, with heat provided by flue gases or a separate furnace.

In a recuperative design, the heat is transferred directly from the hot gases or steam on one side of the surface to air on the other side.

In a regenerative heater, the heat is transferred indirectly from the hot gases to the air through some intermediate heat-storage medium, as in the rotary regenerative, diphenyl, and pebble heaters. Use of the principle of regeneration is not new. Regenerative air heaters were used in other industries before they were used in steam generation. For open-hearth furnaces, it was common practice to control the gas flow from the furnace and the air flow to the furnace by dampers, so that the gas flow would alternately heat one flue and then the other. The air, flowing through the hot flues alternately, would be heated to over 1000 F before entering the

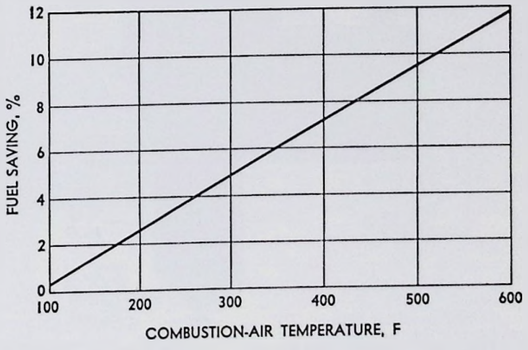


Fig. 17. *Order of improvement in efficiency when heated combustion air is used in boiler units*

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combustion chamber. The brickwork in the flues acted as the heat-storage medium.

Tubular Air Heater

The B&W tubular air heater utilizes the heat from the products of combustion after they leave the boiler, economizer, or superheater. It is essentially a nest of straight tubes expanded into tube sheets and enclosed in a suitably reinforced steel casing. This casing functions as the air or gas flue and is provided with air and gas inlet and outlet openings and the necessary hoppers and structural steel supporting members. The vertical type, suspended from above, is illustrated in Fig. 18. The unit can alternatively be supported from below. Horizontal types are also made. The casing is made of flanged steel-plate panels bolted to stiffening bars and supported by a frame of steel bars, angles, and channels. The frame also carries the flanged air-inlet and air-outlet boxes. Dust and cinder-disposal hoppers as required are provided under the tubes to suit flue arrangements.

In the modern tubular air heater the tubes are

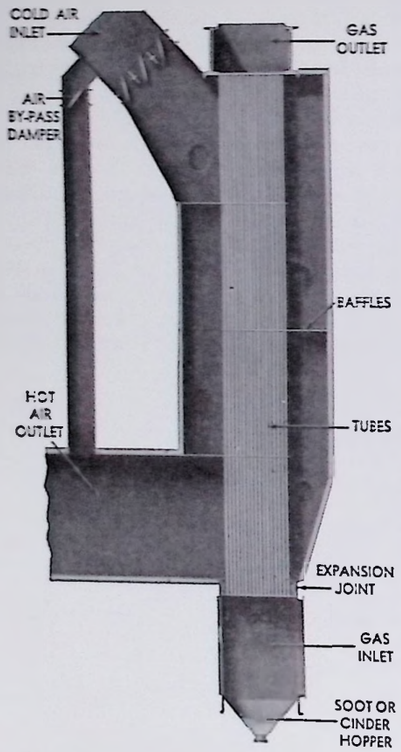


Fig. 18. *Tubular air heater arranged for counter-flow of gas and air, with an air by-pass to control metal temperature at air inlet end*

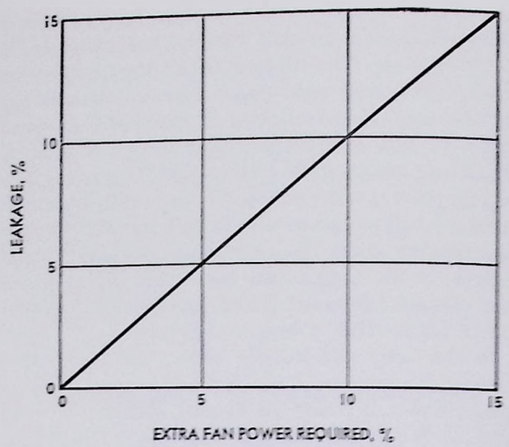


Fig. 19. *Approximate effect of air heater leakage on fan power required for boiler unit, assuming leakage from air inlet to gas outlet*

rolled into tube sheets at both ends. To provide for expansion, one tube sheet should be free to move with respect to the casing, and this must be accomplished without any air leakage. One method is to fit a flexible bellows-type expansion-joint connection as indicated at the bottom tube sheet of the air heater illustrated in Fig. 18. Tests made in actual operation with this type show negligible leakage as reflected in excess air or CO_2 determinations. For over-all tightness of the air heater it is preferable to weld the panel joints of the casing and hoppers. The undesirable effect of leakage on fan power is indicated by the graph in Fig. 19.

Soot blowers are located so that a jet of steam, air, or water in some instances (with the boiler out of service) can be blown through each tube as required (or across the tubes in the infrequently used arrangement where gas crosses the tubes on the outside).

Ample flow paths are provided for the gases, and, where space permits, dust hoppers can be fitted ahead of the heaters. The net result is a material reduction in the amount of dust and fly-ash entering the heaters.

Tube Sizes. In building the first tubular air heaters, tube diameters of 2 in., 2½ in., and even 3 in. and 4 in. were used in various units. Several factors are involved in establishing the size of air heater tubes. With smaller-diameter tubes the surface is used more effectively, but tubes of larger diameter are easier to clean. Greater space is required for the large-tube heater. The over-all cost of the heater, including the manufacture, shipment, and erection of the tubes, casing, and supports, and building costs, tends to be a minimum for some particular size of tube. In stationary boiler practice, the 2½-in. and 2-in. tubes

are now generally used, and in marine practice, where space and weight are limited, the 1½-in. tubes prove to be more economical.

Adaptability for Installation. Adaptability of the air heater to space limitations and to the arrangement of ducts between the heater and the boiler is an important factor in the over-all design of a steam-generating unit. The tubular air heater, as indicated in Fig. 20, can be arranged to suit a wide variety of space conditions and duct layouts. It can be adapted for vertical or horizontal applications and for multipass flow of either gas or air, or both. Where the forced-draft fan is located at a distance from the boiler, the tubular air heater has frequently been used as a flue, thus saving the expense of additional ductwork.

Control of Corrosion. Experience indicates that corrosion is liable to occur in an air heater when its metal temperature falls below certain limits. Sulfur content of the fuel, moisture content of the gases, and type of firing are all important factors to be considered. However, a definite relationship between the minimum metal temperature on the one hand and the gas and air temperatures, the fuel, and the design of the air heater on the other has been established from operating data and engineering tests. It is possible to obtain an optimum design of tubular air heater for every application, with the added assurance of effective control of corrosion, by applying one or more of the following methods:

1. By-pass a portion, or all, of the cool inlet air to increase the metal temperature. This is especially effective at light loads when the gas temperatures are low and the danger of dew point is imminent.
2. Increase the spacing of the first few rows of tubes at the air inlet side of the air heater to decrease the cooling effect (by decreasing mass flow) of the incoming air and thus keep the metal temperature in this zone above the dew point.
3. Recirculate a portion of the hot air from the air heater outlet back to the air heater inlet and thus keep the metal temperature at the cold end of the air heater constantly above the dew point.
4. Use parallel flow in the air heater, often recommended for stoker-fired units and for chemical recovery units in the paper industry, and thus avoid dew point complications (see parallel-flow arrangement in Fig. 20).

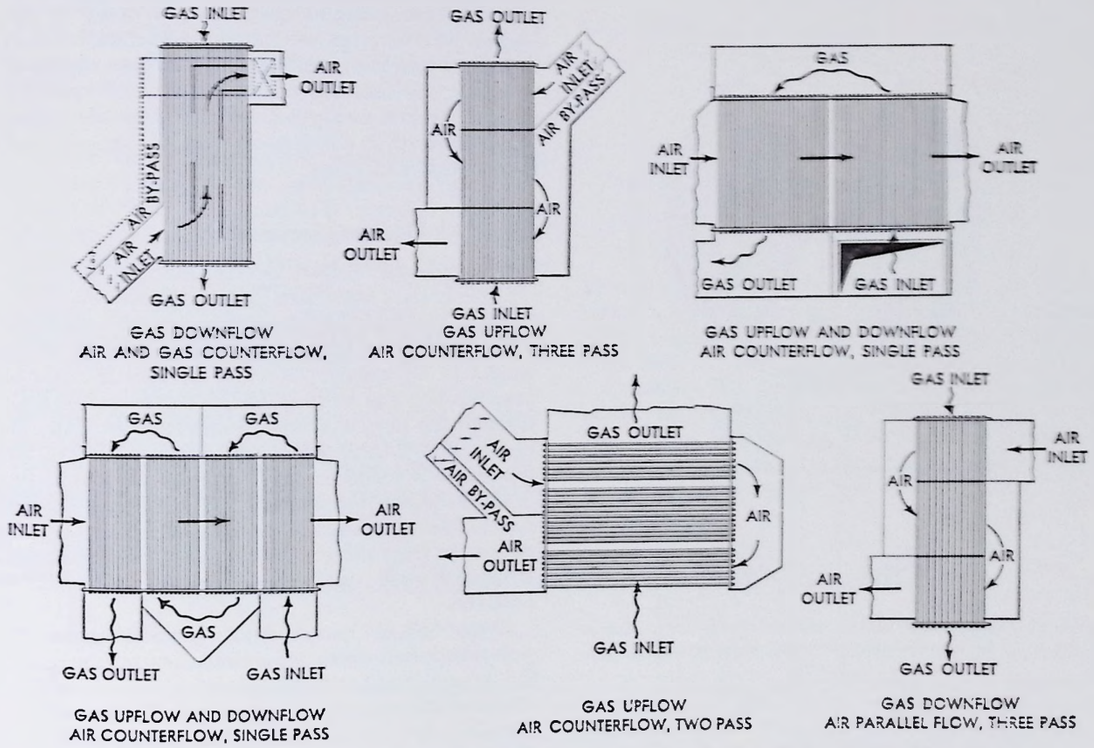


Fig. 20. *Some arrangements of tubular air heaters to suit various directions of gas and air flow*

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Primary-Air Heating. Occasionally, for pulverized-coal-fired units it is desirable to have a separate section in the air heater for heating the primary air to the pulverizers, particularly when the temperature leaving the main heater is lower than that required for drying very wet coal, such as a Midwest coal with more than 10 to 12% surface moisture. The primary-air heating section is then designed to give the required air temperature. This arrangement has the additional advantage of permitting use of a smaller forced-draft fan for the secondary air, since a single fan handling clean cold air is used for all the pulverizers of a unit. However, because of complications in the air heater design, a separate primary heater is seldom recommended when the air temperature leaving the main heater is high enough for the pulverizers.

Rotary Regenerative Air Heater

The principle of using steel plates as the best storage medium in a regenerative air heater was first developed and applied in Europe, and the first rotary heater of this type was installed in the United States in 1923. The design has been greatly modified and improved to meet the ever-changing requirements of the power industry. This type and the tubular (recuperative) type are now the two principal air heater designs used in the generation of steam.

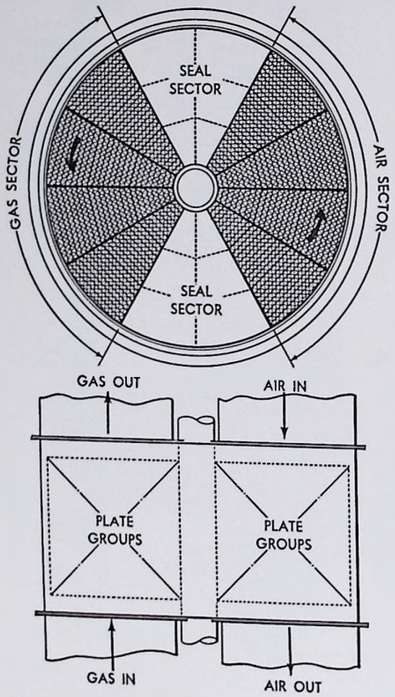


Fig. 21. *Diagrammatic illustration of rotary regenerative air heater (vertical shaft arrangement) with gas and air counterflow*

The rotary regenerative air heater is illustrated diagrammatically in Fig. 21. Slightly separated metal plates, supported in a frame attached to a slowly moving rotor shaft, are arranged edge-on to the gas and air flow. As these plates pass progressively through the gas stream they are heated and then, in passing through the air stream, give up heat to the air before again entering the gas stream, thus maintaining the regenerative cycle. The plates are placed in segmental groupings to fill a complete circle around the shaft. Seals are provided to reduce air infiltration into the gas. Gas and air ducts and the casing around the heater confine the gas and air to the desired flow paths.

Soot blowers are used to clean the heating elements of ash and oil residues. During low-load operation, when deposits tend to increase because of low metal temperatures, the heater can be bypassed on the air side or the air recirculated.

The rotary regenerative air heater is made in a number of standard diameters and in various lengths of elements to obtain the desired gas and air temperatures for different gas and air weights. Air heaters of this type are made with vertical (as illustrated) or horizontal shaft rotors. The choice of one over the other depends to some extent upon the gas and air duct arrangement to and from the heater. Some think that the heater with a vertical shaft rotor is easier to clean.

Other Air Heaters (Recuperative)

Plate-Type Air Heater. Several designs of plate-type air heaters have been developed over the years. They all use thin, flat, parallel plates with alternate wide and narrow spacing to match the ratio of gas weight to air weight. Thus the flue gas is made to pass through the wide-spaced passages and the air through the narrower with countercurrent flow. To obtain more heating surface for a given volume, the gas space is made as small as possible within the limits required for cleaning. Commercial designs have differed chiefly in the method of sealing the air spaces from the gas spaces. Many patented constructions were used but only the welded joint survived.

Steam Coil Air Heater. After the development of pulverized fuel there was some demand to apply this type of firing to existing boilers. This application required hot air, and it was not always possible to install a flue gas air heater. In such cases commercially available steam-to-air heat exchangers were successfully adapted to supply the heat to dry and burn the fuel.

To reduce fouling and corrosion, steam coil heaters are used occasionally today to increase the minimum tube-metal temperatures by heating the combustion air before it enters the air heater. Other means, as cited above, are generally more economical for the control of metal temperature.

Separately Fired Air Heater. Hot air is sometimes required for industrial use. Air at atmospheric or higher pressure may be heated to the desired temperature by using a separately fired air heater. The separately fired air heater usually consists of a refractory furnace with a tubular heater arranged for a number of gas and air passes.

To prevent overheating of the tubes, it is essential to control the gas temperatures entering the heater. This may be done by (a) diluting the gas with excess air or by (b) recirculating the gas from the air heater outlet to the furnace, thus reducing the temperature of the furnace gases. The unit will be more efficient if the latter method is used.

Other Air Heaters (Regenerative)

Diphenyl Air Heater. Occasionally use is made of an air heater operating on the regenerative principle where the intermediate heat-conducting mechanism is diphenyl oxide or a similar suitable liquid. The liquid is heated as it passes through a heat exchanger located in the flue gas stream and cooled as it passes through another heat exchanger located in the air stream. By continuous recirculation of the liquid, heat is transferred from the gas to the air.

Pebble Heater. Regenerative air heaters have been designed to heat air and other gases to higher temperatures than is economically possible using the recuperative principle. Arrangement of a pebble heater, designed for heating air from room temperature to 2300 F, is illustrated by Fig. 22. Various suitable materials may be used for the pebble heat-transfer medium. Spherical pebbles, ½ in. in diameter, of mullite composition (72% Al_2O_3, 28% SiO_2), are commonly used as the heat-conducting medium. The pebbles are alternately heated in the upper chamber, A, by hot gases and cooled in the lower chamber, C, by the fluid to be heated. Close control of the pressures of the gas and of the fluid to be heated is necessary to prevent mixing of the two at B. The whole unit is surrounded by a gastight cylindrical steel casing which forms a rigid support for the brickwork.

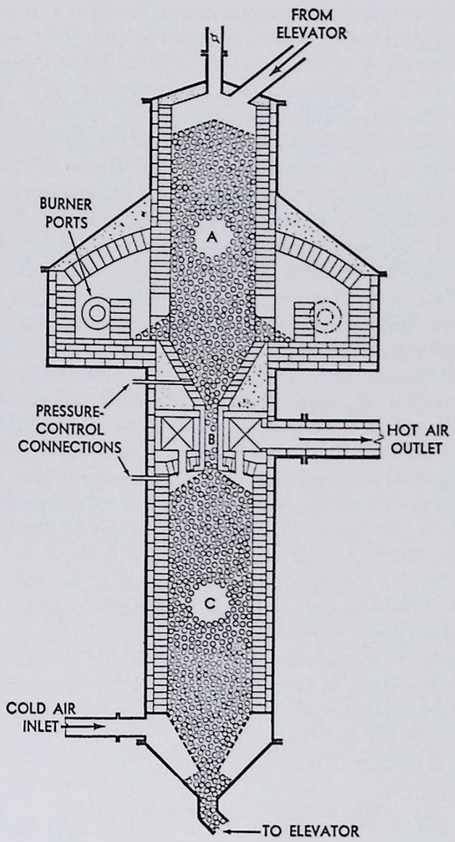


Fig. 22. *Arrangement of pebble heater for delivery of air at 2300 F*

In addition to heating air or steam, the pebble heater has many other applications among which are: heating hydrogen or natural gas for the reduction of metallic oxides, and the production of ethylene from a hydrocarbon feed source.

Temperature, Surface, and Draft-Loss Calculations

EVALUATING the performance of a boiler unit for a given set of conditions is fairly complicated, as indicated in the foregoing. One of the principal problems in boiler design is to attain an optimum interrelationship among three main factors: gas temperature drop, surface, and draft loss. Since surface may be disposed in an infinite variety of arrangements, each affecting the value of temperature drop and draft loss, no single useful equation combining the three has been developed. A practical approach, based in large measure on experience, is therefore necessary in establishing economic surface arrangements for specified conditions.

Thus, in the convection surface of a boiler unit,

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tubes are disposed in banks for the boiler portion, the superheater, the economizer, and the tubular air heater. A rather narrow range of tube diameters has been established as good practice for these components. From experience, tube spacing (side-spacing and backspacing) has also been established within a fairly narrow range. All these details of surface and its arrangement reflect experience based on practical performance. The range of variation in detail has been demonstrated as acceptable in draft loss, use of material, space occupied, and tube size and spacing to maintain cleanliness on the gas side when firing fuels liable to deposit dust on heat-absorbing surfaces.

However, even within the finite limits of variety of surface arrangements accepted as good practice, there are still too many possible permutations and combinations to permit the use of surface in an all-comprehensive formula. The problem must therefore be approached in two steps: 1) to establish for a certain gas mass flow, by trial and error computations, the true relationship between surface and the temperature drop or temperature rise desired, and 2) to establish the draft loss (pressure drop) for this gas mass flow over this surface.

For clean surfaces the relationship of gas temperature drop of the hotter fluid (HF), temperature rise of the colder fluid (CF), and surface (*S*) is given in the general equation:

$$U S \triangle t_m = w c (t_1 - t_2) = w' c' (t'_2 - t'_1) \qquad (1)$$

where:

U = combined conductance, Btu/sq ft, hr, F

$U = \dfrac{U_{HF} \times U_{CF}}{U_{HF} + U_{CF}}$, Btu/sq ft, hr, F

S = total surface, sq ft

$\triangle t_m$ = log mean temp diff, hotter fluid and colder fluid, F

w = wt flow of hotter fluid, lb/hr

t_1 = temp of hotter fluid entering tube bank, F

t_2 = temp of hotter fluid leaving tube bank, F

c = mean sp ht of hotter fluid, Btu/lb, F

w' = wt flow of colder fluid, lb/hr

t'_1 = temp of colder fluid entering tube bank, F

t'_2 = temp of colder fluid leaving tube bank, F

c' = mean sp ht of colder fluid, Btu/lb, F

For boilers, economizers, superheaters, and tubular air heaters, the hotter fluid is gas and the colder fluid is, respectively, saturated steam-water mixture at constant temperature, water at rising temperature, steam at rising temperature, and air at rising temperature. For clean surfaces equation (1) may be expressed as:

For Boiler Surface

$$U S \triangle t_m = w_g c_g \triangle t_g \qquad (2)$$

where:

$U = U_g$ = combined conductance, Btu/sq ft, hr, F

$U_g = U_{rg} + U_{cg}$ = over-all gas-side conductance, Btu/sq ft, hr, F

U_{rg} = radiation conductance (gas side), Btu/sq ft, hr, F

U_{cg} = convection conductance (gas film), Btu/sq ft, hr, F

S = total surface, sq ft

$\triangle t_m$ = log mean temp diff, gas and sat. water, F

$\triangle t_m = \dfrac{t_1 - t_2}{\log_e\left(\dfrac{t_1 - t'_2}{t_2 - t'_1}\right)}$, (Case I, Fig. 23)

t_1 = gas temp entering tube bank, F

t_2 = gas temp leaving tube bank, F

t'_2 = sat. temp of boiler water, F

w_g = wt flow of gas, lb/hr

c_g = mean sp ht of gas, Btu/lb, F

$\triangle t_g = t_1 - t_2$ = gas temp diff, entering and leaving, F

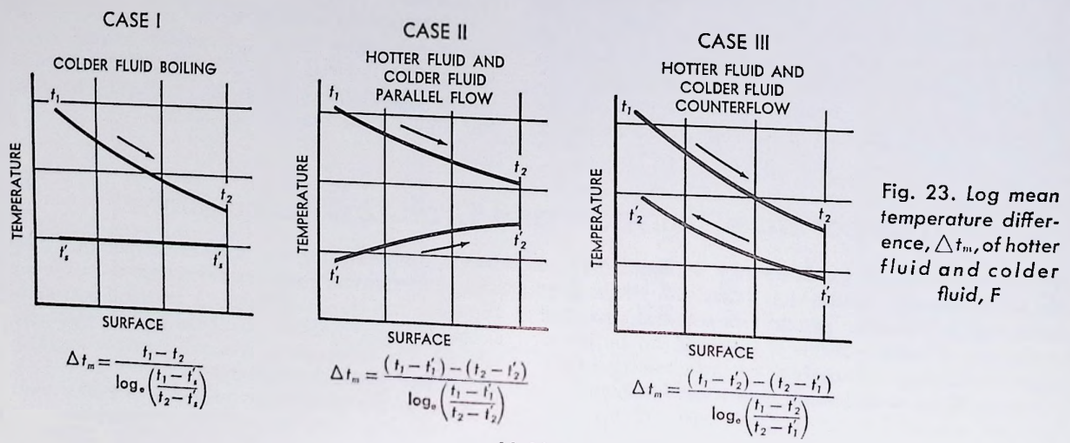


Fig. 23. *Log mean temperature difference, $\triangle t_m$, of hotter fluid and colder fluid, F*

For Economizer Surface

$$U S \triangle t_m = w_g c_g \triangle t_g \qquad (2)$$

where:

$U, S, w_g, c_g, \triangle t_g$ are as defined for boiler surface

$\triangle t_m$ = log mean temp diff, gas and water, F

For parallel flow, Case II, Fig. 23

$$\triangle t_m = \frac{(t_1 - t'_1) - (t_2 - t'_2)}{\log_e\left(\dfrac{t_1 - t'_1}{t_2 - t'_2}\right)}$$

For counterflow, Case III, Fig. 23

$$\triangle t_m = \frac{(t_1 - t'_2) - (t_2 - t'_1)}{\log_e\left(\dfrac{t_1 - t'_2}{t_2 - t'_1}\right)}$$

t_1 and t_2 are as defined for boiler surface

t'_1 = temp of water entering econ, F

t'_2 = temp of water leaving econ, F

Btu Balance for Economizer

$$w_g c_g \triangle t_g = w_w c_w \triangle t_w \qquad (3)$$

where:

$w_g, c_g, \triangle t_g$ are as in equation (2)

w_w = wt flow of water through econ, lb/hr

c_w = mean sp ht of water in econ, Btu/lb, F

$\triangle t_w = t'_2 - t'_1$, temp rise of water through econ, F

For Superheater Surface

$$U S \triangle t_m = w_g c_g \triangle t_g \qquad (2)$$

where:

$S, w_g, c_g, \triangle t_g$ are as defined for boiler surface

$U = \dfrac{U_g \times U_s}{U_g + U_s}$, combined conductance, Btu/sq ft, hr, F

U_g is as defined for boiler surface

U_s = convection conductance (steam film), Btu/sq ft, hr, F

$\triangle t_m$ = log mean temp diff, gas and steam, F

For parallel flow, Case II, Fig. 23

$$\triangle t_m = \frac{(t_1 - t'_1) - (t_2 - t'_2)}{\log_e\left(\dfrac{t_1 - t'_1}{t_2 - t'_2}\right)}$$

For counterflow, Case III, Fig. 23

$$\triangle t_m = \frac{(t_1 - t'_2) - (t_2 - t'_1)}{\log_e\left(\dfrac{t_1 - t'_2}{t_2 - t'_1}\right)}$$

t_1 and t_2 are as defined for boiler surface

t'_1 = temp of steam entering superheater, F

t'_2 = temp of steam leaving superheater, F

Btu Balance for Superheater

$$w_g c_g \triangle t_g = w_s c_s \triangle t_s \qquad (4)$$

where:

$w_g, c_g, \triangle t_g$ are as in equation (2)

w_s = wt flow of steam through superheater, lb/hr

c_s = mean sp ht of steam in superheater, Btu/lb, F

$\triangle t_s = t'_2 - t'_1$, temp rise of steam through superheater, F

The surface of a superheater, determined from the above, will be reduced somewhat (as shown in the superheater example below) when account is taken of the heat radiating from the mass of gas *approaching* across a cavity or from a furnace, where the bank can "see" the gas. Because of the relatively low temperature involved in this example, no account is taken of the radiation to the *rear* of the bank by the mass of gas leaving.

For Air Heater Surface

$$U S \triangle t_m = w_g c_g \triangle t_g \qquad (2)$$

where:

$S, w_g, c_g, \triangle t_g$ are as defined for boiler surface

$U = \dfrac{U_g \times U_{ca}}{U_g + U_{ca}}$, combined conductance, Btu/sq ft, hr, F

U_g is as defined for boiler surface

U_{ca} = convection conductance (air film), Btu/sq ft, hr, F

$\triangle t_m$ = log mean temp diff, gas and air, F

For parallel flow, Case II, Fig. 23

$$\triangle t_m = \frac{(t_1 - t'_1) - (t_2 - t'_2)}{\log_e\left(\dfrac{t_1 - t'_1}{t_2 - t'_2}\right)}$$

For counterflow, Case III, Fig. 23

$$\triangle t_m = \frac{(t_1 - t'_2) - (t_2 - t'_1)}{\log_e\left(\dfrac{t_1 - t'_2}{t_2 - t'_1}\right)}$$

t_1 and t_2 are as defined for boiler surface

t'_1 = temp of air entering air heater, F

t'_2 = temp of air leaving air heater, F

Btu Balance for Air Heater

$$w_g c_g \triangle t_g = w_a c_a \triangle t_a \qquad (5)$$

where:

$w_g, c_g, \triangle t_g$ are as in equation (2)

w_a = wt flow of air through air heater, lb/hr

c_a = mean sp ht of air in air heater, Btu/lb, F

$\triangle t_a = t'_2 - t'_1$, temp rise of air through air heater, F

Mass Flow set by Good Practice

From a long record of experience it is known that for a given set of conditions, such as fuel (amount to be burned and characteristics) and allowable overall draft loss, surface (boiler, superheater, economizer, or air heater) is arranged as tubes of a suitable diameter disposed in a bank and spaced (in both directions) within a fairly limited range. Tube length is generally determined by the drum or header locations. In heat transfer and fluid flow problems it is essential to know the limits set by good practice for mass velocity, or "mass flow" ($G = w/A$). With tube diameter, spacing, and length established, the width of the tube bank is based on an acceptable mass flow of the fluids (gas, steam, water, or air) entering, in which the free-flow area is a factor, since:

$G = \frac{w}{A}$ = mass flow of fluid flowing, lb/sq ft, hr

where:

w = weight of fluid flowing, lb/hr (w_g for gas)
A = free-flow area, sq ft (A_g for gas)

The "depth" of bank, or the number of rows deep, is frequently limited by the usable circumference of the drums or headers to which the tubes are connected. Other considerations, such as the maximum distance for effective cleaning by soot blowing, will sometimes limit the depth of bank. In any case, since it is necessary to start with a tube arrangement, experience will permit a tentative selection of the height and width of the receiving surface and the tube diameter of this bank for a given mass flow of the entering gases. Having thus selected a specific arrangement of surface, it is possible to calculate the total amount of surface required for the desired temperature drop by the use of the general equation (2).

Temperature Drop, Boiler Surface

As an example of a manipulation of equation (2) for the relation between gas temperature drop and surface, a simple case of boiler surface used to cool combustion gases will do. Equation (2) may be rewritten:

$$S = \frac{w_g c_g \triangle t_g}{U \triangle t_m} \qquad (6)$$

Total surface, S, in tube bank, sq ft — to be determined
Depth of tube bank, number of rows — to be determined

To determine the total surface, S, required to cool the gases to the desired temperature, it is necessary, as noted above, to set certain conditions, known from experience to be fairly applicable to the case selected. For this example the known and assumed conditions are:

Fuel = natural gas, 1000 Btu/cu ft (22,000 Btu/lb)
w_g = 185,000 lb/hr, combustion gas (from burning fuel with 115% total air; moisture in flue gas, 11%)
$\triangle t_g = t_1 - t_2 = 610$ F, desired gas temp drop
$t_1 = 1910$ F, temp of gas entering tube bank
$t_2 = 1300$ F, temp of gas leaving tube bank
$t'_s = 378$ F, sat. temp of boiler water @ 176.4 psi
$\triangle t_m$ = log mean temp diff, gas and sat. water, F (Case I, Fig. 23)

Tubes and arrangement in bank:
2.5-in. OD, in line (d_o)
Length = 16 ft
Arranged in setting 99 in. wide (including side lanes)
Avg sidespacing, $l_l = 3.5$ in.
Avg backspacing, $l_{ll} = 4.0$ in.
No. in one row across pass = 28
Surface of one row of tubes = 293 sq ft

Calculations to determine surface, S, using equation (6):

Sum of tube diam = $28 \times 2.5 = 70$ in.
A_g = gas free-flow area through tube bank

$$= \frac{99 - 70}{12} \times 16 = 38.7 \text{ sq ft}$$

$$G_g = \frac{w_g}{A_g} = \frac{185{,}000}{38.7} = 4780 \text{ lb/sq ft, hr}$$

$$\triangle t_m = \frac{1910 - 1300}{\log_e\left(\frac{1910 - 378}{1300 - 378}\right)} = 1200 \text{ F}$$

$c_g = 0.318$ Btu/lb, F, from Fig. 24, for 11% moisture
$U = 12.80$ Btu/sq ft, hr, F (established as explained later under *Solution of Values*, page 11-32 ff)

Substituting the above values in equation (6):

$$S = \frac{185{,}000 \times 0.318 \times 610}{12.80 \times 1200} = 2337$$

$$\text{No. of rows of tubes in bank} = \frac{2337}{293} = 7.98$$

From the calculated value of surface, S, it is evident that, for the assumed conditions of the above example, 8 rows of tubes will be required in the tube bank to give the desired temperature drop.

When the calculated surface differs materially from the initially assumed surface, the values of U_{cg} and U_{rg} must be appropriately adjusted, and the resulting new value of U must be used in a final recalculation of surface.

In the simple illustration of the method used in establishing surface for a desired temperature drop, the conditions assumed in the above example were: crossflow of the gas, in-line tubes, and natural gas as fuel. However, the gas flow may be longitudinal or partly longitudinal and partly crossflow, and the tubes may be arranged in a staggered pattern. A coverage of all possible conditions will not be attempted. The value of the combined conductance, U, can be affected if other fuels are used. The effect on conductance of some ash-carrying fuels is largely a matter of the designer's judgment (see *Absorbing Surface Cleanliness*, page 11-32).

Draft Loss

The draft loss for gas in crossflow through a bank of in-line tubes at sea level, and where there is no stack effect, may be determined from the following equation, partly empirical:

$$\triangle p = f N F_d \frac{460 + 0.95\,(t_1 + t_2)/2}{57{,}500} (G_g \times 10^{-3})^2 \qquad (7)$$

where:

$\triangle p$ = draft differential (draft loss), in. of water
f = friction factor, from Fig. 40, for established Reynolds number, tube diam, and tube spacings
N = number of tube rows in the bank
F_d = draft loss depth factor, from Fig. 20, Chapter 8
t_1 = gas temp entering bank, F
t_2 = gas temp leaving bank, F
$0.95\,(t_1 + t_2)/2$ = approx avg gas temp, F
$G_g = w_g/A_g$ = gas mass flow, lb/sq ft, hr

For each 1000 ft elevation above sea level, the value of $\triangle p$ should be increased about 4% (see Chapters 5 and 8).

With vertical gas flow, the value of $\triangle p$ will be increased by stack effect when the outlet is below the inlet and decreased when the outlet is above the inlet (see Chapter 5). However, in the usual boiler tube banks the total effect is small, in the order of 0.005 to 0.010 in. of water per ft of tube bank height.

For the simple example of boiler tube bank and the conditions noted above in establishing surface for temperature drop, the draft loss may be determined by substituting the known values in equation (7):

$$\triangle p = 0.10 \times 8 \times 1.004 \frac{460 + 1525}{57{,}500} (4.78)^2$$

$$= 0.63 \text{ in. of water}$$

Economizer Surface

The surface of a nonsteaming economizer for (a) desired gas temperature drop, $\triangle t_g$, or for (b) desired water temperature rise, $\triangle t_w$, can also be determined from equation (2) with $\triangle t_m$, however, established from the formula in Case II or Case III of Fig. 23, depending upon whether the arrangement is for parallel flow or counterflow.

A general determination of economizer surface, without a specific example, may be derived from equations (2) and (3) listed above and repeated here for convenience:

$$w_g c_g \triangle t_g = U S \triangle t_m \qquad (2)$$

$$w_g c_g \triangle t_g = w_w c_w \triangle t_w, \text{ Btu balance} \qquad (3)$$

where:

w_g, c_g, $\triangle t_g$ are as in equation (2)
w_w = wt flow of water through econ, lb/hr, known
c_w = mean sp ht of water in econ, Btu/lb, F
$\triangle t_w = t'_2 - t'_1$, temp rise of water through economizer, F
t'_1 = temp of water entering econ, F, known
t'_2 = temp of water leaving econ, F, not known for condition (a), known for condition (b)

For condition (a), $\triangle t_m$ may be determined after establishing the value of t'_2 by solving equation (3) for $\triangle t_w$, since $t'_2 = \triangle t_w + t'_1$. Knowing the value of all temperatures involved, $\triangle t_m$ may then be established (see Fig. 23).

For condition (a), $\triangle t_m$ may be determined after equation (3), and so t_2, the leaving gas temperature, is determined. Since the entering gas temperature, t_1, is known, a trial c_g may be established from Fig. 24 by assuming a temperature for the leaving gas. The true values of c_g and $\triangle t_g$ are determined by successive trials until equation (3) is in balance. Knowing the value of all temperatures involved, $\triangle t_m$ may then be established (see Fig. 23).

After tube diameter, arrangement of surface, and tube bank dimensions suitable for economizer practice have been tentatively set and knowing $\triangle t_m$, the heating surface, S, may be determined from equation (2), for either of the two conditions (a) or (b), by establishing the value of U in the same manner as in the preceding example for boiler surface.

The draft loss across an economizer is calculated from equation (7), as in the example for boiler surface. The feedwater pressure drop through an economizer may be calculated in the same manner (where water is the fluid) as for the steam pressure drop through a superheater (see below).

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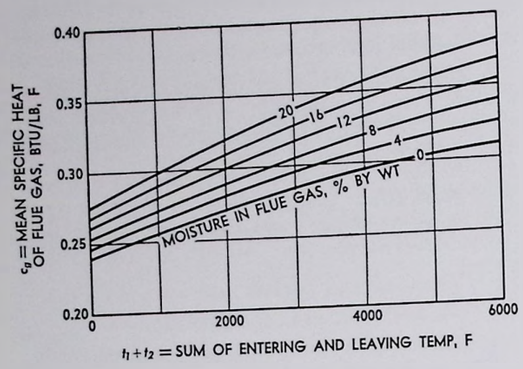


Fig. 24. *Approximate mean specific heat, c_g, of flue gases*

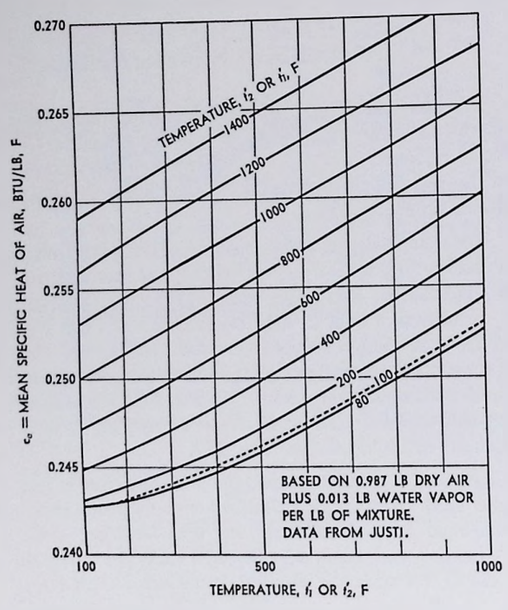


Fig. 25. *Mean specific heat of air, c_a, between any two temperatures, t_1' and t_2', at a constant pressure of one atmosphere*

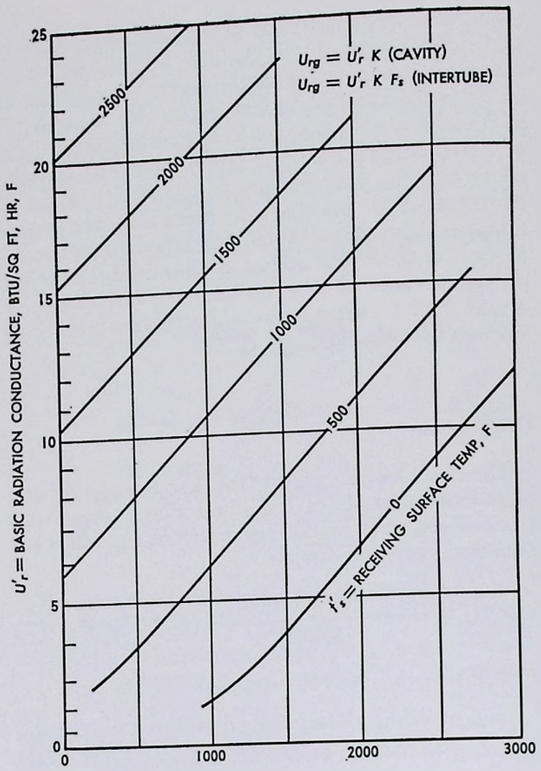


Fig. 26. *Basic radiation conductance, U'_r, as related to log mean temperature difference, Δt_m, and receiving surface temperature, t'_s*

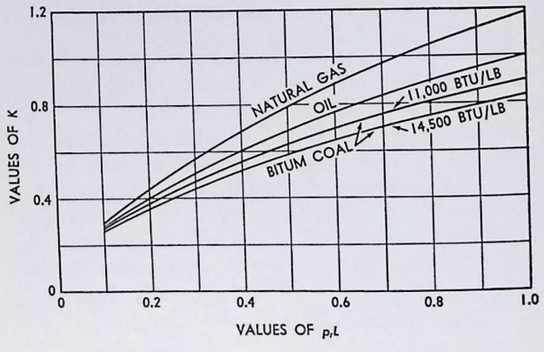


Fig. 27. *Effect of fuel and mean radiating length on radiation conductance*

Superheater Surface

In the relationship of superheater surface, steam temperature rise, and gas temperature drop, the combined conductance, U, in equation (2) is modified by the introduction of a value for steam film conductance as noted above.

From equation (2), total surface for a superheater may be expressed as:

$$S = \frac{w_g c_g \Delta t_g}{\frac{U_g \times U_s}{U_g + U_s} \Delta t_m}, \text{ sq ft} \qquad (8)$$

where:

$\frac{U_g \times U_s}{U_g + U_s} = U$, Btu/sq ft, hr, F

U_g is as defined for boiler surface

U_s = convection conductance (steam film), Btu/sq ft, hr, F

All other terms are as in equation (2)

As noted above and repeated here for convenience, the heat balance equation for a superheater may be expressed as:

$$w_g c_g \Delta t_g = w_s c_s \Delta t_s \qquad (4)$$

where:

w_g, c_g, Δt_g are as in equation (2) or (8)

w_s = wt flow of steam through superheater, lb/hr

c_s = mean sp ht of steam through superheater, Btu/lb, F

$\Delta t_s = t'_s - t'_1$, temp rise of steam through superheater, F

t'_1 = temp of steam entering superheater, F

t'_s = temp of steam leaving superheater, F

For a desired steam temperature rise, Δt_s, it is possible, after finding the values of Δt_g from equation (4), U_g, and U_s, to determine the surface, S, from equation (8) with Δt_m established from the formula in Case II or Case III of Fig. 23, depending upon whether the arrangement is for parallel flow or counterflow.

The surface so calculated, however, will be reduced somewhat (as shown later) when account is taken of cavity radiation from the mass of gas approaching the bank.

As an example of the manipulation of equations (8) and (4) for the relation of steam temperature rise and surface in a superheater, without for the moment considering the cavity radiation, the following conditions have been selected:

A draining-type convection superheater, horizontal continuous-tube, in-line, with gas flow upward through the bank and the steam counterflow.

Fuel = natural gas (as in boiler surface example)

w_g = 185,000 lb/hr, combustion gas (as in boiler surface example)

$\Delta t_g = t_1 - t_2$, gas temp drop, F

t_1 = 1275 F, temp of gas entering superheater

t_2 = temp of gas leaving superheater, F, not known

w_s = 150,000 lb/hr, steam to superheater

$\Delta t_s = t'_s - t'_1 = 200$ F, desired temp rise of steam through superheater

t'_1 = 378 F, temp of entering steam, sat. @ 176.4 psi

t'_s = 578 F, temp of leaving steam

$t'_s = \frac{378 + 578}{2} = 478$ F, avg steam temp

Δt_m = log mean temp diff, gas and steam, F (Case III, Fig. 23)

Tubing and arrangement in bank:

2.5-in. OD, in line (d_o)

2.09-in. ID, for wall thickness to permit bending on 1¾-in. radius (d_i)

Length = 12 ft in each horizontal run (12-ft span between supports)

Arranged in setting 99 in. wide

Avg sidespacing, l_1 = 3.5 in.

Avg backspacing, l_{11} = 3.5 in.

No. of sections across pass = 28

Surface of one row of tubes = 293 sq ft

Calculations to determine surface, S, using equation (8):

Sum of tube diam = $28 \times 2.5 = 70$ in.

A_g = gas free-flow area through tube bank

$= \frac{99 - 70}{12} \times 12 = 29$ sq ft

A_s = steam free-flow area through tubes

$= 28 \times \frac{(2.09)^2 \times 3.1416}{4 \times 144} = 0.668$ sq ft

$G_g = \frac{w_g}{A_g} = \frac{185{,}000}{29} = 6380$ lb/sq ft, hr

$G_s = \frac{w_s}{A_s} = \frac{150{,}000}{0.668} = 224{,}700$ lb/sq ft, hr

Equation (4) is used to establish Δt_g in order to find the value of t_s. The value of c_s, the mean specific heat of steam through superheater, is 0.57 from Fig. 6, Chapter 7°. A trial value of c_g may be used, and thus, by successive trials until equation (4) is in balance, the true values of c_g and Δt_g can be determined.

Substituting the above values in equation (4):

$$\Delta t_g = \frac{150{,}000 \times 0.57 \times 200}{185{,}000 \times 0.30} = 308 \text{ F}$$

and $t_s = t_1 - 308 = 1275 - 308 = 967$ F

then by Case III, Fig. 23:

$$\Delta t_m = \frac{(1275 - 578) - (967 - 378)}{\log_e \left(\frac{1275 - 578}{967 - 378} \right)} = 642 \text{ F}$$

[a] The usual practice is to solve for c_s by dividing the enthalpy difference, obtained from steam tables, by the steam temperature rise.

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In the case of a superheater, since the temperature drop through the tube metal is usually small, the average surface temperature, t'_s, is approximated as being the average steam temperature. The gas film temperature then becomes:

$$t_f = t'_s + \Delta t_m/2 = 478 + 642/2 = 799 \text{ F}$$

Using the values established above, U_g can be determined by following the procedure given below for boiler surface under *Solution of Values*. For this superheater example:

$$U_g = U_{rg} + U_{cg} = 0.85 + 13.15$$
$$= 14.0 \text{ Btu/sq ft, hr, F}$$

In equation (8), $\dfrac{U_g \times U_s}{U_g + U_s} = U$

An empiric equation for determining U_s, the steam film conductance (see Fig. 3, Chapter 12), can be expressed as:

$$U_s = \frac{0.0281}{(d_i/12)^{0.2}} \times G_s^{0.8} \times c_p \times \mu^{0.2} \times \frac{d_i}{d_e}$$

where all values have been established except c_p, the instantaneous specific heat, and μ, the absolute viscosity, both at the average steam film temperature. For their combined value, see Fig. 4, Chapter 12. From the above equation $U_s = 183.7$ Btu/sq ft, hr, F.

The combined conductance, U, then becomes:

$$U = \frac{U_g \times U_s}{U_g + U_s} = \frac{14.0 \times 183.7}{14.0 + 183.7}$$
$$= 13.01 \text{ Btu/sq ft, hr, F}$$

Substituting in equation (8):

$$S = \frac{185{,}000 \times 0.30 \times 308}{13.01 \times 642} = 2048 \text{ sq ft, neglecting}$$

the effect of cavity radiation.

Cavity Radiation

The Btu/hr in the heat balance equation (4) represents the total amount necessary to obtain the desired steam temperature rise. Since part of this heat was already absorbed from the gas mass in the cavity before the superheater, the surface established is too large. This is corrected in the following manner.

The Btu/hr absorbed by the superheater from cavity radiation is the product of the radiation conductance, U_{rg}, the flat projected surface, S_p, of the superheater facing the cavity, and the mean radiating temperature difference, Δt_m, between the average gas temperature and the superheater surface temperature.

The cavity radiation conductance, U_{rg}, is the product of the basic radiation conductance, U'_r (from Fig. 26) and the factor, K (from Fig. 27). For the specific dimensions of the cavity, which in this instance is 12 ft long, 99 in. wide, and 2 ft deep, the factor K is 0.8. The flat projected surface, S_p, is $12 \times 99/12 = 99$ sq ft. For this example the average gas temperature in the cavity is taken as 1285 F, and, assuming that the average tube-surface temperature over the face of the superheater bank is the same as the outlet steam temperature, the mean radiating temperature difference is $1285 - 578 = 707$ F.

The superheater absorption from cavity radiation is:

$$\text{Btu/hr} = U_{rg} \times S_p \times \Delta t_m = U'_r \times K \times S_p \times \Delta t_m$$
$$= 5.4 \times 0.8 \times 99 \times 707 = 305{,}000$$

The total surface of the superheater, as calculated above by equation (8), is reduced by recalculation after subtracting this cavity absorption from the total amount required. The corrected surface, S, then becomes:

$$S = \frac{(185{,}000 \times 0.30 \times 308) - 305{,}000}{13.01 \times 642} = 2010 \text{ sq ft}$$

No. of rows of tubes in bank $= 2010/220 = 9.1$

The required number of rows deep may be specified as 9. In this instance a depth factor need not be considered, as the flow will be disturbed leaving a cavity of this depth (see Chapter 7).

It will be noted, for the assumed cavity dimensions and low mean radiating temperature, that the heat input to the superheater bank from cavity radiation is small, less than 2%. If the cavity is large, and particularly if the mean radiating temperature is high (over 1500 F), the proportion of heat input by cavity radiation to any heat-absorbing surface can be appreciable. A rigorous and complete procedure for calculating such heat input under all conditions will not be attempted, since the influence of cavity dimensions and shape and the effect of radiating temperatures are too involved for the scope of this book.

Steam Pressure Drop through Superheater

The steam pressure drop, Δp, lb/sq in., through the superheater usually is calculated in three parts, 1) entrance and exit loss, 2) straight-flow loss, and 3) loss at bends:

$$\Delta p = \Delta p_e + \Delta p_f + \Delta p_b, \text{ lb/sq in.} \qquad (9)$$

where:

1) $\Delta p_e = 1.5v\ (G_s \times 10^{-5})^2/12$
2) $\Delta p_f = fLv\ (G_s \times 10^{-5})^2/d_i$
3) $\Delta p_b = N_b v\ (G_s \times 10^{-5})^2/12$

where:

v = avg sp vol of steam, cu ft/lb, from steam tables
$G_s = w_s/A_s$, steam mass flow, lb/sq ft, hr
f = friction factor, from Figs. 8 and 9, Chapter 8, for Reynolds number, N_{Re}
L = dev length of tube in one circuit, ft
d_i = internal diam of tube, in.
N_b = total no. of velocity heads lost in bends, from Fig. 16, Chapter 8
N_{Re} = Reynolds number $= d_i G_s/12\mu$
μ = abs viscosity of steam, lb (mass)/ft, hr, from Fig. 12, Chapter 8

Substituting values for the superheater example above:

$$\Delta p = 1.5 \times 2.97 \times 5.05/12 +$$
$$0.0193 \times 108 \times 2.97 \times 5.05/2.09 +$$
$$8 \times 0.59 \times 2.97 \times 5.05/12$$
$$\Delta p = 1.88 + 14.96 + 5.90 = 22.74 \text{ psi}$$

Call steam pressure drop 23 lb/sq in.

Air Heater Surface

In the determination of the amount and disposition of surface for an air heater, cavity radiation is negligible because of the usually low temperature of both the gas and the air. With the frequent arrangement, gas passing through the inside of the tubes (air outside), intertube radiation—while of minor effect on the value of U, the combined conductance—should be included on the gas side. Convection conductance for longitudinal flow of the gas within the tubes must be considered.

Arrangement of the surface is based on good practice from experience. The mass flow of both gas and air should be kept within economical limits—usually from 5000 to 10,000 lb/sq ft, hr for longitudinal flow and from 3000 to 5000 lb/sq ft, hr for crossflow. Tube spacing usually may be closer than for a superheater, since only air passes across the outside of the tubes. As noted above in the general discussion, metal temperatures below the dew point should be avoided. Usually the most economical arrangement is with long tubes and multipass on the air side.

In the relationship of air heater surface, S, to air temperature rise and gas temperature drop, the combined conductance, U, in equation (2) is modified by the introduction of a value for the air film conductance, as noted above.

From equation (2), total surface for an air heater may be expressed as:

$$S = \frac{w_g c_g \Delta t_g}{\dfrac{U_g \times U_{ca}}{U_g + U_{ca}} \Delta t_m}, \text{ sq ft} \qquad (10)$$

where:

$\dfrac{U_g \times U_{ca}}{U_g + U_{ca}} = U$, Btu/sq ft, hr, F
U_g is as defined for boiler surface
U_{ca} = convection conductance (air film), Btu/sq ft, hr, F
All other terms are as in equation (2)

As noted above and repeated here for convenience, the heat balance equation for an air heater may be expressed as:

$$w_g c_g \Delta t_g = w_a c_a \Delta t_a \qquad (5)$$

where:

w_g, c_g, Δt_g are as in equation (2) or (10)
w_a = wt flow of air through air heater, lb/hr
c_a = mean sp ht of air through air heater, Btu/lb, F
$\Delta t_a = t'_2 - t'_1$, temp rise of air through air heater, F
t'_1 = temp of air entering air heater, F
t'_2 = temp of air leaving air heater, F

For a desired air temperature rise, Δt_a, and reasonable mass flow of gas and air, it is possible, after finding the value of Δt_g from equation (5) and the value of U, to determine the surface S, from equation (10), with Δt_m established from the formula in Case III, Fig. 23, for counterflow. The surface so determined must be modified by a correction factor, 1/0.85, to permit the use of the counterflow formula in finding the value of Δt_m for a single-pass crossflow air heater.

As an example of the manipulation of equations (10) and (5) for the relation of surface, air temperature rise, and gas temperature drop in a single-pass air heater, the following conditions have been selected:

Fuel = natural gas (as in boiler surface example)
w_g = 185,000 lb/hr, combustion gas (as in boiler surface example)
$\Delta t_g = t_1 - t_2$, gas temp drop, F
t_1 = 605 F, temp of gas entering air heater
t_2 = temp of gas leaving air heater, F, not known
w_a = 150,000 lb/hr, air to air heater
$\Delta t_a = t'_2 - t'_1 = 270$ F, desired temp rise of air through air heater
t'_1 = 80 F, temp of air entering air heater
t'_2 = 350 F, temp of air leaving air heater
Δt_m = log mean temp diff, gas and air, F (Case III, Fig. 23)

Tubing and arrangement in bank:
2-in. OD, in line (d_o)
1.834-in. ID for wall thickness of 0.083 in. (d_i)

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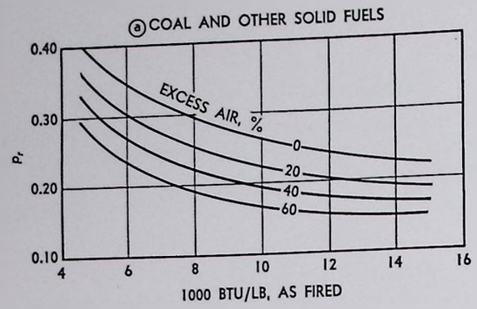


Fig. 28. *Partial pressure, p_r, of principal radiating constituents ($CO_2 + H_2O$) of combustion gases for various fuels, heat values, and excess airs*

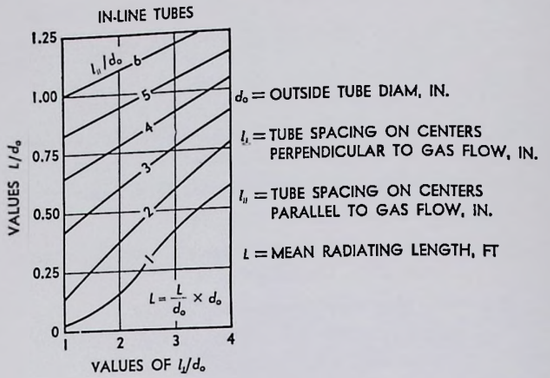


Fig. 29. *Mean radiating length, L, for various tube diameters and arrangements*

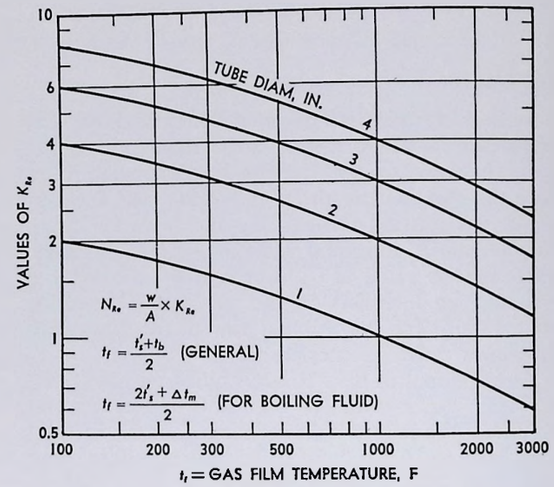


Fig. 30. *Determination of Reynolds number, N_{Re}, (using gas mass flow, w/A) for various tube diameters and gas film temperatures*

Inside cross-section area/tube = 0.01835 sq ft
Arranged in setting 99 in. wide
Avg sidespacing, $l_l = 3$ in.
Avg backspacing, $l_{ll} = 3$ in.
No. of tubes across pass = 33

Calculations to determine surface, S, using equation (10):

Assume:

Gas mass flow, $G_g = w_g/A_g = 8000$ lb/sq ft, hr
Air mass flow, $G_a = w_a/A_a = 4000$ lb/sq ft, hr

then:

$A_g = 185{,}000/8000 = 23.1$ sq ft
$A_a = 150{,}000/4000 = 37.5$ sq ft
No. of tubes = $23.1/0.01835 = 1259$
No. of tubes deep = $1259/33 = 38+$
Actual no. of tubes = $33 \times 38 = 1254$
Sum of tube diam = $33 \times 2 = 66$ in.

$$\text{Length of tube} = \frac{37.5 \times 12}{99 - 66} = 13.64 \text{ ft}$$

Actual total outside surface (air side),

$$S = \frac{2\pi}{12} \times 13.64 \times 1254 = 8960 \text{ sq ft}$$

Actual gas mass flow, $G_g = w_g/A_g$

$$= \frac{185{,}000}{0.01835 \times 1254} = 8040 \text{ lb/sq ft, hr}$$

Equation (5) is used to establish $\triangle t_g$ in order to find the value of t_2. The value of c_a, the mean specific heat of air through air heater, is 0.244, from Fig. 25. A trial value of c_g may be used, and thus, by successive trials until equation (5) is in balance, the true values of c_g and $\triangle t_g$ can be determined.

Substituting the above values in equation (5):

$$\triangle t_g = \frac{150{,}000 \times 0.244 \times 270}{185{,}000 \times 0.278} = 192 \text{ F}$$

and $t_2 = t_1 - 192 = 605 - 192 = 413$ F

then by Case III, Fig. 23:

$$\triangle t_m = \frac{(605—350) - (413—80)}{\log_e\left(\frac{605—350}{413—80}\right)} = 292 \text{ F}$$

In the case of an air heater, the gas and air film temperatures are determined as follows:

$$\text{Gas film temp} = \frac{t'_1 + t'_2}{2} + \frac{3\triangle t_m}{4}$$

$$= \frac{80 + 350}{2} + \frac{3 \times 292}{4} = 434 \text{ F}$$

$$\text{Air film temp} = \frac{t'_1 + t'_2}{2} + \frac{\triangle t_m}{4}$$

$$= \frac{80 + 350}{2} + \frac{292}{4} = 288 \text{ F}$$

Since heat transfer by radiation is so small a part of the total conductance, a value of U_{rg} equal to 0.20 Btu/sq ft, hr, F may be used without significant error in most air heater problems.

The gas-side convection conductance in long flow is calculated from: $U_{cg} = U'_{ct}\ F_{pp}\ F_T\ d_i/d_o$, where d_i/d_o converts the inside gas film conductance to the same outside-surface basis (see Chapter 7) as the air film conductance.

Using the values established above, U_g and U_a can be determined by following a procedure similar to that given later under *Solution of Values*, whence:

$U_g = U_{rg} + U_{cg} = 0.20 + 8.02 = 8.22$
$U_a = 9.0$

and the combined conductance, U, then becomes:

$$U = \frac{U_g \times U_a}{U_g + U_a} = \frac{8.22 \times 9.0}{8.22 + 9.0} = 4.29 \text{ Btu/sq ft, hr, F}$$

Substituting the values established above in equation (10) and including the 1/0.85 factor referred to for a single-pass heater:

$$S = \frac{185{,}000 \times 0.278 \times 192}{4.29 \times 292 \times 0.85} = 9280 \text{ sq ft}$$

required, compared with 8960 sq ft assumed in first trial.

Second Trial—Air Heater Surface

Number of tubes = 1254, same as before
Length of tubes = 14.5 ft, instead of 13.64 ft
Total surface = $8960 \times 14.5/13.64 = 9530$ sq ft
$G_a = w_a/A_a = 4000 \times 13.64/14.5 = 3760$
$U_g = 8.22$, same as before
$U_a = 8.6$
$U = 4.2$ Btu/sq ft, hr, F, combined conductance
$S = 9280 \times 4.29/4.2 = 9480$ sq ft required

This value compares closely with 9530 sq ft of surface assumed in the second trial, and the number, length, diameter, and spacing of tubes selected will satisfy the design requirements.

Draft Loss through Air Heater

The draft loss, $\triangle p$, in. of water, for the longitudinal flow of gas through air heater tubes at sea level, and where there is no stack effect, usually is calculated in two parts, 1) entrance and exit loss and 2) straight-flow loss:

$$\triangle p = \triangle p_e + \triangle p_f \text{, in. of water} \qquad (11)$$

where:

1) $\triangle p_e = 1.5 \dfrac{460 + (t_1 + 2t_2)/3}{172{,}000} (G_g \times 10^{-3})^2$

2) $\triangle p_f = \dfrac{fL}{d_i} \dfrac{460 + (t_1 + 2t_2)/3}{14{,}400} (G_g \times 10^{-3})^2$

where:

t_1 = gas temp entering air heater, F
t_2 = gas temp leaving air heater, F
$(t_1 + 2t_2)/3$ = approx avg gas temp, F
$G_g = w_g/A_g$ = gas mass flow, lb/sq ft, hr
f = friction factor, from Figs. 8 and 9, Chapter 8, for Reynolds number, N_{Re}

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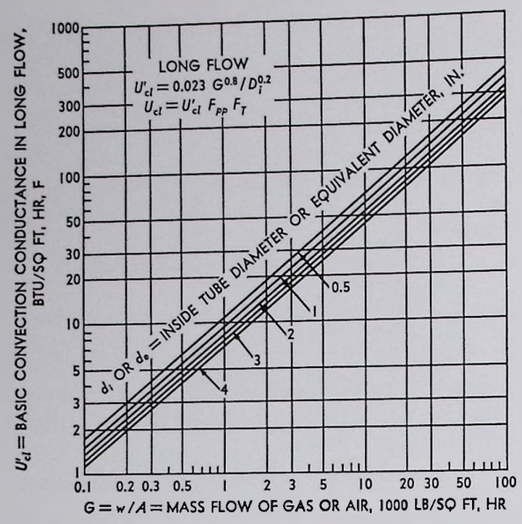


Fig. 31. *Basic long flow convection conductance, U'_{cl}, as related to gas or air mass flow, G, for various tube diameters*

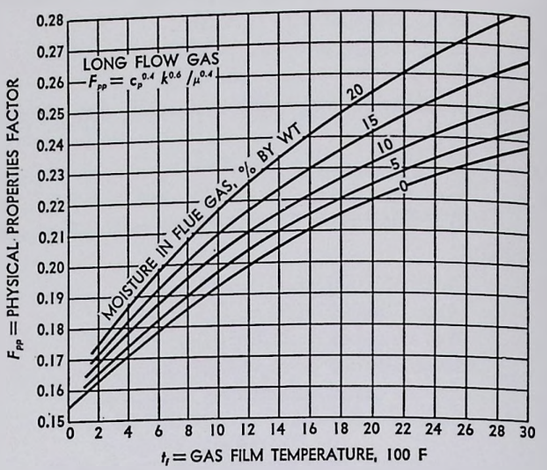


Fig. 32. *Effect of film temperature, t_f, on the physical properties factor, F_{pp}, for gas in long flow*

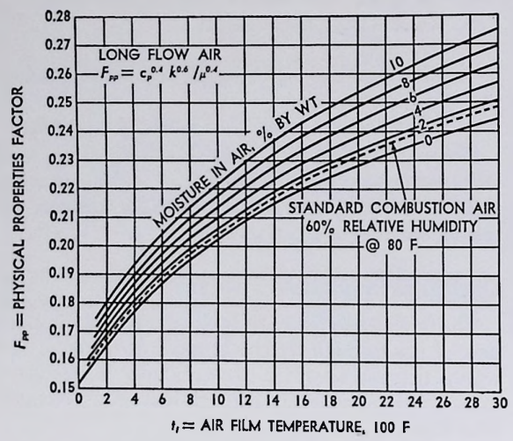


Fig. 33. *Effect of film temperature, t_f, on the physical properties factor, F_{pp}, for air in long flow*

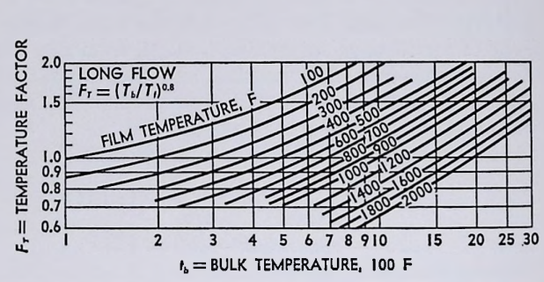


Fig. 34. *Temperature factor, F_T, for converting mass flow from bulk to film basis; long flow gas or air*

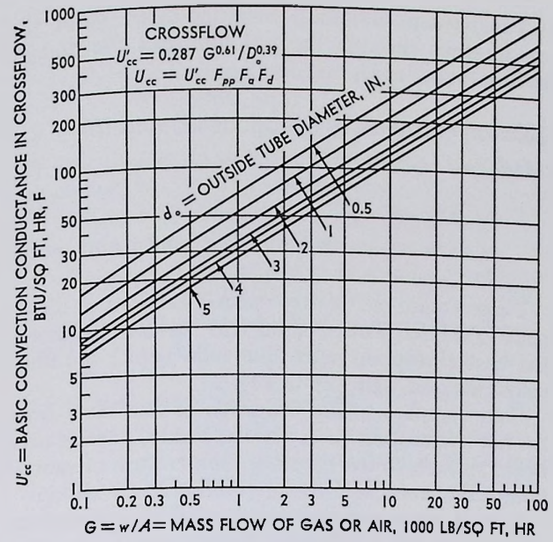


Fig. 35. *Basic crossflow convection conductance, U'_{cc}, as related to gas or air mass flow, G, for various tube diameters*

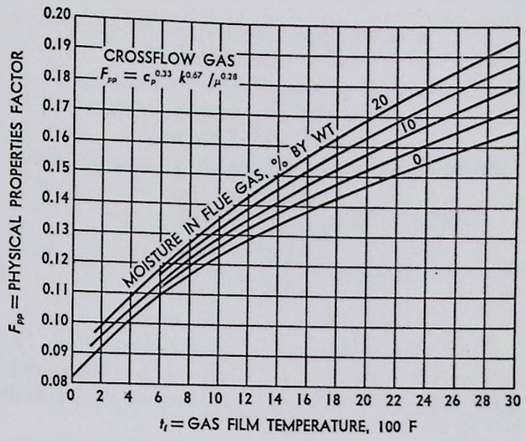


Fig. 36. *Effect of film temperature, t_f, on the physical properties factor, F_{pp}, for gas in crossflow*

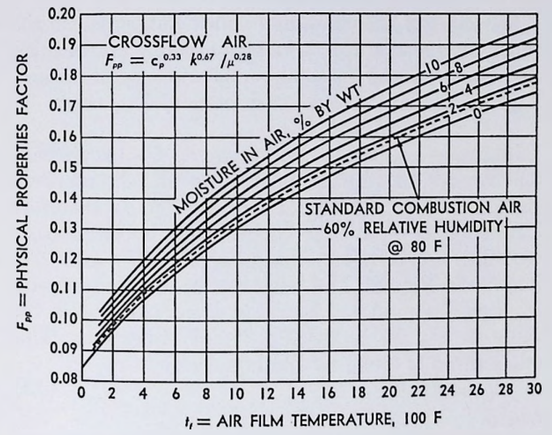


Fig. 37. *Effect of film temperature, t_f, on the physical properties factor, F_{pp}, for air in crossflow*

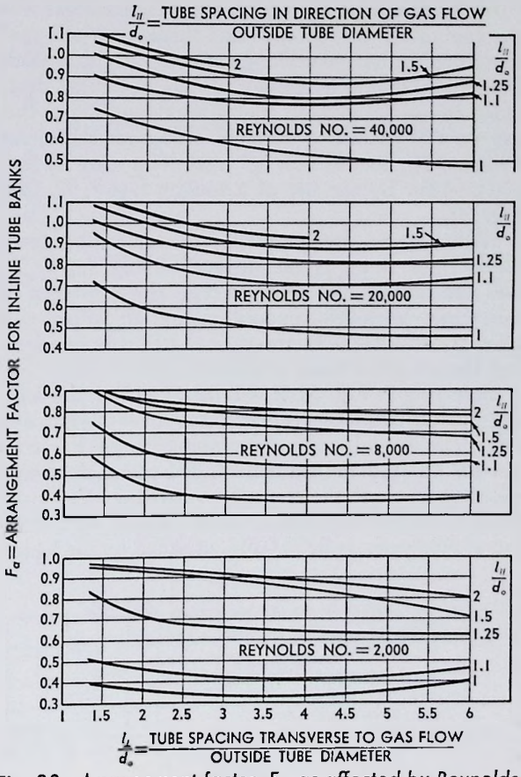


Fig. 38. *Arrangement factor, F_a, as affected by Reynolds number for various in-line tube patterns; crossflow gas or air*

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L = length of tube thru which gas flows, ft
d_i = internal diam of tube, in.
N_{Re} = Reynolds number = $d_i G_g / 12\mu$
μ = abs viscosity of gas, lb(mass)/ft, hr, from Fig. 11, Chapter 8

Substituting values for the air heater example above:

$$\Delta p = 1.5 \times \frac{937}{172{,}000}(8.04)^2 + 0.028 \times \frac{14.5}{1.834} \times \frac{937}{14{,}400}(8.04)^2$$

$$= 0.53 + 0.93 = 1.46 \text{ in. of water}$$

The draft loss obtained by the use of equation (11) is subject to corrections for barometer reading (elevation) and stack effect, as described above for draft loss in the boiler example.

The air resistance for air in crossflow through the bank of in-line air heater tubes may be determined using equation (7) with the approximate average

$$\text{air temperature} = \frac{1}{0.95}\left(\frac{t'_1 + t'_2}{2}\right)$$

Absorbing-Surface Cleanliness

In the examples dealt with above, natural gas was the fuel. With this specific fuel there is no ash, and therefore the heat-absorbing surfaces are clean. With fuels that contain an ash—coal, oil, and many dust-laden industrial gaseous fuels—the deposition of this ash on the absorbing surfaces causes an added resistance, R_f, to the flow of heat. This may be accounted for by the use of a fouling factor, K_f (see Fig. 39).

With air or the products of combustion from clean fuels there is no surface fouling ($R_f = 0$), and the fouling factor, K_f, is one. The resistance of ash varies with its composition and thickness, and allowance for these factors should be made in selecting the proper fouling resistance.

Referring to Fig. 39, it will be noted that the fouling factor is plotted against the over-all gas-side conductance, which is a combination of the convection and radiant components, the parameter being the resistance of the deposit. A general value used for this fouling resistance, when dealing with dust-laden flue gases, is $R_f = 0.015$, as noted on the figure.

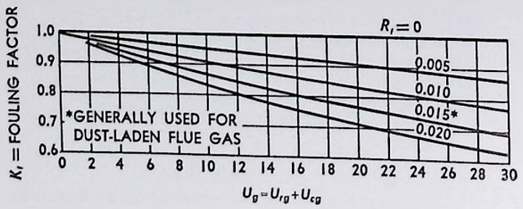


Fig. 39. Fouling factor, K_f, to account for surface cleanliness

Thus, when designing for dust-laden gases the conductance through the gas film and the deposit is the product of U_g and K_f.

Solution of Values in Equations (2) and (6)

$$U S \Delta t_m = w_g c_g \Delta t_g \quad (2)$$

$$S = \frac{w_g c_g \Delta t_g}{U \Delta t_m} \quad (6)$$

Δt_m = log mean temp diff, hotter and colder fluids, F

Three conditions of log mean temperature difference (Δt_m) of "hotter" fluid and "colder" fluid over the heat absorption path (the tube bank) are illustrated diagrammatically in Fig. 23.

Case I applies to the condition of heat flow from the hot combustion gases (hotter fluid), passing over boiler surface, to the constant temperature of saturation of the water and steam (colder fluid) within.

Case II applies to the condition of an economizer, superheater, or tubular air heater where the temperatures of both the hotter and colder fluids change as they proceed in the same direction (parallel flow) along the heat absorption path (the tube bank).

Case III also applies to the condition of an economizer, superheater, or tubular air heater, but in this case the hotter and colder fluids proceed in opposite directions (counterflow) along the heat absorption path.

U = combined conductance, Btu/sq ft, hr, F

The value of the combined conductance, U, for the boiler surface example is determined by establishing the values of the intertube radiation conductance, U_{rg}, and the convection conductance, U_{cg}, since:

$$U = U_g = U_{rg} + U_{cg}$$

Intertube radiation conductance, U_{rg}, represents the transfer of radiant heat from the hotter fluid (flue gases) to the colder surface and is therefore a function of Δt_m. The relationship between basic radiation conductance, designated as U'_r and Δt_m is given in Fig. 26. The basic radiation conductance, U'_r, must be modified by other factors to obtain the value of U_{rg} for a specific set of conditions. This relationship is given by the equation:

$$U_{rg} = U'_r K F_s \quad (12)$$

where:

U'_r = basic radiation conductance, Btu/sq ft, hr, F, from Fig. 26
K = fuel and tube-arrangement factor, a function of p_r and L (see Fig. 27)
p_r = total partial pressure of radiating gas constituents, atm (see diag ©, Fig. 28)
L = mean radiating length, ft (see Fig. 29)
$F_s = \frac{S_b - S_p}{S_b}$ = a factor to correct intertube radiation conductance for cavity radiation received by face of tube bank

S_b = total surface of tube bank receiving heat, sq ft
S_p = projected surface of tube bank face where gas enters, sq ft
$S_b - S_p$ = surface receiving intertube radiation, sq ft

Calculations to determine intertube radiation conductance, U_{rg}, for the boiler surface example above:

$U'_r = 5.70$, from Fig. 26, for $t'_s = 378$ and $\Delta t_m = 1200$
$p_r = 0.25$, from diag ©, Fig. 28, for 115% total air and 22,000 Btu/lb
$L/d_o = 0.13$, from Fig. 29, in-line tubes, for $l_1/d_o = 1.4$ and $l_{11}/d_o = 1.6$
$L = 0.13 \times 2.5 = 0.325$
$p_r L = 0.25 \times 0.325 = 0.081$
$K = 0.26$, from Fig. 27, for $p_r L = 0.081$
$S_b = 2344$ sq ft (assuming bank is 8 rows deep)
$S_p = 16 \times 99/12 = 132$ sq ft
$F_s = \frac{S_b - S_p}{S_b} = \frac{2344 - 132}{2344} = 0.944$

Substituting in equation (12):

$$U_{rg} = 5.70 \times 0.26 \times 0.944 = 1.40 \text{ Btu/sq ft, hr, F}$$

Convection conductance, U_{cg}, depends primarily upon the mass flow of the gas through the tube bank. The relationship between basic convection conductance, U'_{cc}, and gas mass flow, $G_g = w_g/A_g$, for various tube diameters, d_o, is given in Fig. 35 for crossflow. The basic convection conductance, U'_{cc}, must be modified by other factors to obtain the value of U_{cg} for a specific set of conditions. This relationship is given by the equation:

$$U_{cg} = U'_{cc} F_{pp} F_a F_d \quad (13)$$

where:

U'_{cc} = basic convection conductance in crossflow, Btu/sq ft, hr, F, from Fig. 35
F_{pp} = physical properties factor, from Fig. 36, evaluated at gas film temperature, t_f
t_f = gas film temperature for boiling fluids, see Fig. 30
F_a = arrangement factor, from Fig. 38, evaluated at established Reynolds number, N_{Re}
N_{Re} = Reynolds number, see Fig. 30
F_d = heat transfer depth factor, see Fig. 8, Chapter 7

Calculations to determine convection conductance, U_{cg}, for the boiler surface problem:

$U'_{cc} = 93$, from Fig. 35, for $G_g = w_g/A_g = 4780$ and $d_o = 2.5$

$$t_f = \frac{2t'_s + \Delta t_m}{2} = \frac{2 \times 378 + 1200}{2} = 978 \text{ F}$$

$F_{pp} = 0.129$, from Fig. 36, for 11% moisture and $t_f = 978$ F
$K_{Re} = 2.48$, from Fig. 30, for $t_f = 978$ F
$N_{Re} = K_{Re} \times w_g/A_g = 2.48 \times 4780 = 11{,}860$
$l_1/d_o = 1.4$ and $l_{11}/d_o = 1.6$
$F_a = 0.96$, from Fig. 38, for $N_{Re} = 11{,}860$, $l_1/d_o = 1.4$, and $l_{11}/d_o = 1.6$
$F_d = 0.99$, from Fig. 8, Chapter 7, for bank 8 rows deep

Substituting in equation (13):

$$U_{cg} = 93 \times 0.129 \times 0.96 \times 0.99 = 11.40 \text{ Btu/sq ft, hr, F}$$

The combined conductance, U, for boiler example (page 11-22) becomes:

$$U = U_g = U_{rg} + U_{cg} = 1.40 + 11.40 = 12.80 \text{ Btu/sq ft, hr, F}$$

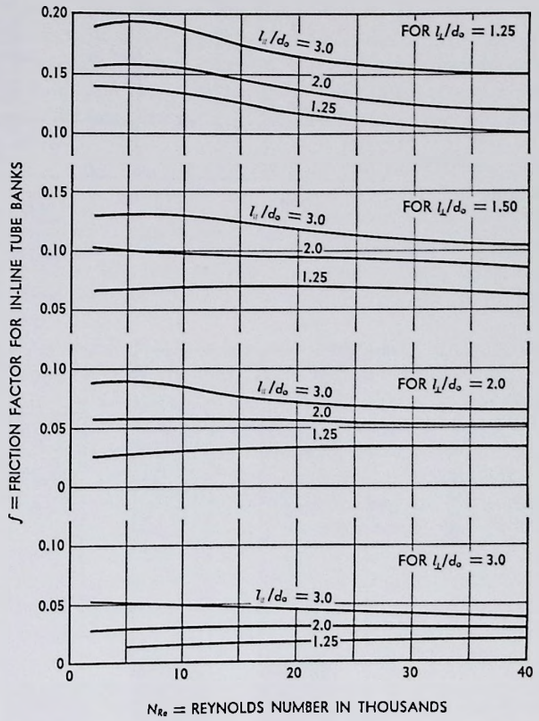
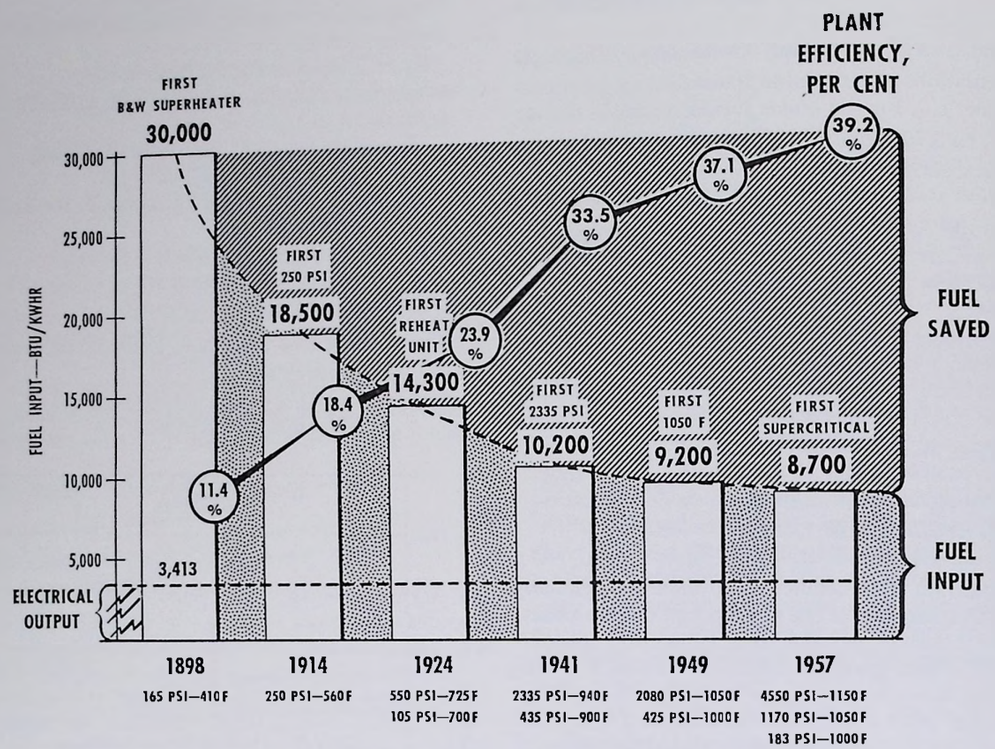
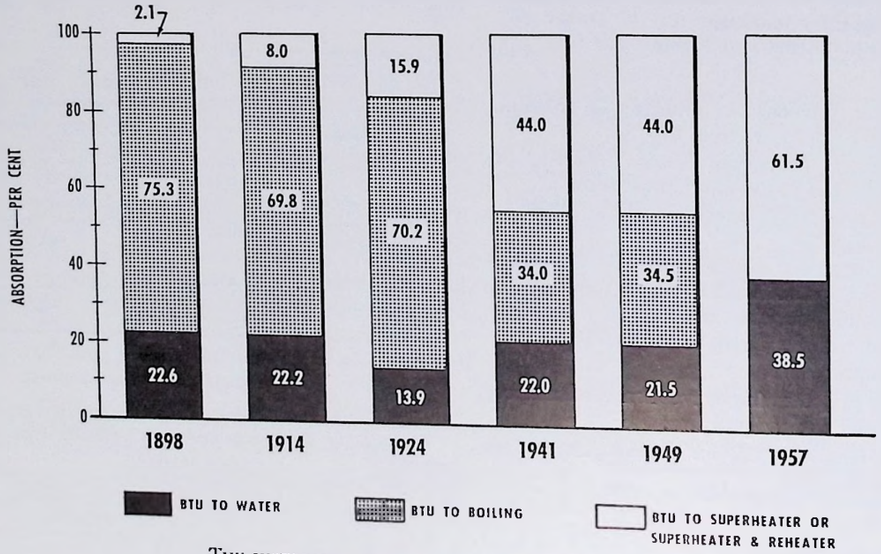


Fig. 40. Friction factor, f, as affected by Reynolds number for various in-line tube patterns; crossflow gas or air

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THE HISTORICAL IMPACT OF TEMPERATURE AND PRESSURE ON THE GENERATION OF STEAM AND ELECTRIC POWER

12

SUPERHEATED STEAM; SUPERHEATERS AND REHEATERS

In 1705 Denis Papin designed a recessed diaphragm, or piston, for an improved form of Thomas Savery's steam engine. Into this recess was placed a heated mass of metal, for the purpose of reducing condensation. Upon cooling, the metal was replaced by another heated mass. Tests of later designs indicated fuel savings of 30 to 50% with engines using superheated steam, and from about 1828 to 1865 much effort was devoted to the development of superheaters. Concurrently, much attention was also paid to the development of compound engines, which gave improved economy without the lubrication difficulties experienced with even the moderate superheats and low steam pressures then in vogue.

The triple-expansion engine was introduced in 1874, and an increase in steam pressure to 60 psi resulted in the temporary abandonment of superheat. However, by 1892 improved lubricating systems using mineral oil permitted the running of engines with superheated steam. Tests showed savings of as much as 20% with 113 F superheat. This provided the necessary impetus for the renewed development of superheaters.

In 1895 Babcock & Wilcox Ltd., England, began building superheaters, and in 1898 the B&W Co. became the first in this country to market superheaters commercially. They were of tubular type, located within the boiler setting. Combustion gases passing over the tubes heated the steam flowing within.

Use of Superheat and Reheat

Commercial development of the steam turbine hastened the general use of superheat. By 1920 steam temperatures of 650 F, representing superheats of 250 F, were generally accepted. In the early 1920's the regenerative cycle, using steam bled from turbines for feedwater heating, was developed to improve station economy without going to higher steam temperatures. At the same time, superheater development permitted raising the steam temperature to 725 F. A further gain in economy by still higher temperature was at that time limited by allowable superheater tube-metal temperature. This led to the commercial use of reheat, where the steam leaving the high-pressure stage of the turbine was reheated in a separate reheat superheater and returned at higher temperature and enthalpy to the low-pressure stage.

The first reheat unit for a central station was proposed in 1922 and went into service in September 1924. It was similar to the unit shown by Fig. 13, Chapter 1 and was designed for 650 psi and operated at 550 psi and 725 F. Exhaust steam from the high-pressure turbine was reheated to 725 F at 135 psi.

A much higher-pressure reheat unit, designed in 1924 for 1200 psi and 700 F primary steam temperature with reheat at 360 psi and 700 F, went into service in December 1925 (see Fig. 15, Chapter 1).

In 1928 live-steam reheaters, in lieu of gas reheaters within the boiler setting, were installed in two stations at Chicago. These, of course, gave only a modest amount of reheat compared with the direct-fired units, but it appeared at the time that the economics would justify their use. A year later, live-steam reheaters in series with gas reheaters were installed in the first of three different central stations, where initial pressures were 1300 to 1400 psi and primary steam temperatures were 750 F. The real intent of the series combination was to use the live-steam reheaters simply for control purposes. Some of the live-steam reheaters gave a poor account of themselves because of leakage of condensate through porous welds. Other units have remained in successful operation, but this method of controlling reheat was found to be a luxury economically, and the use of live-steam reheaters for new central-station plants was generally abandoned in favor of the gas type.

Until the middle 1930's, reheaters represented the best means of improving station economy because

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the temperature of the steam was limited to 750 F. The saving in heat input from reheating to the original high-pressure steam temperature was of the order of 4½ to 5%. To secure an equal gain would require an increase in steam temperature of 150 to 200 F. The development of high-temperature alloys made the design of 900 F steam temperatures practical. Consequently the reheater units lost popularity to the nonreheat high-temperature cycle. Boiler units designed in 1936 for operation in 1937 at 900 F steam temperature relegated reheaters to the background (see Fig. 17, Chapter 1), from which they emerged again in 1941 when the Twin Branch unit, designed for 2650 psi, 940 F high-pressure steam temperature and 435 psi, 900 F reheat, went into operation. By the end of the next decade the development of better metals, which permitted steam temperatures up to 1050 F (see Fig. 16, Chapter 1), rendered reheaters more attractive because, during the same period, the price of fuel had risen substantially and the possibility of any great increase in allowable high-pressure steam temperature seemed quite remote.

Except for the one period in the late 1800's, when lubrication difficulties with reciprocating engines made it unpopular, superheat has demonstrated clearly and forcibly its permanent place in the scheme of power plant economy. Reheat has had a much shorter history, and, while it has only recently emerged from temporary disfavor, it appears that it too has returned to stay, as represented by its overwhelming acceptance in current central-station practice.

Superheater Types

The original and somewhat basic type of superheater and reheater was the convection unit, for gas temperatures where the heat transfer by radiation was very small. With this type the superheat characteristic for any given unit increases with boiler output because of the decreasing percentage of heat input that is absorbed in the furnace, leaving more heat available for superheater absorption. Since convection heat transfer rates are almost a direct function of output, the total absorption in the superheater per lb of steam increases with increase in boiler output. (See also *Effect of Operating Variables*, Chapter 13, page 13-2.)

On the other hand, the radiant superheater receives its heat through radiation and practically none from convection. Because the furnace temperature does not increase in direct proportion to boiler output but at a considerably lesser rate, the curve of radiant superheat as a function of load slopes downward with increase in boiler output.

The two opposite sloping curves have been coordinated by the combination of radiant and convection superheaters to give flat superheat curves over wide ranges in load, as typically indicated in Fig. 1.

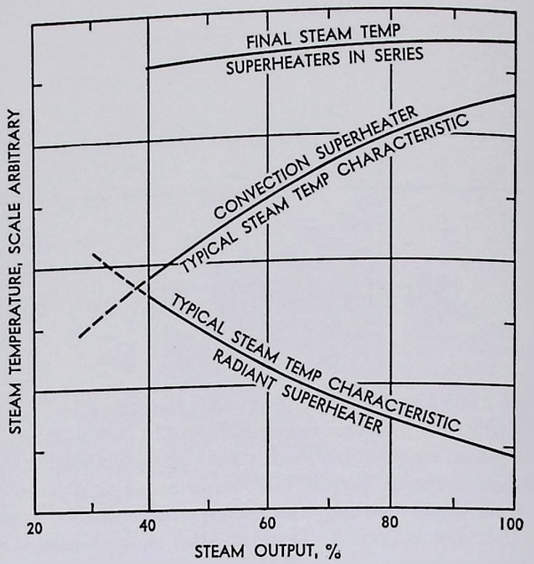


Fig. 1. *A substantially uniform final steam temperature over a range of output can be attained by a series arrangement of radiant and convection superheater components*

A separately fired superheater (see Fig. 1, Chapter 13) has no boiler surface between it and the fire and can be made to function as a combination radiant and convection superheater. To avoid excessive superheater tube-metal temperatures with a separately fired unit, it is necessary to recirculate tempering flue gases from the superheater outlet back to the furnace or to operate with high excess air.

Various arrangements of convection and radiant superheaters as components of boiler units are shown throughout this book and by types in Fig. 2.

Advantages of Superheat* and Reheat

The advantages of superheat and reheat are primarily twofold. The first is the strictly thermodynamic gain represented graphically in Fig. 10, Chapter 10, for the Rankine cycle. The second gain, and a greater one, is the improvement in turbine efficiency resulting principally from the reduction in the moisture formed in the last stages. The total gain in station heat rate from high-pressure superheat alone is approximately 3% for each 100 F superheat (see Table 3, Chapter 10). The gain from reheating

*The term "superheat" is not appropriate in defining the temperature of the working fluid at or above the critical pressure (see Chapter 9). Use of the term in this chapter relates to steam conditions in the more common subcritical pressure ranges.

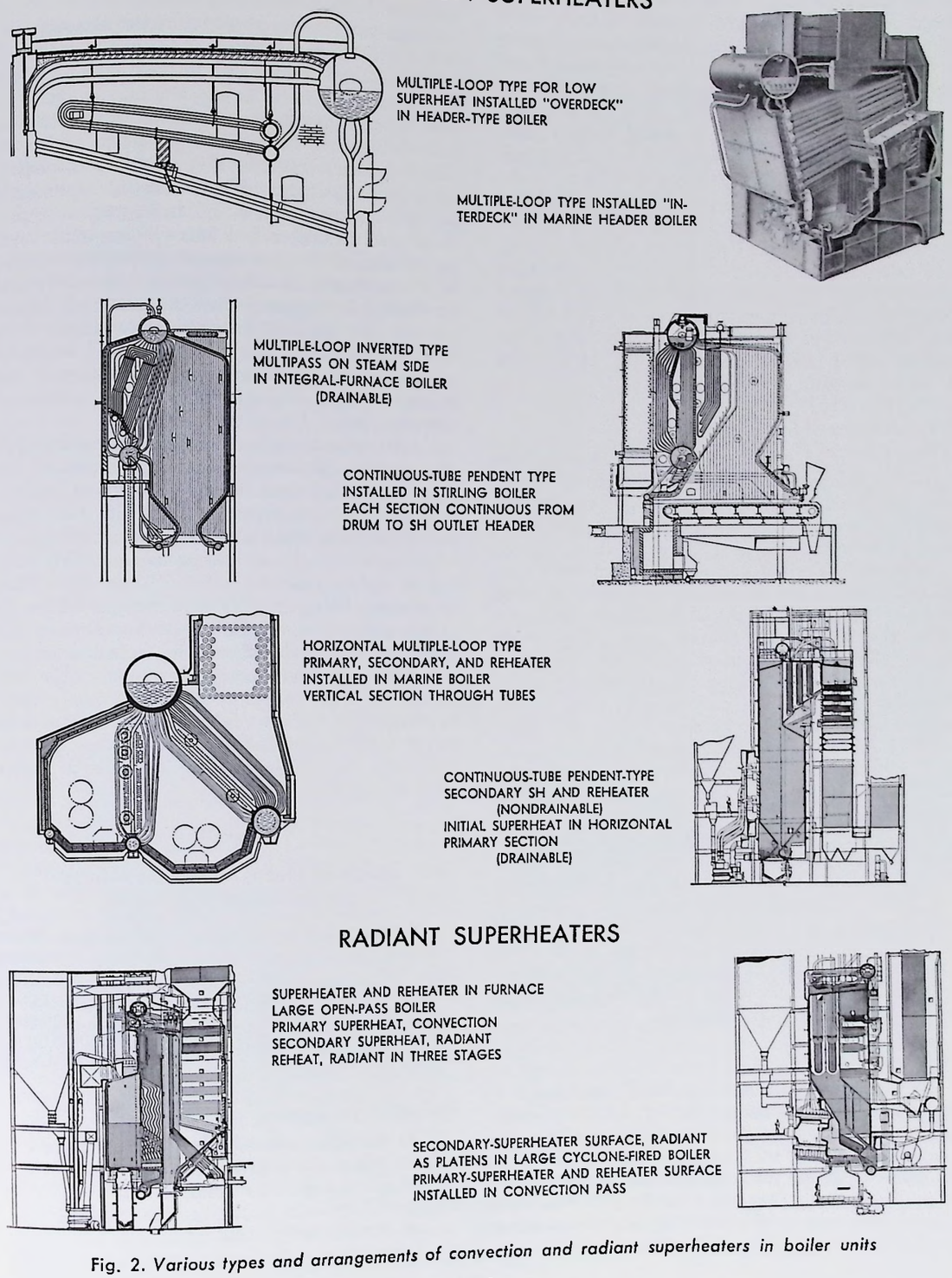


Fig. 2. *Various types and arrangements of convection and radiant superheaters in boiler units*

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to the initial temperature of the high-pressure steam in the range above 750 F is between 4½ and 5%, as noted above. With mounting costs of fuel, much effort is being applied to the development of equipment that will permit higher steam temperatures and higher reheat steam temperatures. Increased costs of materials demand a similar effort toward more effective boiler, turbine, and piping design, particularly since there is an economy loss of approximately 1% for every 10% pressure drop through the reheat system. The pressure drop problem with the single stage of reheating has been solved fairly well by limiting the pressure drop in the reheater, piping, and valves to a maximum of 10%. However, multiple reheating increases economy still further if pressure drop through these subsequent reheat stages can be kept low without a counterbalancing penalty in the cost of piping and reheater. (See Fig. 19, Chapter 1, for an example of double reheat.)

Development of Superheaters

The history of superheaters has, in reality, been a frontier operation. By careful over-all design, progress has been made in overcoming the constant threats of tube overheating, tube leakage, and tube-support failures.

In the early designs, convection superheaters were placed above or behind a deep bank of boiler tubes in order adequately to shield them from the fire or from the highest-temperature gases. They were fitted with flooding connections to fill them with water for protection during the starting-up periods. Flooding caused much more trouble than it prevented. Scale formed, tube seats leaked, and tubes warped and twisted from the quenching of hot spots at the steam pockets caused by partial evaporation of the flooding water. Attempts to use scale-free water and to develop adequate circulation within the superheater tubes proved impractical. These difficulties finally established the principles that water has no place in a superheater and that the tubes must be protected against overheating, during periods of starting-up or of no steam flow, by limitation of the firing rate and therefore the gas temperature to which the uncooled tubes can be subjected. The adoption of these principles, sometime before 1920, eliminated these early starting-up difficulties.

Steam Mass Flow

As steam temperatures increased, the necessary increase in the superheater surface, for the original location, made it more economical to move the superheater closer to the fire, where the reduction in surface justified the required modifications in design. The first big step was taken in 1922, when the superheater was moved from its overdeck position to the interdeck location, one arrangement shown in Fig. 12, Chapter 1. This new location brought into relief many defects in design which were not apparent with the superheaters located in the original lower-gas-temperature zone. Steam distribution difficulties initiated the use of ferrules which increased the tube entrance loss and thereby equalized the flow. Further advances in capacity and temperature duty revealed instances of general overheating of tube metal where "mass flow" (lb steam per hr, sq ft of flow area) was low. This served to emphasize the prime importance of mass flow in superheater design and to introduce the multiple-pass superheater with less steam flow area (higher mass flow) but still dependent upon ferrules or their equivalents for steam distribution. The pending demands and the existing limitations showed the fallacy of adding a ferrule pressure drop to the gainful pressure drop of an increased mass flow which, by itself, was adequate to achieve both uniform distribution to the individual tubes and generally lower tube-metal temperatures.

A reassessment resulted in the fundamental conclusions that the average steam mass flow in the superheater tubes must be large enough to produce a pressure drop sufficient to distribute the steam among the tubes and that each tube must have the requisite individual mass flow to keep it cooled within safe metal-temperature limits. There have been times when these principles were disregarded for expediency, but the experience of many years has demonstrated the value of resisting these temptations. The continuous-tube superheater design, either horizontal or vertical, can, in all but extraordinary cases, be adapted to the mass flow principle without the use of ferrules. The values of steam mass flow for convection superheaters given in Table 1 should serve as a general guide.

TABLE 1

Range of Steam Mass Flow Values
For Convection Superheaters

Temperature, F		Steam Mass Flow,
Steam	Gas	lb/hr, sq ft flow area
Below 750	1200	75,000-150,000
700-800	1600	250,000-350,000
800-900	2400	400,000-500,000
900-1000	2400	500,000-600,000
1000-1100	2400	700,000 and above

Radiant Superheaters

With the continuous-tube design, it has been possible to place the superheaters closer to the fire, behind one row of boiler tubes, as shown in Fig. 25, Chapter 17, or even in the furnace as wide-spaced radiant superheater platens, as in Fig. 10, Chapter 28. Recent increases in the temperature level of both high-pressure and reheat steam have made it apparent that the large amount of heat absorption required in the superheaters and reheaters calls for such an extravagant amount of low-duty surface that, in some cases, it is more economical to put some of the surface in the furnace for radiant absorption.

Experimental installations of radiant superheaters and some commercial installations made under special circumstances have illustrated the additional problems in design which must be met. They have vindicated the accepted practice of not locating the superheater any closer to the fire than the economics of good design makes necessary.

The same principles of sufficient steam mass flow to cool the tubes and slow firing of dry tubes to prevent overheating the tube metal, which were developed for the convection superheaters, apply even more stringently to the design and operation of radiant superheaters. It is paramount to avoid even minor infractions that might occasionally be permitted with convection superheaters. Radiant superheaters installed in furnace walls are subjected to higher temperature conditions and attendant thermal stresses. The latter are augmented by absorption at a higher rate on only the half-tube surface exposed to the furnace gases.

The really new factor in radiant superheater design is the support of the tubes to prevent distortion from temperature stresses which, though higher, may be kept within suitable limits by the more refined application of the convection superheater principles.

Development of Reheaters

The development of reheaters has followed somewhat in the wake of the development of superheaters. The same fundamental considerations governing superheater design apply to reheater design. The pressure drop in reheaters, however, is critical because the gain in heat rate with the reheat cycle can be completely nullified by too great a decrease in pressure. Nevertheless, the same relentless demands of good steam distribution and proper tube cooling by adequate steam mass flow are the important functions to be satisfied by improved designs and evaluations in the less robust reheater.

Difficulties from tube-seat and handhole-cap leakage respond to the same preventive measures taken in the operation of superheaters, namely, keeping the headers thoroughly drained of condensate and preventing other tube-to-header temperature differences, particularly during starting-up, by protection of the headers and several inches of the tubes near the seats from hot gases.

Relationships in Superheater Design

Effective superheater design calls for the resolution of related, and some unrelated, factors. The outstanding considerations are:

1. The steam temperature desired.
2. The superheater surface required to give this steam temperature.
3. The gas temperature zone in which the surface is to be located.
4. The type of steel, alloy, or other material best suited to make up the surface and the supports.
5. The rate of steam flow through the tubes (mass flow), which is limited by the permissible steam pressure drop but which, in turn, exerts a dominant control over tube-metal temperatures.
6. The arrangement of surface to meet the characteristics of the fuels anticipated, with particular reference to the spacing of the tubes to prevent accumulations of ash and slag or to provide for the removal of such formations in their early stages.
7. The physical design and type of superheater as a structure.

A change in any one of the first six items will call for a counterbalancing change in all other items.

Even the steam temperature desired, which, it would appear, must surely be fixed and from which a design can start, is subject to uncertainty. This is so because, in advanced power station design, the steam temperature desired is the maximum for which the superheater designer and manufacturer can produce an economical structure. Economics in this case requires the resolution of two interrelated factors — first, or investment, cost and the later cost of upkeep for minimum operating troubles, outages, and replacements. A higher first cost is warranted if the upkeep cost is thereby reduced sufficiently to cover, in a reasonable time, the extra initial cost. The steam temperature desired is, therefore, based on the complete coordinated knowledge available for the optimum evaluation of the combination of the other five items and the necessities of the particular project.

After the steam temperature desired is actually set or specified, the next consideration is the amount of surface necessary to give this superheat. The amount of superheater surface required is dependent upon the four remaining items, and, since there is no single correlation, the amount of surface must be determined by trial, locating it in a zone of gas temperature that is likely to be satisfactory. In the so-called standard boilers, the zone is fairly well established by the physical arrangements and by the space pre-empted for superheater surface.

The steam mass flow, the steam pressure drop, and the superheater tube-metal temperatures are calculated after the amount of surface is established for the trial location and the trial tube spacing. The proper type of material is then selected for the component tubes, headers, and other parts. If these trial results do not show the best arrangement of all the

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parts, then additional trial arrangements must be made to obtain an optimum combination which will:

1. Require an alloy of lesser cost.
2. Give a more reasonable steam pressure drop without jeopardizing the tube temperatures.
3. Give a higher steam mass flow in order to lower the tube temperatures.
4. Give a different spacing of tubes which will provide more protection against the ash accumulations with uncertain types of fuel.
5. Permit closer spacing of the tubes, thereby making a more economical arrangement for a fuel supply that is known to be favorable.
6. Give an arrangement of tubes which will reduce the draft loss for an installation where draft loss evaluation is crucial.
7. Permit the superheater surface to be located in a zone of a higher gas temperature, with a consequent saving in surface that will compensate for deviation from a standard arrangement.

It is possible to achieve a practical design with optimum economic and operational characteristics and with all criteria reasonably satisfied, but a large measure of experience and the application of sound physical principles are required for satisfactory results.

The methods of calculation for superheater performance are given in Chapter 11. It is appropriate, however, to include here Figs. 3 and 4 to show how tube temperatures are governed by steam mass flow and by the steam specific-heat absolute-viscosity relationship. The curves illustrate the advantage of routing steam with the highest pressure and lowest temperature through the high-duty tubes when the superheater is situated in a zone of high heat input. The application of these factors is illustrated in the superheater surface problem, Chapter 11.

It is established practice to admit saturated or low-temperature steam to any radiant superheater tube elements. A minor portion of this advantage in the use of low-temperature steam is sometimes sacrificed by the addition of a small amount of superheat before the steam enters the radiant or high-duty section. This is done to make dry and nonadhering the very small amount of foreign matter carried over, even in good-quality steam, and thus avoid overheating of the tubes and possible failure by preventing the deposition of this foreign matter in the zone where the steam changes from the wet to the superheated condition.

If the steam mass flow is higher than necessary for adequate safety, there is, of course, a needless penalty in steam pressure loss. This must be avoided. Good practice calls for sufficient steam mass flow to limit the steam-film temperature drop, between the

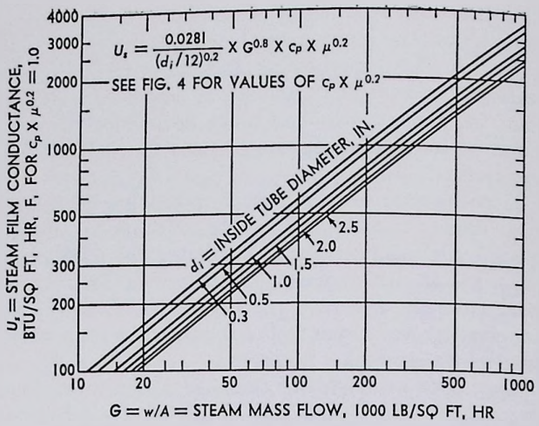


Fig. 3. *The effect of steam mass flow on steam film conductance for several tube sizes*

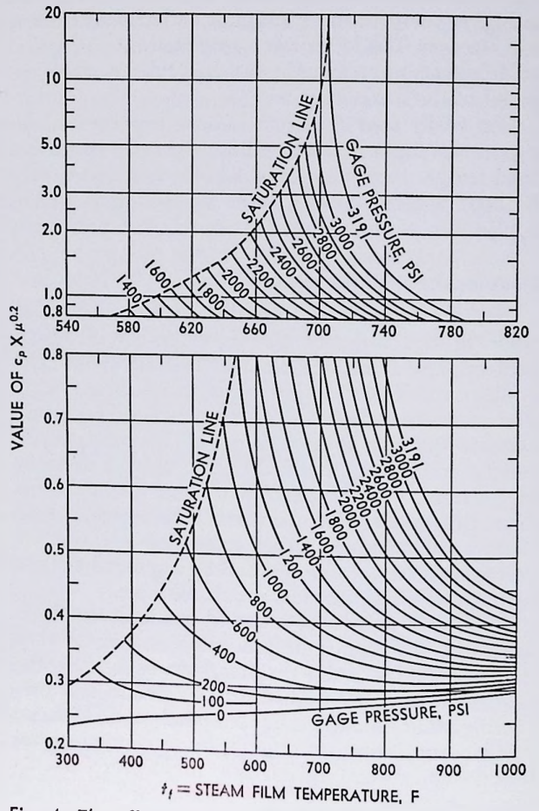


Fig. 4. *The effect of steam temperature and pressure on the value of the specific-heat absolute-viscosity relationship for steam (see Fig. 3)*

inside of the tube and the steam flowing within it, to not more than 100 F. The corresponding pressure drop through the superheater need not exceed 10% of the boiler operating pressure.

Effect of Gas Mass Flow

Tube-metal temperatures are affected in the reverse order by gas mass flow, that is, they increase with increase in gas mass flow. If the transfer of heat is entirely by convection, the superheater tube-metal temperature will change the same amount by gas-mass-flow variation as by steam-mass-flow variation —all at the same gas temperature. Actually in the higher-duty zones of modern superheaters, a substantial portion (about one third) of the heat is transferred by radiation, and, consequently, a given percentage change in gas distribution will have a lesser effect on superheater tube temperatures than the same percentage change in steam mass flow. In the zones of high gas temperature but away from the direct radiation of the high-temperature cavities, a given percentage change in gas mass flow may have a greater effect upon metal temperatures than the same percentage change in steam mass flow. The percentage of heat transferred by convection is high, and a change in gas distribution may result in a higher gas temperature and a higher temperature difference between gas and steam. Because of this, superheater tubes sometimes fail at locations beyond the zone of maximum expected heat input.

The gas mass flow is limited primarily by factors other than superheater metal temperatures. A gas mass flow of about 6000 lb per hr, sq ft of free-flow area is generally the economic limit, from the standpoint of draft loss, in the low gas temperature zones. This gas mass flow, with its accompanying close tube spacing, may be maintained to a gas temperature of 1500 F where the limitations of 1) draft loss economics, 2) bridging of ash deposits from fusion, and 3) tube erosion from high velocities with high dust-loading, pyramid. Above this 1500 F zone, except for units fired with clean fuel, the slag or dirt problem tends to take precedence and, entirely separate from the other limitations, makes the wider tube spacing necessary. The minimum distances considered adequate result in gas mass flows of less than 3000 lb per hr, sq ft of free-flow area and automatically take care of the other two limitations.

Relationships in Reheater Design

The same general similarity exists between superheater and reheater considerations, but the reheater is limited in ruggedness of design by the permissible steam pressure drop. Steam mass flows in reheater tubes should be sufficient to keep the steam-film temperature drop below 150 F. Ordinarily this may be done with less than 5% pressure drop through the reheater itself. This allows another 5% pressure drop for the reheater piping and valves without exceeding the usual allowable total maximum.

Metals for Tubes

Oxidation resistance, maximum allowable stress, and economics determine the choice of materials for superheater and reheater tubes. The use of carbon steel should be extended as far as these considerations permit. Beyond this point, carefully selected alloy steels should be used. The common alloy steels for this application are shown in Table 2. Their composition and properties are given in Chapter 22 and in the Appendix, page 22-A6 ff.

In addition to the immediate economic advantage of using carbon steel to the limit of its capability by keeping tube temperatures and stresses to a minimum, there is another important reason for using carbon steel at low temperature rather than an alloy at high temperature with the same margin of safety in design—the lower the temperature of the tube, the less severe will be any quenching action from soot blowing or other causes of temperature shock.

TABLE 2

Superheater and Reheater Tubes

Maximum Allowable Design Stresses, lb per sq in.

Material	ASME Spec	ASME Grade	For mean-wall metal temperatures not exceeding deg F: 900	950	1000	1050	1100	1150	1200	1250	1300	1350	1500
Carbon steel	SA-210		5,000	3,000									
Carbon-Mo	SA-209	T1a	12,500	8,500									
Croloy 1¼	SA-213	T11	13,100	11,000	7,800	5,500							
Croloy 2¼	SA-213	T22		11,000	7,800	5,800	4,200						
Croloy 3M	SA-213	T21			7,000	5,500	4,000	2,700					
Croloy 5	SA-213	T5				5,200	3,300	2,200					
Croloy 9M	SA-213	T9				5,500	3,300	2,200	1,500				
Croloy 18-8S	SA-213	TP304				10,000	7,500	5,750	4,500	3,250	2,450	1,800	750
Croloy 18-8Ti	SA-213	TP321				13,100	10,300	7,600	5,000	3,300	2,200	1,500	750

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This leads to the principle that the minimum tube temperature with the lowest alloy will, in general, be better than a higher tube temperature with a higher alloy.

Supports for Superheaters and Reheaters

Because superheaters and reheaters are located in zones of relatively high gas temperature, it is preferable to have the major support loads carried by the tubes themselves. In horizontal superheaters, the support load is usually transferred to the boiler or water wall tubes by means of lugs, one welded to the boiler tubes and the other to the superheater tubes. These lugs are, in many cases, made of high chromium-nickel alloy. Conditions make it necessary to have these lugs slide on one another in order to provide relative movement between the boiler tubes and the superheater tubes, as in Fig. 5. Supports of the saddle type, also shown in Fig. 5, provide for relative movement between adjacent components of the superheater itself.

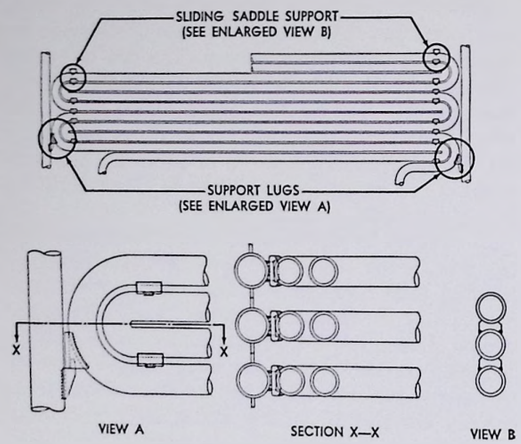


Fig. 5. Support lugs arranged to permit expansion of horizontal superheater both at main supports and within the superheater

With pendent superheaters, the major load is carried by the superheater tubes attached to the top headers, and it is necessary to use lugs only for spacing or for lateral support, all as indicated in Fig. 6.

The same principle applies to the selection of material for supports as for superheater and reheater tubes. It is better to design for the minimum support-metal temperature with the lowest alloy, going to the higher alloys only when proper design requires the use of high temperature. With certain fuels, such as oil, the ash may contain sufficient vanadium to be extremely corrosive at high metal temperatures.

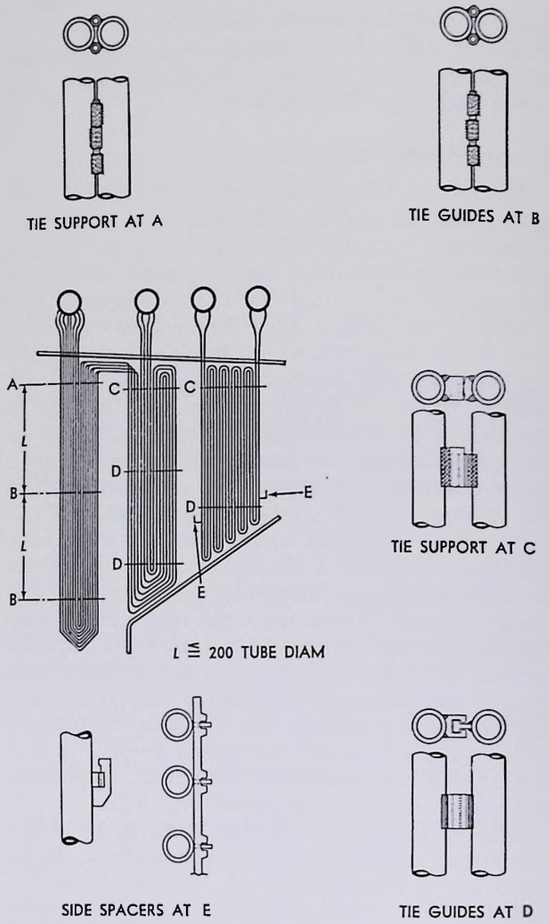


Fig. 6. Pendent superheater supported from above by inlet and outlet tubes to headers. Tie supports, guides, and spacers are also shown

Metal temperatures should be limited to a maximum of 1150 F to avoid excessive corrosion. This consideration often dictates the design of the superheater, in order to insure a support that will be cooled by the tube to which it is attached.

Experience has also shown that it is good practice to adhere to a second principle, i.e., all parts of the superheater or reheater should be free to move in paths recognized as orderly for all tubes, and any tendency to lateral movement should be eliminated. Any remaining tendency toward lateral movement should be resisted by adequate supports or reinforcements.

Cleaning—Internal

In the early designs of superheaters and reheaters, access to tube ends was provided, and the tubes were so arranged that they could be turbined for internal cleaning. Current designs of superheaters and reheaters provide for internal cleaning when such provision does not adversely affect other important features. Access for internal cleaning of the tubes is now subordinated to permit greater freedom in design, so necessary for good performance. The infrequent cleaning required, because of lapses in what is now recognized as good operation, can be accomplished by washing with water or by the use of acid solution, methods which do not require physical access to the internal surfaces. Suggested procedures for cleaning superheaters and reheaters are given in Chapter 21.

Cleaning—External

In the early designs of superheaters and reheaters, little provision was made for external cleaning. The residual space around the superheater was considered sufficient to give the necessary access. In modern units, the maintenance of clean surfaces is so important that a design is not considered successful unless it limits external deposits and provides for the ready removal of minor accumulations.

Tube Spacing

Spacing of the individual tubes is determined by first cost, permissible draft loss, and cost of cleaning to give maximum economy. An arrangement that provides a clear space of 1 in. between tubes transverse to the gas flow, as with 2-in. diameter tubes on 3-in. centers, is adequate for most fuels when the gas temperatures are less than 1500 F. Above this temperature, a 1-in. clear space is usually not enough because of the bridging of slag or of fused-ash residues. The critical temperature varies with the fuel.

Increased tube spacing has often been provided by omitting every other tube, giving a clear space of 4 in., for instance, with 2-in. tubes on 6-in. centers. This space increment is determined by trial and error, a spacing being adopted which does not impose any new unnecessary costs in multiple drilling or in deviation from standard shop practices. Openings of 4 in. have been adequate in most instances, where furnace designs have been conservative and where the gas temperatures entering the superheaters have been below the initial deformation temperature of the ash particles on an oxidizing basis. It has been ample also with oil fuels, where the softening temperature of the cementing materials has been between 1500 and 1600 F (see Chapter 18).

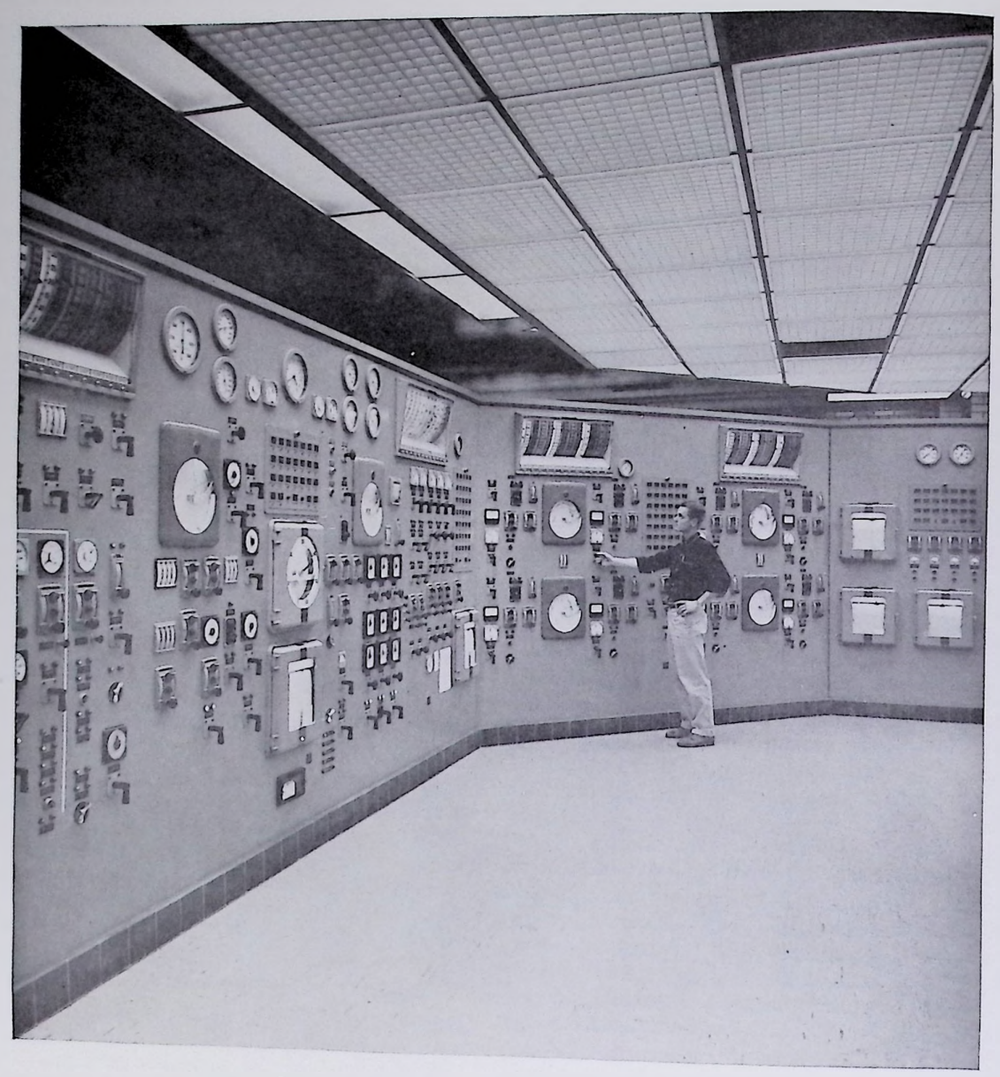
There is no truly rational method of adjusting the clear distances between tubes, or the tube spacing, as a sole function of the slag or dirt problem. Sometimes a clear space of 3 in. would be sufficient, but the manufacturing and other related considerations of spacing 2½-in. tubes on 5½-in. centers might involve so much additional cost that it would be more economical to have the first few rows of tubes on 9-in. centers, or multiples of a 4½-in. center-to-center "standard" spacing in the closer-spaced areas.

Until provision for preventing deposits, or for removing those which do form, can be based on fully developed knowledge of this important subject, it is best to use a tube spacing deemed proper from the many other standpoints of design and function and then to adjust this spacing within practical limits to suit the accumulated experience of cleanliness.

Tube Sizes

Plain cylindrical tubes of 2-in. outside diameter have predominated in the history of superheaters and reheaters. Smaller diameters tend toward higher pressure drops and greater difficulty in the maintenance of good alignment. Larger diameters unnecessarily bring about higher pressure stresses and undesirable rigidity. Recent designs, however, have called for greater spans between supports, for horizontal superheater tubes, and for wider tube spacing or fewer tubes per row to avoid slag accumulation. The 2½-in. tube has met these new conditions with a minimum sacrifice of the smaller-tube advantages. As temperatures increase, the allowable stresses may force a return to the smaller-diameter thinner-walled tube. Extended surface of superheater tubes in the form of fins, rings, or studs not only makes gas-side cleaning difficult, but the added thickness increases metal temperature and stress where they cannot be tolerated. Plain tubes are used almost exclusively in modern superheater practice.

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CONTROL ROOM FOR A LARGE CENTRAL STATION

13

STEAM-TEMPERATURE ADJUSTMENT AND CONTROL

THE use of superheat and reheat with ever higher steam temperatures, in the constant effort to increase efficiency in power generation, makes accurate regulation of steam temperature vital to successful operation.

Increases in temperature have closely followed advances in metallurgy, and it is therefore of the utmost importance to regulate steam temperature within narrow limits, going as high as possible to increase efficiency and still keep within a safe margin to avoid metal failure. Of equal importance is the control of the fluctuations in temperature from uncertainties of operation.

Superheat and reheat steam temperatures in steam generation are mainly affected by load demand on the unit. With drum-type boiler and once-through units employing steam separators, steam output and pressure are maintained constant by firing rate, while the resulting superheat and reheat steam temperatures depend upon basic design and other important operating variables, such as the ratio of convection to radiant heat-absorbing surface, excess air, feedwater temperature, changes in fuel that affect burning characteristics and ash deposits on the heating surfaces, and the specific burner combination in service. In the Universal-Pressure once-through boiler, which has a variable transition zone, steam output and pressure are controlled by the boiler feed pump and steam temperature by the firing rate, leaving reheat steam temperature as a dependent variable.

Improvement in the heat rate of the modern boiler unit and turbine, as indicated in diagrams of Fig. 10, Chapter 10, results in great part from the high cycle efficiency possible with high steam temperatures. The importance of regulating the steam temperature within narrow limits is evident, since a change of 25 F can mean a change of 0.5 to 0.75% in the heat rate.

Other important reasons for accurate regulation of steam temperature are to prevent failures from overheated parts of the superheater, reheater, or turbine, to prevent thermal expansion from reducing turbine clearances to the danger point, and to avoid erosion from excessive moisture in the last stages of the turbine.

Standard performance practice for steam-generating equipment permits a tolerance of plus or minus 10 F in a specified steam temperature. Years ago, when superheat was seldom more than 100 F, this represented approximately 10% of superheater absorption; however, with present-day superheats of 400 F and high pressure, this tolerance corresponds to less than 2% of superheater absorption. Variations in furnace-wall heat absorption will produce some change in steam temperature, but since this is only one of many factors affecting superheater absorption, the desirability of reliable means of steam-temperature regulation is evident.

Meanings of Terms Used

Brief definitions of the various terms used in discussing steam-temperature regulation are given below:

Adjustment. A change in the arrangement of equipment which affects steam temperature but which cannot be used to vary steam temperature during operation. Example — removal of superheater tubes.

Control. Regulation of steam temperature during operation without changing the arrangement of equipment. Example — operation of an attemperator.

Attemperator. Apparatus for reducing and controlling the temperature of a superheated vapor or of a fluid passing through it. Example — a bank of tubes, submerged in the boiler water, through which all or a part of the superheated steam is diverted to give up some of its heat, thereby regulating the final steam temperature.

Radiant Superheater or Reheater. Heating surface

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containing steam and receiving heat primarily by radiation. Example – tubes which form part of a furnace enclosure and within which steam is flowing.

Convection Superheater or Reheater. Heating surface containing steam and receiving heat primarily by convection. Example – a bank of tubes within which steam is flowing and across or along which hot gases pass.

Effect of Operating Variables

Many operating variables affect steam temperatures in drum- or separator-type units. To maintain constant steam temperature, means must be provided to compensate for the effect of such variables, the most important of which are:

Load. As the load increases, both the gas temperature and the weight of gas increase. In a convection superheater, steam temperature increases with rising load since heat absorption by convection, which is augmented by higher furnace exit temperature and greater gas flow, increases more rapidly than steam flow, whereas in a radiant superheater, steam temperature decreases as the load increases. In the case of the radiant superheater, this is so because the furnace absorption rate does not rise in proportion to the steam flow through the radiant superheater elements. Sometimes a convection and a radiant superheater of proper proportions are installed in series in a single steam-generating unit to maintain substantially constant steam temperature over a considerable range of load. (See Fig. 1, Chapter 12.)

Excess Air. For a change in the amount of excess air entering at the burners there is a corresponding change in the quantity of gas flowing over a convection superheater, and therefore an increase in excess air tends to raise the steam temperature.

Feedwater Temperature. Increase in feedwater temperature causes a reduction in superheat since, for a given steam flow, less fuel is fired and less gas passes over the superheater.

Heating-Surface Cleanliness. Removal of ash or slag deposits from heat-absorbing surfaces ahead of the superheater will reduce gas temperature and steam temperature. Removal of deposits from superheater surface will increase superheater absorption and raise steam temperature.

Use of Saturated Steam. If saturated steam from the boiler is used for soot blowers or for auxiliaries, such as pumps and fans, an increased firing rate is required to maintain constant main steam output, and this raises the steam temperature.

Blowdown. The effect of blowdown is similar to the use of saturated steam but in lesser degree because of the low enthalpy of water as compared with steam.

Burner Operation. The distribution of heat input among burners at different positions or a change in the adjustment of a burner will usually have an effect on steam temperature through changes in furnace heat-absorption rate.

Fuel. Variations in steam temperature may result from changing the type of fuel burned or from changes in the characteristics of a given fuel from time to time.

Regulation of Steam Temperature

As noted above under *Meanings of Terms Used*, "Adjustment" and "Control" are the two methods by which steam temperature can be regulated. Everything affecting the regulation of steam temperature belongs in either one or the other of these two classifications.

Adjustment

Why Adjustment is Necessary

A power-generating unit represents a large capital investment, and means should be provided at a reasonable cost for the regulation of steam temperature to meet changed conditions that may be more or less permanent in nature. For instance, if a fuel change that has a considerable effect on steam temperature is anticipated, it is good engineering to design for compensating physical alterations of the equipment. This is not necessarily so, however, if the type and range of control provided are adequate to meet the varied requirements without loss in boiler efficiency.

Adjustment for regulating steam temperature is required when the actual operating conditions depart, as they often do, from the conditions on which the design is based. To provide for ultimate adjustment to meet such variations in operation, allowance should be made in the design for the required change at minimum expense.

Quite often a plant is designed with some future change in mind that would affect steam temperature, as, for example, a different fuel or a higher operating pressure. Here again the superheater (and reheater if used) must be so designed that an economical adjustment can be effected.

Means of Adjustment

The basic method of adjustment for regulating steam temperature is by the addition or the removal of superheater and/or reheater surface. A good design should provide for an economical way of doing so.

Adjustment is also possible by a reduction or increase in the amount of saturated surface ahead of the steam-heating elements. Such alterations will modify the gas temperature at the inlet to these elements. If saturated surface is removed to increase steam temperature, this type of adjustment is relatively simple and, in general, will cost less than the addition of steam-heating elements. However, the addition of saturated surface in order to decrease steam temperature is liable to be difficult and expensive, or even impractical.

The gas temperature at the superheater inlet can be increased or decreased, without altering the convection surface ahead of the superheater, by changing the effectiveness of the heat-absorbing surface in the furnace. This is accomplished by the addition or removal of refractory covering on parts of the furnace surface. While a refractory coating on water-cooled furnace surfaces can have a favorable effect on combustion and carbon loss, refractory should not be added in areas where undesirable ash would deposit.

One of the simplest, least expensive, and most effective means of adjustment in regulating steam temperature is to change the gas mass flow over the steam-heating elements by baffle changes, if the design of the unit will permit such changes. Several of the so-called standardized boilers, especially in the smaller sizes, have an adjustable baffle suitable for steam-temperature regulation. By means of this design it is possible to vary the steam-temperature range as much as 20%. The limit of variation is the effect on draft loss and efficiency. A 10 F change in the temperature of the gas at the exit end of the boiler unit will alter efficiency by 0.25%, approximately.

Control

Why Control is Necessary

Regulating steam temperature within required limits by means of control is necessary, at least in drum-type units for high steam temperature, to correct fluctuations caused by operating variables. Probably the most frequent cause for the need of control in regulating steam temperature is the deposit of ash in some form on the surface of saturated and steam-heating elements. This condition can usually be corrected by simple changes in the operating schedules of soot- and slag-blowers.

To hold the optimum heat rate of a turbine designed to utilize full first-stage pressure over a given load range, it is essential to regulate steam temperature over this range by an effective means of control.

The time in which a turbine may be brought to full load is established by the turbine manufacturer in accordance with a safe steam temperature-time curve. Since the temperature of the steam is directly related to the degree of expansion of the turbine elements, and consequently the maintenance of safe clearances, it must be regulated within permissible limits by some means of accurate control.

The removal of feedwater heaters for servicing may be cited as an instance of variable operation. To maintain the same load, with the feedwater heaters out, will require an increase in the heat input to the boiler unit, with a corresponding increase in superheater or reheater absorption. Unless some means of control is available for regulating steam temperature to cover this condition, a drop in load might be required to protect both the steam-heating elements and the turbine.

Means of Control

Among the means of control for regulating steam temperature are:

Separately Fired Superheater
Excess Air
Gas Recirculation
Divided Furnace Differentially Fired
Burner Selection
Movable Burners
Gas By-Pass
Attemperation

Separately Fired Superheater. A superheater completely separate from the steam-generating unit and independently fired may serve one or several saturated steam boilers. This arrangement is not economical for power generation, where a large quantity of high-temperature steam is needed, and its use is largely confined to process industries, such as chemical manufacture and petroleum refining, to meet special requirements. Since heat absorption can be varied independently of the quantity of steam delivered, the separately fired superheater has great flexibility in operation and a wide range of steam-temperature regulation at all loads. Either oil or natural gas is customarily used as fuel.

A separately fired superheater usually requires some means of reducing the temperature of the combustion gases entering the tube bank, to avoid excessive use of expensive alloys. Entering gas temperatures normally range from 1500 to 1900 F, dependent upon the economies involved. One method of tempering the combustion gases is to operate with high excess air. This increases the stack loss and lowers efficiency. In most instances where a separately fired superheater is used, efficiency is not an important factor. If a higher efficiency is desired and the cost of additional equipment is justified, the combustion gases may be tempered with recirculated flue gas withdrawn from the setting beyond the superheater. Both methods are indicated in Fig. 1.

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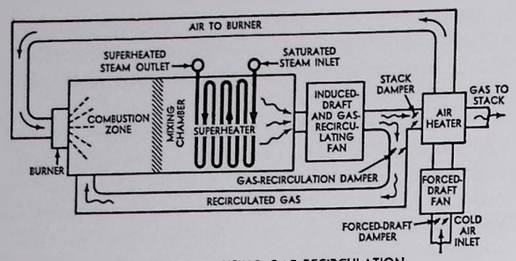


Fig. 1. *Typical arrangements for separately fired superheater*

Excess Air. Boiler operators have long known that the steam outlet temperature of a convection superheater on a drum- or separator-type unit can be increased at fractional loads by decreasing the furnace heat absorption through an increase in the amount of excess combustion air. The resulting greater gas weight to the stack increases the stack loss, but where the fractional-load operation is normally of short duration, the drop in boiler efficiency is usually unimportant since the increase in turbine efficiency may offset this higher loss. Such an arrangement is diagrammatically illustrated in Fig. 2.

Gas Recirculation. One of the most attractive methods of controlling superheat or reheat, both from economic and operational viewpoints, is gas recirculation. As the name implies, this involves a method by which gas from the boiler, economizer, or air heater outlet is reintroduced to the furnace by means of a suitable fan and ducts. For the sake of clarity, recirculated gas introduced in the immediate vicinity of the initial burning zone of the furnace and used for steam-temperature control will be referred to as "gas recirculation," and recirculated gas introduced near the furnace outlet and used for control of gas temperature will be referred to as "gas tempering." Fig. 3 shows a typical application of gas recirculation on a hopper-bottom radiant boiler. In most instances the gas is obtained from the economizer outlet. The point of reintroduction is dictated by the effect which is desired. The recirculated gas should be introduced into the furnace in such manner as to avoid interference with the combustion of the fuel. The amount of recirculated gas is generally expressed as a percentage of the gas that remains after the point of take-off of the recirculated gas.

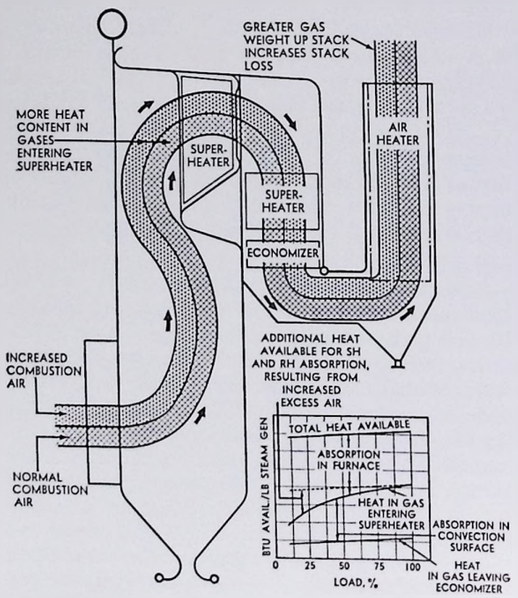


Fig. 2. *Steam-temperature control by use of increased excess air*

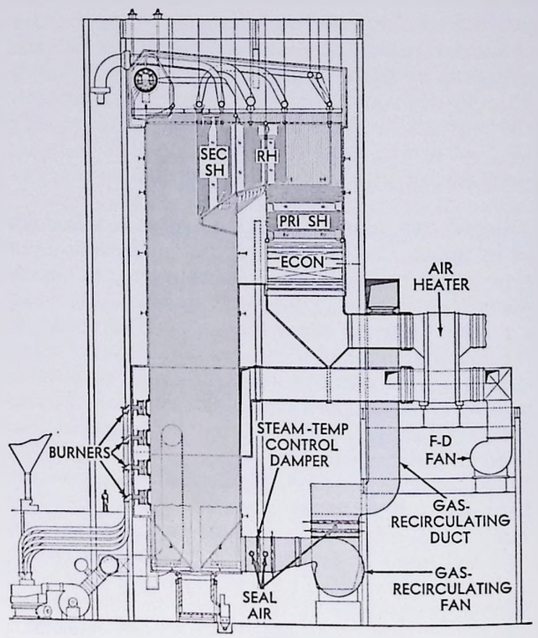


Fig. 3. *One arrangement for gas recirculation*

While recirculated gas may be used for several purposes, its basic function is to provide a means of altering the heat-absorption pattern within a steam-generating unit in a manner advantageous to both designer and operator. Recirculated gas has the special advantage of providing a means of heat-absorption adjustment that may be used both as a design factor in initial surface arrangement and as a method of controlling the heat-absorption pattern under varying operating conditions.

An important feature of recirculated gas is that its use changes only the pattern of heat absorption through a boiler and has a negligible effect on the total boiler heat absorption, the gas weight up the stack, and the boiler efficiency.

The thermal effect of recirculated gas depends upon the amount of gas recirculated, the location of gas introduction, and the furnace rating.

Fig. 4 shows the variation in heat absorption with gas recirculation introduced into the hopper. This location of gas introduction produces a marked reduction in furnace absorption and increases the absorption of the convection section. The total heat absorption remains unchanged.

Furnace heat absorption is primarily a function of the gas temperatures and gas-temperature patterns throughout the furnace, since the heat is mainly transferred by radiation. Hence, the introduction of gas recirculation into the furnace hopper reduces furnace absorption by altering the gas-temperature pattern.

The major portion of the heat absorbed in the superheater, reheater, and economizer is transferred by convection, which depends upon gas temperature and gas mass flow. Both are affected by gas recirculation. Therefore, when the gas mass flow through a convection bank is increased by gas recirculation, the amount of heat transferred may increase, decrease, or remain unchanged, depending upon the changes in the relationship between gas temperature and gas weight entering the bank. Fig. 4 illustrates a condition in which the gas temperature entering the secondary superheater (furnace exit temperature) is relatively unchanged by gas recirculation. Increasing the amount of gas recirculation, therefore, increases the heat absorption in the secondary superheater. The heat absorption in the reheater, primary superheater, and economizer is also increased, with the greatest increase occurring at the cold end of the unit. This is a typical example of variation in the convection-pass heat-absorption pattern by gas recirculation.

While gas recirculation into the hopper always reduces furnace heat absorption, its effect on furnace exit gas temperature depends mainly upon furnace rating. The furnace exit gas temperature may increase, decrease, or, as shown in Fig. 4, be essentially unchanged by gas recirculation. In general, recircu-

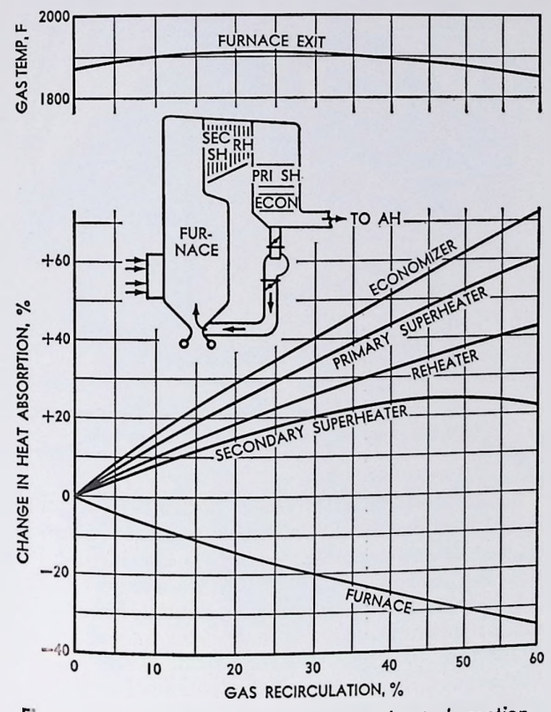


Fig. 4. *Effect of gas recirculation on heat-absorption pattern at a constant firing rate*

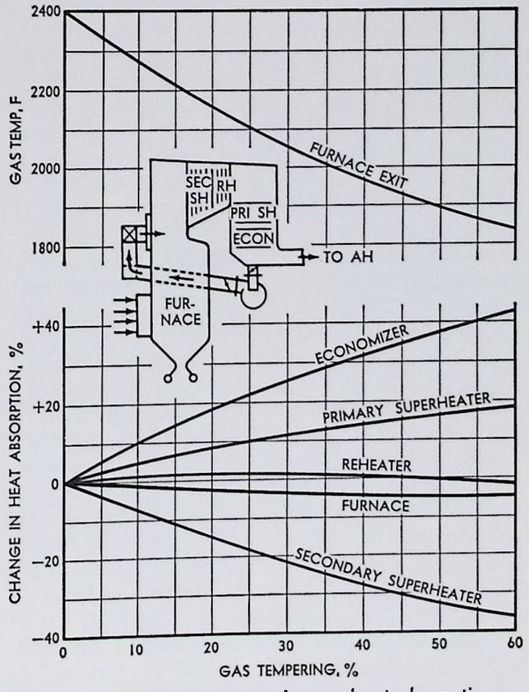


Fig. 5. *Effect of gas tempering on heat-absorption pattern at a constant firing rate*

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lated gas introduced in the hopper will decrease the furnace exit gas temperature of a unit operating at high furnace loading and increase this gas temperature at low loading.

Fig. 5 illustrates the effect of introducing tempering gas at a point near the furnace exit. Since the portion of the furnace in which the bulk of heat absorption occurs is unaffected by the recirculated gas, the furnace heat absorption is decreased only slightly. There is, however, quite a large decrease in furnace exit gas temperature, caused by dilution of the hot combustion gases with colder recirculated gas.

In the case of tempering gas introduced near the furnace outlet, the reduction in furnace exit gas temperature is usually sufficient to overbalance the effect of gas weight increase, and the heat absorption in the secondary superheater is decreased. The effect of gas tempering on the primary superheater and the economizer follows the pattern that was shown in Fig. 4, with the greatest change in heat absorption again occurring at the cold end. Because of the physical location of the reheater in Fig. 5, its absorption remains constant regardless of the percentage of gas tempering.

Figs. 4 and 5 illustrate the effect on heat distribution and gas temperature of introducing recirculated gas either into the hopper or at a point adjacent to the furnace exit. Introduction of gas at intermediate points will result in heat absorptions and gas temperatures between those shown. In order to show the effect of recirculated gas only, Figs. 4 and 5 have been based on constant firing rates.

A modern Universal-Pressure boiler with both gas tempering for gas-temperature control and gas recirculation for furnace heat-absorption control is illustrated in Fig. 6. In addition to gas recirculation, units of this type can employ spray attemperation in the reheater for reheat control.

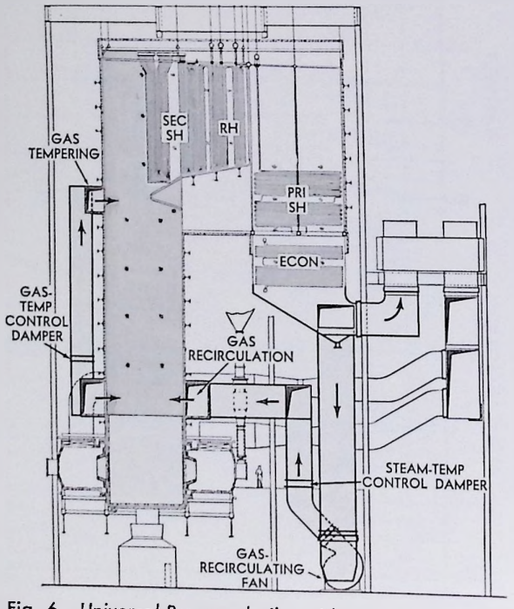


Fig. 6. *Universal-Pressure boiler with gas tempering for gas-temperature control and gas recirculation for control of furnace absorption and reheat temperature*

Divided Furnace Differentially Fired. In some divided furnaces, the superheater receives heat from one section of the furnace only, while the other section of the furnace generates only saturated steam. Steam temperature is regulated by changing the proportion of fuel input between the two furnaces. This arrangement, similar in principle to the separately fired superheater, has won wide acceptance in marine practice, where a considerable amount of saturated steam is required for auxiliaries, for maneuvering at sea, and for port operation when the main propulsion machinery is idle. The differentially fired divided-furnace method of steam-temperature control has also been used in a number of stationary power plants. A typical application of the principle to marine boilers is illustrated in Fig. 7.

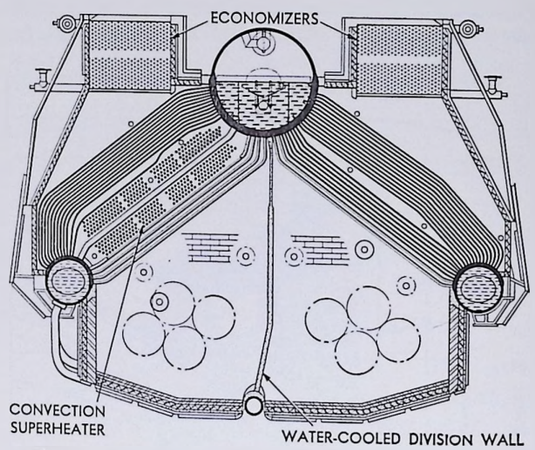


Fig. 7. *Two-furnace marine boiler with superheat controlled by adjusting furnace firing rates*

Burner Selection. It is often possible to regulate steam temperature by selective burner operation. Higher steam temperatures may be obtained at less than full load by operating only the burners giving the highest furnace outlet temperature. When lower steam temperatures are required, firing may be shifted to the other burners. This method of control can be improved by distributing the burners over a considerable height of the burner wall or by installing a special burner near the furnace outlet.

Movable Burners. Regulation of steam temperature by changing the furnace absorption pattern can also be effected by the use of movable burners operated to shift the main combustion zone in the furnace. Tiltable burners are used for this purpose.

Gas By-Pass. If the convection banks of a steam-generating unit are separated by gastight baffles into two or more parallel gas passes isolating portions of the superheater and reheater surfaces, the proportion of gas flow over all or part of the superheater and reheater may be varied by regulating dampers. An example of a two-pass arrangement for the control of superheat is shown in Fig. 8. This method has the advantage of low first cost, and high-purity spray water is not required as with a spray attemperator. Disadvantages may be listed as follows:

1. It is difficult to keep the dampers operable unless they are placed in a cool gas zone.
2. Draft loss through the unit is increased for some designs, particularly if control is desired with alternate fuels.
3. Control is more sluggish than with attemperators.
4. High gas temperature exists at the by-pass dampers unless there is heat-absorbing surface in the pass.

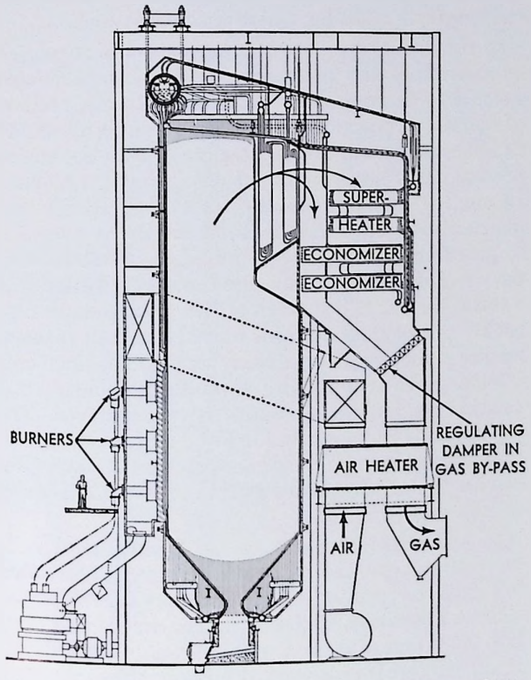


Fig. 8. *Superheat control by damper regulation of gas flow over superheater*

Attemperation. All of the foregoing methods of control in regulating steam temperature are based upon variation in the amount of heat absorbed by the steam-heating elements. In the two general types of attemperators commonly used, steam temperature is regulated by removing heat from the steam or by diluting high-temperature steam with low-temperature water.

The attemperator may be located in one of three places: between the saturated-steam outlet and the superheater; at some intermediate point between two successive sections of the superheater; or at the superheater outlet.

An attemperator located before the superheater will condense some of the saturated steam passing through it and will deliver wet steam of reduced enthalpy to the superheater.

There are important advantages in locating an attemperator between two stages of a superheater. With this arrangement the average steam temperature never exceeds the final steam temperature desired, and the steam from the various circuits of the first-stage superheater is so thoroughly mixed that it enters the second-stage superheater at a uniform temperature. This is, therefore, the most usual location, and the only disadvantage is the amount of piping required—from the first-stage superheater to the attemperator and from the latter to the second-stage superheater.

An attemperator located at the superheater outlet will be somewhat smaller than an interstage attemperator serving the same superheater, since it is handling higher temperature steam. The piping will be simpler. To keep the cost of alloys required in the attemperator, superheater, and piping within reasonable limits, this location should be selected only for moderately high steam temperatures.

Attemperators may be classified in two types—surface and direct-contact. In the surface type, the steam is isolated from the cooling medium by the heat-exchanger surface. In the direct-contact type, the steam and the cooling medium (water) are mixed. The surface attemperator includes the condenser type, the shell type, and the drum type. The direct-contact attemperator is exemplified by the spray type.

In the **condenser** type of surface attemperator (also called header type) illustrated in Fig. 9, loop tubes carrying feedwater are installed in the superheater inlet header. Tubes from the boiler steam drum to the superheater inlet header deliver saturated steam, which passes over the loop and is partly condensed before entering the superheater tubes. Uniform distribution of the condensed water droplets among the superheater tubes is essential, to avoid trouble from excessive temperature variation. Also, droplets must not be large enough to quench the hot superheater tubes and cause cracking. This type of attemperator has a limited range of control.

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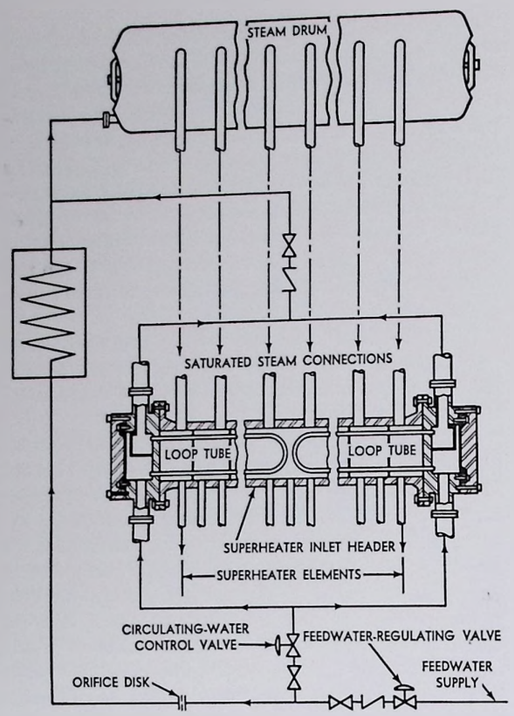


Fig. 9. *Condenser type of surface attemperator*

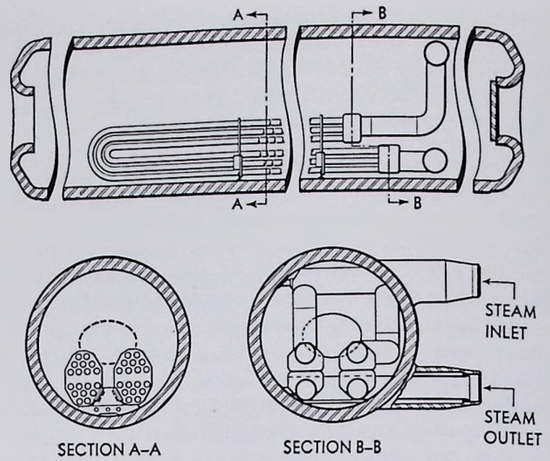


Fig. 10. *B&W drum type of surface attemperator*

When a shell or drum type of attemperator is used interstage, a part of the steam from the primary superheater is diverted to it by an automatic valve. The diverted steam is reduced in temperature by giving up heat to the boiler water, and it is then mixed with the rest of the steam from the primary superheater and passes on to the secondary superheater. The final temperature of the steam is regulated by the position of the automatically operated valve that controls the amount of steam diverted through the attemperator.

In the **shell** type of surface attemperator, a bundle of tubes through which the superheated steam flows is fitted in a cylindrical vessel through which cooling water circulates. A valve in the base of the shell diverts through the tubes the proper proportion of the superheated steam to give the final steam temperature desired. A positive flow of water in the shell at all times is assured by connection to the boiler circulating system. Thermal-sleeve ferrules at the steam-inlet ends connect the tubes to an inlet box which protects the tube sheets from severe thermal stresses. Disadvantages of this type of attemperator are the relative inaccessibility of the tubes and tube sheet for inspection of possible sludge deposition.

A more satisfactory design of surface attemperator is the **drum** type (Fig. 10), in which tube loops are submerged in one of the boiler drums and the required portion of superheated steam passes through the tubes. Difficulties with boiler-water deposits are avoided, and an additional high-pressure shell is not required. It may be necessary, however, to increase the size of the boiler drum to accommodate the attemperator tubes. Where this is not advisable, the amount of attemperation and, therefore, the range of steam-temperature control are limited by the number and length of tubes for which space is available.

In the drum type, steam temperature is regulated by control of the proportion of total superheated steam flow through the attemperator. In one method of control, a proportioning valve receives all of the superheated steam, diverting some of it to the attemperator while the remainder goes through a by-pass. In another method, suitable for comparatively low temperatures, an orifice is fitted in the by-pass line to give the pressure drop required to divert some of the superheated steam through the attemperator, and a throttling valve is located at the attemperator outlet. By this method a simple valve is used in place of a more elaborate proportioning valve, and, since only attemperated steam flows through it, construction suitable for high temperature is not required. The two methods are illustrated in Fig. 11. Steam-temperature regulation with a submerged-type attemperator can be of the same order as that shown for a spray attemperator, Fig. 13.

Of the **direct-contact** type as differentiated from the surface type, the spray attemperator, illustrated in Fig. 12, has proven most satisfactory for regulating steam temperature. Water of high purity is introduced into the superheated-steam line through a spray nozzle at the throat of a venturi section within the line. Because of the spray action at the nozzle and the high velocity of the steam through the venturi throat, the

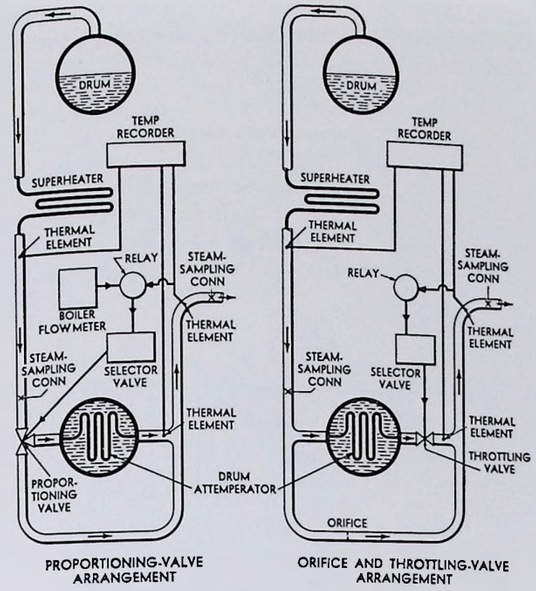


Fig. 11. *Automatic controls for drum-type attemperator*

water vaporizes, mixes with, and cools the superheated steam. An important feature of construction is the continuation of the venturi section into a thermal sleeve, downstream from the spray nozzle, to protect the high-temperature piping from thermal shock that could result from any incompletely evaporated water droplets striking the hot surface.

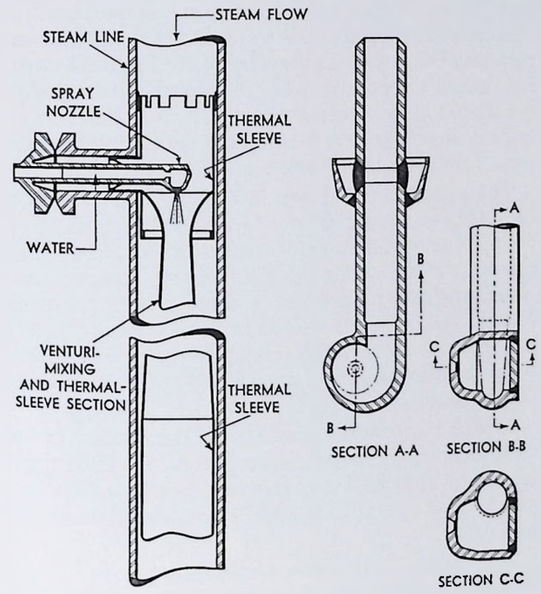


Fig. 12. *Spray attemperator showing thermal sleeve*

The spray attemperator provides a quick-acting and sensitive means of control for regulating steam temperature. An example of steam-temperature regulation using both gas recirculation and attemperation is shown in Fig. 13. It is important that the spray water be of highest purity, since solids entrained in the spray water enter the steam and may, if excessive, cause troublesome deposits on superheater tubes, piping, or turbine blades. High-pressure heater drains are a source of extremely pure water but require a separate high-pressure corrosion-resistant pump if used for attemperator supply. Boiler feedwater may be satisfactory, provided condenser leakage or make-up does not introduce too much contamination. The total solids concentration in the spray water should not exceed 2.5 ppm.

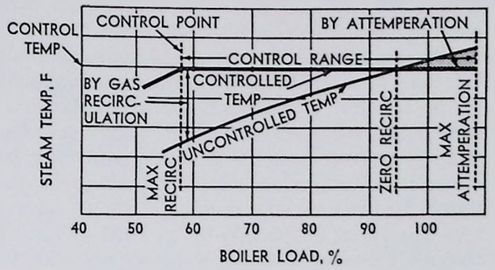


Fig. 13. *Control of superheated steam temperature by both gas recirculation and attemperation*

Reheat Steam Temperature

The need for regulating the temperature of reheated steam and the methods of adjustment and control to do so are, in general, the same as for superheated steam. However, the designer does not have the same freedom of action as when only the superheater is used. For instance, in a drum-type unit the removal of heating surface ahead of the superheater as a means of adjustment to increase superheated steam temperature will also increase the reheat steam temperature, which may be quite undesirable. Furthermore, to reduce gas temperature below slagging limits for convection tube banks and to give the desired steam temperature, such a large proportion of the total input must be absorbed in the furnace and in the superheater and reheater that the boiler is practically eliminated. When this is so, boiler surface ahead of the superheater is not available for adjustment purposes.

Most of the means of control described for regulating steam temperature will also affect the reheat temperature if a reheater is incorporated.

When a spray attemperator is proposed for superheater and reheater steam-temperature control, it is necessary to weigh the convenience and good control

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characteristics of the spray attemperator against reduction in cycle efficiency resulting from diversion of reheater attemperator spray water around the high-pressure turbine.

Instruments and Their Application

Instruments are required to tell the operator how the equipment is performing and to guide him in adjusting the controls to produce the desired results. Among these are instruments for automatic and manual control, indicating, recording, alarm, and for use at test connections.

Instruments for Automatic Control. A single thermal element actuated by the final steam temperature is the simplest and least-expensive type of automatic control for adjusting the means provided to give the temperature desired. The temperature-sensitive element may be a thermocouple, resistance thermometer, or bulblike pyrometer, located in the steam line beyond the superheater outlet. Variation of final steam temperature from the desired standard initiates a correction by adjusting the opening of a gas by-pass damper, moving a proportioning valve to allow a larger or smaller proportion of the total steam flow through a submerged attemperator, or changing the quantity of water supplied to a spray attemperator. A schematic diagram of the elements of such a control system is shown in the view at right in Fig. 11. This type of control is satisfactory for many applications, particularly where steam temperature responds quickly to an adjustment of the regulating mechanism, as in the case of an attemperator at the superheater outlet.

Where the response of steam temperature to the adjustment of the regulating mechanism is slow, as in the case of damper control or because of the thermal inertia in large superheaters, an anticipating type of control is preferable. This feature is included in the two-element control, in which either steam flow or air flow (latter usually preferred) governs the initial adjustment of the control mechanism. The final adjustment must always be by final steam temperature.

A schematic diagram of a two-element control system is shown in the view at left in Fig. 11. With this type of control, quick action follows a change in load because the proportioning valve is immediately set to the approximate new position required without waiting for a change in final steam temperature to take place.

A three-element control (Fig. 14) often used with spray attemperators is an improvement on the two-element control because of a more nearly accurate initial adjustment of the regulating mechanism, in this case the spray-water valve. The steam temperature at the outlet of the spray attemperator is used as the third element of the control. However, final adjustment must always be by final steam temperature.

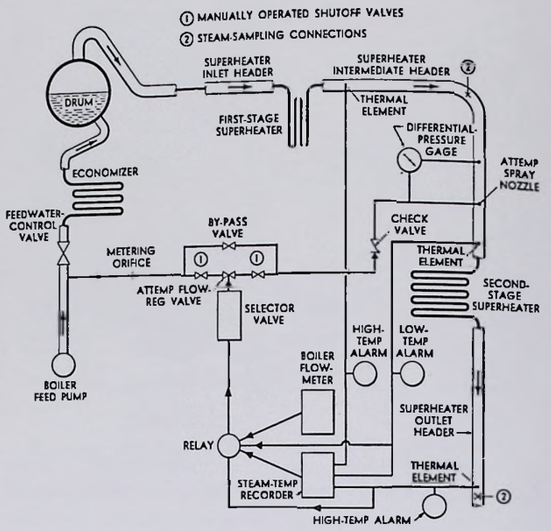


Fig. 14. *Three-element control for spray attemperator*

Indicating, Recording, and Alarm. How far to go in providing instruments for indicating and recording temperature and for alarms depends upon the steam-generating equipment furnished and upon the judgment of the design- or operating-engineer. In a small installation with moderate steam temperature, a simple instrument for indicating final steam temperature may be all that is necessary. For a large unit with high steam temperature, where deviation from design conditions may cause serious loss in efficiency or failure of equipment, a complete set of instruments for all necessary functions should be provided.

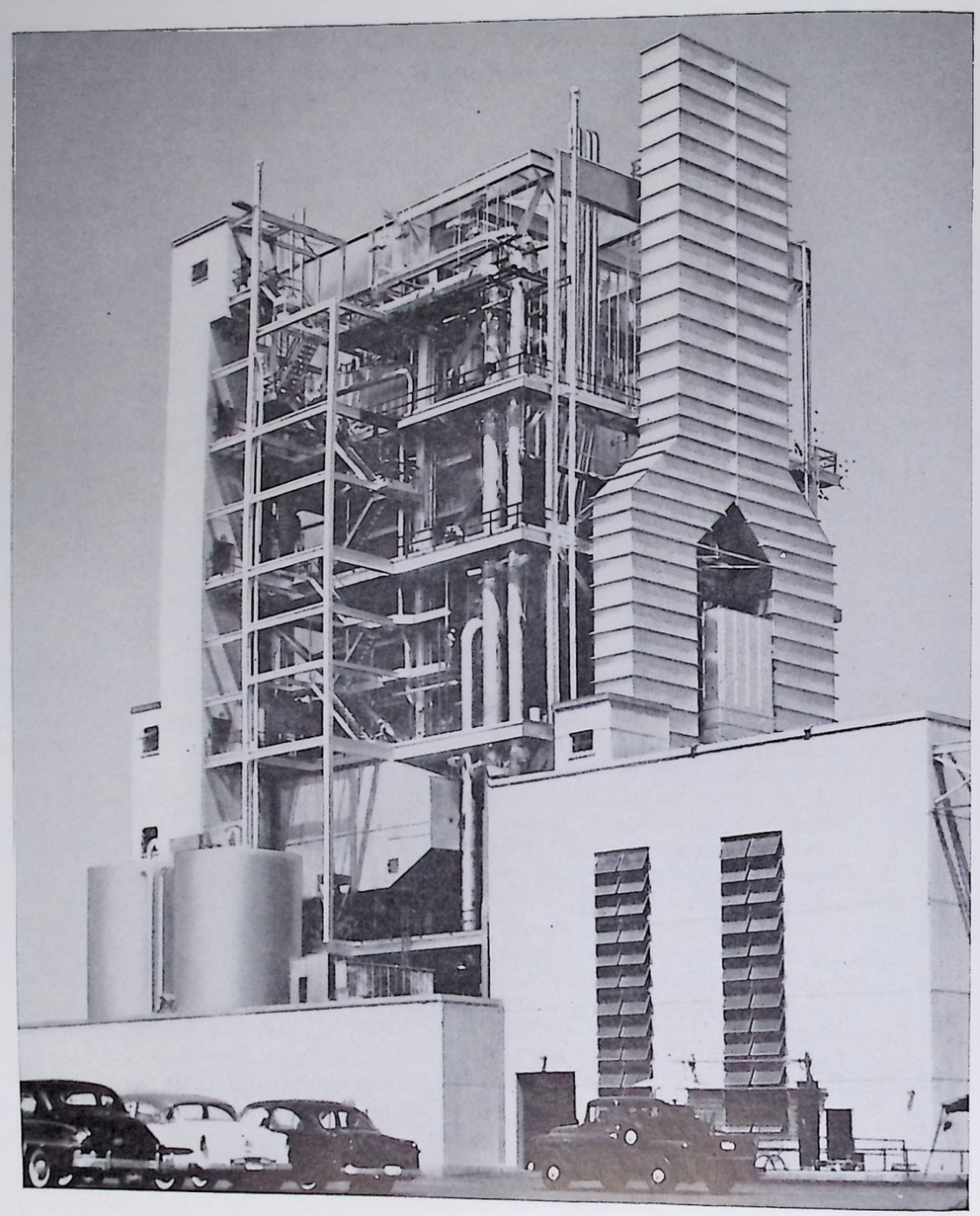
The arrangement shown in Fig. 14 will serve as a basis for illustrating the need for and the functions of a typical set of instruments. Of first importance is the correct determination of final steam temperature, since satisfactory operation and safety of equipment beyond the boiler, as well as protection of the superheater, depend upon this temperature. This measurement is also needed for actuating the automatic control system.

A high-temperature alarm is recommended as an additional safety feature, even though the careful attention given to final steam temperature by both boiler and turbine operators is usually considered sufficient protection.

Measurement of steam temperature with a low-temperature alarm at the spray-attemperator outlet will protect the second-stage superheater from the thermal shock of quenching by preventing overattemperation and the consequent discharge of saturated steam and water into the hot superheater elements. Measurement of steam temperature with a high-temperature alarm at the first-stage superheater outlet protects this stage from overheating should slagging or other upset conditions alter the distribution of heat absorption between first- and second-stage superheaters to such an extent that the primary steam temperature becomes excessive.

It is desirable to have a spare thermometer well in the steam line adjacent to each thermal element of the control system so that these elements may be checked periodically for accuracy. A steam-sampling connection should be installed in the steam line between the steam drum and the attemperator to check the purity of steam leaving the drum. It is also desirable to install a steam-sampling connection in the steam line after the attemperator to check for leakage of boiler water into the line in the case of a submerged attemperator, or for excessive solids carried over in the spray water of a spray attemperator.

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SEMIOUTDOOR GAS-FIRED RADIANT REHEAT BOILER IN A SOUTHERN UTILITY

14

BOILER SETTINGS, INSULATION, AND CASINGS

SETTINGS may be defined as physical surroundings or enclosures. By common usage the term "boiler settings" has been extended to include the physical surroundings of the boiler and its auxiliaries. Thus, boiler-setting drawings show the arrangement of the building steel, boilers, and auxiliary equipment, as well as the arrangement of the several enclosures. For the purposes of this chapter, a more restrictive definition is used, and boiler settings are limited to the enclosures—chiefly the walls—of complete steam-generating units. Fig. 1 illustrates the various types of enclosures with some of the commonly used names for the several parts.

Fig. 1. *Nomenclature of boiler-setting parts*

Design Requirements

Settings must safely contain high-temperature gases and air. Leakage, heat loss, and maintenance must be reduced to the economic minimum. A number of factors must be considered in the design of settings. The more important are:

1. The enclosures must withstand the effects of *high temperature*, ranging up to 3500 F in some cases.
2. The *action of ash and slag* (molten ash) must be considered from the following viewpoints:
   a) Destructive chemical reactions between slag and metal or refractory can occur under certain conditions.
   b) Accumulations of ash on the water walls can significantly reduce heat absorption.
   c) Ash accumulations can fall from a height and cause injury to personnel or damage to apparatus.
   d) High-velocity ash particles can erode the pressure parts.
3. Provisions must be made for the *expansion* of enclosures and for differential expansion of component parts.
4. The design of the *support* for the structure must make provision for the effects of expansion, temperature, and pressure stresses. Wind loads on outdoor units and the possibility of earthquakes must also be considered.
5. The effect of *explosions* must be taken into account, to lessen the probability of injury and damage.
6. *Vibrations* caused by combustion pulsations and the flow characteristics of gas and air must be limited to acceptable values.
7. The insulation of the enclosures should limit the *heat loss* to an economic minimum.
8. The *surface temperature* and the ambient air temperature must not cause discomfort and hazard to the operating personnel.
9. Enclosures must be *gastight*, to minimize leakage into and out of the setting.

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10. The design must be adequate to meet the *corrosive effects* of ash and gases.
11. Settings of outdoor units must be *weatherproof*.
12. Settings must be designed for economical *fabrication* and *erection*.
13. *Serviceability*, including access for inspection and maintenance, is essential.
14. Good *appearance*, consistent with the cost and maintenance requirements, is always desirable.

History

A brief history of the development of settings, considered from the standpoint of design requirements, is helpful in understanding present-day practice.

Early industrial boilers used in this country were of the fire-tube type. The completely water-cooled enclosure of the firebox class of fire-tube boilers (Fig. 2, Chapter 11) was adequate in many respects, but the capacity of the unit was limited by the size of the shell that could be manufactured. The problem of explosions was not solved. Accidental low water, corrosion, poor water circulation, or scale often caused large ruptures of the shell, resulting in a tremendous release of energy. In the year 1880, reports indicate that 254 people were killed and 555 were injured by fire-tube-boiler explosions. It is quite likely that many explosions were not reported.

The need for safety led to the development of the water-tube boiler. The B&W type (Fig. 2) was developed by George Babcock and Stephen Wilcox, with safety as the prime purpose. Good water circulation and improved design and materials minimized ruptures. When failures did occur, they were confined to the tubes, and the small quantity of energy so released could be adequately contained by the brick settings.

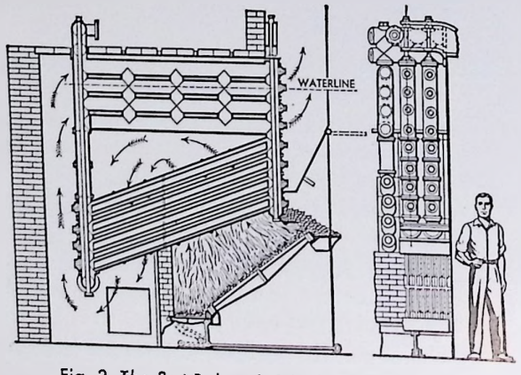


Fig. 2. *The first Babcock & Wilcox boiler, 1867*

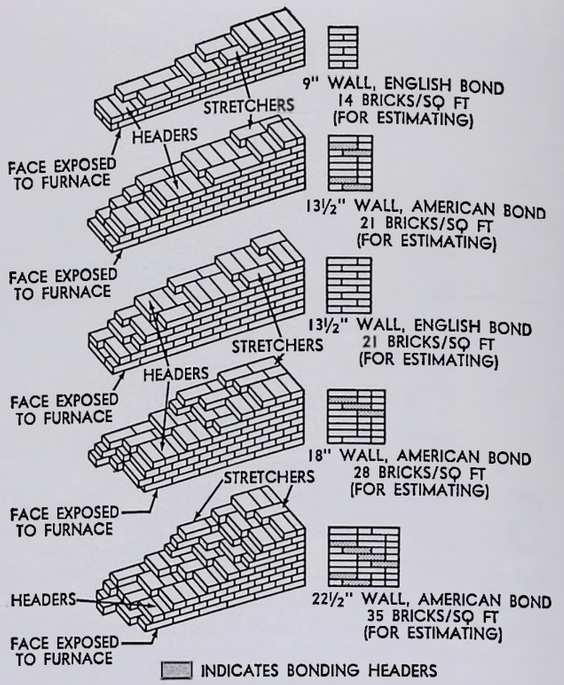


Fig. 3. *Firebrick walls—English and American bond*

The B&W boiler, and other water-tube types, had the boiler walls and the furnace walls made of brick that could withstand high temperatures. These better brick were fashioned from fire clay and were called "firebrick." Fireclay products, today, are generally called "refractories." The term is derived from the root meaning difficult to fuse and resistant to high temperature.

In the days when low steam pressure, high excess air, low boiler efficiency, and hand firing were acceptable, the duty of refractory linings for boiler furnaces was not severe. A fair-quality firebrick, laid in fire clay or even laid dry, gave satisfactory service and life. Air leakage through the walls helped to prolong the life of the refractories. However, the service requirements for refractory linings became more severe with the advent of mechanical stokers, pulverized-coal firing, oil and gas burning, larger furnaces, higher temperatures, and better efficiencies.

Shortly before World War I it became evident that the destructive effects on the then existing refractories would limit the development and construction of larger boiler furnaces for operation at higher temperatures. As a leader in the field of advanced boiler-furnace design, B&W was among the first to realize the need for improved refractories. Exhaustive research was accordingly undertaken and, after thorough study of available materials and processing

methods, a high grade of kaolin was chosen as the basic ingredient. The unusual potential possibilities of this aluminum silicate clay, with its purity, uniformity, and extreme refractoriness, had not previously been completely appreciated. Years were devoted to the development of processes for full realization of the intrinsic benefits. The result was the introduction of B&W 80 firebrick and B&W Junior firebrick with their high fusion points, great stability at elevated temperatures, and exceptional resistance to temperature changes.

With the increase in size of boilers and furnaces, the problems of high temperature and expansion became acute. Various types of wall constructions, as illustrated in Fig. 3, were developed which supported the weight of the wall on the cooler courses of brick at the center and the outer side. The bonds were arranged to facilitate repairs to the furnace lining and to minimize unnecessary removal of brick.

The pressure on the lowest course of brick in any wall is about 1 psi for every foot of height. Since the compressive strength of brick decreases rapidly with increase in temperature, the height of the wall is limited. This led to the development of the slip-wall arrangement, shown in Fig. 4. Here the upper part of the wall is separately supported so that its weight is not imposed on the lower part, which is at higher temperatures. The sectionally supported wall, illustrated in Fig. 5, is a further development of this principle. The structural load is divided into parts, and the expansion is sectionalized by suspending the brickwork from castings, which are, in turn, supported by external steelwork.

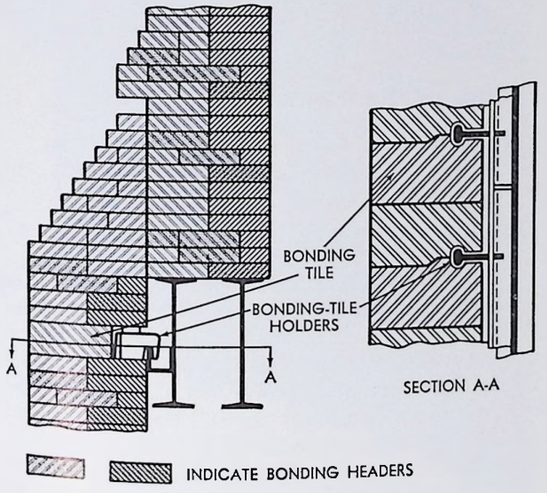


Fig. 4. *Slip wall, showing bonding-tile arrangement*

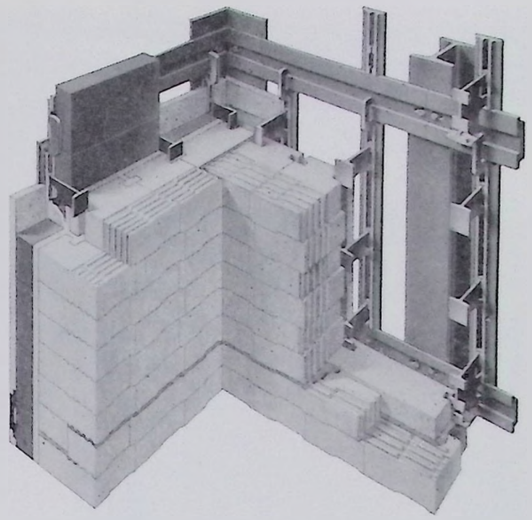


Fig. 5. *Tile wall—sectionally supported from the steel*

Pulverized-coal firing resulted in higher furnace temperatures and increased ash contact with the walls. Firebrick will not satisfactorily resist the combined action of high temperatures, ash, and slag. Ash penetration of the firebrick changes the coefficient of expansion at the surface and causes spalling. Chemical reactions between slag and brickwork at high temperatures result in fusing and washing away the brick. The exterior surface temperature of the brick can run as high as 350 F, the radiation from which makes adjacent areas uncomfortable for workmen. Additional insulation increases the hot-face temperature, further aggravating maintenance problems.

One attempt at a solution is the air-cooled wall, where air for combustion is forced over the walls on the outside surface. However, since the conductivity of brick is relatively low, as is the surface conductance on the air side, heat cannot be transferred at a sufficient rate significantly to reduce the hot-face temperature of the brick. Also, air infiltration into the furnace through expansion joints and cracks in the brick wall is excessive.

A major advance in the development of settings was the introduction of water-cooling. One of the first applications was the use of side-wall water boxes on chain-grate stokers. These water boxes not only reduce the brick temperature but prevent ash from sticking to the side walls and disturbing stoker operation.

Further to resist high temperatures and action of ash and slag, the Bailey block was developed. As illustrated in Fig. 6, these blocks, clamped to the tubes, form a tight wall with a comparatively cold smooth surface to resist ash accumulations.

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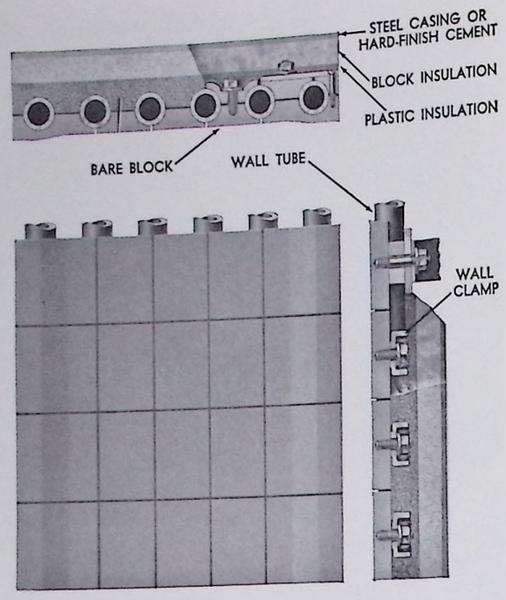


Fig. 6. *Bailey block-covered wall*

With still further increases in furnace temperature, maintenance on Bailey block walls and spaced-tube walls backed by refractory became excessive. Increased knowledge of circulation and improved burners allowed more complete water-cooling. Fig. 7 shows several design approaches made to satisfy the requirements. In the top wall shown, round studs are welded to the tubes at close intervals and packed with plastic chrome ore—a moldable refractory resistant to slags. The studs cool the refractory and hold it in place. The middle section of Fig. 7 represents a flat-stud construction, where small flat bars are welded to the sides of the tubes. Experience proved that there are certain maximum stud lengths that would survive in high-duty furnaces. The lowest section of Fig. 7 illustrates one of the first "tangent" tube constructions. Maintenance on the spaced-tube wall construction of 3¼-in.-OD tubes on 6-in. centers backed by firebrick was extremely high. This undesirable feature was corrected by interspersing 2¾-in.-OD tubes between the 3¼-in.-OD tubes, giving a tube-to-tube construction.

The first commercial slag-tap unit was originally designed for dry-ash removal. Furnace temperatures were higher than the fusion temperature of the ash, which therefore formed a molten pool of slag. To cope with this situation, a hole was cut in the furnace floor, and a water sluice was added under the tap hole to carry away the slag. For use in slag-tap furnaces, the stud-tube wall construction was developed, as shown in Fig. 8. The pin studs retain an insulating layer of slag to reduce heat absorption in

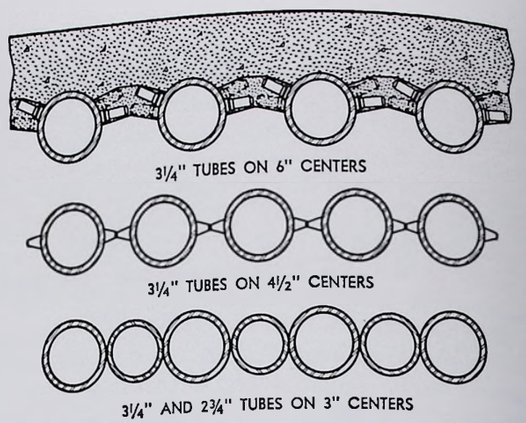


Fig. 7. *Water-cooled wall constructions*

the extremely hot slag-tap furnaces. The full-stud construction also protects against high-temperature corrosion, which can occur under certain conditions.

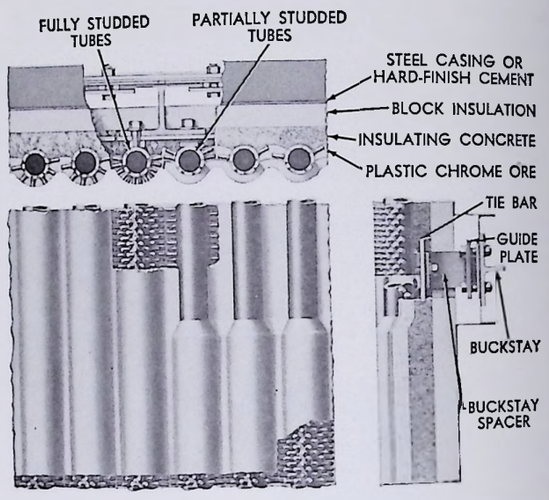


Fig. 8. *Original full-stud-tube wall construction; fully studded (left), partially studded (right)*

Resistance to High Temperature

Over the years, brick and refractory settings have been replaced by water-cooled walls in high-temperature zones. Improved knowledge of combustion and water circulation in the boiler design was necessary for this development. However, not all conditions are severe, and brick settings are still occasionally used for units delivering up to 20,000 lb steam/hr.

For moderate service (under 60,000 lb steam/hr) with gas- and oil-fired units, tube-and-brick walls, as illustrated in Fig. 9, are sometimes used. The tubes

are spaced at intervals and are usually backed by a 2½-in.-thick first-quality firebrick, supported from the wall tubes and by suitable insulation.

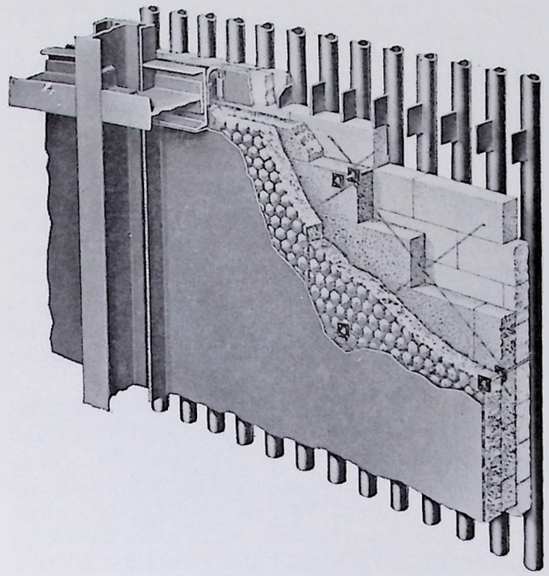


Fig. 9. *Tube-and-brick wall construction*

For more severe service, the close-spaced-tube wall, shown in Fig. 10, is used to form a continuous water-cooled envelope. The flat-stud-tube wall, shown in Fig. 11, is a construction that also presents a practically continuous water-cooled metal surface. The modern full-stud-tube wall, as shown in Fig. 12, is used in liquid-slag zones of slag-tap furnaces and in the lower portion of chemical-recovery furnaces.

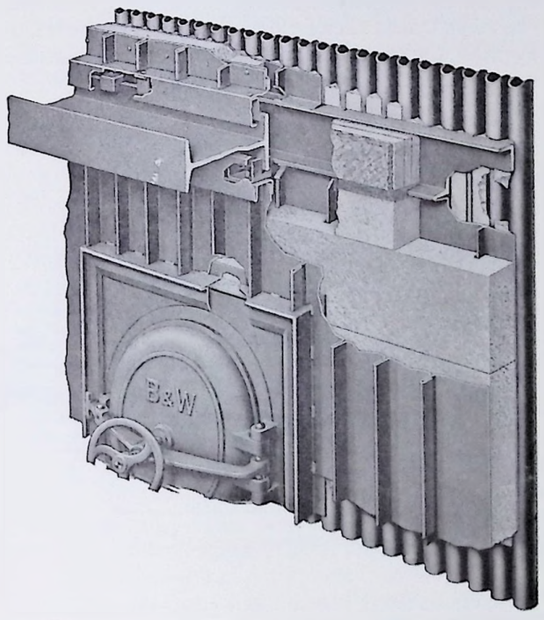


Fig. 10. *Close-spaced-tube wall with steel inner casing and outer metal lagging*

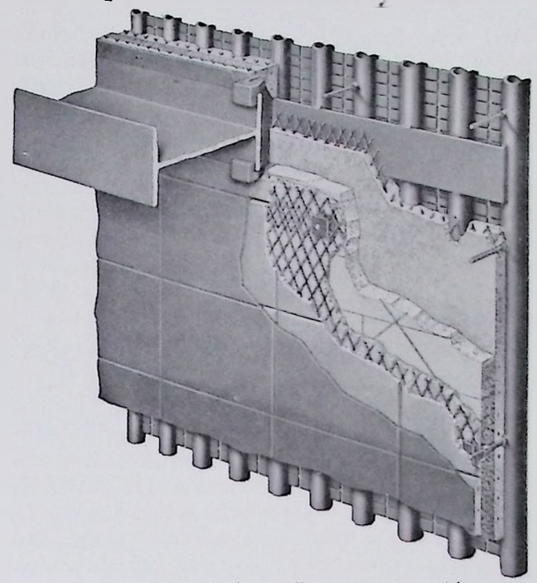


Fig. 11. *Flat-stud-tube wall construction with reinforced insulating finish and seal coats*

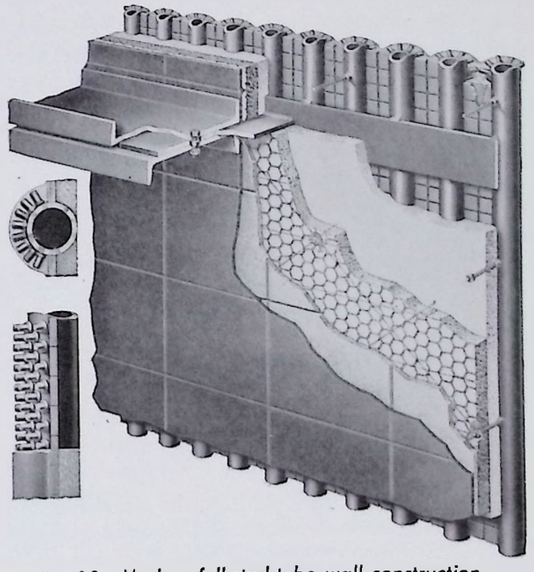


Fig. 12. *Modern full-stud-tube wall construction*

Steam-cooled walls offer certain advantages on high-temperature reheat units, where the superheaters and reheaters absorb a large fraction of the heat re-

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leased. However, considerable care must be used in the design to insure adequate steam flow under all conditions for proper cooling of the wall. Circumferential cracking of the tubes can occur if the cyclic temperature differences between the hot and the cold sides of restrained superheater tubes are large and occur often enough to cause fatigue failures from repeated yielding of the metal. In tubes conveying steam, high-temperature creep and oxidation can be problems resulting from a high temperature drop across the steam film inside the tubes. In general, steam-cooled walls are satisfactory only in the cooler zones of the boiler unit.

In flues that handle gases much above 850 F, refractory linings are required.

Resistance to Ash and Slag

Firebrick and refractories, even of the highest quality, cannot withstand the action of ash and slag in high-duty furnaces. Spalling and actual chemical attack occur, and a furnace lining cannot be maintained. The most satisfactory construction is a completely water-cooled metal surface. The colder the exterior surface of the metal the better. Figs. 10, 11, and 12 illustrate water-cooled-tube wall designs that are used for enclosures where slag and ash are encountered.

Ash has a tendency to shed from a relatively cold metal surface, particularly when the ash itself is relatively low in temperature. Wall-type blowers remove ash in higher-temperature areas, where the ash tends to adhere to the walls.

Extensive areas of exposed refractory should not be used because of the tendency for slag adhesion. Thus damage or injury from the falling of large slag accumulations into the furnace is avoided. Also, crotches formed by tubes bent out of the plane of the wall should be designed to prevent ash accumulation.

Erosion of pressure parts is prevented by limiting the gas velocity through the unit. With the average gas velocity limited, however, local high velocities can still occur in areas where gas by-passes baffles or heating surface. These high-velocity lanes must be eliminated by careful design and installation.

Expansion

The larger the brick settings, the more difficult it is to accommodate the differential expansion between the boiler pressure parts and brick walls and the expansion in the walls themselves. With the completely water-cooled setting, the problems are simplified because all parts expand together.

In the tube-and-brick wall, Fig. 9, the wall tubes are so spaced that the average operating temperature of the refractory is about twice the temperature of the boiler pressure parts. Since the coefficient of expansion of firebrick is about half that of steel, this arrangement permits the use of a continuous refractory envelope around the boiler setting without expansion joints.

For the flat-stud-tube wall, Fig. 11, there is a small theoretical difference, at operating pressure, between the width and height of the tube wall and the corresponding dimensions of the refractory backing. This results from the drying shrinkage of the castable (about 0.3% at 600 F) and the slight difference between the expansions of the boiler steel and the castable, the coefficient of expansion of the latter being about half that of steel. This slight difference in expansion may cause some hairline cracks to appear in the refractory backing. These cracks are insignificant because they are widely distributed by the reinforcing metal lath.

With the inner-cased unit, Fig. 10, small temperature differentials can occur between the casing and the tubes during start-up. Expansion of the wall in the horizontal direction is governed by the temperature of the tie channel. Since the casing and the channel are at the same temperature, they can be welded together. Vertical expansion differences are taken in the slight bending of the casing flange.

With a bottom-supported unit, such as the PFI boiler, Fig. 13, the structure is fixed at one point—at one end of the lower drum. Clearances, seals, and supports can then be designed for known expansions in all directions. With the top-supported unit, Fig. 14, the expansion occurs downwards from one elevation. Unless one of the walls is fixed to the building steel, the expansion will occur outwards from the center of gravity of the unit.

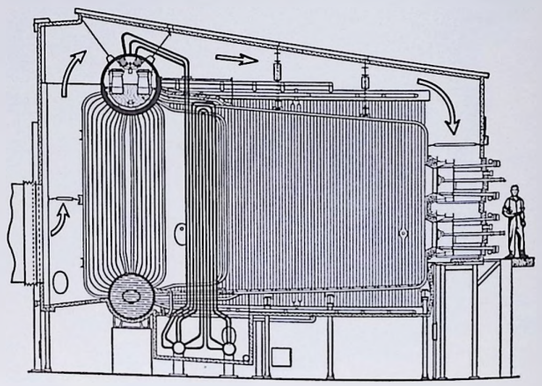


Fig. 13. *Bottom-supported unit*

Flues and ducts, piping, ash tanks, and burner lines must be designed with expansion joints or seals to accommodate the relative movements involved. For

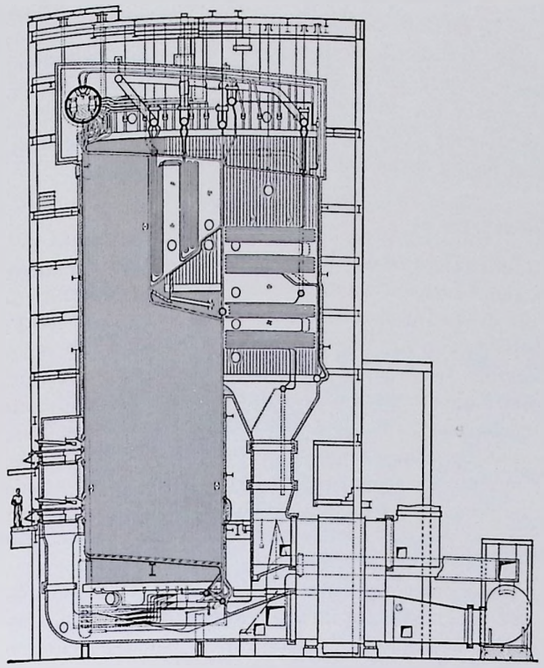


Fig. 14. *Top-supported unit*

this purpose, flexible metal bellows are used in flues and ducts; metal hoses and sliding or toggling gasketed couplings in piping. Water seals are generally used between ash hoppers or slag tanks and the associated furnace. With large units, the expansion may be as much as 7 inches between adjacent parts, yet pressure-tight joints must be provided.

Support

Steam-generating units are supported either from 1) the bottom on steel columns or concrete piers or 2) the top by hanger rods suspended from overhead steel. The bottom support is more economical, but its application is limited by the structural design. Wall tubes and the tubes between upper and lower drums act as columns in compression. There are definite limits to the height of the unit and to the weight that can be supported. Fig. 13 illustrates a bottom-supported unit.

With a top-supported unit, Fig. 14, the hanger rods must be designed for the direct tensile stress resulting from the weight of the unit and the bending tensile stress from the pressure-part expansion that deflects the hanger rods horizontally from the vertical or cold-load position. A common support elevation is normally used, and the effect of temperature on the several lengths of rods must be considered in order to obtain a safe load distribution among them. The use of water-heated rods, steam-heated rods, or compression springs may be necessary to accomplish this.

In the walls of a top-supported unit, the tubes are carrying the lower sections, and the load stress in the tubes must be added to the pressure stress to determine the total stress in these members. The load stress must include the weight of the unit filled with water, the weight of ash and slag, and the weight effect of any other loads that may be imposed on the walls.

Wind loads on outdoor units and earthquake loads for certain geographical areas must be taken into account. On bottom-supported units, the wall enclosures must be of sufficient strength to withstand horizontal forces on the structure. Top-supported units are tied to the building steel at various elevations to prevent pendulum action from overstressing the hanger rods.

The larger the unit, the more involved is the support problem, and the analysis of stress conditions under all operating loads is an intricate and complex procedure. "Rule of thumb" methods which were used in the past are no longer adequate.

Explosions

In the design of settings, the effect of explosions must be taken into account, to eliminate the possibilities of injury and serious damage. Historically, the rupture of pressure parts in boilers was a serious menace. Such disastrous explosions have been largely eliminated by better understanding of the technical problems and the development of adequate design and operating codes. More recently the firing of large units with fluid or fluidized fuels has introduced a new hazard from explosive mixtures of fuel and air within the setting. Such explosions may develop pressures as high as 50 psi in certain parts of the setting. It is not feasible to design the structure to withstand bursting pressures of this magnitude. The enclosure is regularly designed to withstand common puffs and minor explosions which may develop pressures in the order of 2 psi. In the event of a major explosion, the design should provide for failure of studs, stud attachments, and welds rather than the tube walls themselves. This will minimize the danger of releasing large quantities of steam in the event of a furnace explosion.

The loading from furnace puffs and from normal operating negative or positive furnace pressure is contained by the use of bars and channels welded to tubes to form continuous bands around the setting (Fig. 10). Beams, called buckstays, are attached to the tie bars with slip connections and keep the walls from bowing inward or outward.

Since the buckstays are outside the insulation, special corner connections are required (see Fig. 15) that will allow the walls to expand and at the same

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time tie together the corners, where the force of an explosion is concentrated. These corner connections must be tight during starting-up periods when the walls have not fully expanded, as well as at the normal operating fully-expanded position. Furnace puffs caused by incorrect fuel-and-air mixtures are often associated with starting-up operations.

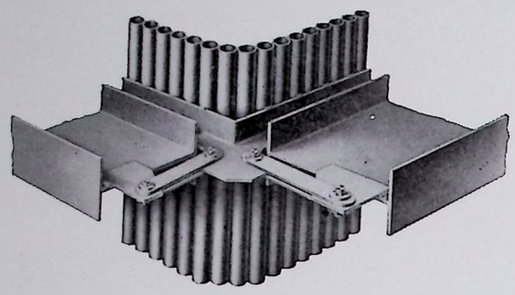


Fig. 15. *Tie bar and buckstay arrangement at corner of furnace*

The tube span between the buckstays acts as a beam to resist the internal furnace pressure. The larger the tube diameter and the heavier the tube wall, the farther apart the buckstays may be spaced. The size of the buckstay beam usually is determined by the permissible deflection.

Explosion doors are now seldom used to relieve excessive internal furnace pressure. They are not feasible except for very small furnaces, because the internal pressure from a fuel explosion builds up rapidly and cannot be significantly relieved by the opening of a door. Actually, explosion doors may be more of a hazard than a safety feature because, in the event of a puff, they may discharge hot gases that would otherwise be completely contained.

Vibration

Excessive vibration in boiler settings can cause failures of the insulation, casing, and supports. This vibration can be produced 1) by external rotating equipment, such as turbines and fans, and transmitted to the setting through building steel, piping, flues, or ducts; 2) by furnace pulsations from the uneven combustion of the fuel; and 3) by turbulence in the flowing streams of air or gas in flues, ducts, and tube banks.

The walls, flues, and ducts are designed to limit vibration to low values for normal operating conditions. For the walls, the section modulus of buckstays is usually selected to limit wall deflection at its midpoint to 1/16 in. with a pressure change of 1 in. of water. Flues, ducts, and casings are similarly stiffened by bars or structural shapes to prevent excessive vibration. This stiffening is particularly necessary in sections of flues and ducts where the flow is highly turbulent, as in the fan-discharge connecting piece. Every effort should, of course, be made to eliminate the sources of severe vibration, such as unbalanced rotating equipment, poor combustion, and highly turbulent air or gas flow.

Heat Loss

In the days when all boilers had solid brick settings without furnace water-cooling, the outside temperature of the furnace walls was often as high as 350 F. In such surroundings, firemen were far from comfortable. On most of these boilers, if insulation had been added to the walls to improve working conditions and to reduce heat loss, the temperature of the brick would have risen sufficiently to shorten its useful life. Therefore the usual way of approaching comfortable conditions was by good boiler-room ventilation, but this was not economical since it increased heat flow through the walls and lowered boiler efficiency.

Water-cooled walls can be covered with any desired thickness of insulation without materially changing the interior temperature. It is usual practice to make the insulation thick enough to assure comfortable working conditions. This results in a reasonable balance between the dollar saving in fuel and the cost of the insulation. The heat conservation and comfort requirements, of course, can be satisfied with practically any arrangement and type of insulation if it is of the correct thickness and if the material will withstand the operating temperature.

Insulation. Materials used in the insulation of water-cooled walls must have a reasonably low conductivity that does not increase in service; sufficient mechanical strength to withstand shipping, application, and handling; resistance to moisture; low shrinkage when heated; good resistance to disintegration or settling by vibration, even after prolonged service at operating temperature; and the ability to retain structural strength at operating temperature.

The basic materials most frequently used in heat insulation and some of the commercial forms of insulation made from these materials and their use are listed below:

1. Diatomaceous-Earth-Base Blocks. A silica composed of the skeletons of simple organisms, uncalcined or calcined, with asbestos fiber and clay molded under heat and pressure. Used principally for boiler walls behind firebrick, in temperature zones from 1200 to 1800 F.
2. Mineral Wool. Molten slag, glass, or rock, blown into fibers by steam or air jet or spun by high-speed wheels.
   a) Mineral-Wool-Base Block. Mineral-wool fibers and clay, molded under heat and pressure. Temperature limit up to 1800 F (claimed by some manufacturers).
   b) Mineral-Wool Blanket. Mineral-wool fibers compressed into blanket form, held in shape by retention between hexagonal wire mesh or expanded metal lath. Temperature limit normally 1000 F. Used on all types of enclosures with external metal lagging or casing.
3. Lime Silica Block. Reacted hydrous calcium silicate. Used on enclosures and piping usually below 1200 F.
4. High-Temperature Plastic. Insulating cement made of mineral-wool fibers processed into nodules and then dry-mixed with clay. To apply, it is mixed with enough water to give good troweling consistency. In final dried condition it is a tough fibrous monolithic insulation. Drying shrinkage is as much as 40%, and there is a tendency to crack upon drying. Temperature limit (controversial) up to 1800 F is claimed by some manufacturers. Used principally on boiler walls, irregularly shaped valves and fittings, and heated tanks.

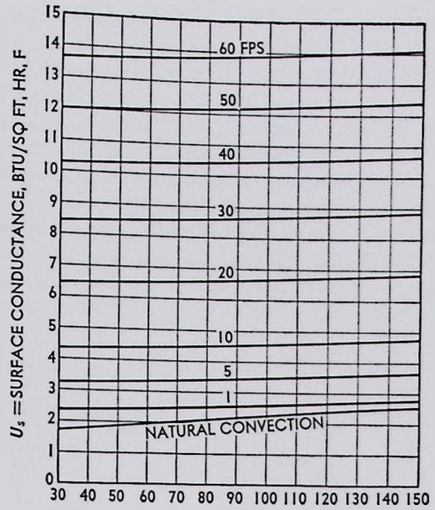
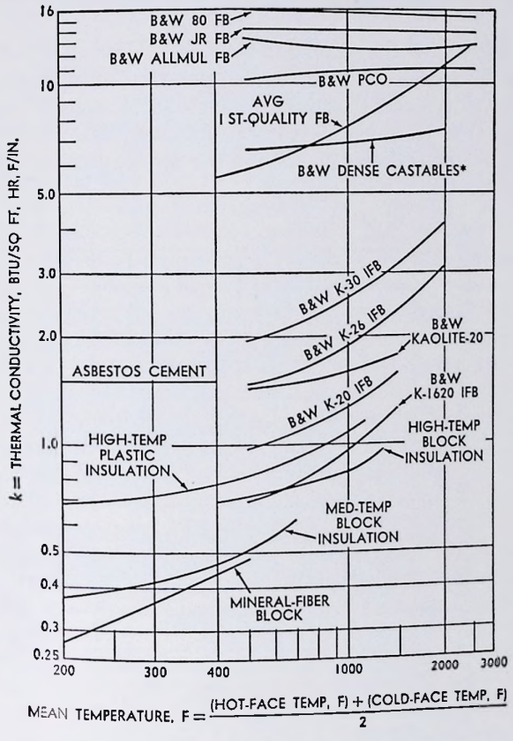


Fig. 17. *Surface conductance at various air velocities (Manual of ASTM Standards on Refractory Materials, 1948)*



*AVG OF KAOCAST & KAOCRETES A, B, & D.

Fig. 16. *Thermal conductivity of various refractory materials*

Heat Loss Calculations. The thermal conductivities of a wide range of commercial refractory materials, at the temperatures for which they are suitable, are given in Fig. 16; surface conductance data in Fig. 17.

The calculation of the over-all heat loss from a steam-generating unit is time-consuming and requires judgment and experience in the estimates and in trial-and-error methods. Generally, the ABMA radiation-loss chart, shown in Fig. 20, Chapter 7, is useful. For detailed calculations of the total heat flow through a composite wall, see example on page 7-3 of Chapter 7.

Ventilation and Surface Temperature

To maintain satisfactory working conditions around a boiler, the insulation must be thick enough 1) to keep the outside surface temperature of the wall reasonably low, and 2) to prevent excessive increase in the boiler-room temperature by heat loss through the wall.

Obviously, insulating a boiler to reduce the heat loss to a value that can readily be absorbed by the total volume of room air does not, in itself, assure comfortable working conditions. Good air circulation around all parts of the boiler is necessary to prevent the accumulation of heat in the areas frequented by the operating personnel. This can be aided by substitution of grating for solid floors, by ample aisle space between adjacent boilers, by the location of forced-draft fans to assist the circulation of air around the boiler, and by the addition of ventilating equip-

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ment to assure adequate air change. Ventilation is, of course, no problem on outdoor units.

Fortunately on modern units good ventilation has little effect on the over-all heat loss. Air velocity affects the surface conductance, as shown in Fig. 17. However, this surface conductance is only a small part of the total resistance to heat flow. For example, an increase in air velocity from 1 to 10 fps, for the conditions given in Fig. 18, will increase the heat loss through the wall only 3 Btu from 144 to 147 Btu/sq ft, hr. This is approximately 2% for a tenfold increase in air velocity.

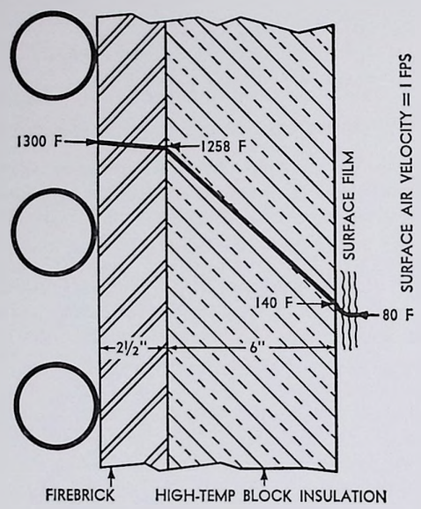


Fig. 18. *Temperature gradients through tube-and-brick wall*

Unlike heat loss, outer surface temperature is affected considerably by surface conditions. This is illustrated by Fig. 19, where two walls of the same temperature are close together. With this arrangement, radiant heat transfer from the wall is eliminated, and the conductance of the surface film, by convection only, is 1.22 at 1 fps, as determined by the equation: $U_c = 1 + 0.225V$, where V is the surface air velocity in fps. The surface resistance, the reciprocal of the conductance, is 0.82. This resistance would have been 0.42 had both radiation and convection applied. As noted in Fig. 20, this change would raise the surface temperature from 144 F to 194 F, even if good ventilation existed in the cavity. In practice, the natural circulation of air through the cavity would be inadequate, and the air temperature would increase considerably, further increasing the surface temperature.

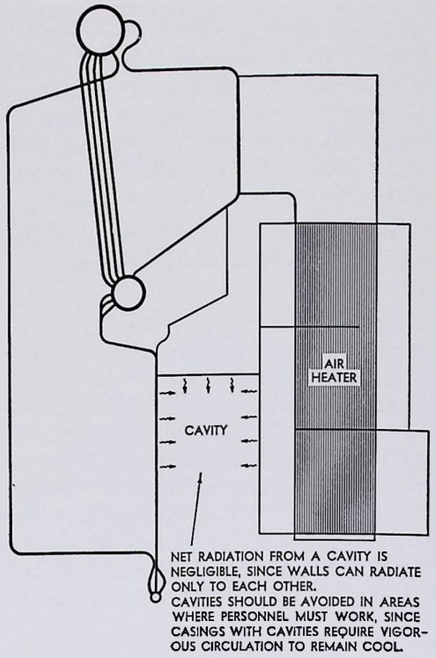


Fig. 19. *Cavities tend to raise wall-surface temperature*

Increasing the insulation thickness does not significantly reduce the surface temperature in a cavity. Therefore cavities should be avoided in areas where operators work. If this is impossible, ventilating ducts should be installed to reduce the air temperature in the cavity.

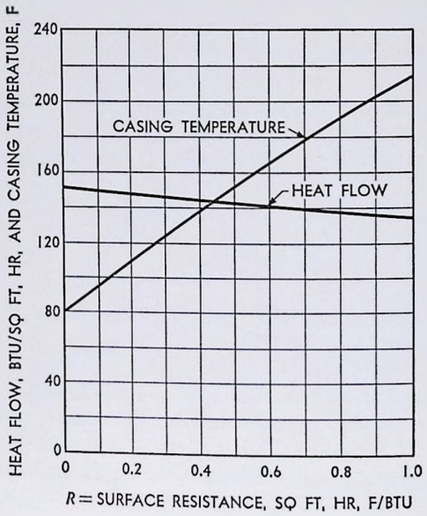


Fig. 20. *Surface resistance has little effect on heat flow but a marked effect on casing temperature (for wall construction and hot-face temperature of Fig. 18)*

Leakage

Continuing efforts have been made over the years to reduce air infiltration into boiler settings. Leakage increases 1) the heat rejected to the stack, thus lowering the thermal efficiency (see *Combustion Calculations—General*, Chapter 4, page 4-7), and 2) the induced-draft-fan power because of the increased gas flow. Despite the best efforts at sealing expansion joints and cracks in brick settings, leakage occurs in amounts as high as 20% of the theoretical air required for combustion.

With the advent of water-cooled walls and welded outer casings, leakage was reduced to 10% approximately. This tight steel casing is shown in Fig. 9. In some cases, particularly during periods of steel shortage, outer plastic finishes with asphalt mastic seal coats have been used over the insulation (see Fig. 11). These uncased units are about as tight as the units with outer steel casing. As an indication of the savings that can be effected, setting repairs on one large utility unit reduced air infiltration 10%, lowered exit gas temperature 20 F, and cut the fuel bill by 1%.

The constant goal of higher efficiency led to the development of pressurized firing. Leakage is thereby reduced to a minimum. The cost of fans and fan power is lowered considerably by the elimination of the induced-draft fan. An all-welded casing, directly behind the tube enclosures with the insulation on the outside of the casing (Fig. 10), was developed for pressurized firing. Since the casing temperature is approximately the same as the temperature of the tube walls, all-welded seals around openings can be used and the pressure contained. For rotating soot blowers and some inspection doors, forced-draft-fan air at a pressure higher than the internal pressure is used to seal the openings against gas leakage and to provide some cooling. This seal air usually amounts to 1 or 2% of the theoretical air required for combustion.

The same type of inner casing is also used for many suction-fired boilers to obtain the benefits of minimum leakage, which is in the order of 1 to 2% of the theoretical air required for combustion.

A rigorous test is made to check the tightness of the inner casing of a pressurized unit. The setting is pressurized to about 20 in. of water, and the unit is "bottled up." Welds are examined for air leakage, and the rate of pressure drop within the setting is noted. A rate of 5 in. of water pressure drop in 10 minutes is usually considered acceptable depending upon the size of the unit. Inner casings on suction units are also tested under air pressure, not by the pressure drop method but by a detailed inspection of all welds.

Gas leakage through walls separating zones of different pressure must be prevented. Such leakage could overheat inner or outer casings as well as other structural parts. Walls that become saturated with ash and sulfur may cause corrosion of outer casings. Enclosures around headers and drums may accumulate large quantities of troublesome ash. A good seal directly behind the tube wall is required to prevent these difficulties. All openings through walls must be sealed from the walls by tight sleeves. Barriers in walls may be necessary to prevent gas flow between zones of different pressure.

A recent development in gastight casings is the membrane wall, shown in Fig. 21. In this construction, tightness is accomplished by welding a bar between the tubes. This actually eliminates the casing and many of its accompanying problems. Insulation is applied directly to the tubes, and metal lagging is attached to give the outer surface durability and good appearance.

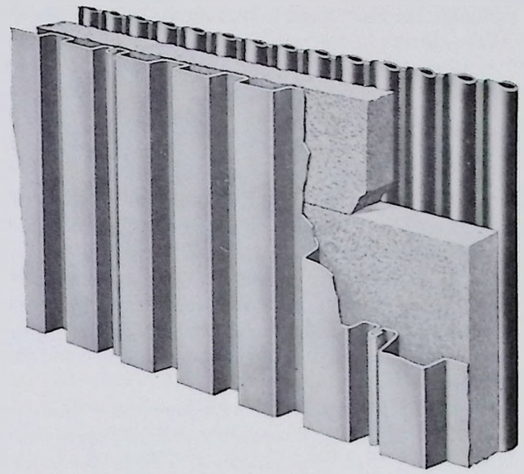


Fig. 21. *Membrane wall construction*

Corrosion

Most flue gases contain sulfur, and metal parts of the setting must be kept either above the dew point of the gases or out of contact with the gases if the metal is below the dew point. The latter ranges generally from 150 to 250 F and is dependent upon the fuel, its sulfur content, and the method of firing.

To prevent corrosion, flues carrying low-temperature spent gases are either insulated on the outside or refractory lined on the inside. This is particularly necessary on outdoor units. Water-cooled doors and slag-tap coils require water temperatures in excess of 150 F to keep the cooling coils above the dew point.

With the reduction of air infiltration into settings, outer-casing corrosion became a problem because of more intimate contact between gases and casing. Despite good refractory seals and barriers, tight outer

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casings, particularly on outdoor units, are subject to rapid corrosion unless asphalt mastic coatings are used to protect the casing on the inside.

With the use of inner casing, corrosion problems are greatly reduced, since the flue gases are completely contained by a metal "skin" that is well above the dew point. Even with the inner casing, however, care must be exercised properly to insulate seals and expansion joints to avoid cold spots and the consequent corrosion.

Resistance to Weather

In recent years, outdoor boiler installations have gained in popularity, particularly in mild climates. Although the initial cost is reduced, maintenance of the boiler and auxiliary equipment must be considered. Severe weather can extend outage time and increase maintenance costs. These units must have sufficient reinforcement to withstand the pressure and suction forces of the wind.

It is relatively simple to make a metal-lagged unit rainproof. Joints and flange connections are overlapped, and flashings are used around openings. Welding of joints or the use of mastic compounds is necessary in areas difficult to seal.

Sloping roofs are required and are particularly important on aluminum lagging, as pockets of water would eventually stain the surface. Direct contact between aluminum and steel must be avoided to prevent galvanic corrosion of the aluminum in the presence of moisture. Copper lines or roof flashings should be so designed that water runoff does not wet the aluminum.

All outdoor jobs should have the connections between the water-gage assembly and the drum insulated, except for approximately 4 ft of the upper connection adjacent to the assembly. This area is left uninsulated to provide for circulation of the condensate. Water columns should also be insulated. In climates where freezing weather may be expected, the drain lines for the water-gage assembly must be insulated and protected from freezing. The water gage and its illuminator will provide ample warmth within the water-gage housing.

Weather hoods should be used to keep rain, snow, and ice from contact with outdoor safety valves. Nozzles and valve necks must be insulated and covered with sheet metal or other waterproof covering.

Outdoor control lines containing air or flue gas, drain and sampling lines, and intermittently operated steam and water lines should be insulated and protected by electric-resistance-heating wires. In some cases steam pipe tracer lines may be substituted. Dry air should be supplied for control lines and soot blowers. It is important that all steam and water lines outside the setting be completely drainable.

Fabrication and Assembly

The setting must be designed for economical fabrication and field assembly. This requires integration of all shop and field methods and practices. Small units can be completely shop-assembled. For larger units the trend has been toward shop subassembly of large components.

Shipping clearances limit the size of shop-assembled wall panels to approximately 10 ft wide and 65 ft long. Shop assembly of components permits better quality control of the more complicated parts, such as burners or Cyclone Furnace throats.

Advances in welding methods and the development of shop machinery have greatly influenced the design of enclosures. Flues, ducts, expansion joints, and inner casings are welded. Tube connections to headers, tie bars, doors, and other attachments are generally welded. New and improved materials and attachment methods are reducing the man-hour requirements for insulating boilers and installing metal lagging.

Serviceability

Many setting-design details must be considered to simplify operation and maintenance. Working areas around the unit should have adequate lighting and comfortable temperatures. Clearances for servicing and removing parts should be provided. Access through the setting is necessary for inspection of boiler internals. Suitable platforms for access doors, soot blowers, instruments, and controls are essential.

Inspection doors permit observation of combustion conditions and the cleanliness of heat-absorbing surfaces. They facilitate good operation and should be safe and easily opened. Fig. 22 illustrates an inspection door for a pressurized setting. Safety is provided by two types of interlocks which insure that compressed air is properly aspirating the aperture before the door is opened. A feature of this door is that the aspirating jet does not restrict the comparatively wide view angle.

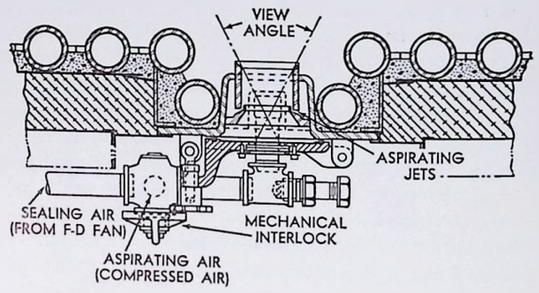


Fig. 22. *Inspection door for pressurized furnace*

The tube bends that form openings in high-duty furnaces must be of the smallest possible radius. In some cases die-formed tube bends are used. The length of the stud-plate closures around the opening is thus minimized, so that the plates can be adequately cooled by welded contact to the tubes.

Fig. 23 illustrates a 15-in. x 21-in. center-pressure type of access door for entering the setting. This door is easily opened and closed and does not require exact centering. By applying the pressure to its center through a yoke and hand wheel, the door is made tight against high pressures. In temperature zones above 2200 F, the door is equipped with a water-cooled coil buried in the refractory lining. A refractory lining is satisfactory for temperatures from 900 to 2200 F, while an insulated liner suffices in temperature zones below 900 F.

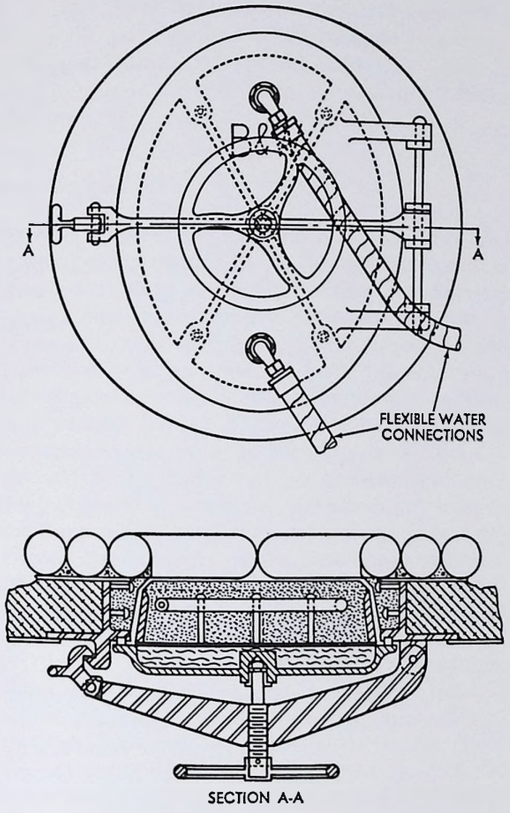


Fig. 23. *Water-cooled access door*

Handhole openings in water-wall headers are necessary for inspection and the flushing after boiling-out and acid cleaning. Gasketed stem-type handhole fittings are used in headers designed for pressures and temperatures up to and including 1200 psi and 850 F, respectively (see Chapter 25, Fig. 21), while welded plugs are used for higher pressures and temperatures. Fig. 24 illustrates a seal-welded handhole plug. This type is called a master handhole plug, because it fits into its own hole and has projections that engage the counterbored seat to withstand the internal pressure.

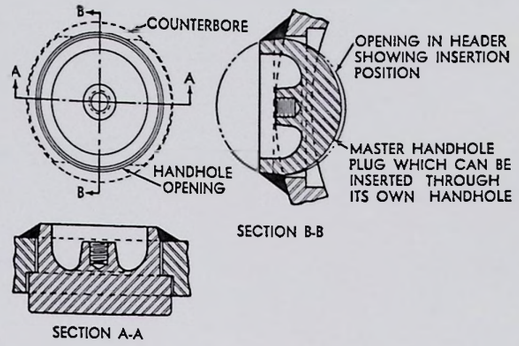


Fig. 24. *Master handhole plug*

Appearance

The setting should present a good appearance initially and be designed so that a good appearance can be maintained indefinitely with a minimum of housekeeping. The outer surface should be of a type that is easily cleaned. Equipment handling flue gas, coal, ash, and oil should be designed to contain these materials without leakage.

The preference for type of outer covering varies considerably. Plastic insulation, cement finishes, canvas and asbestos cloth, welded steel casing, various asbestos lumber casings, and a variety of light-gage metal laggings have been used. With the advent of the inner casing, the outer covering of the boiler and furnace walls no longer has the function of sealing against air infiltration. A welded casing is not required. The most satisfactory cover for inner-cased units is a field-fabricated light-gage sheet-metal lagging. This is particularly true for outdoor units, where it is relatively simple to make the metal lagging watertight.

Figs. 10 and 21 show metal-lagged units. Light-gage galvanized-steel or clad-aluminum sheets form the lagging for most areas. Galvanized steel is generally less expensive than the aluminum, but for outdoor units it may be necessary to paint the galvanized steel after weathering, unless the climate is dry. If much painting is required, the clad aluminum may be preferred, since it does not require painting except under very severe conditions. Since the lagging is fabricated at the job site, the cost is less than it would be for metal casings shop-manufactured from drawings.

For indoor units a plastic insulation finish, made

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from high-temperature plastic insulating cement, makes an acceptable surface when low first cost is important. The relatively rough surface is a dirt catcher unless painted. It is more easily damaged than metal lagging, and random cracks sometimes detract from its appearance.

For outdoor units, asphalt mastic seal coats reinforced by hexagonal wire mesh or glass cloth have been used. This construction is noted in Fig. 12. However, maintenance is high, particularly in areas where the mastic must seal against metal parts, such as door frames and buckstays. The cost of mastic seal coat is not sufficiently below that of metal lagging to justify its use for large areas.

15

COAL PREPARATION, HANDLING, AND STORAGE

THE purpose of coal preparation is to improve the quality of the coal or to make it suitable for a specific purpose by (a) cleaning to remove inorganic impurities, (b) special treatment (such as dedusting), or (c) sizing—either crushing or screening, or both.

Economic Factors

The properties and quantities of impurities in coal are of major importance in the design and operation of steam-generating equipment. Although boilers are often designed and equipped satisfactorily to use a wide range of coals, no boiler installation will perform equally well with all types of coal. All coals have certain properties which place limitations on their most advantageous use. These limitations are particularly stringent for many of the older installations.

To suit the limitations of various types of coal-burning equipment in service, specifications covering several important properties of coal are necessary. For certain types of stokers, for instance, a minimum ash-softening temperature, maximum ash content, and maximum sulfur content of the coal are specified to prevent excessive clinkering. For pulverized-coal firing, it may be necessary to specify a maximum ash-fluid temperature for a unit designed for removal of ash as a liquid slag and a minimum ash-softening temperature for a dry-ash installation. Within these and other equipment limitations there is usually a wide range of coals which can be satisfactorily burned in a specific steam boiler, and the choice depends primarily upon economics—which coal will produce steam at the lowest over-all cost, including cost at the mine, shipment, handling, ash disposal, maintenance, and efficiency. To burn a wide range of coals often requires a larger initial investment than would otherwise be necessary. However, fuel cost is usually the predominating factor because it represents so large a part of the over-all operating cost. The fuel bill for a year, for instance, may equal the initial cost of the entire steam-generating unit.

Cleaning to Remove Inorganic Impurities

Because of a general lowering of quality of raw coal in recent years, the need for coal cleaning has increased. Among the factors contributing to the lowering of coal quality are:

1. Depletion of some of the high-quality coal seams.
2. Greater use of continuous-mining machines.
3. Increased production by strip and auger mining.

Use of mechanical methods in mining coal has increased greatly. In 1925, 1.2% of the underground production was mechanically loaded as compared with 69.4% in 1950 and about 85% in 1958. The most important change that has taken place in recent years was the introduction of continuous-mining machines. Three million tons were produced in 1950 by these machines compared with 56 million tons, or 19.7% of the bituminous coal produced in underground mines, in 1958. Auger mining came into use about 1945, and statistics of coal production by this method indicate 1.5 million tons in 1952 compared with 8 million tons in 1959, or 2% of bituminous coal production. In 1925, 3.2% of the bituminous coal was strip mined compared with 24% in 1950 and 30% in 1959.

Growth of mechanical cleaning at bituminous coal and lignite mines in the U.S. has increased at a continuous rate. In 1925 about 4% was mechanically cleaned; in 1940, 22.2%; in 1950, 38.5%; and in 1959, 63.4%. During the past few years the percentage of coal cleaned at underground mines has increased at a greater rate than that cleaned at strip mines.

In the anthracite fields of northeastern Pennsylvania, 51% of the 21.2 million tons of coal produced in 1958 was from underground mines, 32% from strip mines, 14% from culm banks, and 3% from river dredgings. Practically all of the anthracite produced at underground mines in 1958 was mechanically cleaned.

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The effect of cleaning on the properties of coal cannot be generalized because of differences in coal characteristics. For example, the ash-fusion temperatures of one coal may be lowered by cleaning and those of another raised. Some reduction in sulfur content is usually obtainable and is invariably a benefit, as experience indicates that the higher the sulfur content of coal the greater the likelihood of spontaneous combustion in storage and corrosion of conveyors, bunkers, and heat-absorbing surfaces in the lower-temperature zones of steam-generating equipment.

The principal benefit of cleaning is the reduction in ash content. With ash content reduced, shipping costs and the requirements for storage and handling decrease because of the smaller quantity of coal necessary per unit of heating value. Also, an increase in boiler efficiency may be realized because of less fouling of heat-absorbing surfaces.

Cost of cleaning must be balanced against the benefits obtained. Items included in the cleaning costs are: cleaning-plant operating costs, capital charges, the value of the coal in the discarded refuse, and the cost of disposing of the refuse. In 1946 about 25 million tons of refuse were discarded by bituminous coal and lignite cleaning plants. This amounted to 15.3% of the raw coal supplied to the cleaning plants. In comparison, for the year 1958 the percentage of refuse to raw coal increased to 19.3%, when almost 62 million tons of refuse were discarded. Generally the quantity of coal lost in the refuse increases as the reduction in ash content is increased, hence cleaning costs are directly related to the ash reduction obtained. A careful evaluation of the benefits from cleaning is required to determine whether this loss plus the other cleaning costs are justified.

By balancing shipping costs against cleaning costs, an optimum reduction in ash content can be established which will give a minimum delivered cost of coal, neglecting differences in handling costs and performance at the steam plant. Because of freight costs, it is generally found that an increase in the cost of cleaning is justified as the distance the coal is shipped increases. This is illustrated graphically in Fig. 1 for coal from strip-mine operation.

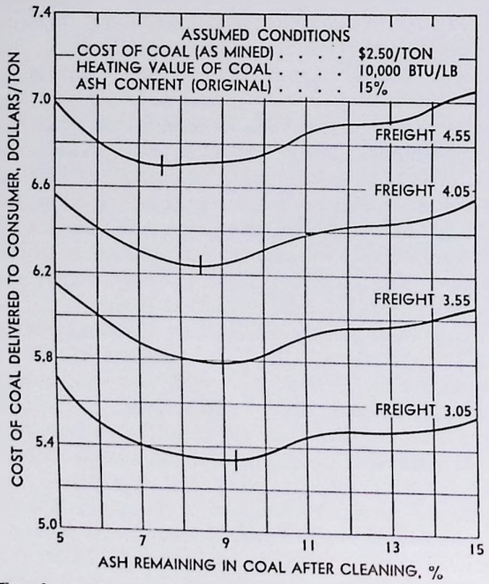


Fig. 1. *Optimum reduction in ash content of coal for minimum delivered cost. An increase in cleaning cost is justified as shipping distance increases*

Nature of Impurities in Coal

As found in seams, coal is a heterogeneous mixture of organic and inorganic materials. Not only are there large differences in the properties of coal from different seams but also of coal removed from different elevations and different locations in a single seam.

Impurities in coal can be divided into two general classifications—inherent and removable. The inherent impurities are inseparably combined with the coal. The removable impurities are segregated and can be eliminated, by available cleaning methods, to the extent economically justified.

Mineral matter is always present in raw coal and forms ash when the coal is burned. The ash-forming mineral matter is usually classified as inherent or extraneous. Ash-forming material organically combined with the coal is considered as inherent mineral matter; this portion came from the chemical elements existing in the plants from which the coal was formed. Generally, the inherent mineral matter contained in coal is about 2% or less of the total ash. Extraneous mineral matter is considered as the ash-forming material that is foreign to the plant material from which the coal was formed; the bulk of this material is from detrital matter which settled into the deposit, crystalline deposits from water that penetrated into the coal through fracture cracks and cleavages both during and after coal formation, or from saline deposits from water before and during formation of peat from which the coal was formed. It consists usually of slate, shale, sandstone, or limestone and includes particles ranging from microscopic size to thick partings. Extraneous mineral matter of mined coal might also include shale, sandstone, clay, etc., from the roof and floor of the mine.

Sulfur is always present in raw coal in amounts ranging from traces to as high as 8% or more. Three forms of sulfur are recognized as occurring in coal. These are pyritic sulfur, which is sulfur combined

Table 1

Chemical Distribution of Sulfur in Various Coals

Location of Mine	Coal Bed	Total Sulfur, %	Pyritic Sulfur, %	Organic Sulfur, %
Washington County, Pa.	Pittsburgh	1.13	0.35	0.78
Franklin County, Ill.	No. 6	2.52	1.50	1.02
Pike County, Ky.	Freeburn	0.46	0.13	0.33
McDowell County, W. Va.	Pocahontas No. 3	0.55	0.08	0.46
Walker County, Ala.	Pratt	1.62	0.81	0.81
Meigs County, O.	8A	2.51	1.61	0.86
Clearfield County, Pa.	Upper Freeport	3.56	2.82	0.74
Boone County, W. Va.	Eagle	2.48	1.47	1.01
Clay County, Ind.	No. 3	3.92	2.13	1.79
Coshocton County, O.	No. 6	4.69	2.63	2.06
Muhlenburg County, W. Ky.	No. 11	5.20	3.20	2.00
Mercer County, N. Dak.	Lignite	1.00	0.38	0.62
Henry County, Mo.	Tebo	5.40	3.60	1.80
Kittitas County, Wash.	Big Dirty	0.40	0.09	0.31

with iron in the form of mineral pyrite or marcasite; organic sulfur, which is sulfur combined with the coal substance; and sulfate sulfur, which is in the form of calcium sulfate or iron sulfate. Sulfate sulfur in coal as mined is usually not over 0.1% and consequently is not of great importance. Table 1 shows the distribution of sulfur in several different coals. Pyritic sulfur in coal may occur in lenses, bands, veins, joints, balls, fossils, or finely disseminated particles; pyrites may vary in size from a few microns to balls several feet in diameter.

Thus, the inherent or nonremovable impurities in coal may be considered to be finely divided pyrite, organic sulfur, and inherent mineral matter.

Moisture, which is inherent in coal, may be considered to be an impurity. It will vary with different ranks of coal, increasing with lower-rank coals, i.e., from 1 to 2% in anthracite to 45% or more in lignite. Moisture that collects on the exposed surfaces of coal is commonly called "surface moisture." See Chapter 2 for reference to the use of the terms "surface," "bed," and "equilibrium" in defining moisture in coal.

Cleaning Methods — General

The principal methods used in the U.S. for cleaning anthracite and bituminous coals are: (a) cleaning at mine face; (b) picking out impurities manually or mechanically; (c) froth flotation; and (d) gravity concentration.

Cleaning at Mine Face

When manual cutting and loading of coal are used, it is possible to minimize the quantity of impurities. In cutting, the miner can more easily distinguish between coal and rock, and in loading he can often discard visible impurities. With mechanical cutting and loading of coal, the miner has much less control over the product loaded; however, it is often possible to follow mining practices that will reduce the quantity of impurities.

Efforts to reduce the impurities loaded in the mine usually result in increased mining costs, hence economics plays an important part in determining the amount of cleaning done. In general, cleaning at the mine face is greatly reduced when mechanical mining methods are employed. The wide use of continuous-mining machines has resulted in further reduction in cleaning at the mine face.

Picking by Hand or Mechanical Means

Hand picking is the earliest method used in cleaning coal at the mine. A shaking table used for hand



Fig. 2. *Shaking table used in hand-picking coal*

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picking is shown in Fig. 2. The run-of-mine coal is screened, and only the oversize material, usually 4 in. or more, goes to the picking table. Refuse from hand-picking tables may run from 50 to 60% coal, and it is common practice to crush the refuse and clean it by mechanical means.

Many mechanical picking devices employ differences in physical dimensions of coal and impurities as a means of separation. Bituminous coal fractures into rough cubes, whereas slate and shale normally fracture as thin slabs. A slot type of flat picker that can be installed on shaker screens, shaking conveyors, and chutes is shown diagrammatically in Fig. 3.

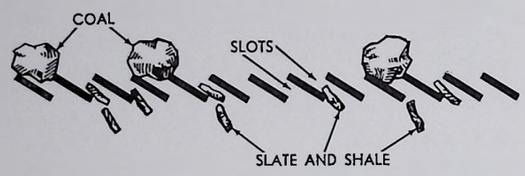


Fig. 3. *Rejection of flat refuse by slot shaker*

Froth Flotation

While froth-flotation cleaning is of long standing for ore separation and has been widely used in Europe for cleaning coal, its application for this purpose in the U.S. is of recent origin. Its use has greatly increased within the last few years. This method can be used in cleaning smaller sizes, $\frac{1}{10}$ in. by 0 and particularly smaller than 48 mesh, a consist for which a market has developed.

The incoming coal feed is agitated in a controlled amount of water, air, and reagents that cause a surface froth to form, the bubbles of which selectively attach themselves to coal particles and keep them buoyant while the heavier particles of slate and shale remain dispersed in the water. The components thus segregated can be suitably diverted to salvage and waste, respectively.

Gravity Concentration

Removal of segregated impurities in coal by gravity concentration is the most widely used method of mechanical cleaning. This method is based on the difference in the specific gravities of coal and the impurities found with the coal. Commercially, the processes used in gravity concentration can be divided into two main classifications—wet and pneumatic. In the first, water is the separating medium, applying the principle of segregation of heavier from lighter particles when settling in a fluid. In the second, generally confined to coal sizes ¾ in. or less, air is used as the separating medium. The specific gravities of bituminous coal and several common impurities are given in Table 2.

TABLE 2

Specific Gravities of Coal and Impurities

Material	Specific Gravity
Bituminous coal	1.12 – 1.35
Bone coal	1.35 – 1.7
Carbonaceous shale	1.6 – 2.2
Shale	2.0 – 2.6
Clay	1.8 – 2.2
Pyrite	4.8 – 5.2

In gravity concentration it is possible to separate the coal into two or more gravity fractions, modified somewhat by size-classification effect. Assuming separation is made at a specific gravity of 1.5, particles with a specific gravity less than 1.5 will be recovered as clean coal and particles of higher specific gravity will go to refuse. The quantity of impurities found in the clean coal and the quantity of coal found in the refuse depend upon the quantity of impurities in the coal, the specific gravity of separation used, and the efficiency of the separation.

In contrast to the values noted in Table 2, the bulk density of crushed coal is much less and is fairly constant even for different size consists. For instance, a sizing of run-of-mine bituminous coals of 60% lump (over 3″), 25% nut (¾″ to 3″), and 15% slack (under ¾″) will weigh between 45 and 55 lb/cu ft. Bituminous coal fines of ⅜″ × 0 sizing weigh about 52 lb/cu ft. Anthracite egg size will run 52 to 59 lb/cu ft, and No. 1 buckwheat will have a bulk density of about 50 to 52 lb/cu ft.

Washability Characteristics of Coal

In the design and operation of coal-washing plants, careful studies are made to determine the properties of the various specific-gravity fractions and how these properties vary with coal sizing.

In most cleaning processes, the low-gravity fractions (below 1.4 sp gr) that represent the relatively low-ash coal and the high-gravity impurities (above 1.8 sp gr) are easily separated. The middlings material (1.4 to 1.8 sp gr) is most difficult to separate.

Where the segregated impurities in the coal are soft compared to the coal, more impurities will be found in the fine-coal sizes. Where the segregated impurities are hard compared to the coal, more of the impurities will be found in the larger-size coal.

There is a general correlation between specific gravity and ash content, although the relation differs for various coals. Ash contents corresponding to various specific gravities for bituminous coals are shown in Table 3.

TABLE 3

Ash Contents of Various Gravity Fractions in Bituminous Coal

Specific Gravity	Ash Content, %
1.3 – 1.4	1 – 5
1.4 – 1.5	5 – 10
1.5 – 1.6	10 – 35
1.6 – 1.8	35 – 60
1.8 – 1.9	60 – 75
1.9 – & above	75 – 90

In actual washery operation it is not possible to realize the results obtained under laboratory conditions of the float-and-sink tests, but data obtained by such tests can be used to rate and control cleaning equipment and to evaluate the quantitative and qualitative efficiencies of the operation.

Cleaning by Gravity Concentration

Wet Processes

In common practice, several well-known types of equipment are used in cleaning coal by the wet process. In recent years four of these, known as (a) launders, (b) jigs, (c) classifiers, and (d) the dense-media method, have been used in cleaning about 92% of all the bituminous coal cleaned in the U.S.

The "Rheolaveur" process is representative of the launder type. It consists of two parts, a short steeply inclined section at the coal-and-water inlet end and a longer more gradually sloped section beyond. A diagrammatic arrangement of a primary Rheo launder is shown in Fig. 4. In passing through the steeply sloped section, relatively high velocities are attained and the particles quickly stratify. In the gradually sloped section, the velocity decreases and the heavy particles settle to the bottom of the trough, where they are removed through two or more Rheo boxes. An important factor in separation is the relatively high coefficient of friction of the impurities as compared with that of the coal, causing the impurities to slide along the trough bottom at a slow speed, the velocity being progressively higher from the bottom to the top of the flowing stream. The difference in velocity between the bottom and top of the stream keeps the impurities below and the coal above the settling velocity; hence the coal is carried through the launder with the water stream. The Rheolaveur launder has a wide range of application for all types of coal.

In jigs, pulsating currents of water pass through a bed of coal resting on a screen plate. At the start of the upward flow of water through the coal bed, the coal is at rest. The upward flow, called the "pulsion stroke," opens up the bed and brings it into a mobile state. The bed is closed and brought to rest during the downward flow of water, called the "suction stroke." Optimum conditions for classification occur at periods during the pulsion and suction strokes, when the mixture is of high density with freedom of movement between particles. Thus the two fractions are separated and finally diverted into separate streams.

Some types of jigs require rather careful sizing of the feed coal for best results, while with others, such as the Baum jig, which take a wide range of sizing, fair results are possible even for the smaller sizes. The water pulsations in the Baum jig are produced by air pressure. The following specific application is typical of the performance of this equipment. In cleaning Roslyn bed coal (from eastern Washington State) on a Baum jig, the ash was reduced from 15.4%

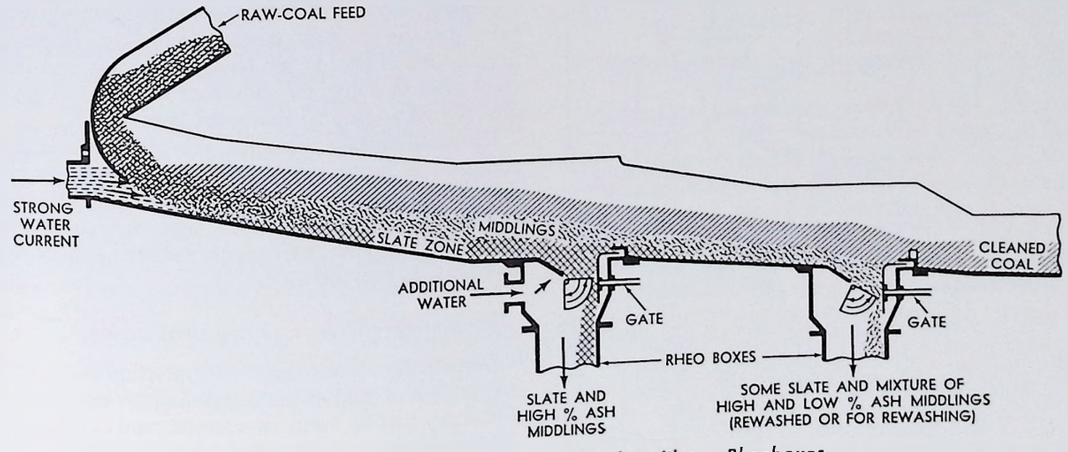


Fig. 4. *Diagram of Rheolaveur coal launder with two Rheoboxes*

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in the raw coal to 10.5% in the washed coal. In this case, 95% of the weight of the raw coal washed ranged in screen size from 20 mesh to 1½ in. and over, and 73% of this weight ranged in size from ⅜ in. to ½ in. and over. Weight of the refuse from the operation was about 18.7% of the raw coal.

In the many forms of classifiers, the principle of free and hindered settling in water is applied to obtain classification. The feed coal is immersed in a flowing current of water (usually upward) with a velocity greater than the settling velocity of the low-gravity fractions (coal) and less than the settling velocity of the high-gravity fractions (refuse), which serves to separate the coal from the refuse. Classifiers are suitable for a wide variety of coals, but efficient operation usually requires careful sizing of the feed coal.

The dense-media method utilizes the principle of the laboratory float-and-sink tests on a commercial basis. Coal is immersed in a medium of specific gravity intermediate between that of coal and of refuse. Hence the impurities sink and the coal floats. A well-known type is the Chance-Cone, shown in Fig. 5. The separating medium is a suspension of fine sand in water. An upward current of water and stirring are provided to keep the sand in the proper suspension to give the specific gravity required for the separation. This method is widely accepted for cleaning anthracite, and its use in the bituminous field is increasing.

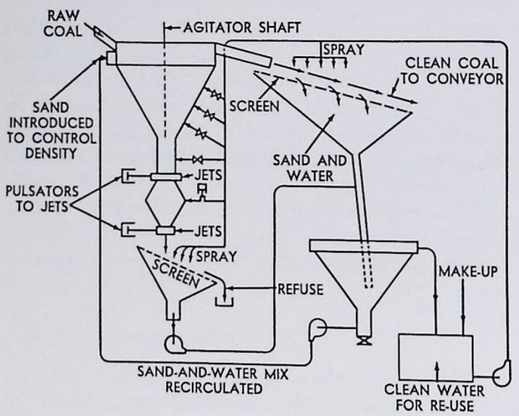


Fig. 5. *Chance-Cone dense-media method of cleaning coal*

Dry Processes

Within recent years, pneumatic or dry processes have been used for 7 to 8% of all bituminous coal cleaned in the U.S. and are generally applied to coal ½ in. or less in size. The raw coal is often screened ahead of the cleaning plant so that the oversize may be cleaned by a wet process and the undersize by pneumatic means. For successful results by pneumatic means, the feed coal should have a uniformly low moisture content.

The principal types of pneumatic-cleaning processes use jigs and tables. Both methods are similar in principle to the corresponding wet processes except that air is used as the separating medium.

In the pneumatic process using tables, air is admitted through holes in the table and is blown up through the bed of coal. The motion of the table plus the air flow segregate the coal and impurities, bringing low-gravity fractions to the top of the bed, whence they move down across the table in a direction perpendicular to the riffles. The heavy impurities settle in the riffles and move along the riffle troughs, discharging at the end opposite the feed. The middlings move diagonally across the bed. This permits separation of two or three products, at desired specific gravities, by the selection of proper points of withdrawal on the table discharge.

Air used in pneumatic-cleaning methods can be passed through cyclone separators and bag filters for recovery of the fines from the coal. The recovered fines can be remixed in the clean coal if their ash content is not excessive.

Special Treatment

Dedusting

Dedusting is a type of air separation, the classification being according to size. Air passed through the coal entrains a large percentage of the fines. The fine coal is recovered from the air with cyclone separators and bag filters.

This process is often employed to remove fines ahead of wet cleaning. The fines, if low enough in ash, may be added to the cleaned coal or disposed of separately. Both wet and pneumatic-cleaning processes provide at least partial dedusting. However, degradation of coal in screening, handling, and shipment after cleaning may increase the percentage of dust so that additional treatment is required.

Increased surface moisture reduces dustiness, and additions of water are sometimes used to prevent dust. To reduce the quantity of water required, small amounts of commercial wetting agents, such as those made from organic derivatives that will reduce surface tension, may be used.

Dewatering—Mechanical and Heat Drying

Dewatering of coal plays an important part in the preparation of coal at the mines. Larger coal (above ⅜-in. size) can be easily dewatered, and occasionally natural drainage is provided by special hoppers or bins, screen conveyors, perforated-bucket elevators,

or fixed screens (radial or bent-sieve screens are finding greater use). However, when lower moisture content is required for the larger-size coal and the fine sizes, mechanical dewatering or heat drying is used.

Mechanical dewatering devices may be shaker screens, vibrating screens, filters, centrifuges, and thickening or desliming equipment. Heat drying is used to obtain low-moisture-content coal, especially for the finer sizes. Various types of heat dryers are rotary, cascade, reciprocating screen, conveyor, suspension (includes flash dryers and venturi dryers), and the fluidized bed.

Practically all heat dryers require some form of dust collector to recover the fines—especially when the smaller-size coal is dried. Inertial-type collectors, such as the centrifugal cyclones or the wet-type collector, may be used. Final cleanup is done with bag filters or a water-spray system. Some bag filters permit temperatures as high as 550 F.

Dustproofing

Oil and calcium chloride are commonly used for dustproofing coal at the mine. When coal is sprayed with oil, the film causes some dust particles to adhere to the larger pieces of coal and others to agglomerate into larger lumps not easily airborne. This treatment, usually applied at the mine, is currently in use for about 43 million tons of bituminous coal yearly, representing about 16% of the cleaned coal. Another 2% of cleaned bituminous coal is treated, for allaying dust, with both calcium chloride and oil. Calcium chloride absorbs moisture from the air, providing a wet surface to which the dust adheres.

The amount of oil required for effective dust treatment depends upon the coal size, its porosity, and possibly its wetness. For a given coal size and dustiness index, somewhat more oil is required for wet coal. Coals of high porosity will require a high-viscosity oil to prevent its absorption into the coal. Small coal sizes will require more treatment than larger sizes. In general, the amount of spray oil used ranges from 1 to 8 quarts per ton of coal.

Effective treatment is accomplished by spraying a stream of coal falling from chutes or loading booms. Well-designed hoods prevent waste of oil and insure better treatment. Larger-size coal may be sprayed when it is on conveyors. In either case, hot or cold oil may be used.

When slack-size steam coal is to be used in high-pressure high-temperature boilers, it is recommended that only oil be applied for dust treatment.

Freezeproofing

The chemical additives usually used to prevent freezing of wet coal during shipment are calcium chloride and rock salt. Mechanically dried or heat-dried coal may not require freezeproofing, but this treatment is customary for slack-size coal. For a given coal, the amount of additives required is dependent primarily upon the moisture content of the coal and the expected low temperature. Table 4 gives approximate quantities of rock salt or calcium chloride required to freezeproof coal.

TABLE 4

Calcium Chloride or Rock Salt to Freezeproof Bituminous Coal

(Lb per Ton of Coal Treated)

Temp, F	Surface Moisture		
	3%	6%	9%
20	3.4	6.8	10.2
10	5.0	10.6	15.0
0	6.6	13.2	19.8
−10	8.2	16.4	24.6
−20	9.2	18.4	27.6

These chemicals are usually applied dry and thoroughly mixed with the coal. In some cases either calcium chloride or salt may be shoveled into the hopper pockets of the coal cars, where the drained water settles.

Spray oil is finding increasing use for freezeproofing coal. It is usually sprayed onto the coal in the same manner as for dustproofing. Occasionally the car hoppers are heavily sprayed with oil. Cost for freezeproofing a slack-size coal usually ranges from about 8 to 15 cents per ton of coal.

Where freezeproofing is required for steam coals that are to be fired in high-pressure high-temperature boilers, the use of oil is recommended. Salt and calcium chloride may cause trouble (see Chapter 18).

Sizing by Crushing and Screening

In America, the earliest practice in the sizing of bituminous coal was to hand-load the lump coal and to leave the small coal in the mine. Until about 1880, when mechanical stokers were introduced, practically all boilers were hand-fired with lump coal, and in early surface-preparation plants at the mines it was the custom merely to screen out the lump coal and to discard the slack. With the adoption of the mechanical stoker, the demand for smaller coal sizes increased, and equipment was installed at the mines to prepare the smaller sizes. The demand for smaller coal sizes again increased with the later development of pulverized-coal firing. As early as 1946, approximately 33% of all the bituminous coal shipped was under 2-in. maximum size. Because of this trend toward

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smaller sizing, the price differential between run-of-mine and slack has been completely eliminated for many coals.

Sizing operations at steam plants are usually confined to crushing and pulverizing. Except for removing large impurities by the use of simple grids, screening at the plant is not generally required. For stoker-fired installations it is customary to specify coal of the proper size to suit the stoker, so that no sizing is necessary at the plant. For pulverized firing, after crushing if necessary, the coal is ground to very fine sizing in pulverizers generally integrated as a part of the firing equipment. Crushers are used for the coarser sizing required for Cyclone Furnaces. When intended for pulverized firing, common practice for steam plants is to specify a maximum coal size with no limitation on the percentage of fines, so that the coal delivered will be suitable for crushing and pulverizing in the available equipment.

To save cost by taking advantage of market conditions, some pulverized-coal or stoker-fired boiler plants have installed crushers to size the coal for pulverizing or, if necessary, for stoker use. If the most economical coal available is a slack that does not require sizing, the crusher may be by-passed. If a larger size, such as run-of-mine coal, is most economical, the crusher is used. In plants where both pulverized-coal firing and stoker firing are used, all the coal supplied may be sized to suit the stokers, with the portion going to the pulverizers crushed if necessary. In this way the arrangement of storage and handling facilities is simplified.

Because of the wide diversity in use and characteristics of bituminous and lignitic coals, no uniformly accepted sizings have been established. The screen sizes generally used in specifications conform to the ASTM Standard D 431, *Designating the Size of Coal from its Screen Analysis*. Standard sizings of anthracite are definitely established, as given in ASTM Standard D 310 (see Chapter 2, Table 15).

Sizing Requirements

Generally acceptable coal sizings for various types of fuel-burning equipment are given in the chapters covering the equipment. The degradation in coal sizing resulting from handling and slacking is an important consideration in establishing sizing specifications for steam plants, where the maximum permissible quantity of fines in the coal is set by firing-equipment limitations.

Crushers and Breakers

Numerous types of crushing and screening equipment are used commercially. Representative types are described below:

Bradford Breaker. The Bradford breaker (Fig. 6) breaks the coal to predetermined size and rejects large refuse and tramp iron. It has a large cylinder rotating at approximately 20 rpm. The cylinder consists of steel screen plates, the size of the screen openings determining the size of the crushed coal. The breaking action is accomplished by dropping the coal. The coal fed at one end of the cylinder is picked up by lifting shelves and carried up until the angle of the shelf permits the coal to drop. Because the gravity force used in shattering the coal is low, the production of fines is small. The Bradford breaker may be used at mine or plant.

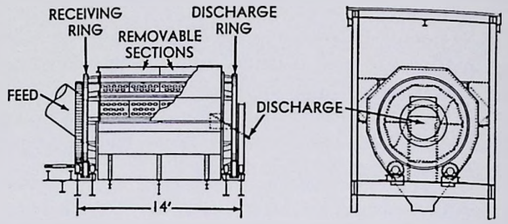


Fig. 6. *Bradford breaker, for use at mine and plant*

Single-Roll Crusher. The single-roll crusher consists of a single roll equipped with teeth designed to force the coal against a plate to produce the crushing action. To prevent jamming by large impurities, such as tramp iron, the roll is permitted to rise or the plate to swing away, allowing the impurities to pass through. The single-roll crusher is one of the older types and is commonly used for reducing run-of-mine bituminous coal to a maximum size of 1¼ to 6 in. Because of the abrasive action between the coal and the plate, the quantity of fines produced is relatively large. The elements of a Jeffery single-roll crusher are illustrated in Fig. 7.

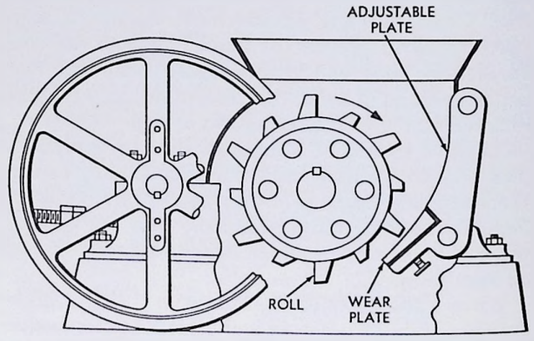


Fig. 7. *Single-roll coal crusher—diagrammatic section*

Double-Roll Crusher. The crushing action is obtained by feeding coal between two toothed rolls. In rotation,

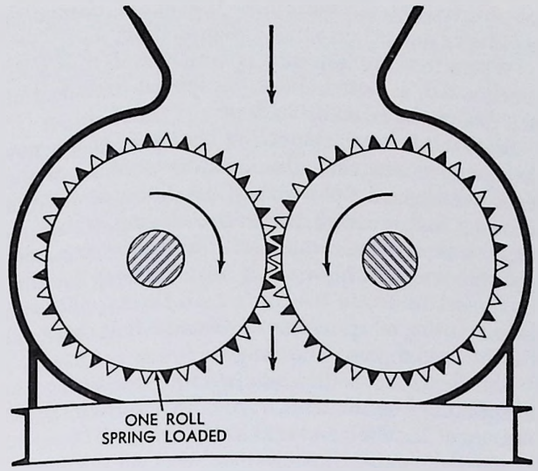


Fig. 8. *Double-roll coal crusher—diagrammatic section*

the faces of both rolls move in a downward direction with the coal. The size of the roll teeth and the spacing between rolls determine the sizing of the coal. Means for passing impurities is provided by spring-loading one of the rolls.

Double-roll crushers are used for reducing run-of-mine coal to smaller sizes at mine-preparation and steam plants. A double-roll stoker-coal crusher is shown in Fig. 8.

Hammer Mill and Ring Crusher. Mills of this type employ the centrifugal force of swinging hammers or rings striking the coal to produce the crushing action. The coal is admitted at the top, and the hammers or rings strike the coal downward. The coal discharges through grate bars at the bottom. The spacing of the bars determines the maximum size of the finished product. Usually a trap is provided for collection and removal of impurities, such as tramp iron.

The crushing action produces considerable fines, consequently these mills are not used when fines are

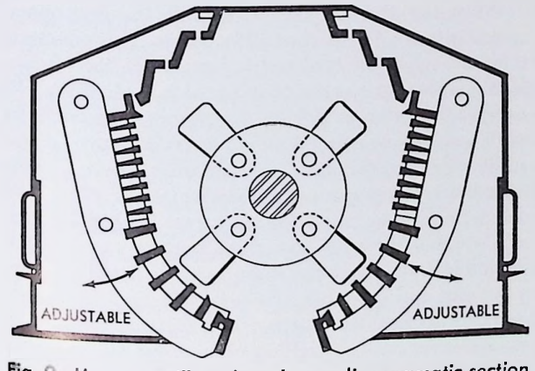


Fig. 9. *Hammer-mill coal crusher—diagrammatic section*

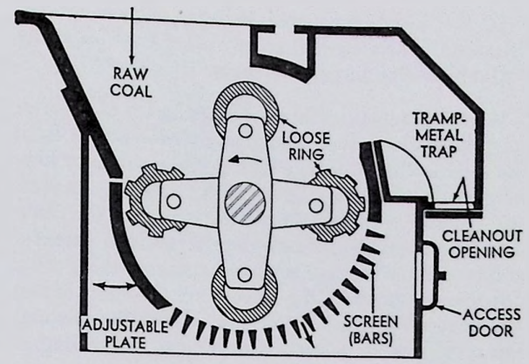


Fig. 10. *Ring coal crusher—diagrammatic section*

objectionable. They are used to reduce run-of-mine coal to sizes such as ¾" × 0 and for recrushing at both mine-preparation and steam plants for sizing feed coal for stokers, pulverizers, or Cyclone Furnaces. A reversible hammer-mill coal crusher is shown in Fig. 9, and a ring coal crusher is shown in Fig. 10.

Screens

Gravity Bar Screen or Grizzly. The gravity bar screen was the first type of screening equipment used in the surface preparation of coal. It consists of a number of sloped parallel bars. The width of the openings between bars, the slope, and the length of the bars determine the separating size. The bars have tapered cross sections with openings between bars smaller on the top side than on the bottom, hence any coal piece entering an opening can drop through freely.

Bar screens are used at mine-preparation plants for production of modified run-of-mine or for removal of undersize ahead of a crusher.

Revolving Screen. A revolving screen consists of a slowly rotating cylinder set with a slight downward slope parallel to the axis. The envelope of the cylinder is a perforated plate or a wire cloth, the size of the perforations determining the separating size.

Because of the repeated tumbling received by the oversize in passing through the cylinder, considerable breakage of coal occurs. For this reason revolving screens are not used for sizes larger than about 3 inches. Since only a small portion of the screen surface is covered with coal, the capacity per sq ft of screen surface is low.

Shaker Screen. A shaker screen is a screen mounted in a rectangular frame. The screen may be horizontal or sloped downward slightly from the feed end to the discharge end. If the screen is horizontal, it is given a differential motion to convey the coal along the screen.

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In the bituminous coal industry, shaker screens are more widely used than any other type for sizing and dewatering larger coal sizes.

Vibrating Screen. Vibrating screens are of gravity-feed type, the screen being sloped downward from the feed to the discharge end. A high-frequency low-magnitude vibration is given to the screen by an electric vibrator or other means. The purpose of the vibration is to keep the meshes clear of wedged particles and to stratify the coal so that the fine particles come down in contact with the screen. For screening of fine wet coal, electrical heating of screen cloth is finding increased use for reduction of binding or packing.

Transportation of Coal

The total cost, the reliability of supply, and the uniformity of the properties of the coal as received by the consumer are vitally affected by the distance shipped and the means of transportation and local distribution.

In the United States, coal is transported by rail, truck, boat, barge, conveyor belts, or a combination of two or more of these methods. Coal is also transported hydraulically in pipelines. Shipping problems differ widely in different parts of the country. For example, a coal consumer in Wisconsin may be using coal that was transported by rail from West Virginia to Lake Erie, where it was reloaded and shipped by boat through the Great Lakes, unloaded, and stored at a dock on Lake Superior. The coal might then be screened to meet a specific requirement, shipped by rail to a local distributing point, and then hauled by truck to the consumer. In contrast to this, a consumer in a coal-producing area might receive coal hauled by truck directly from the mine or receive it via conveyor belt when the mine and plant are within a few miles of each other.

Within recent years the methods of transporting coal have changed. Shipments by barge, truck, and conveyor belts have increased, whereas rail shipments have decreased. In 1957 the Consolidation Coal Co. placed in operation in Ohio a commercial coal-slurry pipeline slightly more than 100 miles in length.

Transportation costs may amount to a significant portion of the delivered cost of coal. Roughly 75% of the coal is shipped by rail, and the freight costs may amount to as much as 60 to 70% of the delivered coal cost. To reduce costs, large coal-consuming plants have been located near coal mines or near rivers where barged coal is available. Electric utilities are now transmitting electricity considerable distances from coal-producing areas to their consumers. These trends are expected to continue, in order to hold down delivered costs of coal or electrical energy.

Transportation may affect the condition of the coal as received by 1) freezing in transit, 2) change in moisture content, or 3) size degradation.

Where freezing in transit is anticipated, it may be ameliorated, as noted above, by special treatment of the coal at the point of loading.

Since most coal is shipped at least part of the way in open railroad cars, the moisture content of the coal will depend upon sizing, condition at time of loading, and weather conditions during transit. It may be received saturated with moisture or so dry that the fines will blow off as dust. To keep the dust down and minimize loss in transit, the supplier can help by using oil sprays and by demanding that tight cars be used to prevent leakage.

Size degradation depends in large measure upon the friability of the coal. It is also affected by the amount of handling and shaking in transit. The extent of size degradation, for instance, is of no moment in pulverized-coal firing. To minimize size degradation for types of firing equipment where size is important, about all that can be done in transit is to reduce handling by avoiding transfer, as far as possible, from one medium to another.

Handling and Storing

The problems involved in handling and storing coal outside and inside the plant of the consumer have increased, one might say, almost in proportion to the increase in the size of boilers. At a small power plant it may be necessary to handle only 30 tons of coal a day, while many of the larger plants use over 3000 tons a day. Intensive efforts have been applied, with fair sucess, to the development of efficient methods for handling and storing a wide range of tonnage. However, while these methods will vary to suit the size of the plant, many of the operations are similar whether the plant is small or large.

Unloading at Consumer's Plant

When the coal is received by rail, the first operation is to unload from the railroad cars. This operation is simple when the coal is dry, because it flows freely. If the surface moisture of the coal is high, it may be necessary to start it flowing by rapping the car sides with a sledge or using a slice bar from above. If this high-moisture coal has been in transit several days in freezing temperatures, it may be frozen into a solid mass, and unloading becomes a real problem. In hot dry weather the coal, on arrival, may be so dry that a high wind will blow the fines away as dust. The supplier can and does help by using oil sprays to settle the dust in hot dry weather and calcium chloride or rock salt for easier handling in freezing weather. The oil treatment does not appreciably affect combustion

in any way and does not cause any trouble in pulverizers. There is some evidence that it tends to reduce hang-ups in bunkers and spouts.

Frozen Coal. The need of special equipment for unloading frozen coal will depend upon the location and size of a plant. For instance, adequate equipment for this purpose must be provided in a plant located where the temperature is below freezing for an appreciable part of the year, while in a plant located where below-freezing temperatures are infrequent and of short duration manual operations will do.

Several successful mechanically operated methods have been developed for unloading frozen coal. For a permanent installation, probably the most reliable and efficient method is a steam-heated thawing shed. However, new installations are seldom made today because they are expensive. Oil-fired thawing pits, arranged to heat the bottom of the cars without flame impingement, give quite rapid and reliable results. When the coal has been thawed at the gates, two devices are available to aid in the discharge of the coal. A heavy shear bar, approximately the width of the car in length, is raised and dropped on the coal by a hoist. The other device is a car shakeout, in which a motor-driven eccentric shaker clamped to the top flanges of the car transmits a vibratory action to the car body. The assembly is hung from a hoist and is readily attached to and removed from the car. The shakeout is also very useful in unloading fine wet coal or other materials that do not flow readily.

For smaller plants with more time available for unloading and where the quantity handled is not large, expensive equipment is not economically justified, and the methods used depend largely upon manual labor. Slice bars, sledges, portable oil torches, and hot water are the tools used, and the effectiveness of the operation depends upon the skill and persistence of the workmen.

Mechanical Handling

Extensive equipment, covering the range of plant requirements from a few tons per day to the largest, is available for handling coal from the cars to either outside storage or to inside bunkers or hoppers. Larger plants are compelled to use mechanical handling. To save money, improve plant appearance and cleanliness, release labor for other tasks, and improve employee morale by reducing the drudgery of manual operation, many small plants have also found it worthwhile to use mechanical handling.

For the very small plants there are portable screw and belt conveyors to unload cars, fill bunkers, reclaim coal from outside storage, etc. Two types are illustrated in Fig. 11.

For plants using above 10 tons of coal per day it is

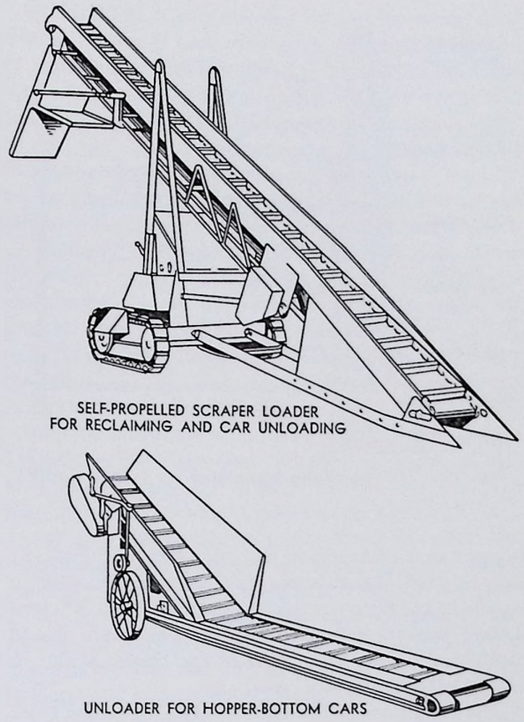


Fig. 11. *Yard coal-reclaiming and -unloading equipment for small plants*

economically necessary to provide complete mechanical coal-handling equipment. The type of equipment depends upon the requirements of the plant. When pulverizers are used, it may be desirable to include a magnetic separator somewhere in the system, but this is usually not considered necessary for most stokers. Frequently a coal crusher will make it possible economically to use locally mined coal that is not screened or sized.

The method of coal delivery, whether by truck or railroad car, may affect the handling arrangements, although a track hopper will serve for either. Some systems are suitable for transferring the coal to either inside or outside storage, while some are limited to one or the other of these procedures. In determining the capacity of the coal-handling equipment, possible future plant expansion should be kept in mind. Practically all systems lend themselves to future extensions provided the original components are large enough. In general, the capacity required depends upon the time allowed for unloading, and in most plants this is set at from 4 to 6 hr a day.

The application of various types of equipment is described as follows:

Where all coal is trucked into the plant, an outside

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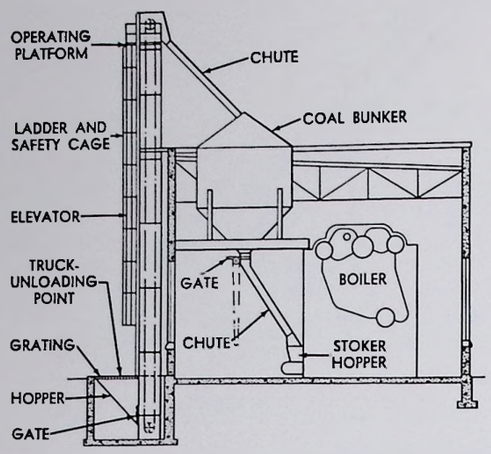


Fig. 12. *Coal-handling equipment for truck delivery*

hopper and elevator to the coal bunker (Fig. 12) may be used. It is a simple effective arrangement for a small- to medium-size plant (burning 30 to 300 tons a day) using a minimum of equipment. Care should be taken to keep the angle of the chutes at 60° or above to minimize coal hang-ups.

The arrangement illustrated in Fig. 13, with a track hopper, feeder, elevator, silo, and weigh larry, has a wide application. The silo is arranged with an internal

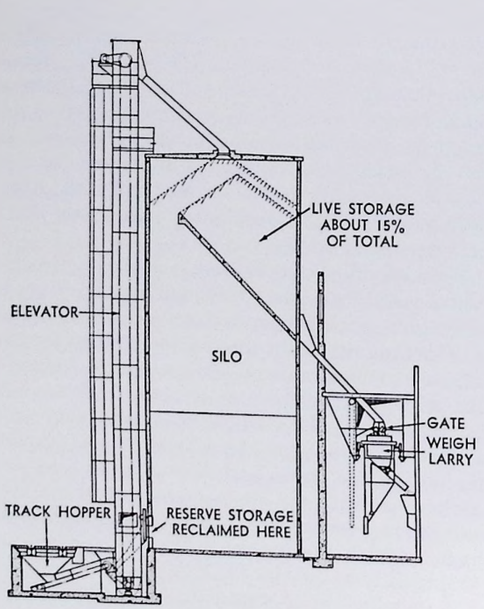


Fig. 13. *Coal-handling equipment suitable for railroad-car delivery. Approximate capacity of silo shown, 600 tons*

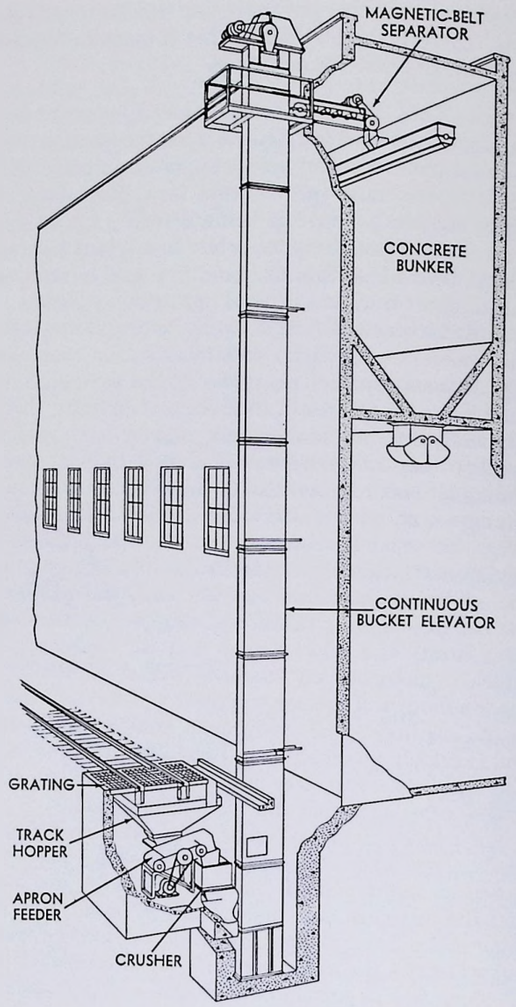


Fig. 14. *Large-plant coal-handling equipment with distribution along length of bunker*

shelf that provides maximum use of storage space. As the coal is used from above the shelf, reserve storage can be reclaimed from the lower part of the silo.

A more elaborate arrangement for larger plants is shown in Fig. 14. This includes a crusher, magnetic separator, and screw conveyor for distribution along the length of the bunker, usually compartmented.

The arrangement illustrated in Fig. 15 is suitable for very large plants, burning 3000 tons per day and over. It includes a car dumper, Bradford breaker, and belt conveyors. In addition to sizing the coal, the breaker will also remove wood and trash. With this arrangement large quantities of coal (200 tons or more) can be moved per hour.

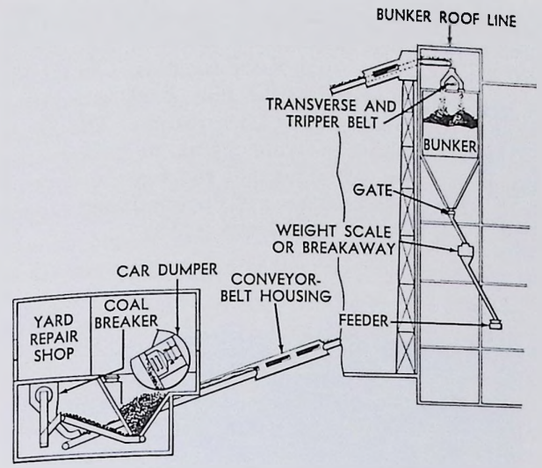


Fig. 15. *Typical coal-handling and -distributing equipment for very large plants*

Outside Storage At or Near Plant

Outside storage of coal at or near the site is often necessary, and in some cases required, to assure continuous plant operation. Coal for an emergency stockpile should be purchased during opportune market conditions. It is important that no more coal be stored than that necessary to meet conditions stated above, because money invested in stored coal cannot be used, changes occurring to the coal during storage may reduce its commercial value, and insurance costs are involved.

The changes that may affect the value of stored coal are loss in heating value, reduction of coking power, reduction of average particle size (weathering or slacking), increase in ignition temperature, and, most important, losses from self-ignition or spontaneous combustion. Reduced caking tendencies may actually improve certain coals for chain-grate or underfeed-stoker firing. Reduction of coking power and sizing will not affect pulverized-coal or Cyclone Furnace firing. Increase of ignition temperature is usually insignificant and has relatively no effect on any firing method. Direct loss of stored coal by wind or water erosion is possible and may, in addition, become a serious nuisance.

Oxidation of Coal. All constituents of pure coal begin to oxidize when exposed to air. This oxidation may be considered a very slow low-temperature combustion process, since the end products, namely, CO_2, CO, H_2O, and heat, will be the same as those of coal burned (rapid oxidation) in boiler furnaces. Although there is evidence that bacterial action causes heating of coal, it is the consensus that heating is effected by a chemical chain process.

Oxidation of coal is affected by the following conditions: length of time exposed to air, coal rank, size or surface area, temperature, petrographic constituents, the amount and size of iron disulfide (pyrites), moisture, and possibly the amount of mineral matter or ash. The rate of absorption of oxygen by coal at constant temperature decreases with time. Higher-rank coals contain less oxygen and oxidize at a slower rate than the higher-oxygen-content lower-rank coals. Oxidation of coal is primarily a surface action, hence the smaller the coal size the greater the surface area and the more rapid the oxidation rate. The latter increases with temperature, doubling for every 20 F rise. The bright components in coal oxidize at a faster rate than the dull. The common banded constituents are listed in probable order of their oxidizability: vitrain or anthraxylon, clarain, durain, and fusain. Pyrites reacts with oxygen and water to form sulfuric acid, iron sulfate, and heat. It is believed that size degradation (exposing more surface) of coal as a result of pyrite oxidation rather than the attendant emission of heat is of greater importance. Moisture content of coal increases with decreasing coal rank. However, the higher oxygen content and greater surface area of very low-rank or young coals are of more importance. Also, the heat of wetting may accelerate oxidation of low-rank coals that had been previously dried. The amount of mineral matter or ash in coal may have a small effect on oxidation, that is, a high-ash coal has less tendency to oxidize than the same coal with a low-ash content, all other conditions being unchanged. The most important factors affecting oxidizability of coal are its rank, surface area, length of exposure to air, and temperature.

It is now possible to draw some general conclusions about storage properties of coal, and these are: high-rank coals, such as anthracite, are easiest to store; fresh small-sized lignite of high pyrite content is the most difficult to store in hot weather; once a safe storage pile is established, the rate of oxidation will have decreased considerably, thus coal should be kept in dead storage until an emergency requires its use; freshly crushed coal has a high oxidation rate; if spontaneous combustion is to be avoided, heat from oxidation of coal should be kept at a low level by retarding oxidation or removing the heat generated thereby.

Protection of bituminous coal or lignite during storage should be considered from the viewpoints of spontaneous combustion and coal deterioration. For example, storing coal loosely in small piles prevents self-ignition, since heat generated by oxidation is released and the temperature of the coal pile does not approach the danger point. However, the coal will suffer loss of both calorific value and coking power as well as size degradation. The safest way to store coal and prevent its oxidation is to submerge it com-

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pletely in fresh water. However, the high cost of underwater storage together with the problems associated with dewatering and handling generally precludes the use of this method. All steam coals may be safely stored, provided proper procedures are used. Coal can be successfully stored by following methods outlined by the Bureau of Mines and practiced by public utilities.

Selection of Storage Site. The first important step in the successful storage of coal on the ground is the selection and preparation of the storage site. If possible, the site should be level solid ground, well drained, and free from all foreign material. Also, natural protection from prevailing winds, tides, flooded rivers that might erode the entire stockpile, and spray from salt water should all be considered, as well as methods of coal delivery and proper access to the site. It should be free of loose fill, such as ashes or cinders, vents from drains, heat from steam or hot-water lines, and even high-pressure gas lines that may rupture and explode. The selected site should be cleared of all foreign matter, properly graded for drainage, and compacted by a bulldozer.

Storage in Small Amounts. No special precautions are required for anthracite.

Sized (double-screened) bituminous coals may be stored in small conical piles from 5 to 15 ft high. If a dump truck is used, the depth should not exceed about 6 ft to maintain better control over segregation.

Slack sizes of bituminous coal may be compacted with a bulldozer. It is not necessary to invest in heavy equipment, since labor and equipment can be contracted for when needed. Some industrial plants may not have sufficient ground available for an emergency stockpile. In some cases an active storage pile may serve both purposes, using a drag-scraper arrangement of moderate cost.

Subbituminous coal and lignite should be stored in small piles and thoroughly packed. For short-time storage (2 to 3 months), where it is undesirable to pack industrial sizes, oil treatment with about one gallon of heavy oil per ton of lignite will permit safe storage in windrows.

Storage in Large Amounts. Anthracite is very safe to store in large high piles that will permit drainage. They should be capped with larger sizes to prevent loss of fines from wind.

Bituminous coal should be stockpiled in multiple horizontal layers (see Fig. 16). The initial layer is spread out from 1 to 2 ft in thickness and thoroughly packed to eliminate air spaces. Well-compacted coal has a bulk density of from 65 to 72 lb/cu ft. Track-type mobile equipment may not give sufficient compaction, and use of a sheepsfoot roller may be required. Rubber-tired self-powered scrapers may be used to advantage where hauls are more than 200 or 300 ft long. When using these scrapers, a bulldozer is required for trimming the pile. Each succeeding layer is compacted in the same manner. The top is slightly crowned and symmetrical, to permit even runoff of water. All sides and the top are covered with a 1-ft compacted layer of fines and then capped with a 1-ft layer of sized lump coal.



Fig. 16. *Coal storage—typical example of thorough packing with minimum of size segregation. This pile contains about 200,000 tons*

Subbituminous coal and lignite should be stockpiled using the same system of layering and compacting, but coal layers should be thinner, not more than 1 ft in thickness, to insure good compaction. It is not practical to seal these piles with coarse coal, since it will weather and slack to small size in a short period of time. The top and sides of the pile are compacted with slack-size coal. Sides should have a shallow slope. To prevent drifting of fines from and around the pile, which might cause trouble from self-ignition, it is recommended that drift (snow) fences be installed across the pile, normal to the direction of prevailing winds, and the top and sides of the pile be periodically trimmed.

Regardless of the coal stockpiled, the use of material, such as asphalt, for airtight sealing is not recommended. Capping methods described above have proved effective. Sealing with asphalt or road-tar coating may be conducive to self-ignition in those areas near any breaks that could cause a "stack effect."

Also, the materials are an additional expense, and they may prove difficult to handle or pulverize when the stockpile coal is used.

Visual Inspection. Visual inspection for hot spots should be made daily for large storage piles and not less than once a week for smaller piles. In wet weather, a hot area is indicated by rapid drying and a lighter color of the surface coal. On a cold or humid day, streams of vapor and the odor of gas are signs of heating or air entering the pile. Hot spots may be located by probing with an iron rod and testing its temperature by hand. If the rod in contact with the coal at the hot spot is quickly withdrawn and found to be too hot to hold in the hand, the temperature is at the danger point. Preventing air from entering the pile, by sealing the top and sides near the hot areas with an airtight coating of asphalt or road tar, will usually stop further heating and actually extinguish the fire. Should this method fail to reduce the temperature, the coal must be removed from the hot spot. Water should not be used except in the case of an actual fire.

Inside Storage and Handling

As the size of single boiler units and the amount of fuel consumed increased, it became more difficult to maintain a steady continuous supply of coal to the burning equipment. Assuming bunkers and chutes of proper size and design, the condition of the coal is the all-important factor in the ease and efficiency of handling inside the plant. Fine wet coal feeding from a bottom-outlet bunker has a tendency to "rat hole" or "pipe" all the way to the top surface. Under these conditions the angle of repose may be 90°. When this occurs, coal flow to the feeders is interrupted and is quite spasmodic. Dry anthracite, on the other hand, often has an angle of repose as little as 15°. Many bunkers and chutes with dents from sledge hammer blows (serving as the only available remedy at the moment) bear evidence of the way fine wet coal has often been made to flow. Casualties, even fatalities, have resulted from men being caught in slides while trying to get the coal moving in bunkers. In the light of past experience it is universally agreed that no one should be permitted to enter a coal bunker to loosen coal hang-ups. Air lances, manipulated from above or below the bunker, are safer and far more effective.

From studies of actual plant operations it has been found that hang-ups of coal are liable to begin when there is a surface moisture of from 5 to 6%. This and other difficulties can best be overcome by improved storage and by careful reclaiming of the coal from storage.

Flow of Coal through Chutes

The following interesting conclusions on the flow of coal through chutes from bunker to pulverizer may be drawn from the results of model experiments, as presented by Wolfe and Van Hohenleiten in a paper entitled *Experimental Study of the Flow of Coal in Chutes at Riverside Generating Station,* published in the ASME Transactions, 1945:

1. The shortest possible path for a chute should be maintained (in the direction of flow).
2. Tapered or uniformly enlarging round chutes should be used whenever possible.
3. Where two streams of coal must be brought together, it should be accomplished with a minimum angle of convergence. If a sudden change of direction cannot be avoided, the introduction of a "breakaway" to help in the realignment of the coal should be utilized. A schematic sketch of such a breakaway is shown in Fig. 17.
4. All angles of chutes should be made as steep as possible, and reductions in cross section should be kept to a minimum.
5. An example of improved layout for bunker discharges, based on the model experiments, is shown in Fig. 18.

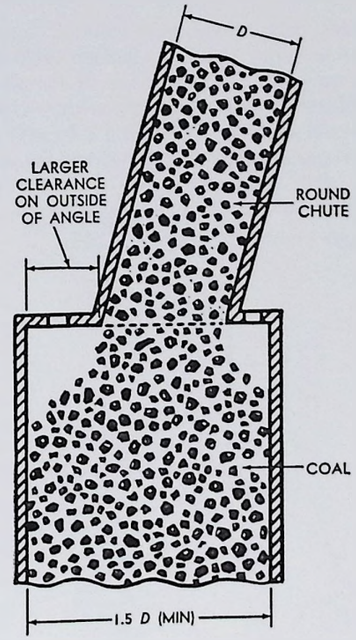


Fig. 17. *Arrangement of "breakaway" in coal chute to relieve lateral pressure and pipe friction*

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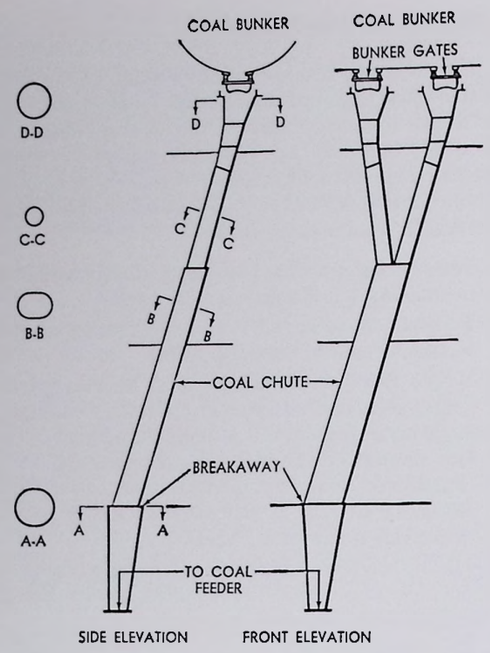


Fig. 18. *Arrangement of bunker discharge with desirable features of breakaways and gradual enlargement of chutes*

Bunker Design

The inside coal bunker, once built, is essentially a part of the building, and there is little likelihood that it will be modified during the life of the plant. The purpose of the bunker is not only to store a given capacity of coal but also to function efficiently as part of the system in maintaining a continual supply of fuel to the pulverizer, stoker, or Cyclone Furnace. In the design of bunkers, careful consideration should, therefore, be given to certain factors, as outlined below:

Capacity. Normally, capacity of the bunker or bunkers should be large enough for a 30-hr supply at top load. For plants where coal-handling labor or equipment is not available for 3-day week ends or where the cost is high because of holidays, it is desirable to increase capacity to cover such a period.

Material. Coal bunkers have been built of tile, reinforced concrete, carbon-steel plate, stainless-steel-clad plate, steel plate lined with acid-resisting concrete, and steel plate lined with rubber. Where unlined carbon-steel plate is used, a periodic check should be made to determine the extent of corrosion. Bunkers have been known to collapse because of corrosion.

Shape. Determination of the shape of a bunker is usually based on a compromise between space limitations caused by the size and arrangement of the building and the optimum design for coal flow. Dead pockets should be avoided because of the danger of spontaneous combustion. Several shapes of bunkers commonly used are shown in Fig. 19. It will be noted that a breakaway is indicated at each bunker discharge. This device is quite effective in reducing hang-ups and also provides access to the coal stream in again starting the flow should clogging occur. Similar breakaways should also be used where there is a material change in the direction of a chute. Many plants use the silo design of coal bunker because, as compared with other shapes, it has been found to be less susceptible to rat-holing and hang-ups. This design is equally suitable for either small or large plants.

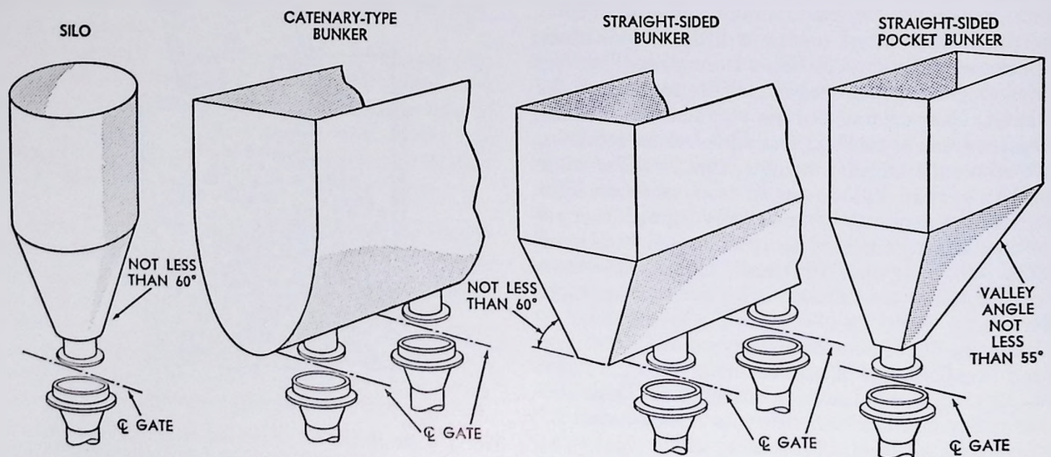


Fig. 19. *Four commonly used shapes in coal-bunker design*

Location. The coal bunker should be so located that the flow to the fuel-consuming equipment is as nearly vertical as possible. It should also be as far away as possible from flues, hot-air ducts, steam pipes, or other external sources of heat that might aid in starting bunker fires.

Bunker Fires—a Serious Danger

A fire in the coal bunker should be recognized at once as a serious danger to personnel and equipment and should be dealt with promptly and adequately. As long as there is fire in the bunker, there is danger of a possible explosion of combustible gases, and steps should be taken to smother the fire. Steam or carbon dioxide may be piped in directly to covered bunkers and to the affected areas in open-top bunkers. While the fire is being smothered with steam or carbon dioxide, the hot coal should be run through the fuel-burning equipment, whether pulverizer or stoker, until the bunker is empty. In feeding the raw coal, special care should be taken to feed uniformly, without interruption, and at as high a rate as the plant load requirements will permit. Continuity of feed at this time is especially important.

If the hot coal from the bunker is fed to a pulverizer, cold air should be used to reduce mill-outlet temperature. Should the feed to the pulverizer stop for any reason, the pulverizing equipment must be shut down immediately, and all valves in connection with the pulverizer should be closed to prevent a draft through the pulverizer. After it has been established by inspection of the pulverizing unit that no fire is present and the cause of the feed interruption has been removed, the unit should again be placed in service until the bunker is empty. The operation after a stoppage should be supervised with the same care as before the interruption. The principal feature in the case of stokers is to maintain a continuous coal flow to the stoker hopper so that the chute is sealed, thus preventing an air flow back into the bunker.

While it is highly desirable to extinguish completely all fire in the bunker before it is emptied, this is rarely possible because of load demands and because it is extremely difficult to do. However, the use of steam or carbon dioxide in smothering the fire will greatly minimize the danger, and it is common practice to continue feeding hot or even burning coal from the bunker at the highest possible rate so that the bunker may be quickly emptied.

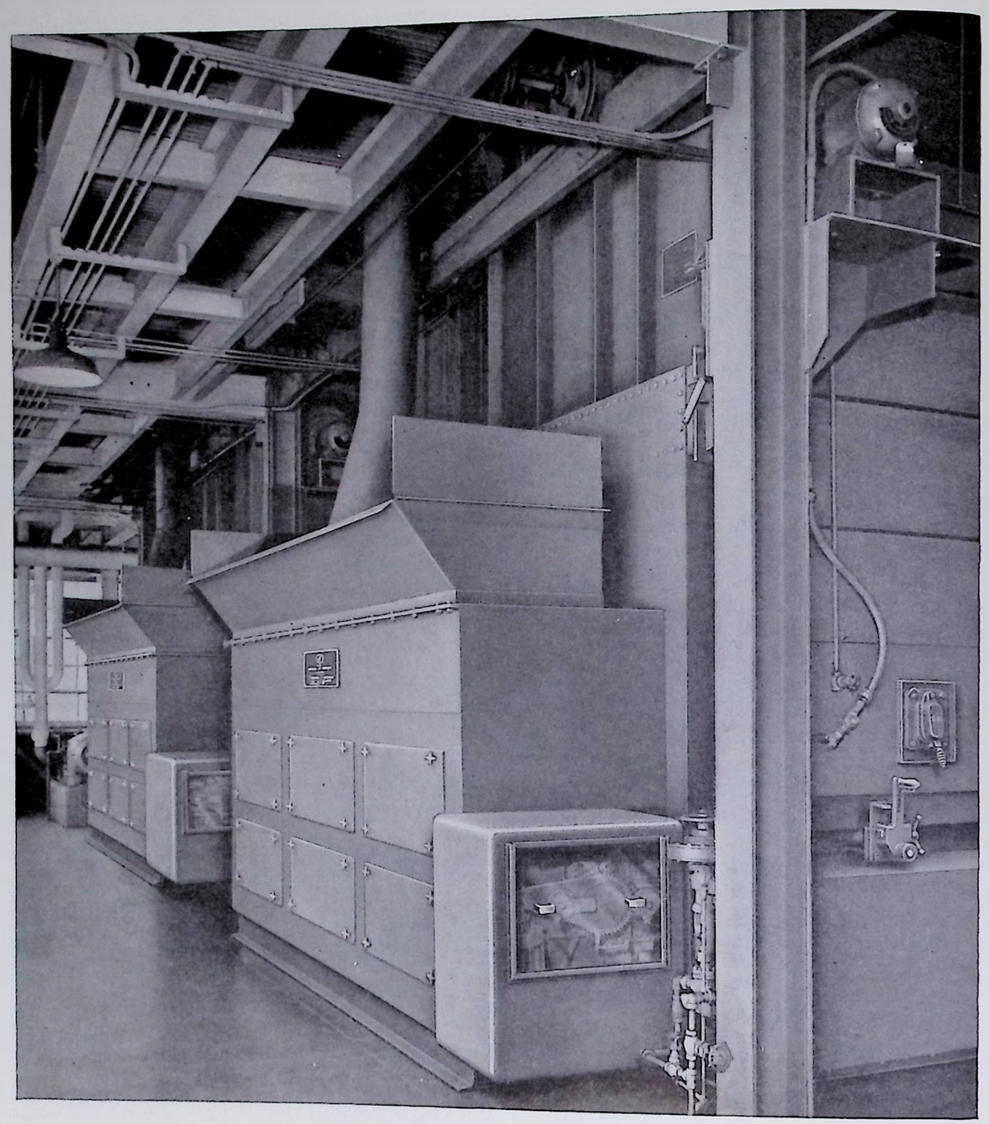
Carbon dioxide is very useful in fighting bunker fires, and in some plants permanent connections for the use of this gas have been fitted to the bottom of bunkers. In other plants Dry Ice, distributed over the top of the fuel in the bunker, has been found quite effective in smothering fires, the heavier carbon dioxide displacing the air.

In a bunker in which a fire has started, all the coal must be worked out before more coal is added, since a few hot pieces will start a new fire. The cause of a bunker fire should be established. The fire may have started from coal already overheated or burning in storage or, more likely, the cause may be spontaneous combustion because of the length of time the coal had been kept in the bunker itself. Dead pockets within a bunker should be eliminated. To reduce the effect of external heat from adjacent steam lines or breechings, insulation should be fitted and ventilation provided around the bunker. Care should be taken to avoid loading the bunker with overheated coal.

Safety of Personnel

No one must be allowed to enter a bunker in which there has been a fire until it is completely empty and the gases generated have been dissipated. Even then, only an experienced person equipped with an oxygen supply should enter the bunker, and he should be attended by helpers outside the bunker to whom he is connected by lifelines.

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JET-IGNITION STOKERS FIRE THIS WESTERN PENNSYLVANIA INDUSTRIAL PLANT

16

STOKERS; OVERFEED, SPREADER, AND UNDERFEED

HAND FIRING of boilers with steam output greater than about 2000 lb/hr is not practical. Historically, the development of the mechanical stoker as an improvement on hand firing began in the early days of the steam boiler, when a patent was issued to James Watt in 1785.

Stokers of various types are all designed mechanically to feed coal uniformly onto a grate within the furnace and to remove the ash residue from the furnace zone. High rates of combustion are possible, and the continuous process of stoker firing permits refined control and high efficiency.

For a successful stoker installation, the stoker selected must be of the correct type and size for the fuel used and the capacity required, and the boiler unit must be equipped with the necessary instruments for its intelligent control. Pulverized-coal and Cyclone Furnace firing are strong competitors of the mechanical stoker, but this is so chiefly because of the comparatively limited capacity of the stoker. Pulverized-coal and Cyclone Furnace firing are suitable for the largest central-station boiler units, with outputs of over 5,000,000 lb steam/hr. The largest stoker reaches its limit at less than one tenth of that figure, and most engineers would limit stokers to not more than 400,000 lb steam/hr. Regardless of type, a stoker requires floor area for its grates, and floor area is limited under any boiler installation.

The mechanical stoker is an important and valued element of modern equipment in the production of steam with small and moderate-size boilers. Almost any coal, from anthracite and coke breeze to lignite, and other fuels, such as hogged wood, bark, and bagasse, can be burned successfully on stokers.

Types of Mechanical Stokers

Most mechanical stokers can be classified in three main groups, based on the direction and method by which the raw coal reaches the fuel bed:

1. Overfeed: chain grate; traveling grate; vibrating and oscillating grate.
2. Spreader.
3. Underfeed.

The trends toward higher-capacity boiler units and lower-quality fuels have had a marked effect on the type of firing equipment used for solid fuels. With the increasing number of large boiler units, the percentage of capacity fired by pulverized coal has increased steadily since 1920. The popularity of Cyclone Furnace firing has grown rapidly since its development, about 1947, because of its ability to handle low-grade low-ash-fusion-temperature coals.

In the small- to medium-size range covered by mechanical stokers, the side-ash-discharge underfeed stoker is used occasionally on small units, but the rear-ash-discharge multiple-retort underfeed stoker has been largely replaced by other types, such as the vibrating- and oscillating-grate stokers. Chain- and traveling-grate stokers are a common method of firing in their size range and are used generally for low-volatile solid fuels, such as anthracite and coke breeze. The spreader stoker dominates the middle to upper size range of stoker-fired units.

Chain-Grate and Traveling-Grate Stokers

The chain- and traveling-grate stokers have assembled links, grates, or keys joined together in endless-belt arrangements that pass over sprockets or return bends located at the front and the rear of furnaces. Coal, fed from the hopper (see Fig. 1) onto the moving assembly, enters the furnace after passing under an adjustable gate that regulates the thickness of the fuel bed. The layer of coal on the grate as it enters the furnace is heated by radiation from the furnace gases and is ignited together with the hydrocarbon and other combustible gases driven off by distillation. The fuel bed continues to burn as it moves along, and, as combustion progresses, the bed becomes correspondingly thinner. At the far end of the travel, ash is discharged over the end of the grate into the ashpit as the components of the grate pass over the rear sprocket or return bend.

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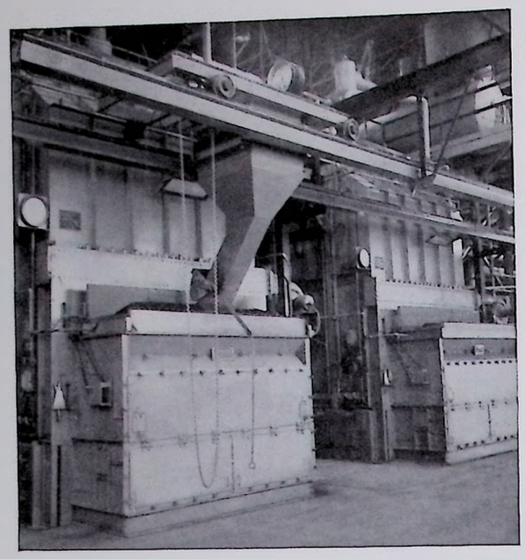


Fig. 1. *Chute and hopper to B&W chain-grate stoker*

This general type of stoker is usually designed for forced draft. Natural-draft designs are not prevalent and are gradually becoming obsolete. Capacity limitations restrict their use to plants where the available stack draft is about 0.8 in. of water and where load demands are fairly steady. Natural-draft stokers have a free air space through the grate of 18 to 22% and operate with a draft loss through the fuel bed of 0.1 to 0.5 in. of water. It is better to use coarse coal (½ in. to 1¼ in.) with few fines, where the available draft is low.

Forced-draft stokers have a free air space through the grate of 6 to 10%, and consequently considerably finer coal (even slack sizing) can be used. A draft loss through the fuel bed of 1 to 2 in. of water is considered normal. Frequently forced-draft stokers are operated at reduced rating on natural draft by opening doors located in the ashpit.

Grate Construction

Chain grates were originally developed for bituminous coals, and traveling grates for small sizes of anthracite. Structural details differ, but the two types perform the same function. However, there is one essential difference in design—the links of the chain-grate stoker are so assembled that they break with a scissorlike action at the return-bend section. This action helps to loosen any clinker which may adhere to the grate surface. In most traveling grates there is no relative movement between adjacent grate sections, hence coals with clinkering-ash characteristics are not suitable.

Chain Grate. In the chain-grate stoker, links of steel and chrome-cast iron are staggered and joined together by steel rods to form an endless chain (Fig. 2).

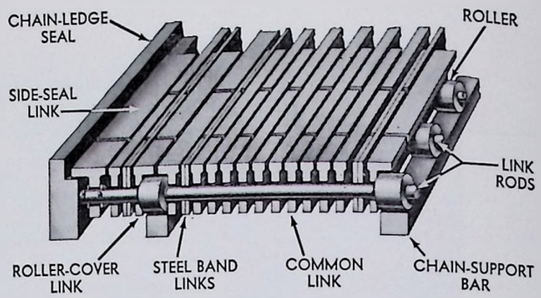


Fig. 2. *Chain and sprocket drive—chain-grate stoker*

The rods, secured with suitable cotter pins at each end, extend across the width of the stoker. Fig. 3 shows a typical sectional view of a chain-grate installation. The driving sprockets are at the front of the stoker, and a return bend is located at the rear. The upper portion of the chain supporting the fuel bed is carried on bars secured to the steel work, which forms part of the air compartment. These bars are continuous from front to rear and are arranged to offer the least interference to air flow through the grates. As the chain returns underneath to the front of the stoker, it is supported by a drag plate. Front and rear drag seals form a closure for the air compartment between the cross girders and the chain.

Traveling Grate. In the traveling-grate stoker, the grate sections are supported and driven through a carrier chain and are under no driving load. The entire assembly is supported on steel rails.

Aside from differences in grate construction, the distinction between chain-grate and traveling-grate stokers is nominal. The following details common to both will be covered by references to the chain-grate stoker.

Stoker Drives

An hydraulic drive unit motivates the grate through a ratchet-and-pawl mechanism applied to the front end of the stoker. The power unit may be a constant-speed motor or a steam turbine.

Arches over Stoker

Front and rear arches over chain-grate stokers were designed to improve combustion. They reflected heat onto the fuel bed, and the front arch also served to break up and mix rich streams of volatile gases that might otherwise travel through the unit unburned.

In 1954, B&W developed the jet-ignition stoker

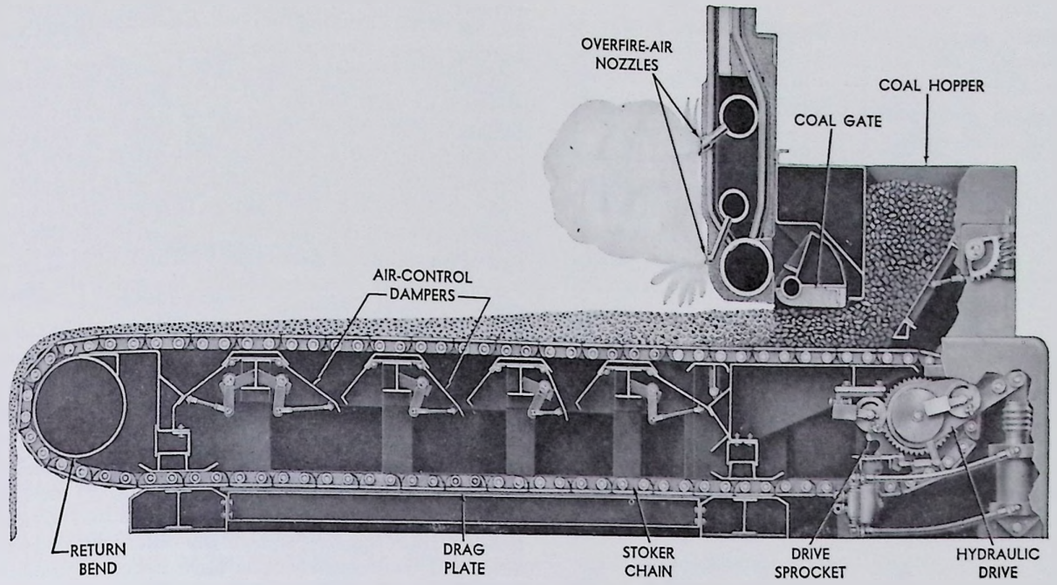


Fig. 3. *B&W jet-ignition chain-grate stoker*

(Fig. 3), which eliminated the need for a refractory front arch. The two rows of overfire-air jets located in the front wall are particularly effective in completing the combustion of rich volatile gases over the first two compartments, which are blasted more heavily than is permissible with an arch. A typical jet-ignition-stoker installation for bituminous coal firing is shown in Fig. 4.

Anthracite and coke breeze fuels, which lack volatile constituents, conventionally require fairly long rear arches to deflect and direct incandescent fuel particles as well as combustion gases toward the front of the stoker, where they assist in the ignition of the incoming raw fuel. This arrangement is shown in Fig. 5, where the rear compartments are regulated as heavy blasting zones. Air admitted to the stoker compartments under the front arch must be regulated to prevent the discharge of carbon particles toward the rear and into the ashpit.

Air Distribution

The quantity and the control of air to the various sections of the stoker are very important. Ignition begins on the surface or in the top layer of the raw coal as it enters the furnace. As distillation increases, burning becomes active until ignition has progressed downward through the bed to the grate, after which overfeed-type burning takes place, carbon is consumed, and only ash remains.

The stoker is zoned or sectionalized and is equipped with individual zone dampers to control air pressure and quantity to the various sections as the fuel travels

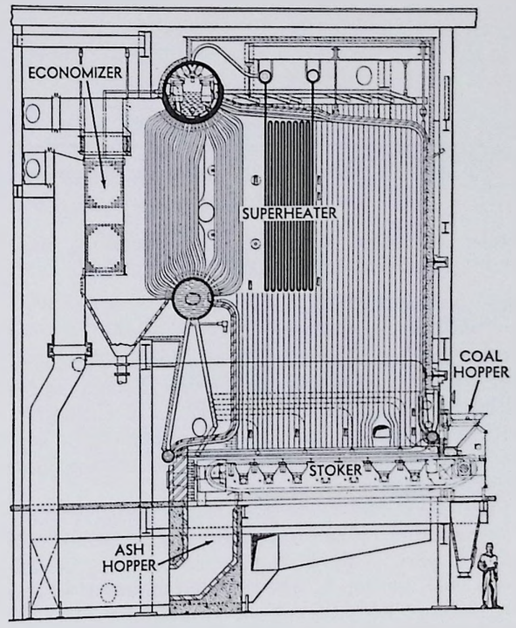


Fig. 4. *Jet-ignition-stoker installation for bituminous coal*

along the stoker length from front to rear. Air for combustion enters the compartments through the side, as shown in Fig. 3.

Air infiltration or leakage into any furnace will adversely affect the process of combustion and the

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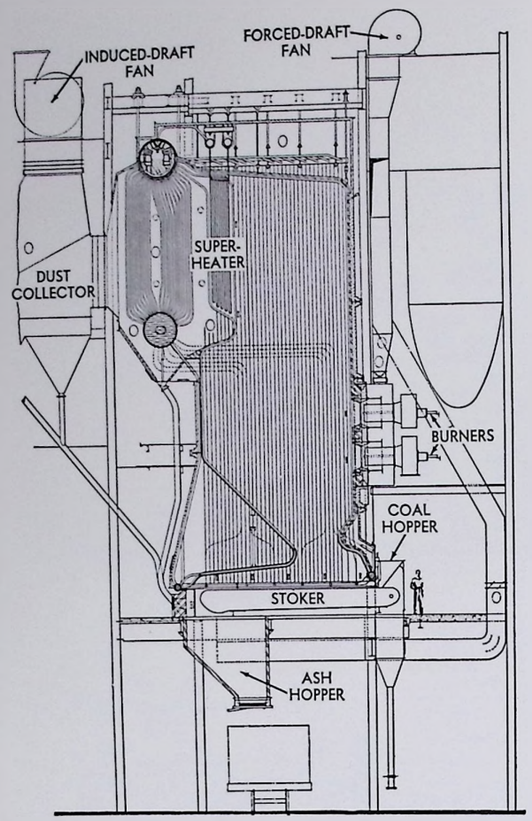


Fig. 5. *Jet-ignition-stoker installation for anthracite and coke breeze*

boiler efficiency. For optimum performance, tight settings and seals are essential.

Ignition

Ignition is the beginning of combustion. For chain-grate firing, the ignition rate must be in step with the combustion rate. In the operation of a chain-grate stoker, the ignition and combustion rates can be kept in phase by controlling the grate speed and the thickness of the fuel bed.

Ignition of fuel on the grate results chiefly from the absorption of radiant heat, although conduction also plays a part. With chain-grate stokers firing bituminous coal, the top of the entering fuel bed is heated by radiation and liberates gas by distillation. This gas first ignites in the atmosphere above the bed but soon flashes back to the bed itself, which continues to give off gas by distillation. Part of this gas burns directly on the surface, but much of it, because of an insufficiency of oxygen, persists as hydrocarbon or carbon monoxide gas until it encounters enough oxygen to complete its combustion. With the jet-ignition stoker this oxygen is supplied by the overfire-air jets, which are positioned to burn the volatile gases in the area just above the incoming fuel bed. Radiation from these burning gases effects stable ignition.

A particle of coal submerged below the top surface receives its heat mostly by conduction, with some radiation from the combustion of adjacent particles previously ignited. Distillation then starts in the submerged layer, and the combustion cycle is repeated.

As already noted, the ignition of low-volatile fuels — anthracite and coke breeze — is aided by the deposition of incandescent particles of carbon, blasted from the rear portions of the fuel bed to the top of the green fuel entering the furnace.

Furnace Construction

An all-refractory furnace does not permit the maximum burning rate; consequently, most stoker-fired furnaces have some water-cooling. Completely water-cooled furnaces are illustrated in Figs. 4 and 5. Such designs give reduced maintenance and greater freedom from slagging so that the older refractory or air-cooled wall constructions have become obsolete. The bituminous-coal-fired boiler shown in Fig. 4 utilizes a short section of water-cooled refractory at the front wall, while the arches for the unit firing low-volatile fuel, Fig. 5, are completely refractory covered.

Fuel-Bed and Raw-Feed Control

Fuel-bed thickness on a chain-grate stoker is determined by the gate position, which is adjustable in service. Normally a given coal will burn most successfully at a particular gate opening. A large-size fuel requires a thicker bed than the smaller-size fuels. For bituminous coal, a thickness of 5 to 7 in. is normal; for small-size anthracite (Nos. 3 and 4 buckwheat), the corresponding thickness is 3½ to 5 in.

Fuels—Wide Variety Suitable

Chain- and traveling-grate stokers can burn a wide variety of fuels. Almost any solid fuel—peat, lignite, subbituminous, free-burning bituminous, anthracite, and coke breeze—of suitable size can be burned on these stokers. Strongly caking bituminous coals may have a tendency to mat and prevent proper passage of air through the fuel bed, with the result that unburned carbon is discharged to the ashpit. Also, a fuel bed of strongly caking coals may not be very responsive to rapidly changing load. However, experience has shown that the following procedures will minimize difficulties with strongly caking coals:

1. The coal may be tempered by adding moisture, either water or wet steam. This makes the fuel bed more porous and permits better burnout of carbon.
2. The coal should be of optimum size. Removal of some of the minus ¼-in. fines from the raw coal will usually permit freer passage of air through the fuel bed.
3. The tendency of coal to swell and cake upon heating may be reduced by weathering. In weathering, coal is exposed to air, and the oxidation that takes place, at a slow rate, will have a tendency to reduce the swelling or caking power of the coal. The stockpiled coal should be checked periodically, especially during the first few weeks after it is placed in storage, to insure against overheating from rapid oxidation.

The jet-ignition stoker is superior to the conventional chain- and traveling-grate types. It was developed to expand the range of suitable coal to include the medium- and high-volatile bituminous coals of strong caking tendencies, such as Pittsburgh Seam coal. Caking and matting are reduced by rapid initial ignition of the coal entering the furnace followed by rapid propagation of the ignition plane through the fuel bed. The jet nozzles produce a turbulent swirling motion of the gases, assuring good mixing and rapid combustion of the volatiles. This concentrated heat zone radiates to the incoming fuel, providing quick ignition. For the strongly caking coals it has been found advisable to limit the amount of fine coal. At least 20% but not more than 60% should pass a ¼-in. screen. Tempering moisture is also used when necessary.

Fuel-Size Limitation

Fuel sizes best suited for the chain-grate stoker are as follows: Bituminous coal should pass through a 1-in. ring, with not more than 60% through a ¼-in. screen. The size for anthracite varies from No. 2, which is all through $\frac{5}{16}$ in. and over $\frac{3}{16}$ in., to No. 5, which is all through $\frac{3}{64}$ in. Coke breeze should pass through ⅜ in., with not more than 20% through $\frac{3}{32}$ in. Subbituminous coal and lignite should be the same size as bituminous coal.

Rates of Burning

Chain- and traveling-grate stokers are offered for a maximum continuous burning rate of 425,000 Btu/sq ft, hr (equivalent to 34 lb of 12,500-Btu coal) with high-moisture (20%) high-ash (20%) bituminous coal, such as that from some districts in Illinois, and 500,000 Btu/sq ft, hr (50 lb coal) with lower-moisture (10%) lower-ash (8-12%) bituminous coal, such as that from Ohio. For anthracite the corresponding rate is 350,000 Btu/sq ft, hr (28 lb coal).

Strongly caking coals are not suitable for the conventional chain- or traveling-grate stokers but may be burned at high rates on the jet-ignition stoker. For this stoker, maximum burning rates are based on heat input per foot width of grate and a coal-rank index, expressed as Btu/lb coal divided by the percentage of volatile matter. Higher index numbers indicate higher-rank coals. The lower the index, the higher the permissible burning rate for bituminous coals. Experience has shown that the free-swelling index of coal is an unsuitable criterion for the establishment of maximum burning rates.

In all cases, coals of low ash content (under 6%, dry basis) do not have sufficient ash residue to protect the grates from overheating and are generally unsuitable for chain- and traveling-grate stokers.

Air Temperature

The use of preheated air on all types of mechanical stokers is not an unmixed blessing, and the chain-grate stoker is no exception. In stokers, some bituminous coals that burn without caking when the combustion air is delivered at room temperature show definite caking tendencies when the air temperature exceeds 250 F. However, it is possible to use 350 F air with some bituminous coals and 450 F with anthracite and coke breeze. The benefits from the use of preheated air are an accelerated ignition with greater flexibility in load and an increase in over-all boiler unit efficiency.

If the preheated air temperature is too high, grates may overheat and warp. Further, with the high velocity of hot air through the grate openings, as is required with fine coal sizes, there is a tendency to carry-over. Nevertheless, the use of preheated air with the chain-grate stoker continues, and with operating refinements and improved stoker design there is a definite net gain in performance.

Ash Discharge and Disposal

Ashes are carried to the rear of the stoker and are dumped into the ashpit as the chain passes over the return bend. From the pit the ashes may be raked out by hand or removed by various types of equipment, such as a water sluice, a drag-conveyor chain, or a sump pump; or they may be dumped directly into a disposal car and quenched. Coal ash from chain-grate stoker is often salable to the building industry for use in special concrete and in the manufacture of cinder blocks.

Change-Over to Oil and Gas

Chain-grate-stoker furnaces are often arranged for auxiliary firing with oil and gas. The oil or gas burners must be so arranged that the flame does not impinge on furnace walls or grate surfaces. A 4 to 5-in. layer of ash on the chain will serve as sufficient protection for short periods of operation. For permanent operation with the auxiliary fuels, it is recommended that the grate be protected with two loose courses of firebrick so laid that all joints are staggered. If there is

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frequent change-over from coal to the auxiliary fuels, care should be taken to prevent coal ash from clogging the gas or oil burners. Normally this is done by admitting a little air through the idle auxiliary burners. For permanent stoker operation, throat plugs are installed in the idle burners.

Stack Discharge—Carbon Loss

The fuel bed of a chain- or traveling-grate stoker firing bituminous coal suffers less disturbance than that of any other type of stoker. While this may be a disadvantage in excluding caking coals as suitable fuel, it is an asset in reducing cinder carry-over, which in many instances is below the requirements of dust ordinances (ASME standard: 0.85 lb/1000 lb of flue gas corrected to 50% excess air), and a dust collector is not required. The carry-over in proper operation with good coal in common sizings is indicated in Table 1. At the same time, the combined combustible loss up the stack and to the ashpit can be very low.

For small-size anthracite, however, particularly Nos. 4 and 5, the carry-over is relatively high. Small burning particles tend to be dislodged from the fuel bed by the strong blast through the grate. Some of these particles fall back on the surface of unburned coal and thereby aid ignition, but some tend to remain in suspension and are carried along with the flue gases. When firing small-size anthracite, dust collectors are often installed, and the unburned particles are pneumatically returned to the furnace.

Response to Load Changes

With favorable coal sizing, a properly designed chain-grate stoker will operate smokelessly from 10% to full load. This very desirable performance results from the effectiveness of the overfire-air jets and the quiescent state of the fuel bed. Generally speaking, if the fuel bed of a chain- or traveling-grate stoker is maintained in suitable condition for fluctuating-load service, a good response to load swings over a wide range is possible without smoke. When there is a sudden demand for an increase in fuel, ignition can be lost if the coal is fed too rapidly. For a drop in load, the principal difficulty is the danger of overheating the grate.

For operation with rapidly swinging loads, the fuel bed is carried somewhat longer than that with steady-load conditions. Usual practice is to reduce the rate of burning at the second and third compartments by lowering the blast pressure slightly in this zone and increasing it slightly in the later compartments. In other words, maximum blast may still be in the second or possibly the third compartment, but there will be more fuel left to burn in the fourth and fifth, and consequently for these compartments the pressures must be increased. Chain or grate speed is usually determined by ignition at the front end.

Combustion Control

No particular problem is involved in the efficient control of a chain-grate stoker, and good combustion control is possible both manually and automatically. The coal-gate opening, which controls the fuel-bed thickness, is usually adjusted by hand, but the forced-draft air pressure and the grate speed are regulated by steam pressure when automatic combustion control is provided. Distribution of air among the compartments under the stoker grate is adjusted manually. The correct air-fuel ratio or air-flow steam-flow should be indicated or regulated by suitable panel instruments, such as a Boiler Meter or O_2 or CO_2 recorders.

Good Practice and Low Maintenance

For chain-grate-stoker operation, maintenance cost is usually low. However, the cost may be increased if operation is not based on sound practice. The chief elements of good practice are outlined below.

Unless care is taken to prevent size segregation of the entering coal across the width of the stoker, the links may be overheated with the attendant difficulties of growth and distortion. Strips or areas of coarse coal tend to burn more quickly, and the areas of fine coal tend to mat. Size segregation produces a "ragged fire," and the chain links are liable to overheat because of reduced air flow through the matted portion. Coal segregation at the gate can be avoided by the use of a traversing coal spout that distributes the coal uniformly across the hopper so that the different sizes are thoroughly mixed. Nonsegregating designs of hoppers are also available.

If all the incoming coal is fine, tempering with water or steam may be required to keep the fuel bed evenly porous while distillation and burning take

TABLE 1

Dust Loadings with Chain-Grate Stoker

Coal Sizing	1", 60% through ¼" screen		1½", 20% through ¼" screen	
Burning rate, Btu/sq ft, hr	400,000	500,000	400,000	500,000
Dust loading, lb/1000 lb gas at 50% excess air	0.63	0.75	0.26	0.39

place. Addition of water for low-sulfur coals may be in the form of a spray applied to coal on the belt conveyor to the bunker. However, much of the coal burned on chain-grate stokers is of medium or high sulfur content, and excess moisture will increase the tendency to corrosion of the bunker. For such coals it is better to do the tempering in the downspout or in the hopper with wet steam. When properly applied to coal requiring tempering, the amount of spray water or wet steam used has an insignificant effect on the over-all boiler efficiency.

Air infiltration to the furnace through leaky settings should be avoided, since air so admitted does not pass through the grate and therefore cannot help to keep the chain cool and provide air at the right point for combustion. Air infiltration at the front of a stoker retards ignition.

The cumulative effect of numerous minor instances of link overheating may be reflected in an over-all growth of the chain after extended periods of operation. This condition is easily remedied by removing one strand of links to restore the necessary lateral clearance in the chain. In other cases link wear may require the addition of links to maintain lateral clearances.

Some wear on the link rods and joints and the drive side of the sprocket teeth will be evident after long continued service. To compensate for such wear, all chain-grate stokers are provided with means for adjusting the tension of the chain, and in some it is possible to reverse the drive sprockets to bring the unworn side of the sprocket teeth in contact with the chain. Lubrication of chain links, driving sprockets, and skid bars is neither possible nor necessary, but the shaft bearings should be kept lubricated at all times in accordance with the manufacturer's recommendations.

Operating Procedure

Lighting the Fire. For bituminous coal, the gate opening is about 5 in. Coal is run in to cover the first two compartments. Kindling placed over the coal is then lighted, and sufficient air is admitted to the compartments covered to cause the kindling to burn briskly. Furnace draft is set at about 0.1 in. of water. When the kindling has ignited the coal, the stoker is started and maintained at a speed slow enough to hold ignition at the gate, thus gradually lengthening the fuel bed. Air is admitted to the additional compartments progressively as required. If necessary, the stoker is alternately stopped and started to stabilize ignition during the lighting-off period.

For anthracite or coke breeze, the gate opening is about 3½ in. Coal or coke breeze is run in to cover the grate back to and including the first full compartment under the nose of the reverse rear arch. Kindling placed over the fuel is then lighted, and air is admitted to the compartments covered by the fuel bed. A dead plate usually covers the first compartment, preventing the admission of air to the fuel bed at this point. As with bituminous coal, the furnace draft is regulated, and, after the fuel has started to ignite from the kindling, the blast pressure to the rear compartment covered by the fuel is slowly increased. When the fuel is definitely burning in this compartment, the blast pressure is adjusted so that it is sufficient to throw some burning particles forward to aid ignition. This may not be possible when burning large-size anthracite, such as No. 2 buckwheat, and for this reason a special front arch is used to aid ignition.

Normal Operation. Three variables must be considered in the operation of a chain-grate stoker:

1. Fuel-bed thickness.
2. Grate speed.
3. Air distribution and compartment pressure.

Fuel-bed thickness is determined by gate position. Generally, for a given coal a certain gate opening determined by trial will give best results. This will remain fixed except for capacities beyond the speed range of the stoker. For bituminous coal a 5 to 7-in. fuel bed is common, and for small-size anthracite and coke breeze it is reduced to 3½ to 5 in.

The speed of the chain grate must be so adjusted that ignition is always maintained at the forward end of the stoker. Fire should not be allowed to burn back to the raw-coal gate, and the line of ignition must not be permitted to recede very far from the vicinity of the first compartment.

Air supplied to the various stoker compartments is varied independently or adjusted between compartments by individual manual settings. When compartment dampers are set for a given fuel, normal changes in capacity require only an adjustment of the forced-draft fan or damper and stoker speed. As noted previously, this may be done manually or by automatic combustion control actuated by steam pressure.

Definite rules cannot be set for the pressure in the various compartments. However, experience indicates that the pressures in the compartments follow a pattern in which a general relationship is evident. The actual pressures in the various compartments must be appropriate for the weight of air (indicated by the Boiler Meter or other means) required to burn the coal as disposed on the grate. The following may be taken as a general guide:

For bituminous coal, the highest pressure usually is required in the second compartment, which may be from 1 to 3 in. of water, depending upon load and fuel-bed thickness. The pressure in the first compartment is then set at approximately 30% and the pressure in the third compartment at about 60% of this value. The pressure in later compartments tapers off gradually to zero in the last compartment.

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For small-size anthracite or coke breeze, the air pressure in the first compartment, blanked off with a dead plate, should not exceed 0.1 in. of water. The pressures in the succeeding compartments usually are gradually increased, with the highest (required for the output) in the next to the last compartment. The relatively high air pressure in the later compartments serves to toss some of the burning anthracite particles forward as an aid to ignition.

Banking Periods. In the normal banking of a chain-grate-stoker-fired boiler, the fire is allowed to burn short as the load demand decreases. When the boiler output is reduced to zero or a minimum, the usual procedure is as outlined below:

1. When burning bituminous coal, the gate is raised to an 8-in. or 10-in. opening, and coal is run in for a length of about 4 ft. With anthracite or coke breeze it is not customary to raise the gate above its normal position, but fresh fuel is run in, as in the case of bituminous coal.
2. The forced-draft fan is stopped, and compartment side doors are opened to permit air to be drawn through the chain grate to prevent its overheating and to supply the small amount of combustion air needed for the banked fire. Furnace draft is regulated by adjusting the induced-draft-fan control or the boiler uptake damper. Incidentally, this procedure should be followed whenever the forced-draft fan is shut down.
3. The induced-draft fan is stopped, or the boiler outlet damper is closed to a point just sufficient to carry off the gases from the banked fire and maintain a slight draft within the furnace.
4. During a banked-fire period, fresh fuel is run in from time to time as the banked bed slowly burns, taking care that the fire does not burn back under the gate where it might damage the gate structure. As a precaution, when burning bituminous coal it is well to run most of the coal out of the stoker hopper, retaining only enough to provide a coal seal at the gate. This will prevent an inrush of cold air under the gate and also damage to the gate.
5. When starting up from bank with bituminous coal, the gate is lowered to the normal position. Forced and induced drafts are set for operating conditions. With anthracite or coke breeze, the live bed is run back to the stoker compartment where heavy blasting is normally applied to toss burning particles forward to aid ignition.

Emergency Shutdown. Sometimes it is necessary to take a boiler unit off the line very quickly, as in the case of electrical-load failures during storms; or some unforeseen contingency may suddenly force the boiler out of service. In such emergencies it is necessary to reduce the burning rate on the stoker to zero in the shortest possible time. Under these circumstances it is usual practice to increase the speed of the stoker chain to maximum and, by raising the gate to its greatest opening, to feed as much coal as possible onto the grate, while at the same time shutting off the air supply. The burning and unburned coal thus run over into the ashpit should be removed promptly to prevent damage to the equipment.

Vibrating-Grate Stoker

Practical application of the vibrating-grate principle to overfeed stoker operation was first realized in Europe, where lignite and brown coals are abundant but difficult to burn satisfactorily. The vibrating-grate overfeed stoker has burned these low-ranking fuels with much higher efficiency than had been possible previously and with exceptionally low maintenance.

Introduced in the U.S. in 1954, this stoker has found acceptance in both industrial and institutional plants and has burned a wide range of American coals successfully.

Grate Construction

The vibrating-grate stoker consists of a grate structure supported on equally spaced vertical plates that are free to move back and forth in a rectilinear direction. The grate structure is made up of water-cooled tubes arranged longitudinally between a pair of headers. Cast tuyère blocks are fitted on top of the tubes and are keyed in position. Fig. 6 is a typical sectional view of a vibrating-grate stoker, and Fig. 7 shows its application to a boiler installation.

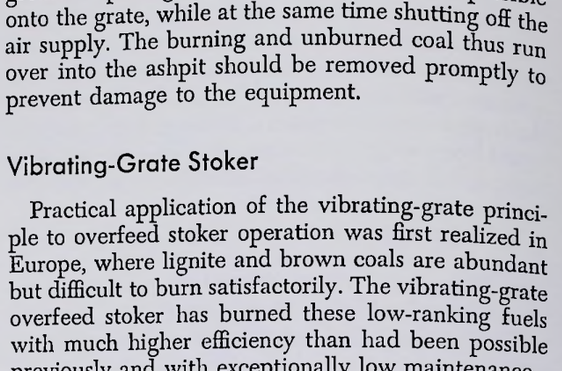


Fig. 6. *Vibrating-grate stoker*

As with chain- and traveling-grate stokers, coal is hopper fed, and the thickness of the fuel bed is regulated by an adjustable gate. The vibratory force is provided by a generator located at the front of the stoker beneath the coal hopper. The vibration generator consists of contrarotating mass weights physically arranged to cancel any motion in a vertical direction.

It is driven by a constant-speed motor that is operated intermittently by a timer control. Air to the stoker is distributed through laterally disposed compartments beneath the stoker, formed by the individual flexing-grate support plates.

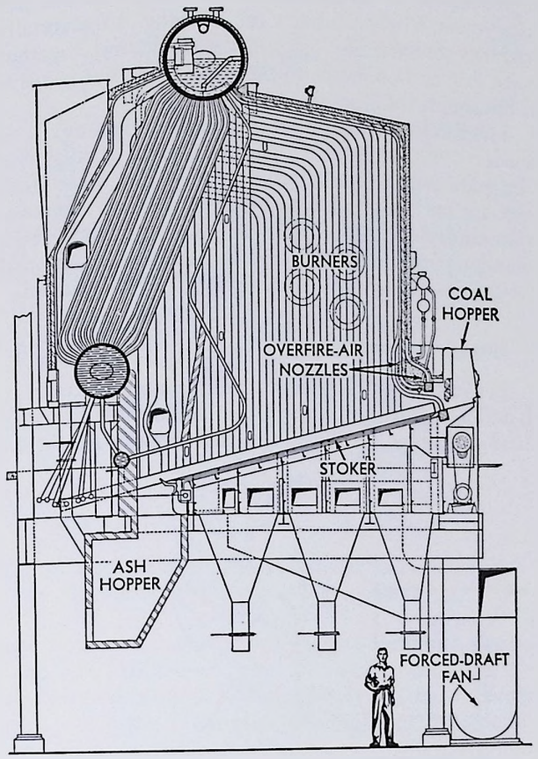


Fig. 7. *A vibrating-grate-stoker installation*

During vibration, new coal is fed to the stoker as the entire fuel bed is simultaneously conveyed down the inclined grate, and ash is discharged into the ash hopper. To complete carbon burnout of the ash, the thickness of the ash-bed portion is controlled by an adjustable ash-dam plate.

Design and Operation

The ability of the vibrating-grate stoker to burn a wide range of fuels is mainly attributable to the gently compacting action produced by the intermittent vibratory motion and water-cooling of the grate surface. Whether the coal is of the free-burning or the strongly caking variety, this compaction of the fuel bed permits uniform air distribution and gentle movement of the fuel on the grate.

Since volatile gases from the fuel bed tend to stratify, consideration is given to the furnace configuration, particularly the front and rear arches. A short front arch, supplemented by overfire-air jets is sufficient for most bituminous coals. The rear arch causes the gases emitted at the rear of the stoker to sweep forward and intimately mix with the turbulent rich gas over the forward portion of the grate; its length is variable, dependent upon the ultimate furnace configuration and the characteristics of the fuel to be burned. The trend in furnace design is toward fully water-cooled surfaces, in order to minimize refractory maintenance and permit the highest steaming capacity for a given space. This requires that consideration be given to the ratio of refractory to cold surface of the furnace walls.

Based upon these and other design parameters, grate release rates on this type of stoker may range from 350,000 to 500,000 Btu/sq ft, hr with medium- and high-volatile bituminous coals and under 400,000 Btu/sq ft, hr with low-volatile bituminous, subbituminous, and lignitic coals. Coal sizing is essentially the same as for chain- and traveling-grate stokers. With water-cooled grates, low ash content is of less consequence. Similarly, water-cooling permits operating with preheated air up to 400 F. In addition, this same feature permits multiple-fuel firing, singly or in combination. If the unit is to be fired with oil, whether for short periods or long, a thin layer of ash on the grate is sufficient to prevent residues from baking onto the grate. The water-cooled grate is tied into the boiler-circulating system, and, with proper design of the supply and riser tubes, the extremely short motion of the grate imposes no significant strain on these connections.

The uniformity of air distribution and the resultant fuel-bed condition provide good response to load swings with minimum cinder carry-over and smokeless operation over the entire load range.

Control

Operation of a vibrating-grate stoker is somewhat similar to that of the chain- and traveling-grate types, differing in that the grate speed is controlled by the frequency of grate vibration. A separate timer control, actuated by the fuel-feed controller of the automatic-combustion-control system is used for this purpose. Standard metering controls are recommended to maintain the ratio of fuel feed to air flow. This affords optimum combustion efficiency throughout the load range and assures immediate response to load changes.

Guide to Efficient Combustion

In normal operation, fuel feed, bed travel, and air distribution are regulated automatically by the combustion-control system. The fuel- and ash-bed thicknesses are governed by the positions of the guillotine gate and the ash-dam plates, respectively, and the intensity of burning over the individual compartments is dictated by the setting of each zone damper.

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The rate of fuel bed travel on the grates depends upon the frequency and length of the vibrating cycles, regulated to follow load demand by the automatic combustion control, and can be reduced to permit a 10 to 1 turn-down ratio. The burning rate is synchronized with travel rate by control of combustion air.

Ash discharge is regulated by the position of the adjustable ash-dam-plate mechanism. These plates are normally positioned to permit ash to flow off the grate during each period of vibrating motion, with the thickness of the ash bed regulated to effect complete carbon burnout.

When burning bituminous coals, the static air pressures for each zone will vary, depending upon the type of coal and its burning characteristics. With most coals, highest air pressure is maintained in the middle compartments while the pressure in the first compartment is reduced and that in the rear compartments is further reduced. The CO_2 content of the flue gas provides a good check on the correctness of these settings.

Emergency Shutdown

For emergency shutdowns, the coal feed and air supply are cut off and the furnace draft is minimized. If the sensible heat of the bed is still excessive, the entire bed may be dumped into the ashpit by operating the vibrating unit continuously. The hot coal and ash may then be quenched or removed from the ashpit.

Spreader Stoker

The essential difference between the spreader stoker and the chain-grate stoker is that the former utilizes a combination of suspension burning and a thin fast-burning fuel bed, whereas the latter employs the mass-bed-burning principle. The spreader stoker is extremely sensitive to load fluctuations, since ignition is almost instantaneous on increase of firing rate and burnout of the thin fuel bed can be rapidly effected when desired.

The principle of spreader firing, though old, did not become practical and popular until the early 1930's, when grates specially designed for spreader feeding were developed. These grates may be of the stationary, the intermittent-dumping, or the continuous-cleaning type (traveling, reciprocating, or vibrating), but it is essential that the design incorporate the high-resistance air-metering principle for best results.

Extremely flexible in its ability to burn a wide range of fuels from high-rank Eastern bituminous to lignite or brown coal, the spreader stoker is also adaptable to the burning of fine coal. Optimum coal sizing for this type of firing is ⅜ in. × zero with about 50% passing through a ¼-in. screen.

Because there is partial burning of the fuel in suspension, spreader stoker firing produces a much higher density of particulate matter in the flue gases than is usual with underfeed or other mass-burning equipment. Dust collectors are therefore required to separate the particles before stack discharge. However, the use of fly-carbon return systems, to reinject carry-over to the furnace for burning, substantially reduces combustible losses to the stack, and, together with low carbon losses to the ashpit, the net result is a minimum total combustible loss.

The modern spreader stoker consists essentially of feeder units arranged to distribute fuel evenly over the grate area, a specially designed grate, forced-draft systems for both undergrate and overfire air, combustion controls to coordinate air and fuel supply with changing load demand, and the dust-collecting and fly-carbon-return equipment previously mentioned.

Spreader Mechanism

Typical feeder-distributor units are shown in Figs. 8 and 9. Fig. 8 shows a variable-stroke reciprocating-feed-plate type in which the fuel is moved from the

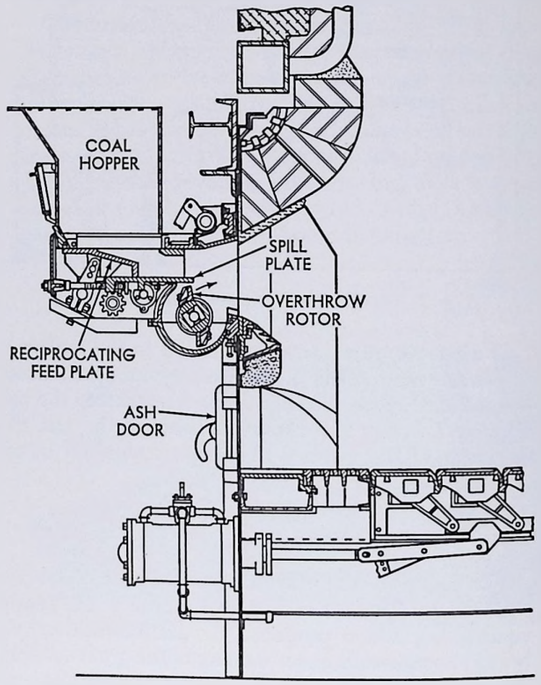


Fig. 8. *Reciprocating-feeder distributor and overthrow rotor for spreader-stoker firing*

supply hopper over a spill plate to fall onto an overthrow revolving rotor with curved blades for uniform distribution of fuel to the furnace. Fig. 9 illustrates a variable-speed chain-conveyor type of distributor

with overthrow rotor. Both types are designed for mechanical connection to the combustion-control system. Less common types of feeder mechanisms

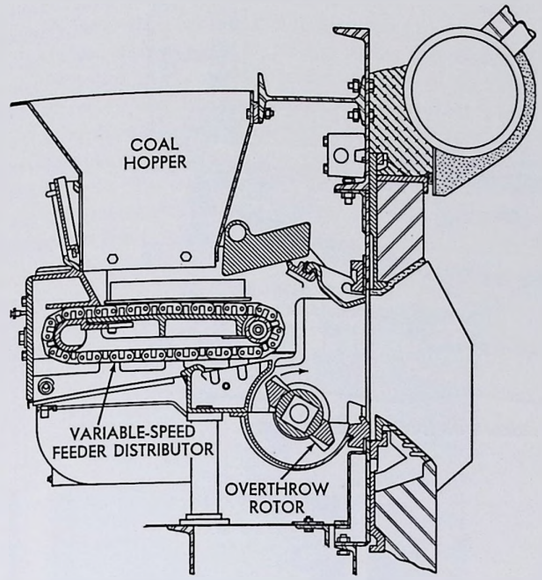


Fig. 9. *Variable-speed chain-conveyor distributor and overthrow rotor for spreader-stoker firing*

have rotary star wheels with worm feed and pneumatic jets. All feeder types, however, endeavor to provide a continuous supply of fuel to the furnace and to vary the feeding rate to conform with load fluctuations.

While the details of the various means used to spread and distribute the coal onto the grate differ, some form of rotor is common. Rotors are designed to overthrow or to underthrow, the former being more popular. Correct longitudinal distribution is obtained through varying the speed of the rotors.

In some instances the fuel is spread by pneumatic means, using air, a mixture of air and hot furnace gases, or steam to project the coal onto the grate. With this method, the coal is fed by a controlled mechanism, and the nozzle at the entrance to the furnace is adjusted to modify the trajectory. A deflector wedge within the nozzle is used to obtain lateral distribution.

Grates for Spreader Stoker

Grates for spreader stokers may be of several different types. For satisfactory operating results, they all employ the basic design concept of high resistance with air metering to avoid blowholes through the thin fuel bed.

The simplest and least expensive type has stationary grates from which ash is removed manually by raking through suitable doors in the front plate. Intermittent-dumping grates, for either manual or power-cylinder operation, are frequently used for small- to medium-size boilers. Both types are sectionalized, with separate undergrate air chambers for each grate section and a grate section for each feeder-distributor unit. Thus both air and fuel supply to one section can be temporarily discontinued as it is being cleaned, without affecting other sections of the stoker. Fig. 10 shows a power-cylinder-operated dumping-grate spreader stoker.

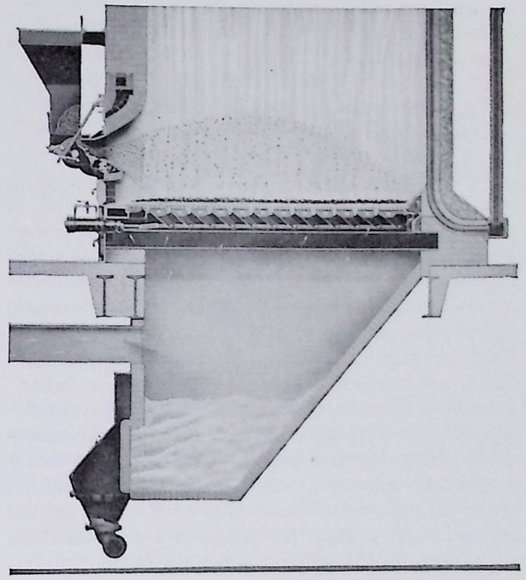


Fig. 10. *Power-cylinder-operated dumping-grate spreader stoker*

For sustained high-efficiency operation, a continuous-ash-discharge or a continuous-cleaning type of grate is necessary. The traveling grate with front ash discharge (rear discharge in rare instances) permits the highest burning rates per square foot of grate. Consequently more steam is produced with less total grate area than would be required with any of the other self-cleaning designs.

With spreader feeding, fresh coal is distributed uniformly over the surface of a burning fuel bed. It is ignited from the fire bed beneath and burns upward. Thus the grates are always protected from high temperatures by a layer of ash. With correctly designed metered-air openings to cool the metal, grate maintenance is extremely low.

Traveling-grate spreader stokers are economically sound for boiler capacities up to 400,000 lb steam/hr. For larger capacities, the required grate area may run the boiler cost beyond that for other methods of coal

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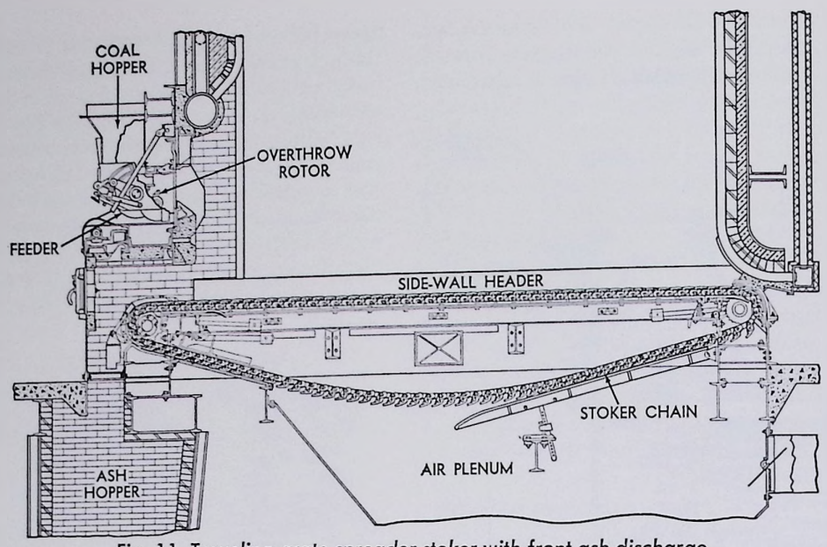


Fig. 11. *Traveling-grate spreader stoker with front ash discharge*

firing. The stoker shown in Fig. 11 includes self-adjusting air seals at the front and rear to minimize air leakage around the ends of the grates, since leakage at these areas will allow stratification along the front and rear walls of the furnace and add excess air that is not used effectively for combustion purposes.

Another advantage of the traveling grate over other types of self-cleaning designs is that the grate movement carries the fuel bed through the highly active combustion area, where it is reduced to ash without much disturbance. This enables coals with very low ash-fusion temperatures to be burned with less clinkering. Also rapid ignition permits the burning of caking coals without difficulty.

Other designs of self-cleaning stokers include reciprocating types, where a level grate has alternating rows of reciprocating bars to serve as the conveying medium for the fuel bed, as shown in Fig. 12, and undulating or oscillating grates, slightly sloped, using the vibrating-conveyor principle to move the fuel bed and the ash to the discharge point. The practical top capacity for continuous-cleaning spreader stokers other than the traveling grate is probably about 75,000 lb of steam/hr.

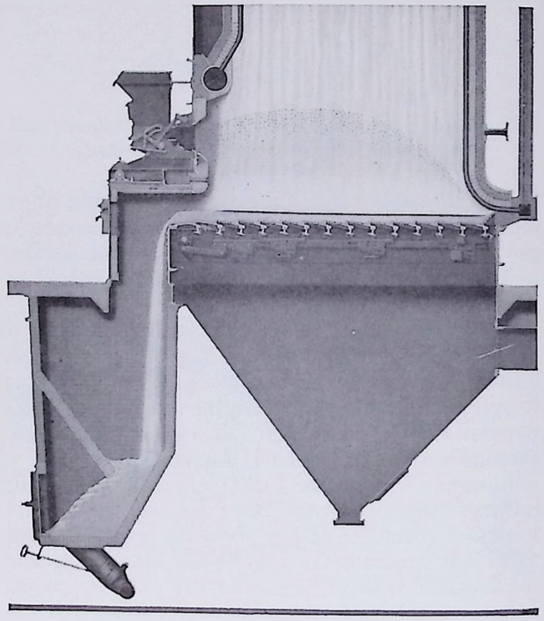


Fig. 12. *Reciprocating-grate continuous-cleaning spreader stoker with front ash discharge*

Fly-Carbon-Return Systems

An important adjunct of spreader-stoker firing is the system for return and burning of the fly carbon that is trapped out of the flue gas stream. Pneumatic systems with pressurized air as a conveying medium have been used extensively. Carbon particles are picked up from the outlets of settling hoppers under the boiler passes and heat traps and from the dust-collector hoppers. This material is reinjected into the furnace in the high-temperature zone above the grate for burning. Steam jets have been used for this purpose, but steam systems are subject to plugging if condensate collects and mixes with the fine particles in the carry-over.

The most modern system of fly-carbon return for use with traveling-grate spreader stokers is the gravity-flow return shown in Fig. 13. When physical layout permits the collecting-hopper outlets to be located at a sufficient height, the fly carbon flows by gravity to a distributing hopper directly behind the rear wall of the furnace. From this hopper a series of fine streams of carbon particles is deposited, again by gravity, onto the rear end of the forward-moving grate. When the internal air seals of the stoker are effective, a quiet zone exists at this point, and these particles are buried in and become a part of the fresh fuel bed that is continually being formed through the spreader-feeder action. Recirculation is minimized with this system, and the combustible in the particles carried over is consumed on the grates, the ash being deposited in the front ash bunker.

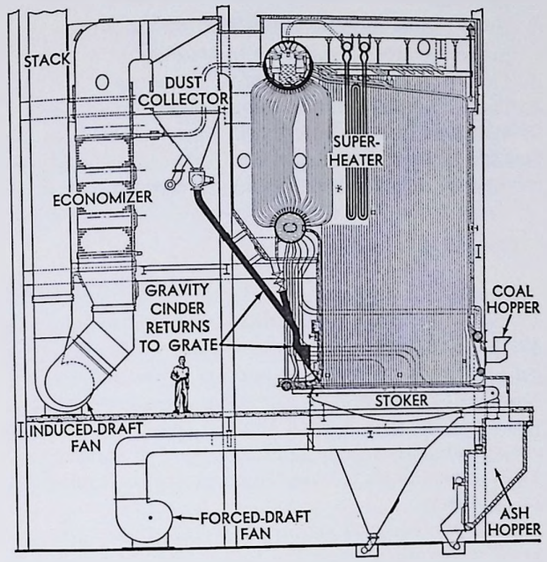


Fig. 13. *Spreader-stoker installation with gravity fly-carbon return*

When collection-hopper outlets are too low to permit flow by gravity, partial-gravity systems are sometimes used, with pneumatic or mechanical conveying of the carry-over to the distributing hopper, from which it flows by gravity to the stoker grate.

Effective use of overfire air, to promote and to aid the combustion of fuel in suspension, contributes in large measure to efficient spreader-stoker firing and reduction in the amount of fly-carbon carry-over.

Furnace Design

An example of good furnace design for spreader-stoker firing is shown in Fig. 13. Water-cooling of the walls is quite generally used and is, in fact, a necessity for traveling or continuous-cleaning grates, where slag or clinker formation adjacent to the stoker would interfere with movement of the fuel bed. Walls are vertical, or nearly so, as arches are neither necessary nor desirable. Air jets through the furnace walls create the turbulence required for the completion of combustion.

Drive Arrangement

Drive arrangements vary considerably with various stoker manufacturers and may be either mechanical or hydraulic. Multiple feeder drives are so arranged that any one feeder can be taken out of service without disturbing the operation of the remaining feeders. Fuel-feeding rates are regulated by the combustion-control system, and provision is made for manual adjustment to compensate for variations in coal characteristics.

Traveling-grate spreader stokers may have separate drives for the feeder distributors and for grate activation, or both may be driven from the same prime mover with suitable drive-reduction units. Fuel-feeding rate and grate speed are usually synchronized and varied in step with the combustion control to follow steam demand. Change in ratio of grate speed to fuel-feeding rate is effected manually to conform to ash content of the fuel.

Air Distribution

Control of both the quantity and the distribution of combustion air is important, as it is with all methods of firing, and air supply is generally apportioned by the combustion-control system through dampers at the fan or in the supply duct. Since undergrate air distribution is a function of the grate, it is important that grate design be such that air is metered uniformly over the entire active burning area. The undergrate plenum chamber is usually divided into compartments for dumping-type or stationary-grate stokers to allow shutoff of air for the period during which the individual sections of grate are being cleaned. Traveling-grate spreader stokers are generally installed with one large plenum or air chamber under the entire grate surface.

Fuels and Fuel Bed

The spreader stoker is extremely flexible in its ability to burn fuel with a wide range of burning characteristics, including caking tendencies. High-moisture free-burning bituminous and lignite coals are commonly used, and coke breeze has been burned in a mixture with higher-volatile coal. There are many successful installations of spreader stokers burning wood waste, such as bark from pulp-mill operations and chips from turpentine and other extraction processes. Bagasse from cane-sugar mills is a satisfactory fuel, whereas anthracite is not.

Coal-size segregation is a problem with spreader

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stokers as well as with other types of stoker firing. Spreader stokers can tolerate a small amount of segregation because the feeding rate of the individual feeder distributors can be varied, but it is not possible entirely to overcome all problems inherent in coal-size segregation. The coal should be a mixture of fine and coarse particles of approximately even proportions from ¾ in. maximum to zero.

The fuel bed on a spreader stoker is relatively thin (from 2 to 4 in.) although burning rates are high, from 450,000 Btu/sq ft, hr (36 lb coal) on stationary or dumping grates to as much as 750,000 (60 lb coal) on traveling grates when burning coal. Higher releases are possible with some of the wood wastes.

It is more difficult to maintain satisfactory combustion with coal at low loads with lower heat-release rates than it is with higher rates, as the fire tends to smoke below 125,000 Btu/sq ft, hr (10 lb coal). Proper application of overfire-air jets or even steam jets is helpful in preventing low-load smoke. In some instances dampering devices are provided to reduce active grate area in order to maintain satisfactory heat release for load ranges that exceed the normal variations.

Refuse Handling and Disposal

With stationary grates ash is removed manually through doors in the front at grate level. Intermittent-dumping grates discharge ash to a pit, as do the reciprocating- and vibrating-grate continuous-cleaning stokers. This pit may be shallow, permitting cleanout at the operating-floor level if a basement is not available. Traveling-grate continuous-ash-discharge spreader stokers deposit ash into a suitable pit or hopper usually located at the front of the stokers. Final disposal from this pit is by steam or water jets, ash conveyors, or sluices.

Control

The thin active fuel bed of a spreader stoker requires close regulation of fuel and air supply. Various types of automatic combustion controls are suitable. Of these the air-flow steam-flow type serves to proportion air and fuel more accurately than does the simple positioning type. Coal feed and air flow are reduced when there is a drop in load. Since the fuel bed is thin, there is little tendency to overrun steam pressure. For a load pickup, coal feed and air flow are increased, and the response is immediate. In this respect the action of the spreader stoker is somewhat like pulverized-coal firing, since a considerable portion of the fuel is burned in suspension. Distribution of coal on the fuel bed is usually controlled manually by regulation of the individual feeders and spreader rotors.

Aids to Operation and Maintenance

Observance of the practical suggestions listed below for the operation of a spreader-stoker unit will improve performance and reduce maintenance costs:

1. With stationary- or intermittent-dumping-grate stokers, clean the fires as rapidly as possible so that the entire grate is back in service in the shortest possible time. With continuous-cleaning or continuous-ash-discharge traveling-grate stokers, adjust the speed of the grate to suit the ash content of the fuel, so as to maintain the optimum ash bed for protection of the grate.
2. Do not operate the stoker for long periods with one section clean and another dirty, as this tends to impair the uniformity of air distribution.
3. Do not allow an excess accumulation of ash; follow the grate-cleaning instructions of the stoker manufacturer.
4. Keep grates in good condition. Replace burned or broken grates at earliest opportunity.
5. Do not permit coal hopper to run empty.
6. Do not permit ashes to remain long in ashpit.
7. Wet down and remove ash.
8. Maintain good fuel distribution on the active grate area by adjustment of spreader controls.

Operating Procedure

Starting Fire. Preparatory to lighting-off, the spreader mechanism is operated to distribute coal evenly over the grate to a depth of approximately 1 in. Kindling is then placed over a large portion of the coal surface and ignited. The stoker spreaders are started at slow speed. As the fuel ignites, the fan speeds or dampers are adjusted to supply sufficient air for combustion. The furnace draft is maintained at approximately 0.1 in. of water.

The coal feed and air supply to the stoker are regulated to permit placing the boiler unit on the line in accordance with established procedure specified by the boiler manufacturer. After this is done, the steam demand becomes the guiding factor in the operation of the stoker. With a traveling-grate spreader stoker, the stoker drive is started at this time or slightly before the boiler goes on the line. In normal operation the thickness of ash carried by the grate at the point of dumping is the governing feature in adjusting the speed of grate travel. Normally, with common bituminous coals, ash thickness is between 3 and 5 in. The cinder-return fan and return lines are put in operation whenever coal is fed.

The optimum fuel bed for spreader-stoker operation exists when the combustion rate over the entire active grate area is uniform and a minimum of excess air is used. Excess air is commonly between 25 and 40%, which is somewhat higher than the normal practice for the chain-grate stoker.

Banking Periods. When a boiler is to be banked, the coal feed and air supply are reduced to conform to the diminishing load until the boiler is at or near zero output. Stationary or intermittent-dumping grates should not be cleaned just prior to banking, and continuous-ash-discharge grates should be stopped before any part of the grate becomes bare. Air supply, including overfire air, is shut off, and enough coal is fed to form a bed approximately 1 in. thick over the entire grate. By positioning the boiler outlet damper, the furnace draft is regulated to provide only enough draft to carry off the gases. The banked fire is maintained by adding fresh coal.

To start up from bank, overfire air is admitted, and, in the case of continuous-ash-discharge stokers, the grate is started; draft, air supply, and coal feed are manually increased until normal load conditions prevail, after which automatic combustion control is placed in service.

Emergency Shutdown. For an emergency shutdown, the coal feed and air supply to the fuel bed are shut off. This reduces the burning rate to practically zero almost immediately, because of the small quantity of active fuel burning on the grate at any one time.

Underfeed Stoker

The underfeed stoker is classified in two types — horizontal feed and gravity feed. In the horizontal-feed type, as the name implies, the coal travel in the retort within the furnace is parallel to the station floor, while in the gravity-feed type it is inclined at an angle of 20 to 25°. The latter type consequently requires a basement or tunnel under the floor for ash disposal; for the horizontal-feed type, a shallow depression or pit suffices. Only the multiple-retort gravity type is suitable for the higher capacities within the range possible with an underfeed stoker.

Side-Ash-Discharge Underfeed Stokers

In the side-ash-discharge underfeed stoker, coal is force-fed to the fuel bed in small increments intermittently by ram or, for very small stokers, continuously by a screw. The coal moves in a longitudinal channel, known as a retort, usually assisted by an auxiliary push rod with small pusher blocks at the bottom of the retort, which operate in conjunction with the ram. After the retort is full, the fuel is forced upward and spills over the top on each side to form and to feed the fuel bed. Air is supplied through tuyères at each side of the retort and through air openings in the side grates.

As the coal rises, heat travels downward from the burning fuel above, and volatile gases are distilled off and burned as they pass through the incandescent fuel bed. The rising fuel then ignites from contact with the actively burning bed. The pressure exerted by the incoming raw coal moves the fuel bed gradually over the tuyères and side grates while burning continues. Combustion is completed by the time the bed reaches the side-dump grates from which the ash is discharged to shallow pits.

Grates. Some designs of side-dump underfeed stokers rely on pressure from the incoming raw fuel to achieve distribution over the side grates. This is satisfactory for single-retort stokers up to a width of 8 ft, as shown in Fig. 14. For wider side-dump underfeed stokers, distribution is obtained from reciprocating tuyère-block action between double-retort sections (Fig. 15). Other designs have a single center retort and depend upon agitating grates for distribution.

Underfeed stokers for domestic use and for very small commercial application have round retorts and stationary grates from which ash is removed manually in the form of clinker. Larger side-dump underfeed stokers are normally constructed with long center retorts, adjacent to which are tuyère grates. These are followed by side grates and dump grates with air openings, which permit a final burnout before ashes are discharged to the shallow pits for cleanout through doors in the stoker front.

Air Distribution. Air for combustion is supplied, in quantities suitable for the fuel-bed conditions, from a plenum chamber or windbox and is admitted to the

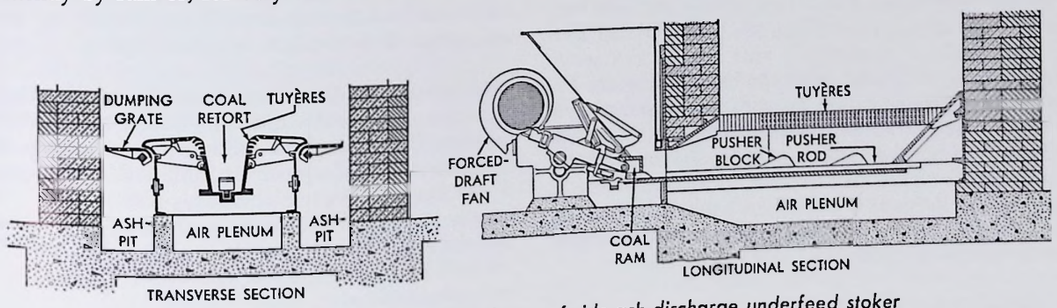


Fig. 14. *Single-retort horizontal-feed type of side-ash-discharge underfeed stoker*

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Fig. 15. *Double-retort horizontal-feed type of side-ash-discharge underfeed stoker*

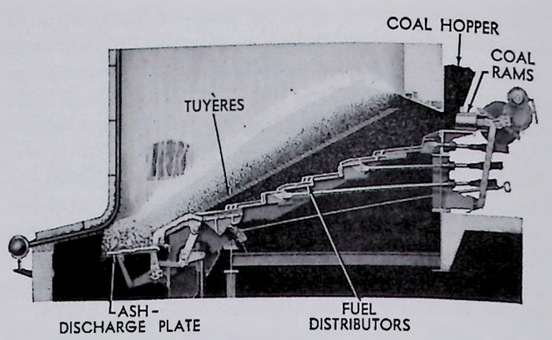


Fig. 16. *Multiple-retort gravity-feed type of rear-ash-discharge underfeed stoker*

fire through the tuyère openings and the perforated grates. Manually operated dampers admit air from the central plenum chamber to the ashpits under the side-dump grates for final burnout before the ash is dumped.

It is quite common for overfire air to be used with underfeed stokers to provide some combustion air in the flame zone directly above the active fuel bed. This air may be drawn from the windbox, from a separate overfire-air fan, or from both. Overfire air is quite effective in preventing smoke, especially at low loads with a lazy fire and on sudden increase of coal feed with the distillation of large quantities of volatile gases.

Rear-Ash-Discharge Underfeed Stokers

The rear-ash-discharge gravity-feed type of underfeed stoker is usually longer than the side-ash-discharge horizontal-feed type and is always fitted with multiple retorts. The multiple-retort gravity-feed stoker was a natural development of the principles successfully applied in the single retort. The multiple-retort stoker, Fig. 16, consists of a series of inclined single retorts placed side by side with tuyères between and sloped from front to rear for constant ash discharge. Each retort is equipped with a round or square primary ram, which feeds the coal into the ram box at the head of the retort. From this point the fuel bed is moved slowly toward the rear and, at the same time, is forced upward over the bank of tuyères by secondary pushers or by the moving bottom of the retort. Most of the air for combustion is admitted through the boxes supporting the tuyères and located between the retorts. An overfeed or cleanup section is provided at the rear end of the bank of tuyères for completing combustion before the refuse is discharged to the pit for disposal.

Grates. It is evident from Fig. 16 that the grouping of retorts and tuyères occupies about two thirds of the combustion surface. The more active burning lanes of the fuel bed are over the tuyère rows or air-admission ports, while the relatively inactive fuel lanes are over the retorts. As the fuel bed completes its movement over the retorts and tuyère banks and reaches the extension-grate section, the burning lanes gradually widen. The constant agitation of the grate section keeps the fuel bed broken up and porous, permitting the gradual completion of the combustion of carbon. The ash then moves to the ash-discharge plates, where it is ejected in a continuous discharge for removal.

The extension grates permit the effective burning of coke in the refuse before it reaches the dumping grates. These extension grates are usually equipped with a reciprocating-action drive, either separately or in series with the pushers. The grates may be flat or stepped. The agitation of these grates keeps the fuel bed active, breaks up any large clinker formation, and moves the refuse to the dump grates.

The dump grates, or ash-discharge plates, located directly aft of the extension grates, rid the stoker of refuse after combustion of the fuel is completed. They may be arranged to lower at intervals to dump the accumulated ash and clinker into the pit, or they may be given a reciprocating motion by connection to the stoker mechanism for continuous ash discharge. Air is admitted to most dump grates for the last possible stage of combustion before dumping.

Air Distribution. Air for combustion is supplied from a plenum chamber (windbox), and the quantity and pressure for the various sections are controlled by dampers. Most of the air is admitted through the tuyère blocks. A more elaborate system of controls is usually provided for these longer stokers of the gravity-feed type. The pressures for the various sections are indicated by gages, so that a closer control can be maintained for greater efficiency. With closer control of combustion air it is also possible to obtain

higher burning rates without serious increase in maintenance cost.

Features Applicable to Both Types

Fuel-Bed Contour. As the coal leaves the ram boxes and enters the furnace, a cross section of the flow of the fuel bed assumes a sine-wave contour in both types of multiple-retort underfeed stokers. The crest of each wave is directly over the retorts, which are the channels of feed, while the trough will be over the tuyère sections. Most of the air for combustion is admitted through the tuyères. By lateral flow from the adjacent retorts, the coal is distributed over the tuyères, and these zones become the active burner lanes, with much slower burning over the retorts. In the gravity-feed type, as the fuel bed approaches the lower end of the grate, particularly the overfeed or cleanup section, the contour tends to level out.

Stoker Drives. The type of mechanical drive for the underfeed stoker will depend upon plant load conditions and the flexibility desired. Motors are commonly used when the application is for low-pressure boilers or when mechanical power is not produced at the plant. Where sufficient steam pressure is available, a steam-turbine or a reciprocating-steam-engine drive may be used to advantage, particularly when the steam required is a factor in heat-balance conditions. The speed range and flexibility of stoker drives may be increased by variable-speed transmission. To protect stoker parts from damage, the driving mechanism is equipped with a shear pin.

Two types of hydraulic-drive systems are used. In one a constant-speed motor or a turbine drives a variable-displacement oil pump, fitted with proportioning valves, which supplies oil pressure for all drives and controls required for operating the stoker. In the other the oil pressure from the pump acts directly on the hydraulic cylinder driving the rams and the mechanism for agitating the grates.

Rams and Ram Boxes. Most rams are cylindrical but some are rectangular. They may be horizontal or inclined at the angle of the grate. Usually the ram boxes have renewable wear plates. In the case of the rectangular boxes, the wear plates may be adjustable. By the use of separate adjustments for each ram and distributor or retort pusher, the quantity of coal fed to the stoker as well as its distribution from front to rear within the furnace can be closely regulated. These adjustments are made while the stoker is in operation. Therefore, regardless of the variation in the kind and condition of the raw feed, the fuel bed can be maintained at the proper thickness and distribution for efficient operation.

Tuyères. Tuyères are mounted between the retorts and, as assembled in banks, provide a system of graduated openings for admission of the proper quantity of air for combustion. Long webs provide effective air cooling, which, with minimum exposure to heat, assures long life. Side tuyères are usually provided to prevent clinker adhesion to the side wall.

Ash Discharge. In the single- or double-retort horizontal-feed stoker, the ash discharge is usually at the sides. The ash may accumulate on the side-dump plates to be disposed of periodically, or, for units equipped with moving dump-grate sections, it may be discharged continuously into a pit.

In the gravity-feed multiple-retort stoker, ash is disposed of at the rear. Dump grates that open intermittently are usually operated by a steam or air cylinder. The period of intermittent dumping will depend upon the percentage of ash in the coal and the combustion rates per retort. Water sprays are normally installed in the ashpit to cool the refuse immediately after dumping. Ash hoppers of ample capacity should be provided for this type of stoker. Higher efficiency is possible with continuous ash discharge because of the reduction in carbon loss and lower excess-air requirements. With this method it is also possible to obtain higher capacities with a more flexible load range. In this type of installation, particularly where the ash is to be removed by sluicing, clinker grinders are usually fitted. The grinders may be single or double rolls driven by a separate motor, a drive chain from the stoker shaft, or an hydraulic cylinder, depending upon plant preference or conditions. Grinders usually have replaceable teeth.

Use of Water-Cooling. In the early days of stoker application, furnaces were of all-refractory construction. As furnaces became larger and burning rates increased, it became necessary to improve furnace-wall and grate construction to meet the new requirements. This was done by the use of water-cooling. Water-cooled tubes absorb radiant heat and protect the refractory. Water-cooled retorts and tuyères lower the fuel-bed temperature and chill the ash. This helps to prevent the closing of air ports by slag and thereby to lengthen the life of the parts.

Suitable Coal Characteristics. A fairly wide range of coal can be used with either type of underfeed stoker. The horizontal-feed type is better suited for free-burning bituminous coal, with a size consist of 1¼-in. × zero nut, pea, and slack in equal proportions being most desirable. It can also satisfactorily burn anthracite separately or mixed with bituminous coal.

A comparison between stationary- and agitating-grate underfeed stokers shows that stationary grates can burn coals with lower ash-fusion temperatures

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without serious clinker problems whereas the agitating-type grates can more readily burn coals with high caking tendencies. On the other hand, the gravity-feed stoker gives good results burning caking coals of moderate to high ash-fusion temperature as well as high-volatile bituminous coals of the free-burning variety. With water-cooled stationary tuyères, the capability of this type of stoker is extended to include low-fusion coals.

The size of the coal has a marked effect on the relative capacity and efficiency of a stoker. A nonstratified distribution of the raw coal across the hopper is important, particularly for a stoker with multiple retorts. A reduction in the percentage of fines helps to keep the fuel bed more porous and extends the range of coals with a high coking index.

In general, underfeed stokers are able to burn caking coals. The range of agitation imparted to the fuel bed in different stoker designs permits the use of coals with varying degrees of caking properties. There is no rule of thumb by which the degree of agitation required can be correlated with the caking properties of the coal. The only certain way to select suitable coal is by actual field tests for a given unit. The ash-fusion temperature is an important factor in the selection of the coals. Usually, the lower the ash-fusion temperature the greater the possibility of clinker trouble.

Size Limitation. The single- or double-retort horizontal underfeed side-cleaning stoker is generally designed to service boiler units generating up to 30,000 lb steam/hr continuously, with slightly higher peaks. This type of stoker with moving grates should be designed for a maximum rate of 425,000 Btu/sq ft, hr (equivalent to 34 lb of 12,500-Btu coal) in furnaces with water-cooled walls. For refractory-walled furnaces, the maximum rate should be reduced to about 25 lb coal/sq ft, hr.

The multiple-retort gravity-feed stoker can be designed in capacities suitable for boiler units generating from 20,000 to 500,000 lb steam/hr. This type has burning rates up to 600,000 Btu/sq ft, hr (48 lb coal), with peaks of 720,000 (58 lb coal). It is apparent that it is not only these higher rates but also the much greater fuel-bed area possible with this type that boosts the maximum capacity to over 10 times that of the horizontal-feed stoker.

Burning rates are directly related to the ash-softening temperature. For coals with ash-softening temperatures below 2400 F, the burning rates will be progressively reduced.

Preheated Air. The development and extensive use of air heaters have had an effect on underfeed-stoker design and performance. The gain in efficiency from preheated air is not confined solely to the heat recovered. With some coals the result has been a thinner and more uniform fuel bed with a decrease in excess air and lower undergrate pressure.

Since grate temperature depends upon the temperature of the combustion air, it is apparent that the latter should not exceed the allowable use limit of the metal of which the grate is made. Generally, the top limit for preheated air, as set by the manufacturer, is 350 F.

Combustion Control. Many small underfeed stokers handling relatively steady heating loads operate with a start-stop control. This consists of a pressure regulator set for a steam-pressure differential of 2 to 5 psi, which starts the stoker on pressure drop and stops it as the steam pressure reaches the preset high point. For such operation it is desirable to include a hold-fire timer in the control system.

Stokers operating to suit a varying load should be equipped with modulating combustion controls that will vary the coal feeding rate and air supply in step with the steam demand and should include control of the furnace draft through operation of the boiler outlet damper. Positioning-type controls, either electric or hydraulic, are quite satisfactory for this purpose when manual adjustment is available to establish the correct fuel-air ratio.

Maintenance. Maintenance cost for any type of firing equipment is always of utmost importance. Like other items of maintenance, any excess over the recognized and legitimate cost is subject to the control of the operator if the design and construction are sound to begin with and the particular unit is suitable for the service intended. Maintenance costs, therefore, may be said to depend upon the design of the stoker, the correctness of the application, the type of fuel used, and the skill of the operator.

Probably the greatest single factor in high maintenance is the burning of stoker metal. If the fuel bed is maintained in a porous condition, stoker parts can be held at a safe temperature with an adequate flow of air to remove the heat as fast as it is generated. Use of inferior grades of coal, unsuitable for the equipment as designed, will increase maintenance costs. Similarly, maintenance for units using preheated air is higher than that experienced on units designed for ambient temperature air.

Refuse Handling. The horizontal-type side-cleaning underfeed stoker usually discharges ash from dumping grates adjacent to furnace side walls into shallow ashpits from which it is removed by hoe and shovel through suitable doors in the stoker front. With this arrangement no basement is required, which is sometimes important when considering installation costs. If basement space is available, deeper pits will provide more ash-storage space. Final disposal of the ash is by wheelbarrow or any of several types of mechanical ash conveyors.

The gravity-feed multiple-retort stoker, because of its inclination and its larger capacity, requires a basement and mechanical equipment for handling the ash. Where the ash is to be removed by sluicing, a rotary ash-discharge arrangement with a clinker grinder is a desirable feature, especially in localities where there is a market for sized clinkers.

Cinder Emission and Fly-Ash. Cinder emission from the underfeed stoker depends upon the combustion rate, the caking properties, and the size consist of the coal fired. For burning rates under 500,000 Btu/sq ft, hr, cinder emission is relatively low. Free-burning fine coal will increase the dust loading of the gases leaving the furnace. A multiple-retort gravity-feed stoker burning a fine coal at high rates may have a combustible loss of 15 to 20%.

General Operating Aids

For good performance in operating an underfeed stoker, the following precautions should be observed:

1. A furnace draft of not less that 0.1 in. of water should be maintained at all times above the fire.
2. The grates should, from time to time, be inspected through the windbox doors to make certain that they are cool.
3. A fairly uniform air-fuel relation should be maintained over the entire stoker area. Localized overheating of the tuyères and grates and waste of fuel may result if the stoker is operated with too little air at one portion of the fuel bed and too much air at another.
4. Except in an emergency, combustion air should be shut off shortly after the coal feed is stopped, to prevent the fire from burning down into the retorts.
5. Air leaks through the setting must not be allowed, since this may cause a deficiency of air through the fuel bed.
6. The fuel must be properly distributed over the grate surface to assure active burning over all areas where air is flowing. The fuel bed should be kept at the correct thickness for an intimate mixture of the coal and air without blowholes.
7. Ash and clinker should be handled as follows: Some ash should be kept on the stoker grates as insulation and protection against overheating. The period between cleanings will depend upon the clinkering tendencies of the ash and the quantity of ash in the coal. Cleanings must be timed to preclude interference of clinker accumulation with combustion or stoker operation. The rolls and cutters of clinker grinders should be checked periodically, and broken or badly worn parts replaced. The stoker windbox should be inspected frequently, and siftings should be removed before any appreciable amount has accumulated.
8. The various sizes of the coal should be distributed uniformly, since segregation produces uneven fires and a loss of capacity and efficiency.
9. For the correct operation of a large underfeed stoker, in addition to observation of the fire it is essential to give close attention to the readings of draft gages and panel-board instruments and to act promptly in making the adjustments indicated by these readings. The depth of the fuel bed above the top of the tuyères may vary between 12 and 30 in., depending upon the burning characteristics of the coal.

Operating Procedure

Starting Fire. To start the fire with a horizontal-feed type of stoker, sufficient coal is slowly fed to cover the grate surface to a uniform thickness of 3 in. after manual spreading. Kindling or oiled waste is placed over the coal on each side of the retort. Just before igniting the kindling, the boiler damper and the windbox door are opened. When the coal is well ignited, the stoker is operated at the speed required to replace the coal being burned. When load is to be placed on the boiler, the windbox is closed and forced draft is supplied.

When raw coal is initially fed into the stoker, an even distribution of fuel is maintained by observing the contour of the fuel bed. As the stoker picks up load, fuel and combustion air are adjusted for the thickness of the fuel bed. The correct thickness depends somewhat on the width of the stoker and the type of coal being used. The depth of the fuel bed over the tuyères may vary within the limits noted above. Usually low-volatile coals can best be burned with a heavy fuel bed. Coarser-size coal also calls for a thick fuel bed.

The gravity-feed type of stoker requires about 8 or 9 in. of fuel distributed over the tuyères of the underfeed section. A fire is started by lighting oiled waste or kindling placed on top of the coal close to the front wall. When ignition is well established, light air pressure can be applied to hasten burning. After sufficient ash has accumulated to protect the tuyères and the moving and dump-grate sections, the stoker can be put into general operation. The stoker should not be forced until the fuel bed is in a good coked condition and evenly distributed.

Normal Operation. The best fuel bed is the one which carries the load with just enough thickness to burn freely throughout the length of the stoker when operating with the desired quantity of excess air. Distribution and agitation of the fuel bed are controlled

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by adjustments to the stoker mechanism. Speed will control the fire for proper thickness and length.

Banking. Ash should not be dumped just before banking, since it is best to leave as much ash protection on the stoker as possible. Preparatory to banking, the fire should be burned down thinner than it is during normal operation. The air should then be shut off and the fuel feed operated for a few minutes to build the fuel bed to normal depth. With a good bed of coal in the furnace, the coal feed should be shut off and the boiler outlet damper closed to a position just sufficient to carry off the gases. This damper is closed gradually as the furnace cools. If it is necessary to bank with coal in the hopper, the stoker should be run a few minutes every hour until the hopper is empty or the boiler is again under load.

Emergency Shutdown. The object of an emergency shutdown is, of course, to stop the heat release of combustion as quickly as possible. For a sudden loss of load the best procedure would be as follows: The stoker should be stopped. The forced-draft fan should be shut off and the damper closed tight. All doors in the boiler above the grate line should be opened. Under these conditions burning of the fuel bed will be retarded, and, with the admission of air above the grate, the furnace will be permitted to cool gradually. Water should not be used on the fire, as the rapid expansion of gases and steam or the generation of hydrogen could cause an explosion.

In the case of a water-wall-tube or a boiler-tube failure with steam and water discharged into the furnace proper, the steam and water so discharged usually extinguish the fire. Under such circumstances the problem is to keep the gases and steam confined within the setting to protect the personnel. Furnace doors should have locking latches for this reason.

For an emergency shutdown because of low water within the boiler, the operating procedure, so far as the stoker is concerned, should be the same as for a sudden loss of load. Detailed procedure for the protection of the boiler is covered in Chapter 21.

Change-Over to Oil or Gas, All Types of Stokers. Oil or gas burners can be applied to underfeed-stoker-fired furnaces. Certain precautions must be observed, however, when oil or gas is to be fired alternately with coal. If the burning of the alternate fuel is on a temporary basis, a good layer of stoker ash will serve to protect the grates. If the change-over is to be for a period of several weeks, the grates should be protected in a more permanent manner by using firebrick. For an extended period it may be expedient to remove the stoker and to build a false refractory floor over the steel work. On the other hand, if the gas or oil burners are to be idle for some time, it may be desirable to brick-in the burner openings to prevent plugging of the parts by slag.

17

PULVERIZED-COAL EQUIPMENT

Of the three commercial fuels—coal, petroleum, and natural gas—found in abundance and widely distributed in the United States, coal is predominant and indispensable for the production of power and industrial use. Because of its distribution and availability, coal is the basic fuel in the generation of steam for light, heat, and power. The production of bituminous coal in the United States in 1959 was 410 million tons, the consumption 366 million tons. Its usage was divided approximately as follows:

TABLE 1°

U.S. Consumption of Bituminous Coal

Use	Per Cent
Electrical utilities	45.3
Railroads (Class 1)	0.7
Metallurgical coke	21.6
Steel and rolling mills	1.8
Cement	2.3
Other industries	20.1
Retail	7.9
Bunker foreign trade	0.3

° According to Natl Coal Assn, *Bituminous Coal Data 1959.*

About 80% of the coal for the generation of electricity and practically all the coal for the manufacture of cement was burned in the pulverized form. This is equivalent to 40% of the total annual consumption, or about 150 million tons, and demonstrates the importance of pulverization equipment and processes.

The preparation and the burning of coal in pulverized form are covered in the present chapter; stoker firing in Chapter 16; and Cyclone Furnace firing in Chapter 28. Hand firing is practically obsolete.

Selection of Coal-Burning Equipment

The selection of the most suitable burning equipment for a particular job consists of balancing the investment, operating characteristics, efficiency, and type of coal to give the most economical installation.

Investment. Experience shows that stoker firing is best suited for steam-generating units with design capacities less than 100,000 lb steam/hr, and pulverized-coal or Cyclone Furnace firing for units above that size. For kilns and industrial furnaces of all sizes, pulverized-coal firing is most commonly used.

Operating Characteristics. Operating characteristics of the installation may be the determining factor. For a small plant with inferior labor or for a plant with a wide load range, the stoker is best suited. For large plants with skilled operators, pulverized-coal or Cyclone Furnace firing is preferable.

Efficiency. Small process plants can frequently tolerate the lower efficiency of a stoker-fired unit. In larger plants, the fuel cost is such a large fraction of the operating cost that efficiency may be the controlling factor and dictate the selection of pulverized-coal or Cyclone Furnace firing.

Type of Coal. The type of coal will influence the choice of the method of burning with the primary considerations as follows:—

Stoker firing: rank of coal, moisture, volatile matter, ash, and ash-softening temperature.

Pulverized-coal firing: grindability, rank, moisture, volatile matter, and ash.

Cyclone Furnace firing: volatile matter, ash, and ash viscosity.

Convenient approximations for the selection of bituminous coals for the firing of boiler furnaces are given in Table 2.

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TABLE 2

Coal Characteristics Related to Method of Firing

Method of Firing	Stoker	Pulverized Coal	Cyclone Furnace
Max total moisture° (as fired), %	15-20	15	20
Min volatile matter (dry basis), %	15	15	15
Max total ash (dry basis), %	20	20	25
Max sulfur (as fired), %	5	—	—
Max ash-softening temperature, F	—	—	2400

°These limits may be exceeded for lower rank, higher inherent-moisture-content coals, i.e., subbituminous and lignite.

When firing such equipment as metallurgical furnaces and kilns, other considerations may determine the suitability of a specific coal. The rank of coal and the end use govern the fineness to which coal must be ground. The data of Table 3 are helpful in this specification.

Fundamental Considerations for Pulverized-Coal Systems

1. Pulverizer Requirements

A) The pulverizer may do grinding only and leave the feeding, drying, classifying, circulation, and transport to component equipment; or it may do the complete job.

B) Modern steam-generating units require systems which:
- a) Are conservative in space consumption;
- b) Handle a wide variety of coals;
- c) Are responsive to load changes and capable of fully automatic operation;
- d) Maintain performance throughout the useful life of the components;
- e) Are easy to maintain and have the fewest number of parts;
- f) Remain in uninterrupted service for extended periods.

2. Design Fundamentals

A) Feeding. The fuel rate must be coordinated with pulverizer air-flow characteristics in order to furnish the velocities necessary for transportation and the air-coal-mixture proportions required by the burners. It must be automatically controlled by the boiler load demand.

The B&W Co. uses a feeder controlled by air-flow and coal-delivery rates. Air flow is measured by a pitot tube or an orifice in the air supply line and coal flow by the static-pressure differential across the grinding zone.

B) Drying. This is essential to furnish 1) the proper dryness of material to the grinding surfaces for efficient grinding and 2) the proper air-coal-mixture temperature to the burner for ignition without coking. Thus an adequate heat supply to, and an adequate residence time in, the pulverizer are necessary.

In B&W pulverizers these requirements are met by the blotting action created by the dry internal recirculation of material ahead of the grinding faces and the flash drying by hot air after grinding.

C) Grinding. The pulverizer must do adequate work on each passage of the material through the grinding zone and without the production of excessive superfines. This is best accomplished by internal recirculation of coarse material. The pulverizer should maintain its grinding ability over the life cycle of the grinding elements and be able to reject foreign matter

TABLE 3

Required Pulverized-Fuel Fineness at Maximum Rating

Per Cent Through 200 U.S.S. Sieve°

	ASTM Classification of Coals by Rank					
	Fixed Carbon, %			Fixed Carbon below 69%		
Type of Furnace	97.9-86 Petroleum coke	85.9-78	77.9-69	Btu above 13,000	Btu 12,900-11,000	Btu below 11,000
Marine boiler furnace	—	85	80	80	75	—
Water-cooled furnace	80	75	70	70	65†	60†
Cement kiln	90	85	80	80	80	—
Metallurgical	(As determined by process, generally from 80 to 90%.)					

°The 200-mesh screen (sieve) has 200 openings per linear inch or 40,000 openings per square inch. From U.S. and ASTM sieve series, the nominal aperture for 200 mesh is 0.0029 in. or 0.074 mm. The ASTM designation for 200 mesh is 74 microns.

†Extremely high-ash-content coals will require higher fineness than indicated.

that enters with the feed. These objectives should be attained without excessive wear and power consumption.

The grinding elements of B&W pulverizers are spring loaded to provide the necessary grinding force throughout the life of the elements. Shallow cupped rings furnish adequate grinding surface, allow rapid material flow, and facilitate the discharge of foreign material to the counterweighted relief gate. The spiral rotation of the pulverizer balls keeps the balls spherical and allows the ring surfaces to wear to the proper contour.

D) Circulating. Circulation within a pulverizer should dry the material, remove it from the grinding zone, and deliver it to the classifier for rejection and return of oversized material to the grinding zone.

These functions are all performed in the B&W pulverizer where the controlled inlet-air jet system lifts the material as it passes through the grinding zone, sends it through deflectors for immediate return of oversize material to the grinding zone, and at reduced velocity delivers slightly oversize material to the classifier.

E) Classifying. The function of a classifier is to permit passage of material of desired fineness and to return oversize material for further grinding. If separation is not precise, fine material returned for further grinding produces unnecessary superfines. Separation must be effective over a wide load range, be reasonably adjustable for product size, and permit uniform division among multiple burner lines.

The stationary cyclone-type classifier, used in the B&W pulverizer, meets these requirements. The shrouded discharge prevents re-entrainment of rejects. A flat fineness characteristic is obtained since classification ability is proportional to velocity and particle-size loading. The movable inlet vanes permit adjustment for the desired product fineness.

F) Transporting. The velocity in pulverizer discharge lines should be sufficiently high to prevent fall-out of material in the lines. A uniform coal-air mixture to the burners must be at suitable velocity.

The B&W system, by a variable controlled air-coal relationship, increases the air-coal ratio at lower loads and decreases it at higher loads to provide a maximum load range. The symmetrical take-off of burner lines assures equal distribution.

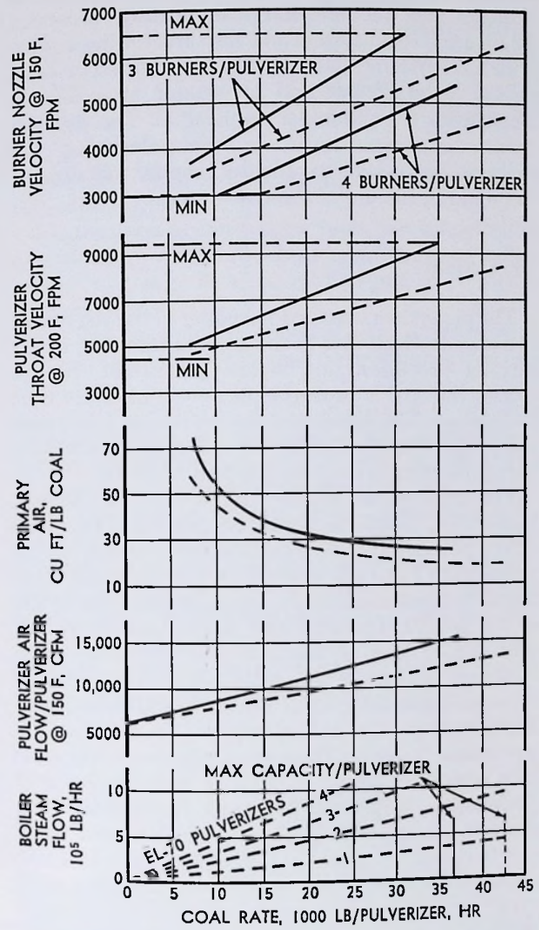


Fig. 1. Coordination curves for pulverizers and burners

3. Adjustments for B&W Pulverizers

A) Calibration of Air-Metering Device. This is necessary to provide proper air-coal relationship for efficient pulverizing, since this ratio controls the internal circulation and insures optimum burner performance.

The metering equipment consists of a modified pitot tube or orifice to measure the primary-air flow to the pulverizer. Taps at the pulverizer air entrance and exit permit measurement of the clean-air pressure drop and the circulating-coal load. This facilitates the calculations for construction of pulverizer curves (Fig. 1).

B) Selection of Fuel-Air Ratio for a Particular Installation. This is dependent upon the coal grindability, the desired fineness, and the load range.

C) Development of Coordination Curves. The following steps show the development of the coordination curves:

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a) After the number of pulverizers has been established, the steam flow vs coal rate per pulverizer can be determined. The normal load requirement should be known, as well as the pulverizer control capacity.

b) The curve for pulverizer air flow cfm vs coal rate (lb/pulverizer, hr) can be drawn when the grindability, desired fineness, and pulverizer size are known.

c) The cu ft of primary air per lb of coal vs coal rate (lb/pulverizer, hr) curve can then be drawn.

d) The curve for throat velocity vs coal rate can be drawn after the pulverizer size and air flow have been determined.

e) The burner nozzle velocity can then be calculated, and the operational overlap for multiple burners can be established for the load range.

D) Adjustment of Feeder Controller. The feeder controller is set for maximum load based on fineness and output, and the minimum load is based on burner-line and -nozzle velocities, burner stability, and pulverizer throat velocity.

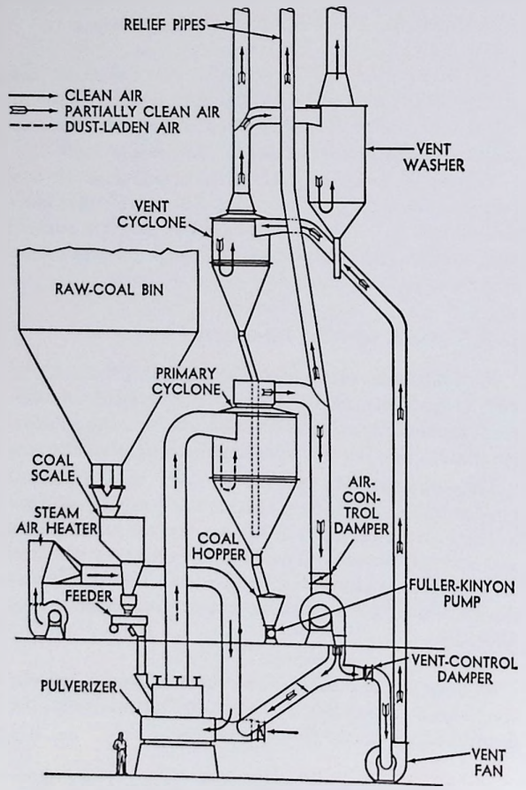


Fig. 2. *An early bin-system installation (1930). Coal dried within pulverizer*

No adjustment is required for changes in grindability of less than 20 points. A wider variation requires readjustment to obtain proper drying, fineness, and load range.

No adjustments are normally required for changes in volatile matter, heating value, or coal size. No adjustment is required for moisture as long as outlet temperatures remain within allowable limits (see Table 7).

Major Pulverized-Coal Systems

In general, there are two systems currently in use in processing, distributing, and burning pulverized coal for the generation of steam, the manufacture of cement, and for metallurgical heating processes. These are the bin system and the direct-firing system.

In both systems, the coal may be predried or dried during pulverization and is ground to a fineness suitable for the particular application. The development and improvement in coal-pulverizing and -burning equipment have had a marked influence on the trend in the use of the two systems.

Bin System

The bin system was the forerunner of the large-scale use of pulverized coal in the cement industry, metallurgical processes, and steam generation. In this system the coal is processed at a location apart from the furnace. The distance of the processing plant from the point of application is limited only by available means for transporting the finished product.

Functions of the modern bin system are drying, pulverizing, classifying, and transporting. The coal is dried, pulverized, and classified continuously in the pulverizer. The end product is pneumatically conveyed to cyclone collectors, where approximately 95% of the coal is separated and discharged by gravity into storage bins or hoppers. The remaining 5% is recovered in filters or high-efficiency cyclones, and the clean moisture-laden air is vented to the atmosphere. From storage, the pulverized coal is conveyed by pneumatic transport through pipelines to utilization bins sometimes as far as 5000 ft from the point of preparation. In the older systems, the coal was predried in external thermal dryers, which were dangerous to operate from the standpoint of flash fires and explosions. The increased demand for coal-pulverizing equipment in the generation of steam and in the metallurgical field led to the development of an improved bin system, where the coal is dried, pulverized, and classified in the pulverizer. This required the use of preheated air in the pulverizer and partial or total venting, dependent upon the moisture in the coal and the temperature of the air. A bin system of this type

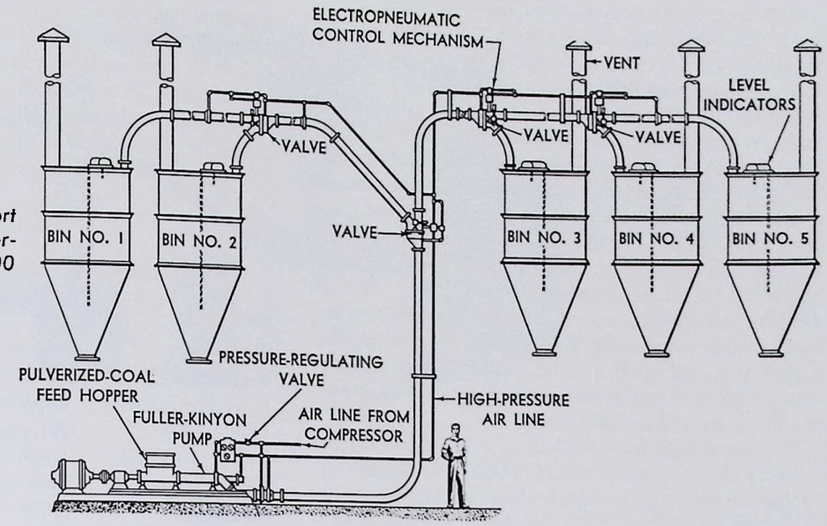


Fig. 3. *Pneumatic transport system for conveying pulverized coal. Capacity, 1 to 100 tons per hour*

is illustrated in Fig. 2. A typical transport system is shown in Fig. 3.

Modern design bin-system vents are connected to dust-cleaning devices, such as small bag-type collectors, or discharge to a combustion process to reduce dust nuisance.

For the coal-air transport system, a differential-pitch screw pump is provided to feed pulverized coal continuously into a pipeline, at the entrance of which the coal is aerated or fluidized so that it will flow through the pipe somewhat like a heavy fluid. A feed-hopper screw-pump assembly is shown in Fig. 4. The transport line may run in any direction, and, through a system of two-way valves (Fig. 3), the coal can be distributed to any number of bins. The system may be arranged for manual, remote, or completely automatic control. Air-transport systems (see Fig. 5) are built in sizes from 1 to 100 tons of coal per hour. For successful operation, the moisture in the pulverized coal must not exceed 3% (surface) and the fineness should not be less than 90% through a 50-mesh sieve.

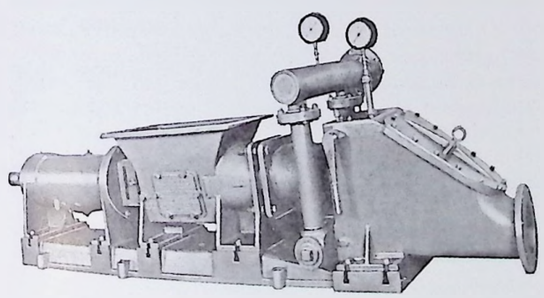


Fig. 4. *Fuller-Kinyon pump for transporting pulverized coal*

Features of the Bin System

Pulverized coal was used extensively before pulverizing equipment had reached the stage of development where it could be relied upon for continuous uninterrupted operation, flexibility, and consistent performance. In that period the bin system served as a means of insuring a continuous supply of pulverized coal in the plant.

Where the coal has to be predried in this system it is possible to handle high-moisture coal, which cannot be done with other means of pulverized-coal preparation except at reduced capacity. It has some advantage in pulverized-anthracite operations since the low-volatile and -moisture contents of anthracite reduce the hazards of bin fires. Stability of ignition in the burners can be maintained over a wide load range because of greater flexibility in the fuel-air mixture.

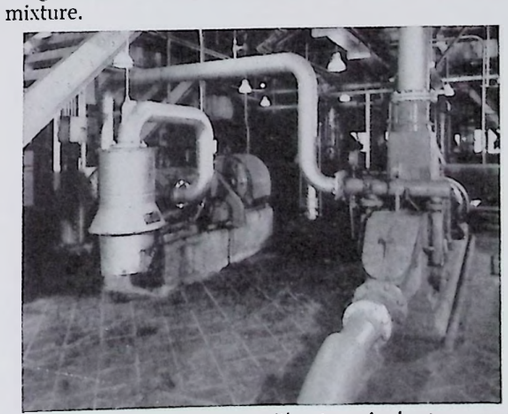


Fig. 5. *Air-transport system with rotary single-stage compressor at left and Fuller-Kinyon pump at right*

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However, there are inherent disadvantages in the use of the bin system. Except with anthracite, storage of pulverized coal for any length of time involves fire hazard from spontaneous combustion. Under certain conditions there may even be serious explosions of coal gas emanating from the fuel stored in the bin. Freshly pulverized coal is hygroscopic in character, and, after a time, it will pack and hang in the corners of the storage bin. That is, to a minor extent, true even with anthracite. In processing coal for storage, at least one third of the air circulating through the pulverizer must be vented. Dust-cleaning devices must be provided to reduce the vent nuisance and the loss of coal. Separately fired dryers require careful supervision and control of temperature to avoid unwanted ignition conditions in the dryer. The exhauster or blower must be explosionproof.

The bin system for installations in the metallurgical industry or for steam generation has been superseded by the direct-firing system because of improvements in safety conditions, plant cleanliness, greater simplicity, lower cost of operation, lower initial investment, and less space requirement.

Direct-Firing System

In the period 1920 to 1925 there was an increasing demand in the electric utility field for steam-generating units of greater capacity. This led to the development of pulverizing equipment to permit direct and continuous firing of coal processed as received without predrying and storage as in the bin system. A direct-firing installation is simple and efficient. The raw coal is dried and pulverized simultaneously in the mill. The cost of operation and the initial investment are lower. Typical direct-firing installations are illustrated in Figs. 6, 7, 20, and 25.

Components of the direct-firing system, in general, are as follows:

1. Steam or gas air heater to supply hot air to the pulverizer for drying the coal as pulverized.
2. Pulverizer fan, known as the primary-air fan, arranged either as an exhauster or as a blower.
3. Pulverizer arranged to operate under suction or pressure.
4. Automatically controlled raw-coal feeder.
5. Coal-and-air conveying lines.
6. Burners.

There are two direct-firing methods in use, 1) the suction and 2) the pressure type. In the suction method, the air and entrained coal are drawn through the pulverizer under negative pressure by an exhauster located on the outlet side of the pulverizer. In this case the fan handles a mixture of coal and air, and distribution of the mixture to more than one burner is controlled by a distributor beyond the fan discharge. In the pressure method, the fan, located on the inlet side of the pulverizer, either forces, under positive pressure, cold air through a steam-air heater or handles preheated air from a gas-air heater. In this case there is no dust-loading passing through the fan. Mixing of the coal and air is accomplished within the pulverizer housing. The principal differences between the two methods are summarized in Table 4.

TABLE 4

Direct-Firing Suction and Pressure Systems

Method	Suction	Pressure
Type of fan	Exhauster	Blower
Location of fan	Pulverizer outlet	Pulverizer inlet
Fan construction	Explosionproof	Standard
Fan handles	Pulverized coal and air	Air only
Relative fan eff	Low	High
Fan wear	High	Low to none
Pulverized-coal distribution	Tends to be difficult	Good

The feeding of coal and air to the pulverizer is controlled in two ways: 1) The coal feed is adjusted so that it is proportional to the load demand, and the primary-air supply is regulated to the rate of coal feed; 2) The primary air through the pulverizer is controlled so that it is proportional to the load demand, and the coal feed is automatically adjusted to the rate of air flow. In either case, a predetermined air-coal ratio is maintained for any given load.

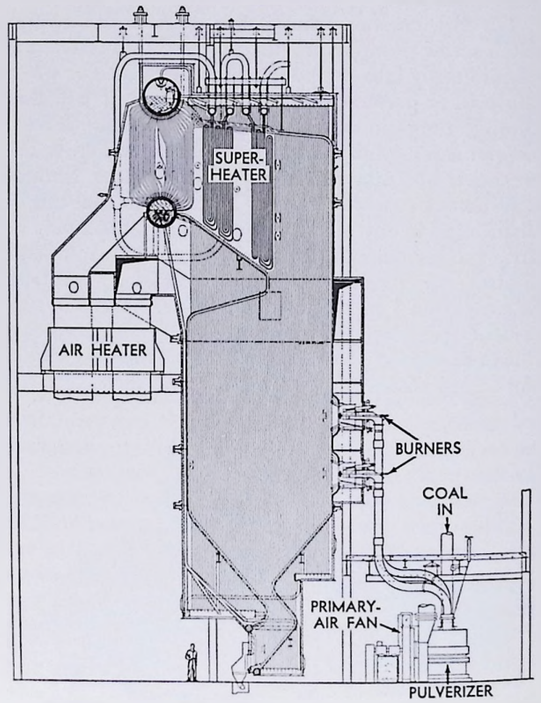


Fig. 6. *Stirling Two-Drum boiler (B&W) direct-fired with pulverized coal*

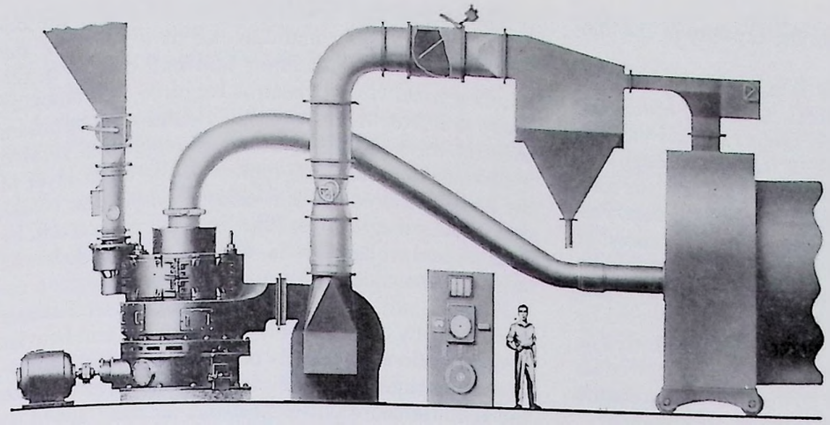


Fig. 7. *Rotary cement kiln direct-fired with pulverized coal*

Features of the Direct-Firing System

In the direct-firing system, the coal is dried in the pulverizer during pulverization and the finished product moves to the burners in a continuous uninterrupted pattern. The direct-firing system is widely used for steam generation, cement manufacture, and in processing nonferrous metals. The advantages of this system are:

1. Reliability in operation.
2. Flexibility in control.
3. Safety from fire hazard.
4. Compactness of the installation.
5. Low operating cost.
6. Low first cost.

Disadvantages of the direct-firing system are few and minor. The operating range, without changing the number of burners in service, is usually not more than 3 to 1 because the air velocities in lines and other parts of the system must be maintained above the minimum values to keep the coal in suspension (see Fig. 21). This limitation rarely obtains in practice because of the prevalence of multiple burners and multiple pulverizers on a single boiler unit. If the inlet air to the pulverizer is hot enough, coals with total moisture of 20% (surface moisture 15%) can be dried in the pulverizer. For such extremely high-moisture coals, an inlet air temperature of 600 F is practical. The predrying of coal before its introduction into the pulverizer is largely obsolete. High-moisture lignites, containing 20 to 40% moisture, are pulverized and fired with residual moisture contents above 10%.

Types of Pulverizers

The reduction of materials to a fine particle size for countless human needs is one of the oldest arts of civilization. Coal-pulverizing equipment may be said to be based, in general, on rock and mineral-ore grinding machinery. The principles involved in all pulverizing machinery are grinding by impact, by attrition, by compression, or by a combination of two or more of these.

The principal types of coal pulverizers may be classified under high, slow, and medium speed:

High Speed (above 225 rpm):
- Impact pulverizer

Slow Speed (below 75 rpm):
- Tube-mill pulverizer

Medium Speed (75 – 225 rpm):
- Roll-and-race pulverizer
  - a) Standard roll and race
  - b) Bowl mill
  - c) Disc and roll
- Ball-and-race pulverizer

High-Speed Pulverizers

In the high-speed impact pulverizers, the grinding elements consist of hammerlike beaters revolving in a chamber lined with wear-resistant plates. There may be one or more rows of beaters, and the shaft may be horizontal or vertical, the former being the more common. An exhauster is generally mounted on the pulverizer shaft with its inlet communicating directly with the pulverizing chamber. In the inlet end of the pulverizing chamber, where primary crushing takes place, means are provided for removal of tramp iron and other heavy foreign material. These pulverizers may be equipped with integral or external classifiers. Since the grinding efficiency is affected by wear of

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the grinding elements, either the capacity or the fineness decreases with wear.

Coal fed to the machine is pulverized and dried before delivery to the burners. A feeder, usually an integral part of the machine, feeds raw coal into the crushing chamber, whence it travels through the grinding zone, and then, induced by the fan, the mixture of coal and air is discharged to the burners. Preheated air is used to dry the coal during the process of reduction. Since the fineness of the finished product is affected by the air velocity through the grinding zone, the total heat available for drying is limited by the low primary-air to coal ratio and temperature. The absence of load circulation in this type of pulverizer imposes a further limitation on the amount of moisture tolerated in the coal for satisfactory performance.

Slow-Speed Tube-Mill Pulverizers

One of the oldest practical pulverizers is the tube mill, in which a charge of mixed-size forged-steel balls in a horizontally supported grinding cylinder is activated by gravity as the cylinder is rotated. With few modifications, the tube mill is still being used almost exclusively for wet and dry grinding of mineral ores and cement raw materials, and for the dry grinding of the cement kiln product (clinker). Its continued use in these industries, notwithstanding its low mechanical efficiency compared with some other types operating on entirely different principles of grinding, stems from its simple and sturdy construction, low maintenance costs at rated capacities, and ability to grind very abrasive materials.

Because of its dependability, the tube mill was considered a suitable unit for the direct-firing of pulverized coal when this system was first introduced. However, as the use of pulverized coal for power generation and in the metallurgical industry increased, the tube mill was gradually replaced by more efficient modern machines for grinding coal.

Much has been done to improve and adapt the tube mill for grinding coal, and it is successfully applied for this purpose where power is not a governing factor in the operating cost of the system and where there is ample room for installation.

A grinding cylinder, a ball charge, internal or external classifiers, and a raw-coal feeder are the principal components of the machine. Air is circulated over the charge to convey the finished product to the classifiers. The coal rejected in the classifiers returns to the grinding zone by gravity. This machine is used for direct-firing systems by positioning the pulverizer fan so that it draws from the classifier outlets and discharges to the burner lines. The conical-end feature of the grinding cylinder causes size segregation in both ball charge and material within the grinding zone.

The outstanding features of the tube mill are dependability and low maintenance as a result of the simple arrangement of liners and ball charge in the grinding zone. No provision is required for removing tramp iron from the raw coal. To obtain the optimum effect of the grinding elements, the speed is limited to produce the maximum impact of the outer layer of the balls on the toe of the charge, where the coarse coal usually segregates. The coal is pulverized by attrition and impact as the ball charge ascends and falls with the coal.

The tube mill is a slow-speed pulverizer. For any given capacity it is relatively larger in size and heavier in construction than high- or medium-speed pulverizers using different principles of grinding. The tube mill requires more power per unit output than other types. Because of the absence of load circulation within the mill, the excessive amount of fine product within the ball charge, and circulation of preheated air over the top of the charge, it is not as efficient in drying the coal as the medium-speed pulverizers with high air circulation through the grinding zone. This results in capacity reduction when handling wet coal.

Medium-Speed Roll-and-Race Pulverizers

In the medium-speed pulverizer there are two groups, classified as roll-and-race and ball-and-race. The same principle of pulverizing, by a combination of crushing under pressure and attrition between grinding surfaces and material, is used in each group, but the method is different. In both groups air is the predominant means for circulating the material through the grinding zone and conveying the finished product to the burners or cyclone collectors. The first group is operated under suction; the second group may be operated under either suction or pressure.

The roll-and-race class of pulverizer is subdivided into two types, known as: 1) standard roll-and-race pulverizer, where the ring is stationary and the rolls rotate; and 2) bowl mill and disc-and-roll mill, where the bearing rolls are stationary and the ring rotates.

Standard Roll-and-Race Pulverizer. In the standard roll-and-race pulverizer, the grinding elements of four or more rolls, suspended from driving arms, revolve in a horizontally positioned race. The pressure exerted by the rolls against the race is proportional to the centrifugal force resulting from the speed of the driving shaft. Roll-and-race pulverizers of this type have been used extensively in processing coal for the bin system and for grinding various materials in the ceramic and chemical industries. Air is the principal means for load circulation, for conveying the fine coal to the classifier, and for collecting the finished product. By the introduction of preheated air it is possible, within certain limits, to pulverize coal without pre-

drying. However, this type of pulverizer is not usually considered suitable for direct-firing conditions, as it has to be taken out of service to lubricate the internal roll journals.

A classifier above the grinding elements is an integral part of the pulverizer. It is a high-sided chamber with an internal cone classifier provided with adjustable vanes and a reject discharge trap at the bottom.

The feeder is usually driven direct from the main drive. The air-coal ratio is adjusted, as required, by an automatic control actuated by the pressure differential across the pulverizer.

Bowl Mill. A rotating bowl equipped with a replaceable grinding ring, two or more tapered rolls in stationary journals, an automatically controlled feeder, a classifier, and a main drive are the principal components of the bowl mill. As noted above, the bowl mill is another type of the roll-and-race class of pulverizer. The principal differences from the standard roll-and-race pulverizer are the methods of applying grinding force on the material and of obtaining load circulation within the grinding zone. In the standard roll-and-race pulverizer, the grinding action depends entirely upon the centrifugal force which presses the rolls against the race, while in the bowl mill the rolls are held in the desired position relative to the grinding ring by mechanical springs and the centrifugal force acts only to feed the coal between the race and the rolls. The bowl mill is suitable for the direct-firing system since the roller journals can be lubricated and the rolls can be adjusted without shutting down. The fineness of the coal is externally controlled by adjusting the entrance vanes in a classifier of the same general design as used with the standard roll-and-race type of pulverizer and by controlling the pressure of the rolls on the material by adjusting the compression springs.

The bowl mill is usually designed to operate under suction, and the pulverizer fan is placed on the outlet side of the classifier. Heavy scroll liners and rotor blades are generally provided to withstand the abrasive action of the material on the fan. These mills are used extensively in direct-firing systems, for steam boilers, and for cement kilns.

Disc-and-Roller Mill. In construction and operation this machine is similar to the bowl mill. The principal differences are that the bowl has been flattened out into a disc and the replaceable ring driven by the rotating disc is essentially flat. There are two or more tapered rollers loaded by external springs or air cylinders to produce the desired pressure between the rollers and the race. These pulverizers are equipped with an integral classifier of the cyclone type or a rotating member driven by a separate motor-gear reducer unit.

Medium-Speed Ball-and-Race Pulverizers

The principle of pulverizing by compression, impact, and attrition is the same, in both the ball-and-race and the roll-and-race medium-speed pulverizers. The distinguishing feature is the grinding elements. In the ball-and-race pulverizer these elements are balls and one or more races. High load circulation through the grinding zone, very desirable for effective drying and classification of the finished product, is a feature of the ball-and-race pulverizer.

The ball-and-race pulverizer works on the ball-bearing principle. The B&W coal pulverizers are of the ball-and-race type and are classified as Types B and E. Type B pulverizers are further divided into what is known as the 100 Series and the 200 Series, with one and two sets of balls, respectively. A single row of balls is used in all Type E and Type EL pulverizers. (See Figs. 8, 9, and 10.) The Type EL pulverizer is similar to the Type E except for modified grinding rings and throat construction.

B&W development is toward new designs of pulverizers for commercial application to pulverized-coal preparation and firing, with due consideration to high capacity, first cost, space limitations, dependable service, low power consumption, and low maintenance.

Type B 100-Series Pulverizer. In the Type B 100-Series pulverizer (Fig. 8) there are two races, one revolving and one stationary, with one set of balls interposed, rotating in a horizontal plane. The revolving race (the top grinding ring), driven by a vertical shaft through a yoke-and-spring assembly, provides traction and rotating motion to the balls. Raw coal is fed to the outer ball periphery, and the pulverized coal, coarse and fine, is conveyed by air through the radial ports in the driving ring. Classification is effected in the housing above the grinding zone, and fineness is con-

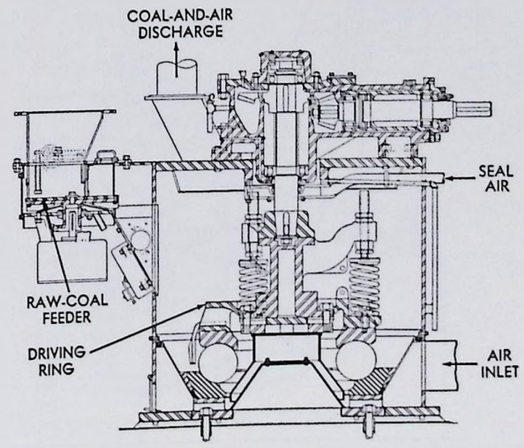


Fig. 8. *B&W Type B single-row ball-and-race pulverizer*

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trolled entirely by varying the air flow through the pulverizer. Because of the high circulating load and high air-coal ratio possible through this type of pulverizer, high-moisture coal can be pulverized satisfactorily. Capacities for this series range from 0.25 to 7.5 tons/hr with bituminous coals of 50 grindability (ASTM) and a product fineness of 70% through a 200-mesh sieve.

Type B 200-Series Pulverizer. In the Type B 200-Series pulverizer (Fig. 9) an intermediate revolving grinding ring is interposed with two sets of balls between the races of two stationary rings (an upper and a lower). The intermediate ring provides traction and rotating motion to the two sets of balls. The bottom ring, fixed in position, carries the entire thrust load. Through the top ring (with race on under side), supported from the housing but free to move up and down within limits, force is applied to the grinding-element assembly by four or more multiple-coil springs. Spring adjustment, without shutting down the pulverizer, is made from the outside of the housing by screw-down bolts.

Raw coal is fed inside the grinding rings to the table attached to the intermediate revolving grinding ring. As it passes through the top row of balls, the coal is partially pulverized, and the fine particles are entrained and conveyed by air to the classifying section. The coarse particles fall behind the lower row of balls and flow by gravity through the bottom grinding zone. Preheated air from the windbox, flowing through the throat orifice between the bottom grinding ring and the driving yoke, entrains all the coal, coarse and fine, as it passes through the lower row of balls. This mixture of air and coal flows into the pulverizer housing through the radial ports in the rotating drive ring and then into the classifier for final separation.

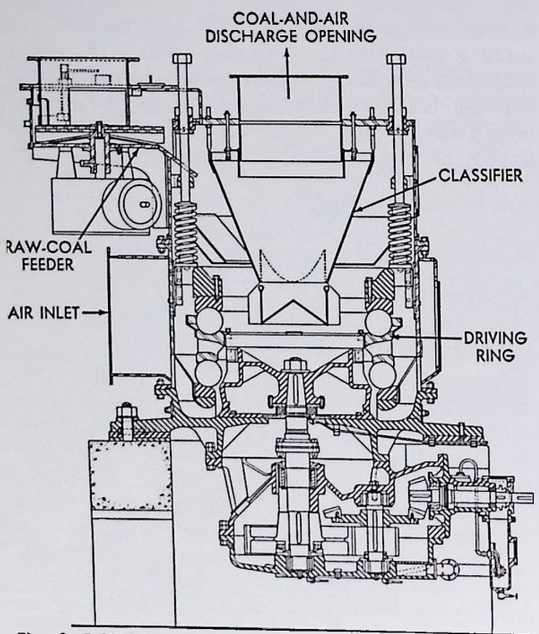


Fig. 9. B&W Type B double-row ball-and-race pulverizer

The classifier is a concentric cone provided with adjustable inlet vanes and a reject trap at the bottom. Within a limited range, the fineness of the finished product is determined by the position of the outlet sleeve and inlet vanes. Low air velocity through the mill housing is one of the controlling factors for a high degree of fineness. Capacities for this series range from 7.5 to 30.5 tons/hr with bituminous coals of 50 grindability (ASTM) and a product fineness of 70% through a 200-mesh sieve.

The revolving grinding ring is propelled by the main shaft driven by a spiral bevel gear-and-pinion assembly in the gear housing. The housing also supports the top section and functions as the base area on the foundation. Both main and pinion shafts are mounted in roller bearings, and the thrust from the weight of the drive and yoke is carried by a step roller bearing integral with the gear housing. The entire gear assembly and bearings are force lubricated. The oil pump, driven from the pinion shaft, is connected to the main oil reservoir in the gear housing with an intermediate oil strainer. No lubrication is required in the grinding zone, thus avoiding shutdown on this score.

Type EL Pulverizer. The Type EL pulverizer, illustrated by Fig. 10, has one revolving and one stationary race with one interposed set of balls propelled in a horizontal plane. Through the nonrotating top ring (with race on under side), grinding force is applied to the lower rotating ring (with race on upper side) by mechanical springs externally adjusted for more or less pressure. The raw coal fed to the grinding zone is swept into the path of the balls by the centrifugal force from rotation. The yoke, with attached grinding ring, is driven by a vertical shaft extending from the gear housing through the air seal above the upper radial bearing.

The stationary ring, known as the top grinding ring, is held in position by the dual-purpose single-coil springs. The bottom ends of the springs are secured to lugs on the grinding ring, and the upper ends are secured to the adjusting screw-down bolts. The springs are designed for the dual purpose of providing the necessary pressure on the balls and resisting the turning moment of the ring.

Finely pulverized coal is first separated from coarser particles by the swirling action of the air and coal imparted by the rotation of the grinding ring and balls. Final separation takes place in a classifier attached to the pulverizer top. Oversize material is returned by

gravity through rotating seal vanes to the grinding zone, where it mixes with the raw coal.

The classifier is a stationary cyclone that has inlet ports made up of flat blades with openings counter to the direction of rotation. Coarse particles are removed from the air stream by inertia. This arrangement utilizes the swirling action of the coal-laden air outside the classifier. The fineness of the finished product is controlled by the length of the adjustable blades forming the inlet.

Foreign material, such as tramp iron and pyrites, that may enter with the raw coal is collected in a "pyrites trap" and periodically removed.

The raw-coal feeder of the rotating-table type may be integrally mounted on the pulverizer or separately

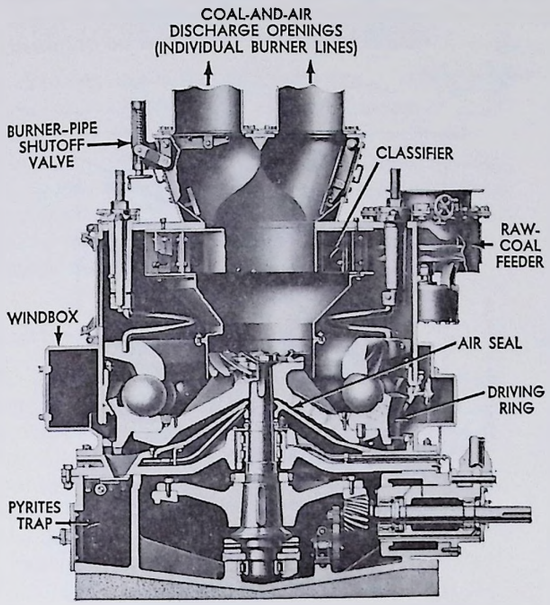


Fig. 10. B&W Type EL single-row ball-and-race pulverizer

mounted above it. The feeder is independently driven and controlled by a two-speed motor. The rate of feed is controlled either by the speed of the feeder and fixed plow blade or by a variable-opening gate at the discharge from the table.

The main drive, housed in the base of the pulverizer, is completely separated from, and sealed against, the grinding zone. A spiral bevel gear is driven by a pinion shaft mounted in roller bearings. The main shaft, on which the bevel gear is mounted, is supported on a thrust bearing and is guided in two vertical roller bearings, one at the base of the shaft and the other in the gear-housing cover plate. The entire thrust load from the grinding pressure and the weight of the grinding elements is supported by the thrust roller bearing at the bottom of the shaft. Forced lubrication is provided for the entire gear assembly and bearings by an oil pump submerged in the oil reservoir and driven direct from the pinion shaft. Except for the feeder gear-reduction unit, no other lubrication of the pulverizer is necessary. Since the applied force on the grinding elements is adjusted externally, it is possible to operate the pulverizer continuously for long periods of time, shutting down only for inspection and periodic maintenance.

The Type EL pulverizer is made in sizes from EL 17 (17-in. pitch diameter at the race) to EL 76 (76-in. pitch diameter at the race) with capacities from 1½ to 20 tons/hr, respectively, based on bituminous coals of 50 grindability (ASTM) and a product fineness of 70% through a 200-mesh sieve.

One of the important features of the ball-and-race type of pulverizer is that it operates under positive pressure with drying and conveying air supplied by the primary-air fan, thus eliminating wear on the fan rotor and housing. This allows a wide latitude in the selection of a high-efficiency fan. A further advantage is better coal-and-air distribution to a plurality of burners in a direct-firing system, since each burner take-off is connected to a common outlet wherein there is a uniform mixture of coal and air.

TABLE 5

Characteristics of B&W Type EL Ball-and-Race Pulverizers

Bituminous Coal at 50 Grindability (ASTM) and 70% Through 200 Mesh

Type	Ball-and-race
Size range	EL 17–EL 76
Output, max tons/hr	1½–20
Speed	Medium
Rpm of main shaft	220–82
Operates under	Pressure or suction
Fan location	Pulverizer inlet or outlet
Load circulation	High
Classification	Internal
Classification control	Mechanical adjustment
Drying, range of moisture	Up to 20% total, 15% surface
Effect of moisture on output	None up to 15% surface if air is hot enough
Effect of wear on fineness	None
Method of control	Control primary air, which controls coal feed
Response to load change	Fast
Power input per unit output, including fan	Low, 14 kwhr/ton
Maintenance, material only	Low, 0.5 to 3.8¢/ton
Noise level	Medium

Summary of Pulverizer Characteristics

There are many variations in the design of pulver-

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izers currently used in different industries. In their application to the process of grinding various materials, consideration should be given to the design characteristics as they influence dependability in performance, economy in operation, and initial investment. The optimum selection would naturally be the design that produces the desired results at least cost per unit output. Operating characteristics and the principles of design of the B&W Type EL ball-and-race pulverizers are summarized in Table 5.

Selecting Pulverizer Equipment

A number of factors must be considered when selecting pulverizer equipment. If selection anticipates the use of a variety of coals, the pulverizer should be sized to the coal that gives the lowest "base capacity." "Base capacity" is the product of desired capacity, fineness factor, and grindability factor. The extent of drying within a given pulverizer depends upon its design and the method used to introduce preheated air into the grinding zone. Raw coal with very high surface moisture (about 15%) can be efficiently dried when fed into the grinding zone of a pulverizer designed for a high internal-circulating load. On the other hand, where the design does not permit rapid mixing of raw coal with the partially dried coal within the grinding zone, the pulverizer is not suitable for handling high-moisture coal, and the grinding capacity will drop as the surface moisture increases.

The two controllable elements in drying are the temperature and the weight of air, since the total heat input is proportional to the temperature and the weight of air injected with the coal. However, for a given pulverizer performance the weight of primary air is usually a fixed quantity, which leaves the temperature as the only variable.

The percentage of volatile matter in the fuel has a direct bearing on the probability of premature ignition of the fuel-air mixture at the burners. The generally accepted safe values for pulverizer exit fuel-air temperatures are given in Table 6.

TABLE 6

Pulverizer Exit Fuel-Air Temperature

Fuel	Exit Temp, F
Lignite	120
High-volatile bituminous	150
Low-volatile bituminous	150–175
Anthracite	200
Petroleum coke	200–250

The temperature of the entering air may run as high as 600 F, depending upon the amount of surface moisture and the type of pulverizer.

Fineness is accepted as a measure of the new surface produced in the pulverizing operation. It is expressed as the percentage of the product passing through various sizes of sieves, graded from No. 16 to No. 325 in the ASTM designation.

Fine grinding of coal is necessary to assure complete combustion of the carbon for maximum combustion efficiency and to avoid the deposit of ash and carbon on the heat-absorbing surfaces. This applies not only in the firing of steam boilers but also in other applications where close temperature control and the avoidance of carbon contamination are important. Chemical and metallurgical processes using pulverized coal as a source of thermal and/or chemical energy generally require very finely ground coal to insure the optimum reaction in a limited combustion zone and often under difficult firing conditions. The coal classification by rank and the end use determine the fineness to which it should be ground. This can be obtained from Table 3.

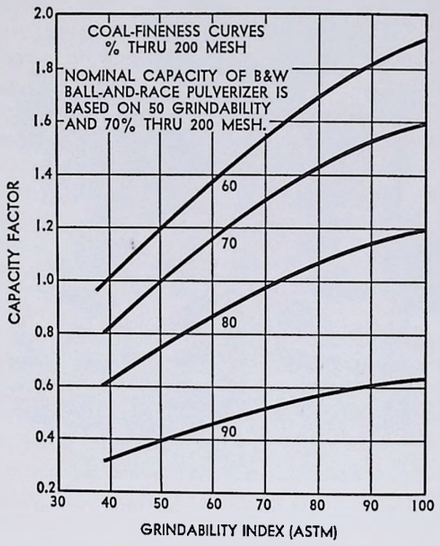


Fig. 11. *Effect of grindability and fineness on pulverizer capacity*

The effect of fineness on capacity is indicated in Fig. 11, where the capacity factor is the product of the fineness factor and the grindability factor. "Base capacity" for a B&W pulverizer is the product of the desired capacity and the capacity factor.

The range through which the equipment will operate must be considered in the selection of pulverizing equipment for direct firing. The range through which a single pulverizer can operate is an inherent feature of the pulverizer. For B&W Type B 100-Series and Type EL pulverizers it is of the order of 4 to 1. However, the range through which it may be safely operated depends upon the type of burner, number of burners, type of fuel, and whether furnace is "hot" or "cold."

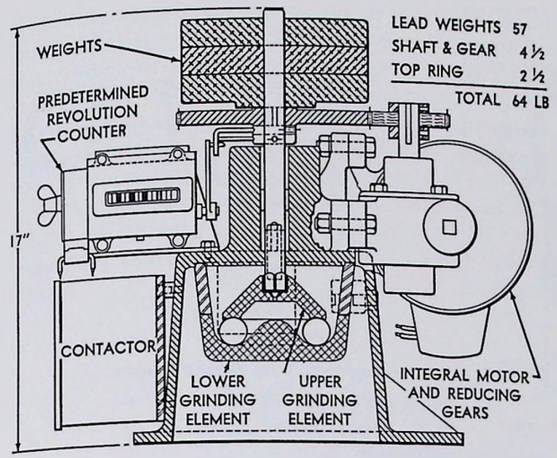


Fig. 12. *Hardgrove Grindability Machine*

When the B&W pulverizers are operated at fractional loads, the fineness of the product increases automatically as the air flow is decreased. This is beneficial to low-load operation as it assists in ignition and flame stability.

Grindability Index (ASTM)

The grindability of a coal is expressed as an index showing the relative hardness of that coal compared with a standard coal chosen as 100 (unity) grindability. Thus a coal is harder or easier to grind if its grindability index is less or greater, respectively, than 100. The capacity of a pulverizer is related to the grindability index of the coal.

A method of testing to determine the grindability index of coal is described in ASTM Standard D 409. The method is based on the premise that "The work done in pulverizing is proportional to the new surface produced." A definite amount of grinding energy is applied to a prepared sample in a miniature pulverizer (Fig. 12), and the new surface is determined by sieving.

Auxiliary Equipment

Pulverizer Feeder Controller. All ball-and-race pulverizers operating as air-swept mills have a common characteristic in that the material-air ratio should be maintained at some definite relation. The ratio depends upon the type of pulverizer, installation, and material characteristics. For efficient and consistent results, operation should be kept at the established ratio.

For any fixed set of conditions there is a precise relation between the pressure drop across the pulverizer and the pressure differential of a pitot tube or orifice in the air supply to the pulverizer. This relation is the principle on which the feeder-controller system, illustrated in Fig. 13, is based. Only the air flow through the pulverizer need be varied (manually or by combustion control) to meet load requirements. The feeder controller then automatically adjusts the coal feed to the air flow. Hence, by the use of this controller the correct material-air relation can be maintained automatically over the entire operating range of the pulverizer while at the same time keeping (a) the proper velocity through the conveying lines at the lowest practical load and (b) satisfactory fineness at maximum output.

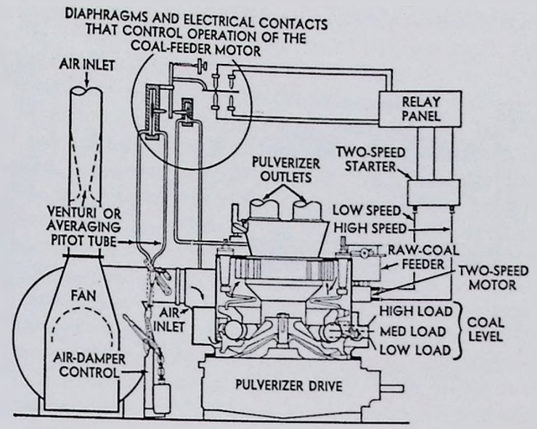


Fig. 13. *B&W automatic feeder controller applied to Type EL pulverizer for regulating coal supply*

Sampling Equipment. There is no universally adopted procedure for sampling pulverized coal, but the "Test

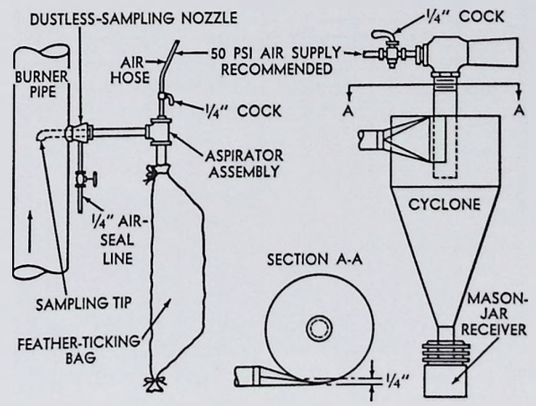


Fig. 14. *Recommended arrangement for sampling pulverized coal in a direct-fired system using aspirator and bag or cyclone collector*

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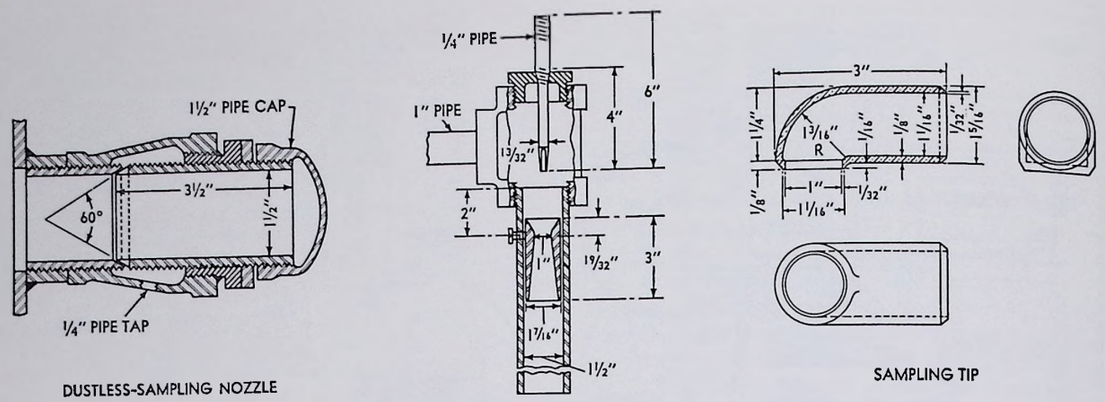


Fig. 15. *Pulverized-coal-sampling equipment*

Code for Coal Pulverizers" (ASME) and the ASTM Standard D 197 may be considered as good practice.

Two general arrangements of the equipment required for sampling pulverized coal in a direct-fired system are shown in Fig. 14, the principal difference being that in one (shown at the right) the bag collector is replaced by a cyclone collector.

Details of the important parts of the equipment used for sampling pulverized coal in burner lines are given in Fig. 15.

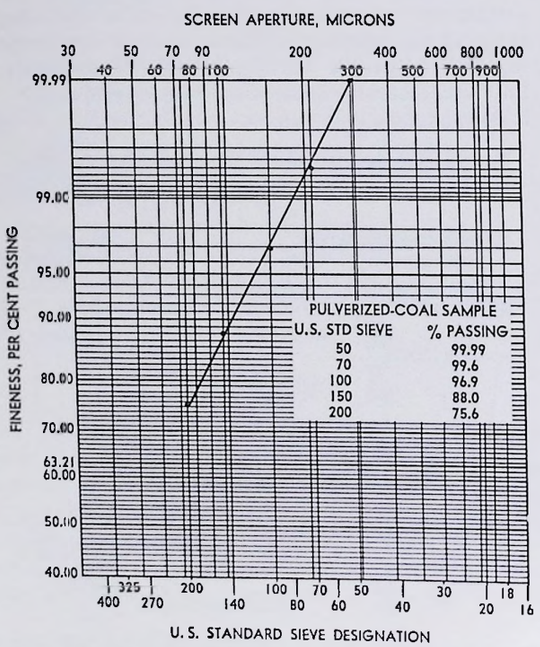


Fig. 16. *Rosin and Rammler chart for plotting pulverized-coal-sample sieve analyses*

The results of sieve analyses of pulverized-coal samples should be plotted on a Rosin and Rammler chart, Fig. 16. This graphical form of presenting size distribution is explained in The Bureau of Mines Information Circular 7346 (1946). If the points fall on a relatively straight line, the sample can be considered as representative. If there is a sharp or abrupt break in the curve, the sample should be discarded.

Exhausters and Blowers. In direct-firing, primary air is required for classifying and conveying the finished product to the burners.

Primary air to the pulverizer is supplied by either (a) a standard commercial blower handling clean air and discharging directly into the pulverizer or (b) an exhauster handling pulverized-coal-laden air from the pulverizer. With a blower, the pulverizing system is under pressure, and the fan is not subjected to abrasion by the pulverized coal. In this case a high-efficiency fan can be used since the conditions permit an efficient rotor design and high tip speed. When the fan or exhauster is handling pulverized-coal-laden air, the pulverizing system is under suction. To comply with the National Fire Protection Association, the exhauster housing must be designed to withstand a pressure of 200 psi in case of an explosion within the fan. Furthermore, since the exhauster is subject to excessive wear, the design is limited to a paddle-wheel type of heavy construction and hard-metal or other protective-surface coatings. All of these construction features are detrimental to the mechanical efficiency of the fan (see Table 4).

The speed of the pulverizer fan depends upon the pressure requirements of the system but is generally about 1800 rpm. The fans are usually independently driven, but in small sizes one motor may be used to drive both the pulverizer and the fan.

Power consumption of the pulverizer fan is an important item in the operating cost, and the selection

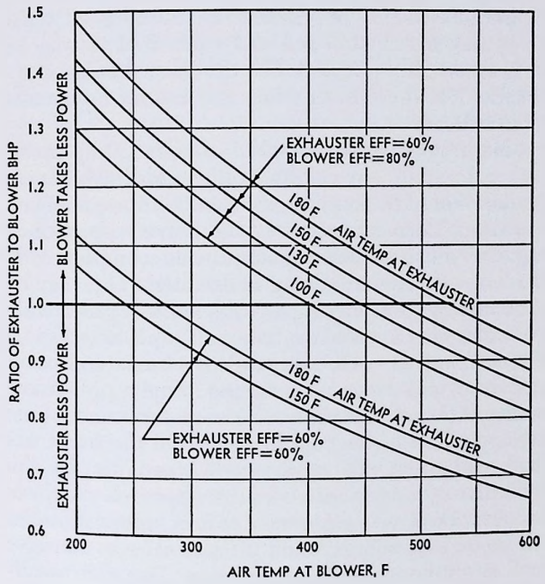
of the fan should receive careful consideration. For a direct-fired installation, the fan must overcome the flow resistances through the pulverizers and burner lines as well as provide the pressure required at the burner. The resistance through the pulverizer depends upon the inherent characteristics of the pulverizer and the amount of drying accomplished in it. The resistance through the discharge lines is a function of their length and the air-coal velocity that must be maintained to prevent settling. Table 7 shows average figures.

Table 7

Average Velocity of Primary-Air-and-Coal Mixture Through Conveying Pipes

	Air-Coal Ratio, 2 Lb Air/Lb Coal		
Pulverized-coal fineness, % through 200-mesh sieve	70	80	85
Normal velocity, fpm	5000	4500	4000
Minimum velocity, fpm	3000	3000	3000

Fig. 17 shows average relative power consumption, under given conditions, for a standard commercial blower handling clean air and a fan used as an exhauster handling coal-laden air.



NOTE: EXHAUSTER BHP INCREASES 15% OVER CLEAN-AIR POWER BECAUSE OF SEPARATION OF COAL ON FAN BLADES AND DIRECT ACCELERATION OF THE COAL. ASSUMED SAME PRESSURE DROPS FOR BOTH SYSTEMS.

Fig. 17. *Comparison between power consumption of blower and exhauster for given set of conditions*

Cost of Pulverizing Coal

The operating costs per unit of output are affected by (a) size of the installation, (b) characteristics of the coal, and (c) fineness and distribution of the finished product.

The life of the grinding elements of the ball-and-race type of pulverizer varies from 6000 to 32,000 hours of operation. For bituminous coal the rate of wear is generally a function of the volatile matter and the sulfur in the coal. The operating cost per unit of output is a function of the total coal pulverized in the life of the grinding elements. The maintenance cost is usually expressed in cost per ton of coal pulverized. For direct-fired systems, the power consumption per ton of coal delivered to the burners is the sum of the power required to drive (a) the auxiliaries delivering coal to the pulverizer, (b) the pulverizer, and (c) the fan delivering air to the pulverizer and coal to the burners. When pulverized coal is delivered to a group of furnaces located at some distance from the pulverizer, the power consumption of the fan is considerably higher than that for the standard direct-fired system.

Burners—General

For optimum utilization of the heat released in burning a substance, the fuel burner is surrounded by a combustion chamber or furnace to (a) provide space to burn the fuel and (b) form a flue or duct for conveying the hot gases to the remainder of the heat-absorbing surfaces or processes. This discussion is concerned particularly with coal fuels burned in suspension under steam-producing equipment. (See Chapter 28 on *The Cyclone Furnace* and Chapter 16 on *Stokers* for comparison with other methods of burning coal.)

The appropriate fineness to which coal must be reduced is summarized in Table 3 for different classifications of coal by rank and for different types of furnaces.

The quantity of air required to burn any fuel depends upon its constituents. For a discussion of combustion-air requirements and the usual amounts of excess air supplied to fuel-burning equipment, see Chapter 4.

It is essential to control the quantity of combustion air, since its use in excessive amounts decreases efficiency. It is now general practice to measure (with a boiler meter or other means) both the air and the coal injected into a combustion chamber, and it is recognized that, even in the best apparatus used to combine air and coal for combustion, some air in excess of theoretical is required.

Pulverized coal is dispersed in an air stream in the pulverizer for the direct-firing system and immediately following the pulverized-coal feeder for bin-system firing. The mixture of fuel and primary air is conveyed through suitable pipes to the burner. Pri-

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mary air represents only a small portion of the air required to burn the fuel. For direct firing, its quantity depends upon the pulverizer characteristics and ranges from 15 to 20% of the total air required for combustion. The remainder is known as secondary air. The burner is the device by which the primary air, fuel, and secondary air are mixed just prior to, or immediately after, their introduction into the furnace.

Mechanism of Burning Fuel

Fundamentally, the mechanism of burning a particle of coal in its quota of air is the union of oxygen with the substance of the outer-surface layer of the particle to form CO_2, leaving residual ash and residual nitrogen. The inert CO_2 (plus the inert nitrogen) must be displaced and the residual ash must be permeable if more oxygen is to penetrate to unite with a new outer-surface layer of carbon in the original particle. Thus, to continue the process of burning until all the carbon has united with oxygen, prompt and continuous removal of the outer envelope of inert gases is necessary. This requires a scrubbing action. An example of this action is the moving stream of air over a stationary piece of coal on a stoker, scrubbing off products of combustion and bringing new air (oxygen) to the new surface. Another striking example of scrubbing action is in the Cyclone Furnace (Chapter 28), where the coal particle adheres to the molten slag surface and is burned and scrubbed to the vanishing point.

With pulverized coal this necessary scrubbing action is not as evident as in the Cyclone Furnace. In the coal-and-primary-air stream there is little relative velocity between the coal particle and the conveying air since the velocity of the air is regulated to keep the fine pulverized-coal particles in suspension and moving with the air. However, secondary air for various types of burners is introduced to coal particles carried by the primary air at a slightly different direction of travel and at a different velocity (usually higher). Thus the inert envelope repeatedly forming around each coal particle is displaced by continuous scrubbing, and oxygen is admitted to successive surface layers of the particle until only the ash remains.

Experience has shown that the angle of confluence of the primary-air-and-coal stream with the secondary-air stream is quite important. Even two burners can interact with each other to produce disappointing results. If the secondary air is introduced into the furnace in the same direction as the coal particles, the whole mass tends to assume the same velocity with a reduction in scrubbing effect so necessary for prompt burning.

Standards of Burner Performance

Operators of pulverized-coal equipment should expect burner performance to meet the following conditions:

1. For most modern applications, ignition of the pulverized coal should be stable, without the use of an auxiliary fuel, over a load range of 3 to 1. Since the majority of steam boilers is equipped with a multiplicity of burners, considerable range of capacity in excess of 3 to 1 may be obtained by varying the number of burners in use.
2. Unburned-combustible loss should be less than 2%. With most modern installations of well-designed equipment it is possible to keep the unburned-combustible loss under 1% with excess air in the range of 15 to 22%.
3. Only minor repairs should be required at annual overhauls. Burner parts subject to abrasion, that may require replacement at more frequent intervals, should be easily renewable at reasonable cost. Parts subject to high temperature, that cannot be protected by other means, should be made of alloy steels.
4. Burners should not require adjustments to maintain flame shape and freedom from deposits that will interfere with the continued efficient and reliable performance of the boiler unit in the normal range of operating load.
5. Burners should be efficient in burning all types of coals anticipated and such other fuels as may be economically justified. The designer should be apprised of these facts when the burner equipment is ordered.

Measurement and control of air (including excess air) and coal for any combustion process for heat generation should be considered a minimum requirement. However, in many instances the correct proportioning of air and coal does not alone insure complete combustion, as intimate mixing is essential. This may be illustrated by considering an early form of pulverized-coal burner — a round or flattened pipe located in a furnace wall or arch. Although the sum total of air and coal may have been in the proper proportion, some portions of the air were always too remote from some portions of the coal to unite, and the result was both unburned coal and uncombined air. In this primitive arrangement, even to approach complete combustion it was customary to use values of excess air as high as 50%. Ignition of the fuel was sensitive and might be lost without warning. The unit usually had to operate at or near a base-load condition. The loss in efficiency from unburned combustible was in the order of 4%, and maintenance costs were high. This was wasteful, and in addition there were the cost of moving the air and the cost of otherwise usable heat lost to the stack.

Stability and Ignition—Velocities

Perfect dispersion of the fuel in the air, while desirable for completeness of combustion, does not insure stability of ignition. To obtain ignition stability it is necessary that a small part of the fuel stream should not be so thoroughly dispersed in the air and that it should move at a lower velocity than the main stream. To attain this needed low-velocity rich stream is not difficult since there is a certain amount of drag in the different parts of the burner. On the other hand, once coal dust actually settles out because of too low a velocity in transportation, it is not easily put in suspension again. Rather it tends to "drift" along to the burner. Such drifting frustrates any attempt to subdivide the mixture and can be hazardous if slugs of pulverized coal are suddenly and erratically allowed to dribble into a hot furnace.

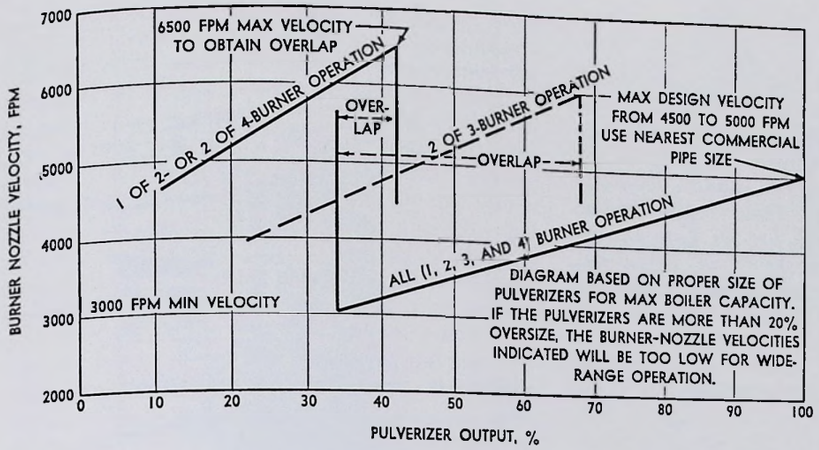


Fig. 18. *Burner-nozzle velocities for various numbers of burners per pulverizer*

Current practice is to use primary-air burner-tip velocities above 3000 fpm for horizontally arranged burners and above 2000 fpm for vertically arranged burners. Experience has established that these velocity limitations are necessary to avoid settling of the fuel particles and consequent formation of coke within the burner ports. Maximum velocities range from 4500 to 6500 fpm for all burners in service. The upper limit depends upon a number of factors, the most important of which are: pressure drop (reflected in the additional power requirements for the primary-air fan), stability of ignition, and wear of the burner parts. Common burner-nozzle velocities for different loads are indicated in Fig. 18. The diagram represents operation with one, two, three, or four burners per pulverizer. Fig. 1 shows a complete coordination chart. For large multipulverizer-fired boilers, an entire pulverizer with all its burners is normally put into or taken out of service as required by the load, instead of shutting down or lighting off individual burners.

Most pulverized-coal burners of all types are equipped with an impeller, or equivalent device, which performs the following functions:

1. When the coal-air stream, at the velocity required to keep the fuel in suspension within the burner proper, strikes the impeller, the velocity of a portion of the fuel particles (the finer components) is reduced so that ignition can take place.
2. The impeller (or equivalent) deflects the fuel particles into the stream of secondary air, and thus a more intimate mixing of air and coal is obtained.

The velocity of the secondary-air stream varies for different burner types, the temperature of air involved, and the construction and arrangement of furnace envelopes. In the following description of several different types of burners, the secondary-air velocity required for each is discussed.

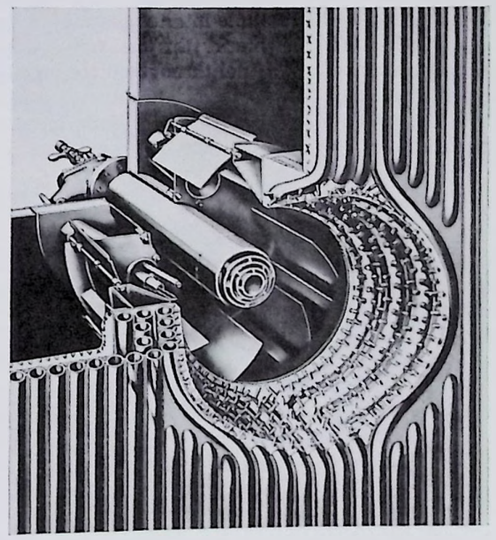


Fig. 19. *B&W circular burners for pulverized coal*

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Modern Burner Types

Circular Burner. The most frequently used burner is the circular type, illustrated in Fig. 19. This burner has several desirable characteristics. It does not require much attention to obtain reliable and efficient performance. It can be equipped to burn efficiently either gas or oil as alternate fuels. The individual burners of this type, shown in Fig. 20, have a maximum capacity of approximately 165 × 10^6 Btu/hr.

In the circular type of burner, the tangentially disposed vanes built into the air register mix the fuel and air and produce the necessary scrubbing action on the fuel particles. Thus, while the fuel is introduced to the burner in a fairly dense mixture at the center, its dispersal by the impeller is such that it penetrates the secondary-air stream. This burner is generally used on dry-ash-removal furnaces. However, it is occasionally used in slag-tap furnaces with the burner wall formed entirely of furnace-cooling tubes.

For burners of this type installed in dry-ash-removal furnaces, the velocity of secondary air, as measured in the annular space surrounding the end of the coal nozzle, ranges from 4000 fpm with unheated secondary air to 6400 fpm with 300 F air. For burners of this type installed in slag-tap furnaces, the velocity of 600 F secondary air, measured at the same location, is 7500 fpm. The velocities noted are all at maximum burner capacity.

Multiple-Intertube Multitip Burner. Another successful burner design is the multiple-intertube multitip type, shown in Fig. 21. The burner tips are arranged for optimum distribution of fuel throughout the secondary-air stream. In the type shown, the fuel stream is divided into 12 smaller streams, each introduced to approximately 1/12 of the secondary air. There are

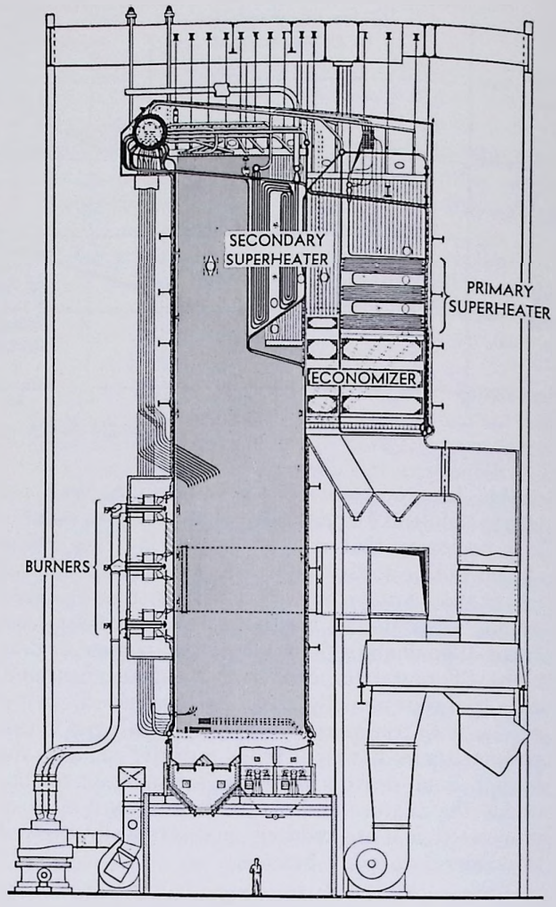


Fig. 20. *B&W circular burners installed in large direct-fired boiler*

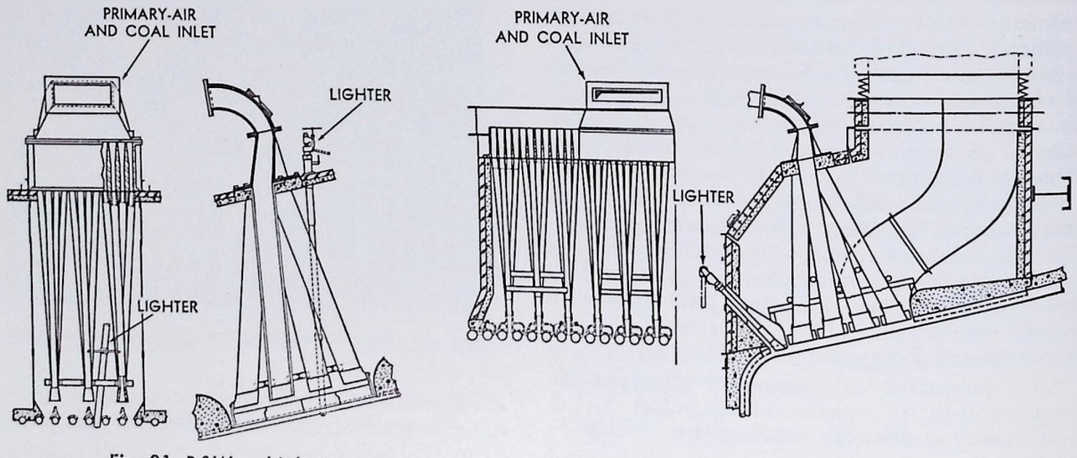


Fig. 21. *B&W multiple-intertube multitip pulverized-coal burners. Form (a) left, form (b) right*

two general forms of this burner type. In one form, Fig. 21 (a), the fuel leaving the burner tips strikes a tube protector and is deflected laterally into the secondary-air stream flowing between the tubes. In the other form, Fig. 21 (b), the direction of the fuel ports is into the spaces between the water-cooled tubes of the furnace roof, and the secondary air, admitted through inclined ports, meets the fuel stream about a foot below the furnace roof tubes. The first form is used for coals that are not highly abrasive, such as those from the eastern part of the United States, whereas the latter is used for the more abrasive coals prevalent in the central part of the country.

In the burner shown in Fig. 21 (a), the tube protector functions similar to the impeller for the circular burner, deflecting the fuel into the secondary-air stream and protecting the tube directly under it from abrasion. The tube protectors also serve as a "drag" on a small part of the fuel stream, creating a low-velocity zone necessary for stable ignition. Tube protectors usually have smooth surfaces, but, for a coal difficult to ignite, the surfaces are ribbed to increase the number of slower-moving streams in the ignition zone.

The burner shown in Fig. 21 (b) employs high-velocity secondary air to mix with the lower-velocity fuel stream to achieve good combustion. This burner is used primarily with Midwest coals that ignite easily but are also highly abrasive. Since the fuel is not abruptly deflected by any wearing surface in its passage into the furnace, burner maintenance, even with a highly abrasive coal, is nominal.

Both forms of this burner were developed initially for the slag-tap furnace, to fire downward on the slag pool to facilitate melting and tapping. These burners are still used for this service, particularly in large boiler installations. Secondary-air temperatures are usually about 450 F. The burners are shielded from direct furnace radiation by water-cooled tubes that form the burner ports and the roof of the furnace. This protection from direct radiation permits a construction with a minimum of heat-resisting materials.

In recent years this type of burner has been successfully applied to the dry-ash-removal furnaces, where there are many satisfactory installations. This burner is available for alternate firing with oil or gas. The range in size for both the combination fuel burner and the all-coal burner is from 50 × 10^6 to 165 × 10^6 Btu/hr.

Cross-Tube Burner. Still another type is the cross-tube burner (Fig. 22), in which the primary-air-and-fuel mixture is injected into the furnace in a thin wide layer. The furnace end of the fuel port is shielded from radiation by the water-cooled tubes in front of it. The fuel is kept from impinging on the pressure parts by protectors. Just as the fuel enters the furnace, it is directed upward and downward by a deflector

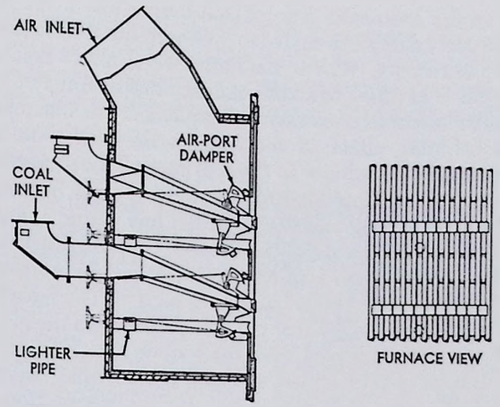


Fig. 22. *B&W cross-tube pulverized-coal burners*

block that serves the same purpose as the impeller on a circular burner. Secondary air is admitted above and below the fuel port through adjustable dampers that control the quantity of air as well as its direction. At full capacity, the velocity of secondary air for this burner is from 10,000 to 12,000 fpm. While not restricted to large-boiler applications, the cross-tube burner is widely used in the large-boiler field, where air temperatures of about 450 F are common. The range of sizes is from 50 × 10^6 to 150 × 10^6 Btu/hr.

Although it was designed initially for slag-tap units, this burner is used successfully on dry-ash-removal furnaces. On slag-tap units the proximity of the burners to the slag level and the ability to regulate the flame elevation (by adjusting the secondary-air dampers) permit tapping slag when it would be impossible to do so with other burner types. The cross-tube burner can be equipped to burn either oil or gas, or both, if desired.

Distribution of Coal and Air

In the foregoing, the ability to effect complete combustion of the fuel with a minimum of excess air has been emphasized. For efficient fuel consumption, each particle must be intimately associated with the proper amount of air, and the fuel entering the furnace must be uniformly dispersed into a well-distributed secondary-air stream. Where pulverized-coal burners are concerned, uniform distribution of the fuel stream is rather difficult since the primary-air-and-coal mixture is not homogeneous. Bends in the coal pipe between the source and the burners may deflect the coal-air stream with the result that a nonuniform mixture arrives at the burners. Similarly, the secondary air may be concentrated in one part of the duct, resulting in poor distribution in the burner windbox.

In the circular pulverized-coal burner, a distributor is installed in the coal nozzle. This distributor is adjusted for optimum performance.

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In the cross-tube type of coal burner, the shape of the coal nozzle is designed to give uniform distribution at the furnace. Where there are several severe bends in the coal pipe (to avoid some external obstruction), it is sometimes necessary to supplement the inherent distributing effect of the coal nozzle itself with a correcting distributor in the coal pipe.

In the multiple-intertube multitip type of burner, the arrangement and design of the fuel piping ahead of the burner are rigidly controlled. Uniform distribution of fuel to the burner tips is obtained by a series of elbows and changes in piping section. The burner tips are so arranged that the fuel stream from the piping is well dispersed over the burner area. The distribution of primary air and coal is somewhat better in this burner than it is in the circular burner. However, this is realized at the expense of higher pressure drop.

Good secondary-air distribution is very important for optimum combustion. A nonuniform flame characteristic can be caused by maldistribution of either the primary stream or the secondary air. Observation of the fire should show a uniform and symmetrical pattern, predominantly bright with some short dark streaks. Where there is faulty distribution, a fire will be particularly bright in one sector of the flame and quite dark in another. This effect might also be caused by a deficiency of secondary air. Even though the duct arrangement to the burner windbox may appear ideal, improper distribution of air at the burners is still possible from excessive velocities entering and within the burner windbox or deflection of the air because of obstructions in its path.

Air flowing in a duct has a strong tendency to continue in its direction and will not uniformly follow a new direction caused by some change in duct contour unless it is properly directed. Therefore, both the inside and outside corners of all bends should have generous radii, and the bends should be fitted with turning vanes to distribute the air uniformly around the turn. In addition to improving air distribution entering a burner windbox, proper turning vanes will also reduce air resistance in the duct. In general, the velocity of the secondary air entering the burner windbox should be uniformly distributed and not exceed 2500 fpm. Under these conditions secondary-air distribution is usually good. Generously proportioned ducts and windboxes assist good distribution.

Wall Impingement and Tube Wastage

It is not good practice to have pulverized-coal-burner flames blast against furnace walls — either refractory or water-cooled. Intensely hot ash particles (always molten when newly formed) have the ability to dissolve most refractories, hence heavy "traffic" of ash over a refractory wall may seriously damage the

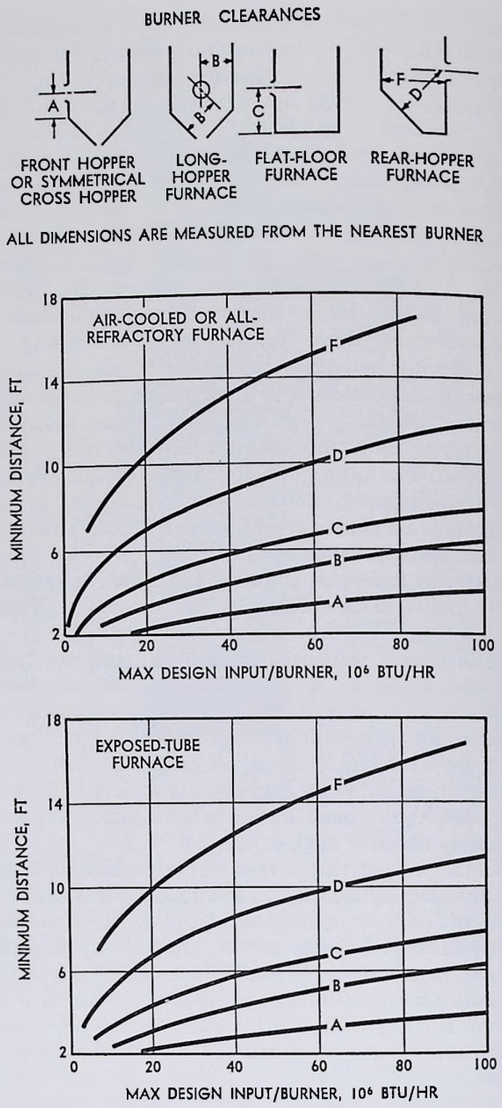


Fig. 23. *Typical burner clearances to confining furnace walls for circular type of pulverized-coal burners*

brickwork. An active flame impinging on, or in close proximity to, furnace wall tubes may also cause damage, but of a somewhat different kind. Even if the boiler water within the tubes is good and truly non-scale forming, certain conditions, such as 1) high heat release per sq ft, 2) coals high in sulfur and alkali content, and 3) localized areas deficient in oxygen, may induce a slow attack on the external surface of the furnace tubes. This is known as "tube wastage," a phenomenon which seems to occur where a reducing condition exists. Aside from the possibility of damage, the prompt chemical union of coal and air is delayed

by the proximity and attendant chilling effect of the comparatively cold surface of furnace tubes. Experience is the best guide in this matter. Conservative clearances between various circular-burner arrangements and furnace surfaces at different rates of heat input are given in Fig. 23.

There are instances where the flame tends to impinge against a particular surface or is directed in the wrong way. If burner clearances are adequate, the chances are that the deflection of the fire is caused by the secondary-air stream. The importance of proper secondary-air distribution within the burner as well as between burners is evident since more than 80% of the total weight of combustion air is secondary air.

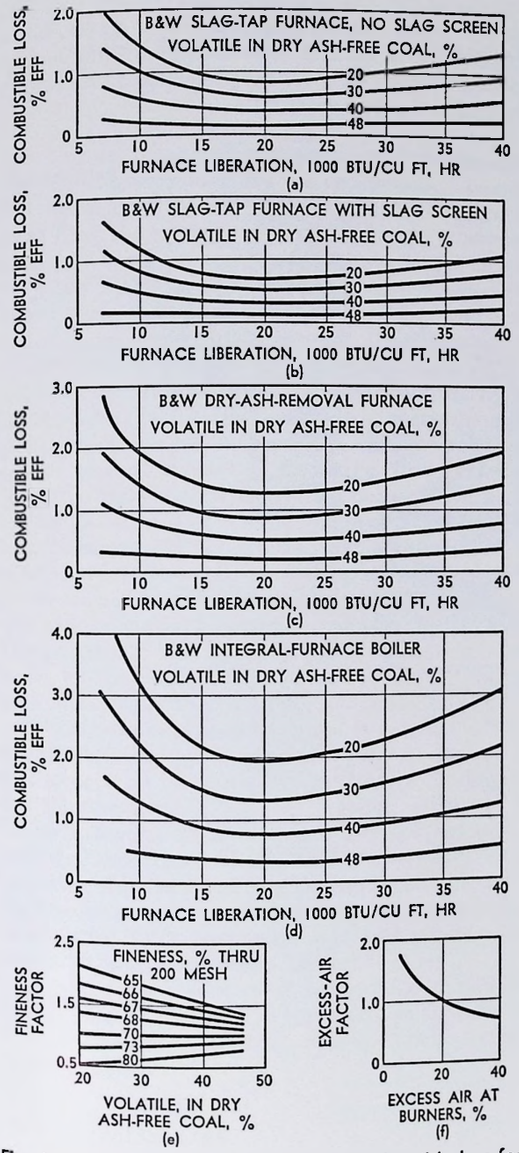


Fig. 24. *Principal factors affecting combustible loss for pulverized-coal firing in various types of furnaces*

Unburned-Carbon Loss

The primary objective of a pulverized-coal burner is to combine air and fuel ideally, without an excess of either. This result is closely approached in the present state of development. With approximately 15% excess air, a carbon loss of 0.5% can be expected under the most favorable conditions. The principal factors affecting combustible loss in pulverized-coal firing with various conditions of fineness, excess air, and types of furnaces are noted in the diagrams in Fig. 24. The base loss is multiplied by factors for percentage of fineness (diagram e) and percentage of excess air (diagram f). As an example, the percentage carbon loss for two furnace types operating under identical conditions may be established from Fig. 24 as follows:

Conditions		
Furnace liberation, Btu/cu ft, hr		20,000
Volatile (dry ash-free coal), %		40
Fineness (through 200 mesh), %		75
Excess air at burners, %		15
Solutions		
Furnace	Dry-Ash Removal	Slag-Tap with Slag Screen
Diagrams	c, e, and f	b, e, and f
Carbon loss	$0.5 \times 0.8 \times 1.2 =$ 0.48%	$0.3 \times 0.8 \times 1.2 =$ 0.29%

Pulverized Coal for Other Uses

Application of pulverized-coal firing to copper- and nickel-ore smelting and refining has been standard practice for many years. In using pulverized coal, high-purity metal can be obtained since the furnace atmosphere and temperature can be easily controlled. In this industry, where the price differential between coal and natural gas or oil is in favor of coal, the other advantages of pulverized coal are the high rate of smelting and refining and the ability readily to oxidize the sulfur.

Copper reverberatory furnaces for both smelting and refining are fitted with waste-heat boilers for the generation of steam. These boilers supply a substantial portion of power for auxiliary equipment and also provide the means for cooling the gases leaving the furnace. A typical copper-reverberatory-furnace and waste-heat-boiler arrangement is shown by Fig. 25.

An interesting application of pulverized coal is in the nonferrous industry as a reducing agent in the recovery of zinc from lead blast-furnace and copper reverberatory-furnace slags. Lead-bearing ores contain an appreciable amount of zinc and other metals

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that remain in the slag after smelting. The slag-fuming process by which these metals are recovered is not new. However, by applying the newly developed methods of preparing, distributing, and firing pulverized coal, the fuming can now be done directly, automatically, and more efficiently.

Approximately 65 to 70% of all the malleable-iron tonnage produced in this country is made from white cast iron that is melted in a low-head air furnace of the reverberatory type, burning pulverized coal in a direct-firing system with forced-draft burners.

In producing malleable iron castings, the hard white iron is converted to a tough, ductile, machineable iron by annealing, which transforms the combined carbon (iron carbide) to free carbon, known as temper-carbon. Pulverized coal is especially suitable for this purpose because of better heat distribution and greater economy than with oil or gas.

In the production of cement, fuel cost is one of the major items of expense. The first consideration, therefore, in deciding on the type of firing in the manufacture of cement is the price of fuel. In general, except in locations where the cost differential is in favor of fuel oil or gas, pulverized coal is universally used in this industry.

Direct firing, with a single pulverizer delivering coal to a single burner, is common practice in the majority of the cement plants in this country (see Fig. 7). The waste-heat air from the clinker cooler, taken from the top of the kiln hood through a suitable dust collector, usually serves as the preheated air for drying the coal in the pulverizer.

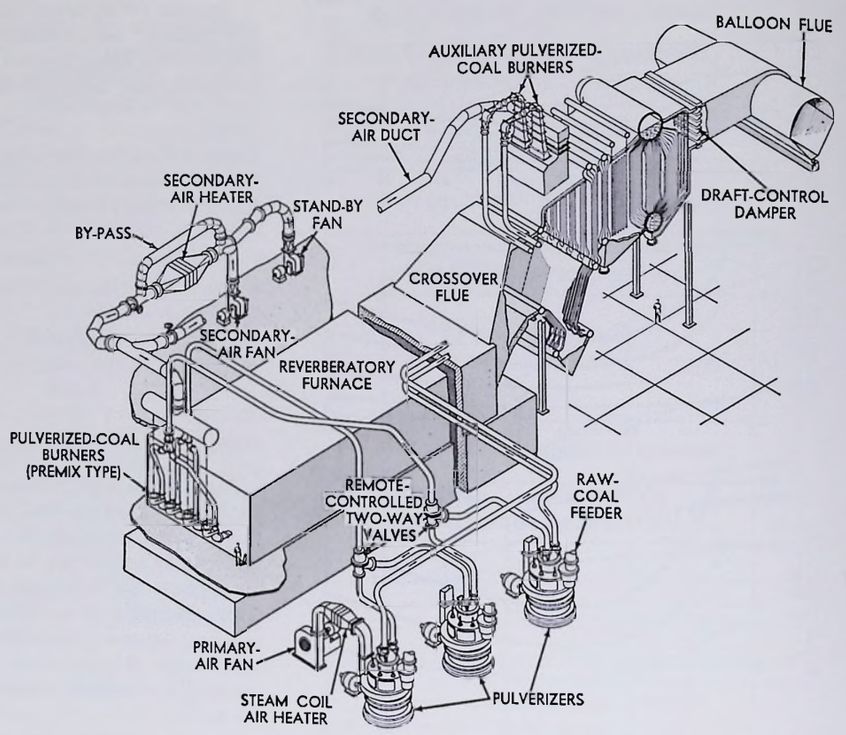


Fig. 25. Direct-fired copper-reverberatory-furnace and waste-heat-boiler arrangement

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FUEL-ASH EFFECTS ON BOILER DESIGN AND OPERATION

From the beginning of the application of combustion of fossil fuels for the production of power, much attention has been devoted to the problems created by the residues of such combustion, broadly known as "ash." The complexity of these problems has increased with the size and rating of modern units. When burned, all but a few fuels have solid residues, and in some instances the quantity is considerable.

For continuous operation, removal of ash is essential to all methods of firing. With the fuel bed, this is accomplished by intermittent shaking or dumping of grate sections, or by continuous movement of the ash residue toward a point of discharge. In suspension firing, the ash particles are carried out of the furnace by the gas stream or retained in part by settling or by adherence to boiler surfaces. Retained material of solid form is removed by periodic cleaning. If temperatures are sufficiently high, the retained ash is molten and may be drained continuously from the furnace. The portion of the ash that is carried through the boiler by the flue gas may cause fouling of furnace walls and gas passes and, under some conditions, may lead to corrosion of these surfaces. It is therefore important that the nature and the amount of ash in the fuel to be used be of major concern to the designer and the operator.

The following discussion is concerned primarily with the importance of coal ash in the design and operation of boilers, but consideration is also given to the importance of ash residue from petroleum fuels. Commercial fuels containing ash, and others with little or no ash, are listed in Table 1.

At best, some means must be provided to handle and dispose of the ash; at worst, ash in its various forms may seriously interfere with operation or even cause shutdown.

Table 1

Commercial Fuels for Power Production

Fuels Containing Ash	Fuels Containing Little or No Ash
All coals	Natural gas
Fuel oil—"Bunker C"	Manufactured gas
Refinery sludge	Coke-oven gas (clean)
Tank residues	Refinery gas
Refinery coke	Distillates (most)
Most tars	
Wood and wood products	
Other vegetational products	
Waste-heat gases (most)	
Blast-furnace gas	
Cement-kiln gases	

Ash Content of Coal

The ash contents of coals vary over a wide range. This variation occurs not only in coal from different parts of the world or from different seams in the same region but also in coal from different parts of the same mine. Some rock and earthy materials find their way into the mined product. Before marketing, most commercial coals are cleaned or washed to remove a portion of what would be reported as ash in laboratory determinations (see Chapter 15). In any case, the ash determinations of significance to the user are those made at the point of use, and the values noted below are on that basis.

The bulk of bituminous coal used for power generation in the U.S. has an ash content within the range of 6 to 20%. Low values of 3 or 4% are encountered infrequently, and such coals find other commercial uses, particularly in the metallurgical field. On the other hand, some coals may have an ash content as high as 30%. Many high-ash fuels are successfully burned in the Cyclone Furnace as well as in pulverized-coal-fired boilers. Their use is increasing in

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localities where the fuel costs indicate a favorable over-all economy.

Nature of Coal Ash

The presence of ash is accounted for by minerals associated with initial vegetal growth or those which entered the coal seam from external sources during or after the period of coal formation. Appreciable quantities of inorganic material may be contributed to the commercial fuel by partial inclusion of adjacent rock strata in the process of mining.

Although the character, quantity, and distribution of minerals may vary considerably with the coal seam, much of this diversity is masked in the as-fired fuel as a result of cutting, transporting, size reduction, washing, classifying, storing, and pulverizing, which operations tend to bring the mineral-matter content into a less heterogeneous condition. Large differences may exist, however, in the ash characteristics of coals from different mines or regions. To some extent, they can be identified with the coal and with methods of mining and preparation, thus providing a rough basis for selection of fuel or choice of burning equipment.

Since quantitative evaluation of mineral forms is extremely difficult, the composition of the coal ash is customarily determined by chemical analysis of the residue produced by burning a sample of coal at a slow rate and at moderate temperature (1350 F) under oxidizing conditions in a laboratory furnace. It is thus found to be composed chiefly of compounds of silicon, aluminum, iron, and calcium, with smaller amounts of magnesium, titanium, sodium, and potassium. The analyses of coal ash in Table 2 indicate what may be expected of some coals from the eastern United States.

The element sulfur is present in practically all coal, and its effect on equipment performance has been given much attention. Sulfur itself burns as a fuel with a relatively low heating value (4050 Btu/lb when burned to SO_2), but its reputation, which is nearly all bad, results from the effect of its chemical combination with other elements. Under certain conditions some of these compounds corrode boiler components; others contribute to the fouling of gas passages and heating surface.

Some of the sulfur in coal is in combination with iron as FeS_2. Sulfur may also be present in the form of complex organic compounds and, in minor amounts, in combination with the alkali metals (sodium and potassium) and alkaline earths (calcium and magnesium). When the fuel is burned, the sulfur compounds are normally converted to more or less stable mineral oxides and sulfur dioxide gas, SO_2. A very small part of the SO_2 thus formed is further oxidized to SO_3. These sulfur gases are carried along with the other combustion gases, and their presence, under certain conditions, can contribute to corrosion of boiler heating surfaces and to air-pollution problems.

Ash-Fusion Temperatures

The accepted procedure for the determination of ash-fusion temperature is outlined in ASTM Standard D 271 and is described briefly in Chapter 2. The fusion temperatures of ash, noted in Table 2, are of special

TABLE 2

Coal-Ash Analyses and Ash-Fusion Temperatures

Coal: Rank	Semibituminous	Bituminous	Bituminous	Bituminous
Seam	Pocahontas #3	Freeport	Pittsburgh #8	#5
Mine	Kilpoa, Va.	Longans Ferry, Pa.	Richland, O.	Springfield, Ill.
Ash in coal, %	5.3	12.15	11.3	11.7
Sulfur in coal, %	0.6	1.15	3.5	4.1
Analysis of ash, % by wt				
SiO_2	53.1	50.8	44.8	39.8
Al_2O_3	33.8	30.6	21.4	15.1
Fe_2O_3	9.1	11.0	28.3	32.9
CaO	1.0	5.0	3.3	8.4
MgO	0.8	0.6	0.7	0.8
$Na_2O + K_2O$	2.2	2.0	1.5	3.0
Ash-fusion temperatures,°				
ASTM standard cone				
Initial-deformation temp, F	2725	2400	1880	1880
Softening temp, F	2795	2490	2080	1935
Fluid temp, F	2850	2680	2325	2190

°Reducing atmosphere.

interest to the designer, the operating engineer, and the coal-purchasing authority. Generalities usually are inappropriate when applied to coal or to any of its parts or performance, and ash is no exception. However, just as it is customary to consider ash a nuisance and the "more ash the more nuisance," so it has become customary to consider a low-fusion ash less desirable than a high-fusion ash. That the latter is not always so will be evident later on, but the practical effect of the property known as "ash-fusion temperature" should be recognized and understood.

Ash melts when heated to a sufficiently high temperature. Following combustion, individual ash particles are generally in the form of tiny spheres (cenospheres) that appear hollow when viewed under a microscope, as illustrated in Fig. 1. The form of the ash particles indicates that, during combustion of the coal, the particles were actually liquid and the spheres were formed as tiny bubbles by evolved gases trying to escape. What happens to these particles depends upon their physical and chemical characteristics and upon furnace conditions. If cooled promptly and sufficiently, the result is a dusty ash that may travel through the equipment, lodge on heating surfaces, drop out in soot hoppers and along flues, or collect at the base of the stack. Those particles that remain in suspension are carried out with the exit gases and contribute to the visible discharge from the stack.

The individual ash particles do not, however, always cool quickly to a solid state. If insufficiently cooled, they remain molten or sticky and tend to coalesce into large masses on the boiler or other heat-absorption surfaces. This problem is dealt with by adequate design of burners and furnace arrangement for the fuels to be burned and by proper attention to boiler operation.

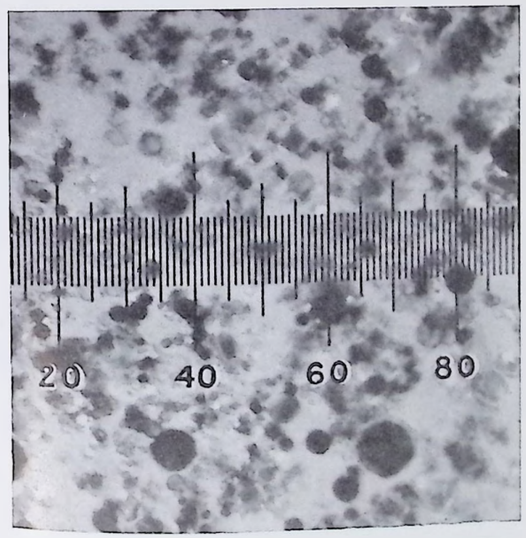


Fig. 1. *Ash cenospheres (485x) formed in burning pulverized coal*

Determination of "ash-fusion temperatures" is strictly a laboratory procedure, developed in standardized form, which experience shows can be duplicated with a fair degree of accuracy. The accepted procedure is outlined in the *Standards on Coal and Coke* issued by the ASTM, D 271 (see Appendix, page 18-A1 ff).

A simplified account of the relation of ash-fusion temperatures to actual operating phenomena follows:

Initial-Deformation Temperature. In the laboratory, the initial-deformation temperature is the temperature at which a prepared standard cone of ash just begins to fuse or show evidence of deformation at the top of the cone as it is being heated in a reducing atmosphere in a standard test furnace. This would correspond to the temperature in an operating furnace at which the particles of coal ash, in transit through the setting, have been cooled to a point where they still retain a slight tendency to stick together or slowly to build up on heat-absorption surfaces. When the temperatures in an operating furnace are such that the ash particles have cooled to a temperature lower than their initial-deformation temperature, they tend to accumulate as a "dry" product.

Softening Temperature. The softening temperature (commonly called the fusion temperature of an ash) corresponds to an observed condition between initial deformation and fluidity. To the laboratory technician, the softening temperature is the temperature at which the test cone under observation has deformed still further to a point where it assumes a spherical form on the plaque. To the operating engineer, this temperature is related to that at which the fuel ash shows a greatly accelerated tendency to mass together and stick in large quantities to heat-absorbing surfaces.

Fluid Temperature. In the laboratory determination, the fluid temperature is the temperature at which the test cone flattens out to a pancake shape on the plaque. To the operating engineer, the fluid temperature is related to the temperature at which the coal ash may be expected to flow in streams or drip from heat-absorption surfaces or the temperature at which heavy clinker forms on the grates under a fuel bed.

The full significance of ash-fusion temperatures is not quite as simple as the above might indicate, although it is general practice in design to specify or limit the values of gas temperatures from foreknowledge of ash-fusion temperatures.

Even dry or dusty ash, deposited at points of repose in a gas stream, slowly sinter at temperatures con-

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siderably lower than the initial-deformation temperature to form a consolidated mass, the strength of which depends upon the particular ash and the temperature and duration of sintering (see *Ash Fouling*, page 18-11). The difficulty of removing the sintered mass by soot blowers or other means is, in turn, related to its strength.

The fluid temperature of a given coal ash is not a satisfactory criterion for slag-tapping in a wet-ash furnace. Another characteristic, the viscosity of the liquid ash, which varies for different coals (see *Viscosity of Coal-Ash Slag*, page 18-7), has proven to be a reliable guide to good performance in slag-tapping.

The ash-fusion temperatures given in Table 2 are not particularly noteworthy, but they do show values of the three temperatures described above for coals selected from rather widely separated parts of the country. The Virginia coal ash happens to have relatively high values for all three temperatures, whereas the Illinois coal ash has relatively low values. However, not all Virginia coals have high-fusion ash, although the tendency is toward higher values than for the general run of coals from the Illinois fields.

Some of the most troublesome ash for the designer and the operator is that which has a wide spread between the upper and lower values. Reduced to simple terms, such an ash remains "sticky" for a longer period in the gas stream. The Virginia coal ash listed in the table has a comparatively small over-all spread of 125 F between fluid and initial deformation, whereas the particular Ohio coal ash listed has a spread of 445 F between the high and low points. While the latter spread is fairly wide, spreads of nearly 600 F are sometimes noted in coal ash from some particular region or mine. Such a coal may require special consideration because of its adverse effect on furnace-wall cleanliness.

Effect of Iron on Ash Behavior

Iron is one of the constituents in almost all coal ash and tends to form at least two stable oxides, ferric, Fe_2O_3, and ferrous, FeO. Other elements in the ash can also exist as compounds in more than one oxide form and may have their effect on ash properties. However, the relatively large proportion of iron present makes it particularly significant, and experience indicates that iron has a dominating influence on the behavior of the ash in the furnace, as indicated by its effect on the ash-softening temperature.

The specific effect of the presence of iron in coal ash is interesting and can be simply stated as follows: In completely oxidized form (Fe_2O_3) iron tends to raise all three values of ash-fusion temperatures, while in the lesser oxidized form (FeO) it tends to lower all three of these values. The effect of iron, in each of the two forms, on initial-deformation and softening temperatures is indicated in Fig. 2, plotted for a large number of ash samples from coals in widely scattered regions of the U.S.

Under actual operating conditions of a boiler furnace, complete combustion of fuel requires an excess of total air and therefore excess oxygen. However, not all parts of the furnace have excess combustion

Fig. 2. *Influence of iron on coal ash-fusion temperatures*

air in equal amounts at all times, and therefore the iron oxide finally produced is a mixture of the ferric and the ferrous forms, with a tendency toward more of the former. Ash-fusion-temperature determinations made solely on a reducing basis may therefore be less than adequate as a guide in predicting the performance of a given coal in an operating boiler unit. Since it is important to know the effect of the state of oxidation of iron on the behavior of a specific ash, ash-fusion temperatures on both reducing and oxidizing bases are made regularly in the B&W laboratory.

Furnace Design

A properly designed furnace has two functions, 1) to burn the fuel completely and 2) to cool the products of combustion sufficiently so that the convection passes of the boiler unit may be maintained at a satisfactory degree of cleanliness with a reasonable amount of soot blowing. Experience and tests indicate that, when the average gas temperature leaving a coal-fired furnace exceeds the oxidizing initial-deformation temperature of the ash, the need for cleaning of the ash from the convection surfaces becomes excessive. This is, of course, subject to some latitude, because what constitutes excessive cleaning depends largely upon the opinions of the individual operating crews.

In establishing the proper functional design of a furnace, two methods of cooling the products of combustion must be considered, namely, radiation and gas tempering.

Cooling Gas by Radiation

For many years the method used to cool the products of combustion has been that of providing radiant furnace-cooling surface. Sufficient surface is installed to insure a reduction in gas temperature to a level where convection passes can be kept reasonably clean. As indicated previously, there is a general relationship between the average desired furnace exit-gas temperature and the softening temperature of the coal ash. There can be a considerable variation in the gas temperature at the furnace outlet, as shown in Fig. 13, page 7-18, and the maximum gas temperature could be considerably higher than the average. A better design criterion of furnace size when using this method of cooling gases is "release rate," which is defined as heat available per hour per square foot of effective-cold furnace-cooling surface (see Chapter 7, page 7-20). Heat available, in turn, is defined as the amount of energy released in the furnace and available for increasing the temperature of the products of combustion. Mathematically, it is equal to (Heat input in fuel + heat in air) − (Heat required to evaporate moisture in fuel + ½ radiation loss).

The relationship between furnace release rates and furnace exit-gas temperatures has been developed from field test data. Factors affecting this relationship include type of furnace design, method of firing, fuel, burner location, and type of furnace-wall construction.

As a consequence of years of experience, desirable release rates have been established for different types of coals. Along the eastern seaboard, where coals have an initial-deformation temperature of about 2400 F, the furnaces for Cyclone-fired, pulverized-coal-fired, and stoker-fired units may be proportioned for heat-release rates of 100,000 Btu/hr, sq ft of flat-projected area up to the convection surface, at maximum continuous capacity. For similar units located in West Virginia, western Pennsylvania, Ohio, eastern Kentucky, and the Carolinas, furnaces may be proportioned for release rates of 90,000 Btu/hr, sq ft. Units located in the Midwest which burn coals (except those from Fulton and Christian Counties, Illinois) that have deformation temperatures as low as 2200 F on an oxidizing basis may be designed for furnace heat-release rates of 80,000 Btu/hr, sq ft at maximum continuous capacity. Because of the high alkali content of the Fulton and Christian County coals, heat-release rates are limited to 43,000 Btu/hr, sq ft. A boiler designed to burn these coals is shown in Fig. 3.

Furnaces fired with oil are usually designed for furnace heat-release rates above 135,000 Btu/hr, sq

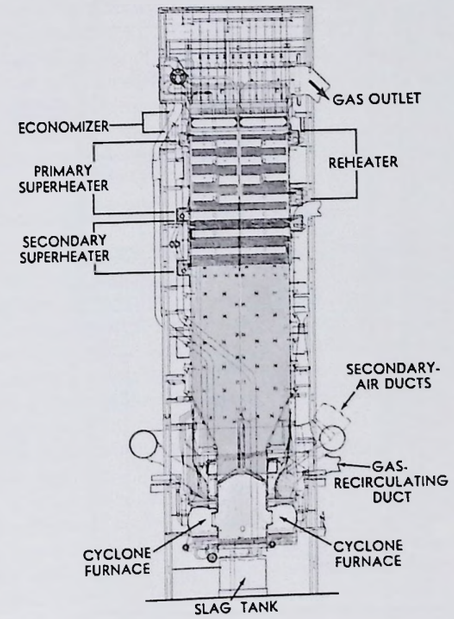


Fig. 3. *Cyclone Furnace-fired unit in Midwest burning high-alkali-ash coal*

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ft. Since there are few high-temperature ash-pluggage problems with this fuel, the release rate is limited only by the consideration of economic factors and high-temperature corrosion.

Gas Tempering

The second method of cooling hot furnace gas is that known as "gas tempering," in which relatively cool gas from the economizer outlet is mixed with the hot furnace gas near the furnace outlet (Fig. 4). It provides the operator with a control of furnace exit-gas temperature with little or no effect on superheater and reheater absorption. Thus, if a low-fusion slagging coal is fired, the amount of tempering gas is increased to lower the gas temperature leaving the furnace to a level at which the convection surface is free from slagging. On the other hand, if a high-fusion nonslagging coal is fired, the amount of tempering gas is decreased, thereby reducing fan power. With some coals, tempering may be reduced to zero if experience so warrants.

Gas tempering thus provides a positive means of controlling gas temperature at the furnace outlet regardless of furnace size. With this method of design, furnaces may be held to a reasonable size, consistent with burner clearances and flame travel. Over-all building requirements, steam piping, structural steel, and platforms are all decreased with no sacrifice in quality or performance of equipment. Rather, performance has been upgraded, since the gas temperature leaving the furnace is subject to positive control and is not dependent upon furnace cleanliness. When gases are cooled only by radiation, the furnace cooling surfaces may be covered with slag from low-fusion coal and will not do the required cooling.

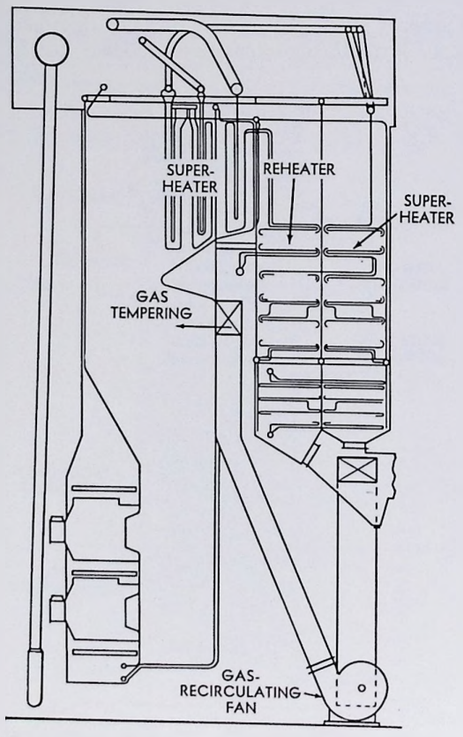


Fig. 4. *Gas tempering provides positive control of furnace exit-gas temperature*

With gas tempering, the combustion gases are cooled uniformly across the width of the furnace, minimizing the large temperature variation customary with older designs. The possibility of localized slagging and excessive superheater metal temperatures is therefore reduced.

With high steam temperatures and pressures it was formerly necessary to resort either to steam-cooling the walls or locating steam-cooled platen surface in high-gas-temperature zones. Gas tempering eliminates these requirements and thus improves the reliability and availability of units.

Furnace exit-gas temperature rather than release rate is the design criterion for furnaces with gas tempering. For example, the Universal-Pressure boiler, Fig. 5, has operated very successfully for a number of years with a release rate of 180,000 Btu/hr, sq ft, a furnace exit-gas temperature of 1900 F, and 80% gas tempering at full load.

Gas tempering permits engineers to design high-capacity units that occupy less space and have greater reliability, lower maintenance, safer operation, and reduced over-all plant costs.

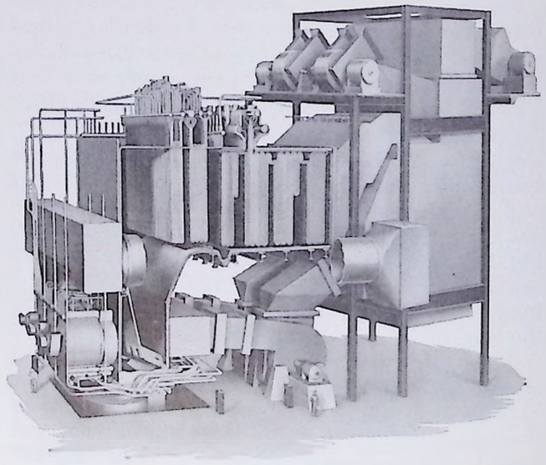


Fig. 5. *Gas tempering enables high furnace release rates in this Universal-Pressure boiler*

Ash from Pulverized-Coal Burning

No matter how fine the pulverization, the fuel fed to the furnace still has all its original ash. However, the final ash product ejected from a pulverized-coal-fired unit differs in appearance from the refuse of a stoker-fired unit. This is because, with pulverized coal, nearly all of the ash particles are formed in suspension and tend to remain in suspension in a dry-ash furnace, with the individual particles well dispersed and very much smaller than those from stoker firing. In the slag-tap furnace, a portion of the ash particles coalesces on walls or other suitably designed surfaces and drains to the furnace bottom.

Where pulverized coal is burned in a dry-ash furnace (as in Fig. 10), it is not unusual for the ash escaping toward the stack to be 80% of the ash originally in the coal. On the other hand, with pulverized coal burned in a slag-tap furnace, as much as 50% of the ash may be retained, part in the furnace as fluid slag and part in the gas-pass hoppers in the form of dry dust. The other 50% of total ash in the coal leaves the unit, also in the form of dust. Collectors are frequently placed ahead of the stack to prevent the ejection of large quantities of this ash to the atmosphere.

Although these figures show the effectiveness of the slag-tap arrangement in reducing atmospheric pollution by dust, even more impressive is the performance of the Cyclone Furnace. This special slag-tap furnace, described in Chapter 28, permits less than 20% of the total ash in the coal to escape from the boiler unit, and the stack discharge is correspondingly lower. If dust-collecting equipment is installed for a Cyclone Furnace boiler unit, the ash escaping from the stack to the atmosphere may be diminished to 1 or 2% of the total.

Ash Produced by Stokers

In the fuel bed of a stoker, whether of the underfeed, chain-grate, or spreader overfeed type, ash particles tend to become fused together. In a properly operated stoker burning a suitable coal, the passage of air and the agitation of the fuel bed serve to keep ash accumulations more or less porous, and the ash is discharged to an ashpit in fairly large pieces varying from popcorn size upward.

Not all the ash is retained in a fuel bed. With the types of stokers mentioned, and particularly the spreader stoker, some of the fuel is burned in suspension. A considerable quantity of ash particles, containing some unburned and some still burning fuel, is consequently "carried over" with the gases. With the spreader stoker this material is usually collected in hoppers provided for the purpose and is reinjected into the furnace for further burning of any combustible that it contains (see Chapter 16). Reinjection is also used occasionally with other types of stokers.

Use of Dry-Ash or Slag-Tap Units

Considered answers to the following pertinent questions relating to fuel are essential in initiating the design of a successful coal-fired unit:

1. Has the particular coal selected been burned extensively, so that its combustion and ash characteristics are well known?
2. Will the source of coal and the characteristics of the coal remain substantially constant during most of the life of the equipment? What is likely to be the "worst" fuel supplied?
3. What auxiliary fuel may be made available during periods of temporary suspension of coal delivery?
4. What type of equipment is best suited to the experience of the operating personnel?
5. In the public interest, for better public relations, or to comply with civil regulations, what is required to limit atmospheric pollution from dust?
6. What method of final refuse disposal is available?

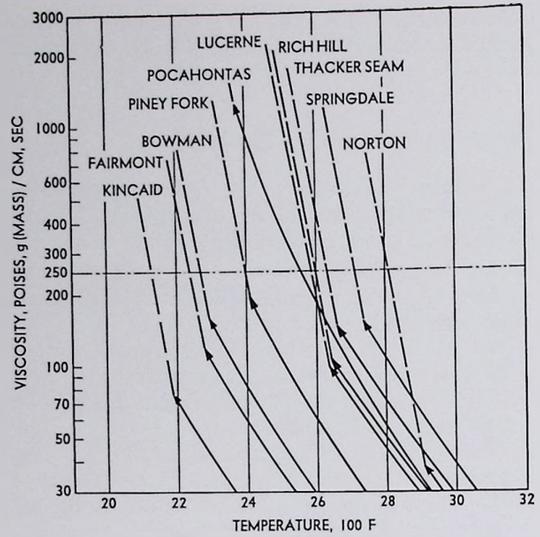
Answers to the above questions make it possible to select proper equipment for the specific method of utilizing the coal. For instance, coal should not be burned in a slag-tap furnace if the ash-fusion temperature and viscosity are higher than experience indicates they should be for successful tapping. Slag-tap furnaces are rarely suitable for ash-fluid temperatures in excess of 2600 F (ASTM reducing basis) or for an ash viscosity greater than 250 poises at 2600 F. If the ash-fluid temperature or the ash viscosity of the coal selected is too high or marginal, equipment permitting the ash to be removed in dry form is indicated.

Viscosity of Coal-Ash Slag

The fluid temperature of coal ash is considered by some to be a fairly reliable guide to the ease of slag-tapping. However, experience with various slag-tap pulverized-coal furnaces and Cyclone Furnace-fired units indicates that this is not always true and that the viscosity rather than the fluid temperature of the coal ash is the significant criterion. Even though the gas temperature in the slag-tap zone exceeds the coal-ash fluid temperature, the slag may not flow freely if the viscosity is too high.

The coal-ash viscosities in Fig. 6, calculated from the chemical analyses of the ash by the method developed by the Bureau of Mines, correlate well with the observed tapping characteristics of the coal tested. Difficulty in tapping the slag has been experienced when the calculated coal-ash viscosity at 2600 F exceeds 250 poises. Slags from coals of the Norton, Springdale, and Thacker seams, with calculated coal-

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NOTE: FROM COAL-ASH ANALYSES AND DATA OF BUREAU OF MINES. BELOW 250 POISES SLAG TAPS FREELY.

Fig. 6. *Relation between calculated viscosities and temperatures for various coal slags*

ash viscosities above the 250-poise limit at 2600 F, tend to tap rather sluggishly. On the other hand, Kincaid, Fairmont, Bowman, Piney Fork, Pocahontas, and Lucerne coal slags, where the viscosities at 2600 F are below this limit, tap easily. It is not yet possible to accept with complete confidence the calculated viscosities for coals of ash composition outside the range of those covered in the work of the Bureau of Mines.

Viscosity of coal-ash slag is actually measured in a high-temperature viscometer (Fig. 7). The control panel is at the left, and the furnace with its measuring element is at the right. A section through the furnace is shown in Fig. 8. The molten slag sample is placed in a platinum crucible. There is an optical electronic device for measuring the torsional deflection of the calibrated wire, and the torque is recorded on a chart. Provision is made for controlling the atmosphere within the furnace.

Ash is introduced into the crucible at an elevated temperature (2600-2800 F) and held at that temperature until it becomes uniformly fluid and all decomposition gases have been expelled. The temperature is then decreased in predetermined steps, and the viscosity is measured at each temperature.

The viscosities of a Central Illinois and an Allegheny County, Pa., coal ash as measured by the high-temperature viscometer are given in Fig. 9. The viscosity of the latter is not only higher than that of the Central Illinois slag but it increases rapidly in the midrange. This may be an undesirable characteristic

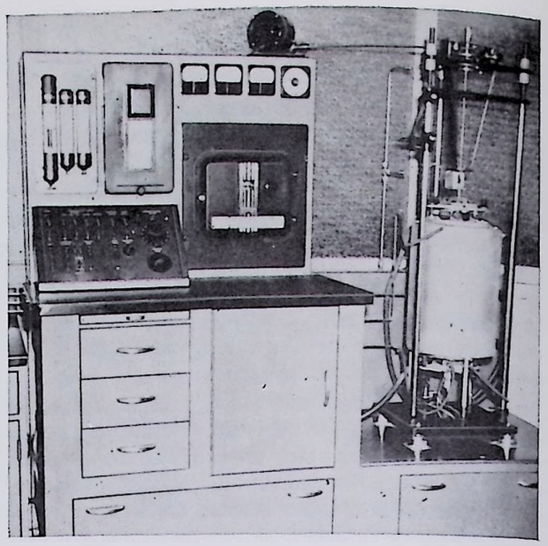


Fig. 7. *High-temperature viscometer for determining the viscosity of molten coal ash*

in the operation of a slag-tap furnace. The Pennsylvania coal ash taken as an example is definitely not suitable for slag-tap furnaces.

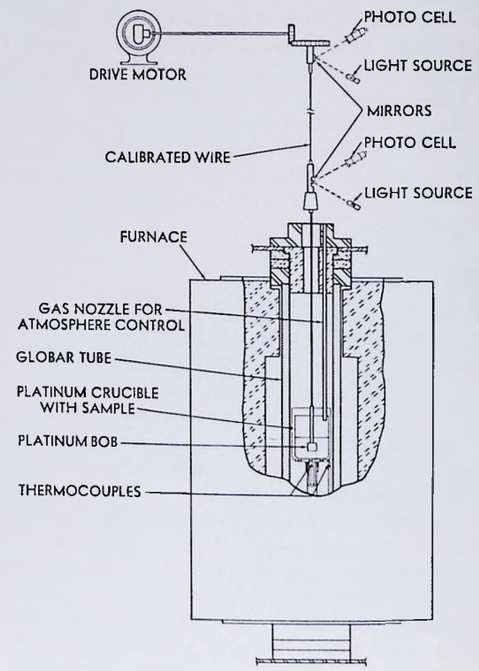


Fig. 8. *Section through furnace of high-temperature viscometer*

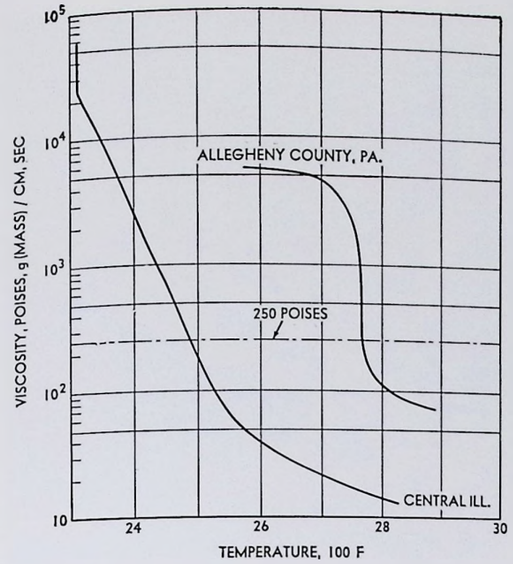


Fig. 9. *Viscosity-temperature relationship for two coal slags of widely different tapping characteristics*

By the use of synthetic ash, as well as typical coal ash, to explore the range of ash composition not covered by the work of the Bureau of Mines, the tapping characteristics of any coal ash may ultimately be predicted with fair accuracy from its chemical composition.

Ash Erosion

Ash erosion must not be ignored even though coal-ash particles may be exceedingly fine. Where ash particles are concentrated in a local region, erosion is a potential problem. The induced-draft fan is a well-known example, and erosion is an important factor to be considered in selecting the type of fan and its performance characteristics. Blade wear in high-speed fans is more pronounced at 1800 rpm than at the lower and more usual speeds of 1500 and 900 rpm and is even less for variable-speed fans. Local high concentrations of ash through the boiler setting may cause erosion of highly critical pressure parts. This action is liable to be found where ash tends to concentrate in streams, as at turns formed by baffles within boiler banks. In the latter case the remedy is to limit gas velocities and to eliminate baffles if possible. The elimination of baffles has been the trend in the development of coal-fired boiler units, as illustrated in Fig. 10, where the gas flow continues horizontally through the tube banks without turns.

Maximum allowable gas velocities with pulverized-coal, blast-furnace-gas, and Cyclone Furnace firing are given in Table 3.

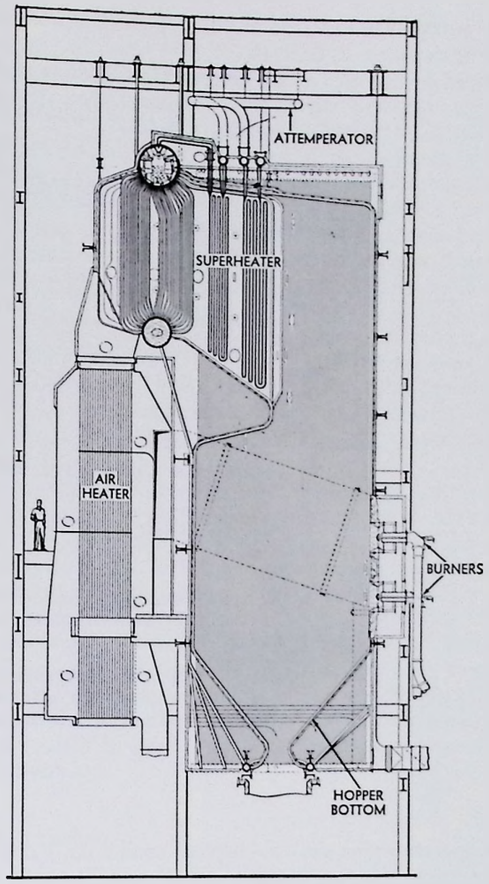


Fig. 10. *Boiler with dry-ash furnace designed for horizontal gas flow through tube banks without baffles*

Erosion may also be a problem in slag- and ash-handling equipment. If the unit is of the slag-tap variety, the molten refuse or slag tapped from the furnace must be cooled to be conveniently handled. This is done with water, and the quenched product is usually small sharp-edged particles, like coarse sand, and, regardless of the mode of transport, tends to wear the material of the conveying system. When water is used for transport, the pump, pipe lining, and turns in pipes or sluices are especially subject to localized wear and are usually protected by replaceable linings of special alloy steel or chilled cast iron.

Design and Operation of Slag-Tap Furnaces

The method of disposal of coal ash by tapping from a boiler furnace was evolved in 1926, more by accident than by design, in a boiler furnace originally expected

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TABLE 3

Design Gas Velocity, fps, through Net Free-Flow Area in Tube Banks to Prevent Flue-Dust Erosion

Type of Firing and Fuel	Baffle Arrangement	
	Multipass	Single Pass*
Pulverized coal	75	100†
Spreader stoker	50	60
Chain-grate stoker, anthracite	60	75
Chain-grate stoker, coke breeze	60	75
Chain-grate stoker, bituminous	100	100
Underfeed stoker	75	100
Blast-furnace gas	75	100
Cyclone Furnace	No limits	No limits
Wood or other waste fuels containing:		
Sand	50	60
Cement dust	—	45
Bagasse	60	75

* Allowable velocity with single-pass baffle arrangements applies to Single-Pass, Radiant, and Open-Pass boilers, continuous-tube economizers and superheaters, and Gas-Tube boilers with tube entrances formed by belled ferrules.

† Use a maximum velocity through the net free area of 75 fps for units that burn fuels with volatile matter less than 20% (dry basis), or ash content 20% and over (dry basis), or ash-softening temperature above 2600 F (oxidizing basis).

to operate under dry-ash-disposal conditions. During operation, the low-fusion ash melted, collected in a pool at the bottom of the furnace, and was drained at intervals by means of an improvised arrangement.

Many features of this furnace, the forerunner of later slag-tap furnaces, were totally inadequate for the formation, retention, and disposal of fluid ash. The furnace bottom was of ordinary brick that quickly disintegrated under the chemical action of the molten slag. It was also apparent that the proper handling of a stream of molten slag was a difficult undertaking. Nevertheless, this early experience opened the way to the development of a method for easier handling and disposal of refuse in furnace operations. Some of the important requirements for adequate slag-tap furnace design that evolved from this development are:

1. The temperature of the slag in the furnace must be kept above its fluid temperature. It follows that the furnace temperature must be even higher. Therefore a slag-tap furnace should be built to withstand the maximum temperature reached during combustion, which is usually in excess of 3000 F.
2. Since fluid slag is heavy as well as extremely hot, it must be securely contained in those regions of the furnace where it tends to collect.
3. The interior surface of the furnace must be chemically inactive to the constituents of the hot slag.
4. Means must be provided to drain slag from the furnace as fast as it is formed, or at least at frequent intervals.
5. Once the molten slag has left the furnace, it must be cooled to a temperature that renders it suitable for ultimate disposal.

To withstand the high temperatures noted above, all sides and the floor of the furnace are water cooled. More or less unsuccessful attempts were made at different times to employ other arrangements, including solid refractories in brick and molded form, air-cooled refractories, and water-cooled tubes on various centers embedded in a variety of refractories.

A common arrangement for withdrawing molten slag from the furnace is illustrated in Fig. 3. An early type of slag removal is illustrated in Fig. 11. These two illustrations indicate the two methods of slag removal, and they are conventionally termed "continuous tap" and "intermittent tap," respectively. In both cases the molten slag is disintegrated in water to the consistency of sand, and the final product is conveyed either to storage sites or to ash cars.

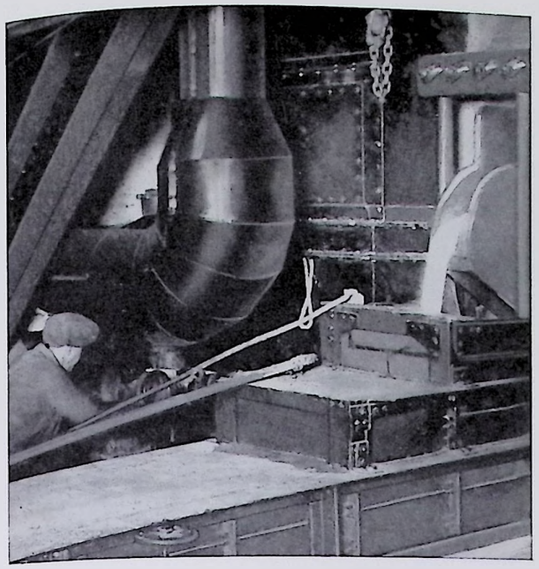


Fig. 11. *Typical flow during intermittent removal of molten slag*

The difficulty in tapping ash of high fluid temperature is evident during low-load operation. Under these conditions, even a coal with a medium ash-fluid temperature may not be suitable for slag-tapping, since the furnace temperature may not be sufficiently high to furnish the temperature differential required to attain the degree of fluidity necessary for tapping.

In variable-load operation, with periods of high and low furnace temperature, it is sometimes possible to use intermittent tapping successfully. Here the slag is permitted to accumulate in the furnace bottom for a period of time which in practice may be up to 20 hr. From the daily-load curves it is possible to anticipate when the steaming load will be high enough to attain the proper degree of slag fluidity to assure its removal through the tap hole. In other words, for intermittent tapping the operator synchronizes the time of tapping with a well-established high-load period in a particular schedule of operation.

In modern power plant designs, equipment selection is based on single-boiler single-turbine arrangements. For optimum investment returns, the heavy-duty and expensive equipment must be operated continuously at its maximum output. These conditions require continuous tapping in slag-tap designs.

One of the most important attributes of the slag-tap furnace is the coating of sticky ash that covers a portion of the furnace walls near the bottom. The sticky surface of molten ash, deliberately maintained in selected high-temperature zones, serves to entrain other transient particles. The ash so collected drains continuously toward the furnace bottom and is removed in due course through the tap hole. The consequent reduction in the quantity of dust and ash leaving the boiler unit has a definite practical value, since it decreases the amount of dust to be intercepted by collectors and therefore decreases the size and cost of the dust-collecting equipment. With the Cyclone Furnace-fired boiler unit it is possible to go even farther in the direction of reducing ash discharge to the atmosphere (see above and Chapter 28).

It may be possible to reduce building costs in housing units with slag-tap furnaces, since they are usually built without hoppers, thus requiring less building height. In planning new buildings, this may be an important factor in cost saving. Even in outdoor installations the height of the supporting steel structure can be reduced. For the same reasons, slag-tap units may be installed in older plants where it would be difficult, because of the greater height required, to install a unit equipped with ash hoppers.

Attempts have been made to utilize slag from slag-tap furnaces. It has been proposed that it be used as fill for low wasteland, as an aggregate for cinder or concrete blocks, and as a source of the metals it contains. Perhaps the chief obstacle to utilizing slag as a by-product of some value is that all operations at a power plant are subordinated to its prime function of producing electrical energy. It has not been found generally practical to coordinate the use of the equipment and procedures necessary for the reclamation of a minor by-product with the major function of a power plant.

Ash Fouling

With all methods of coal firing, a portion of the ash is carried through the unit by the combustion gases. The amount and type of material depend, in part, upon the method of firing and also somewhat upon the quality and characteristics of the coal fired. The degree to which this suspended ash, or fly-ash, causes operational difficulties is a function of the characteristics of the fly-ash itself, the design of the heat-absorbing surfaces in the unit, and the conditions of operation.

The fouling of a unit by fly-ash can assume several forms. Fused-ash or slag deposits can occur on furnace walls and other surface exposed to high gas temperatures. These deposits are caused primarily by the entrainment of fused ash and slag in the combustion-gas stream. Impingement upon furnace walls results in cooling and solidification and can cause the formation of a heavy layer of dense slag. In the transition temperature range between liquid and solid condition of the ash, semifused deposits can form as large clusters.

Ash or slag deposits on furnace walls decrease heat absorption in the furnace and cause an increase in the temperature of gases entering superheater banks.

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Large accumulations of ash or slag may break loose and fall, causing damage to floor or screen tubes below.

Sintered or bonded deposits are found on closely spaced convection surfaces where gas temperatures are still relatively high. This type of deposit occurs at temperatures appreciably below the initial-deformation temperature of the ash. It is usually characterized by a bonding agent that is present in proportionately greater amounts than it is in the parent fly-ash. The bonding agent may consist of compounds of sodium, potassium, phosphorous, calcium, silica, or combinations of several of these.

The typical bonded deposit consists of a relatively thin inner layer next to the tube surface. This inner layer is rich in the bonding agent and is firmly attached to the tube. The major part of the deposit builds upon this layer and becomes fused as it extends farther out into the gas stream. The outer part of the deposit has essentially the same composition as the fly-ash.

Deposits in the convection passes are difficult to remove from closely spaced surfaces and often obstruct the gas path sufficiently to require shutdowns for cleaning.

In the cooler parts of the unit (economizer and air heater), deposits are more generally a result of the condensation of sulfuric acid from high-dew-point gases, which causes fly-ash particles to stick together and to adhere to wet surfaces. The insulating effect of the deposits on the cooled metal surfaces tends to promote condensation, and the deposits become compacted through chemical and physical actions. The deposits are difficult to remove from the narrow gas-flow passages and can ultimately obstruct these passages completely. These deposits, because of their acidic nature, are associated in many cases with corrosion (see *External Corrosion*, page 18-14).

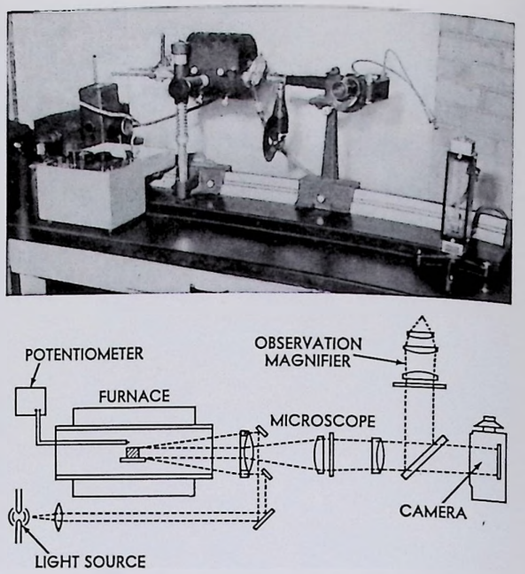


Fig. 12. *Microscope, light source, and furnace for laboratory determination of the behavior of ash under simulated furnace conditions*

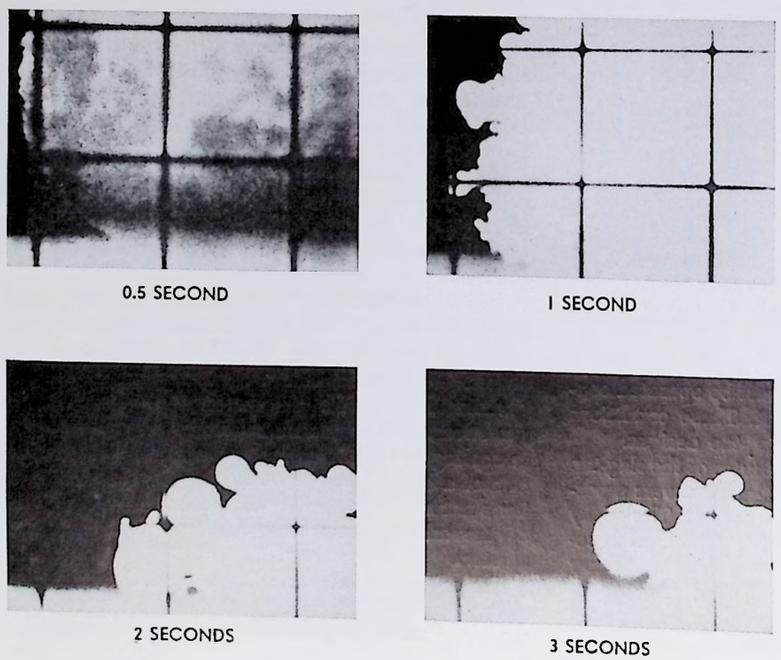


Fig. 13. *Four frames of a motion picture recording the combustion of a coal particle at 2700 F furnace temperature, obtained with the equipment shown in Fig. 12*

Microscope Studies

The Leitz heating microscope, shown in Fig. 12, is an aid in studying the ash-deposition problem. Components of the instrument are a furnace capable of maintaining a temperature of 2800 F, a 5x microscope, and a light source. The instrument was designed primarily for determining the melting behavior of coal ash, clinker, and other materials. It is also applied to other phases of this work, such as combustion experiments to determine the behavior of ash under simulated furnace conditions. In these experiments a single particle of 18-mesh coal is placed on a platinum stage that can be moved rapidly into or out of the furnace. Prior to insertion of the coal particle, the temperature of the furnace is raised to 2700-3000 F. The history of burning, which takes only a few seconds, is recorded in a motion picture. Four frames of such a film, at intervals of ½, 1, 2, and 3 seconds, are shown in Fig. 13. After ½-second exposure at furnace temperature, swelling of the coal particle and active combustion are well under way. At the end of 2 seconds, small spheroids and cenospheres of fly-ash are formed and are carried away by the combustion air. At the end of 3 seconds, large spheroids of ash coalesce to form a completely liquid droplet of slag.

Sintering of Fly-Ash

The combustion of some coals results in troublesome ash deposits that accumulate on furnace walls and convection tube banks. These deposits cannot be correlated with the fusion temperature of the coal ash nor with the gas temperature through the tube banks, since they occur in areas where the flue gas temperature is several hundred degrees lower than the ash-fusion temperature. The effect of "sintering" (the forming of a more or less solid mass by heating without thorough melting) on the compression strength of fly-ash from various coals appears to be a rational explanation supported by experience.

It has been established that the compression strength of test pellets of fly-ash from various coals, which have been sintered for 24 hr under conditions similar to those in a superheater, is a good index of fouling tendencies. The compression strength of sintered fly-ash from Central Illinois coal, which causes troublesome ash deposits, is ten to thirty times greater than that of sintered fly-ash from an eastern Ohio coal, taken for comparison, which does not produce troublesome deposits (see Fig. 14). The compression strength of the fly-ash from both coals increases rapidly with time and the temperature of the sintering treatment.

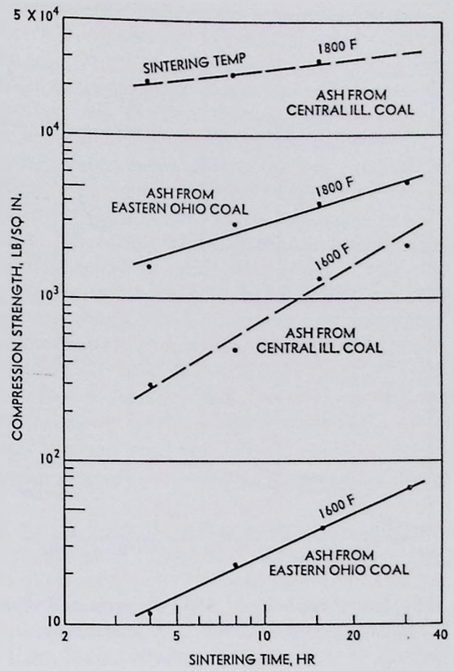


Fig. 14. *Variation of compression strength with sintering time for two fly-ash samples at two sintering temperatures*

Dry-Ash and Slag-Tap Furnace Installations

For the 25-yr period 1936 through 1960, B&W sold 429 slag-tap and dry-ash boiler units in sizes of 300,000 lb steam/hr and above, for a total capacity of over 336,000,000 lb steam/hr. Of these, 167 units for 130,899,000 lb steam/hr, or 39%, were of the slag-tap type, and 262 units for 205,706,000 lb steam/hr were of the dry-ash type. In the size range considered, the steam output of B&W dry-ash and slag-tap units by years is shown graphically in Fig. 15. It will be noted that, in the earlier years of the period, the greater part of the steam output was produced with slag-tap units. In the middle years of the period, the proportion of dry-ash units steadily increased, and in 1948 virtually an equal amount of steam was produced by each type. In the years 1948 through 1952, more steam was produced by dry-ash units than by the slag-tap type. The last period (1955 on) again shows a significant increase in the output from slag-tap units sold.

The statistics plotted in Fig. 15, for the units included in the study, indicate a steep increase in the application of the dry-ash method of burning pulverized coal during the middle period as compared with the earlier years. Not the least of the reasons for this increase is the fact that boiler manufacturers, by improvements in burners, furnace proportions, and the

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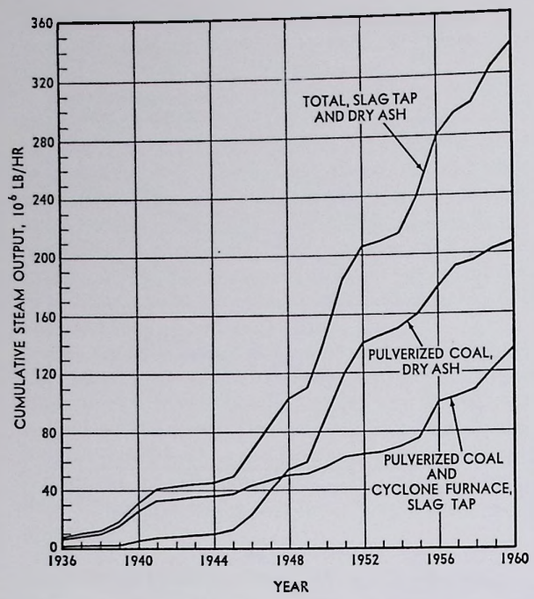


Fig. 15. *Steam capacity of B&W slag-tap and dry-ash boiler units of 300,000 lb/hr and above, for a 25-yr period*

arrangement of heating surfaces to preclude fouling, largely overcame the earlier difficulty of keeping all portions of the dry-ash unit free of slag or troublesome ash.

The last 5-yr period, however, again shows a significant increase in the use of slag-tap units. The inherent advantage of reduced air pollution with slag-tap furnaces exerted considerable influence on the use of these units in the years following 1955. The major contributing factor was the extensive development and acceptance of the Cyclone Furnace, a slag-tap design by nature of its operation.

External Corrosion

One of the problems that arise as a direct result of the constituents present in coal ash is external corrosion of heat-absorbing surfaces. External corrosion falls into two general categories: 1) low-temperature corrosion of economizers, air heaters, and flue gas ducts and 2) high-temperature corrosion of furnace walls, superheaters, and reheaters.

Corrosion and loss of metal require periodic repair or replacement of tubes, which is not only costly but also results in loss of boiler availability. Corrosion, along with attendant deposits, limits boiler capacity, reduces efficiency, and often leads to unscheduled outages. The economic consequences of this loss of availability can become very serious, especially for the electrical utilities.

Low-Temperature Corrosion

In recent years increased attention has been given to the problem of low-temperature corrosion in the economizers and air heaters of oil-fired and coal-fired boilers. Insofar as the latter are concerned, the problem is confined, for the most part, to stoker-fired boilers and in particular to underfeed stokers. Pulverized-coal and Cyclone Furnace-fired boilers are comparatively free from low-temperature corrosion, even when burning high-sulfur coals and operating with stack temperatures below 300 F. The problem is, however, fairly serious in boilers burning high-sulfur oils.

The basic cause of low-temperature corrosion is well known, although the precise mechanism is still a matter of some controversy. During combustion, most of the sulfur in the fuel burns to sulfur dioxide (SO_2), with a small part (1–3%) forming sulfur trioxide (SO_3). Passing through the boiler, the SO_3 combines with water to form sulfuric acid vapor. In most cases this sulfuric acid vapor makes up about 10 to 50 ppm of the flue gas composition. The presence of sulfuric acid vapor raises the dew point of the flue gas above that of water, which is normally 100 to 115 F, to an acid dew point of some 250 to 350 F (see Fig. 16). If metal temperatures within the boiler fall below the acid dew point, sulfuric acid (70-90%) condenses, and acid corrosion results.

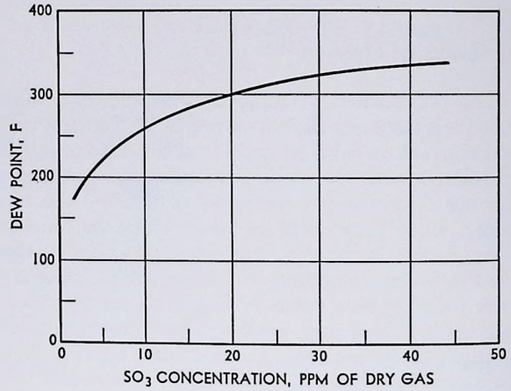


Fig. 16. *General relationship between dew point and SO_3 content of clean flue gas*

Generally, low-temperature corrosion is accompanied by plugging and ash deposition. Fly-ash particles act as condensation nuclei for sulfuric acid vapor, become wet, and stick to air heater surfaces. In addition, surfaces wetted by acid condensation collect and hold fly-ash. Acid will react with fly-ash in the cooler areas of the boiler to form hygroscopic salts, such as ferrous sulfate, aluminum sulfate, and bisulfates of sodium and potassium, which adhere to surfaces and promote corrosion even above the acid dew point.

High-Temperature Corrosion

Corrosion of the high-duty surfaces of boilers, superheaters, and reheaters has become more prevalent in recent years with the continuing trend toward higher operating pressures and temperatures. Superheat and reheat temperatures of 1050 F are now common. This has required higher heat absorption in superheaters and reheaters, and the additional heat has been obtained, for the most part, by placing part of the superheater and reheater in the higher-gas-temperature zones of the secondary furnace. This has resulted in higher metal temperatures and has required the more extensive use of austenitic materials.

The increased incidence of corrosion in these areas is apparently the result of a number of factors, none of which is sufficient to suggest a positive solution to the problem. In most boilers there is a layer of ash next to the surface of superheater and reheater tubes that is rich in alkali sulfates, irrespective of the alkali content of the coal burned. This alkali-rich ash can be very corrosive to some alloys at metal temperatures above 1100 F, common in boilers producing 1050 F steam.

Although the exact mechanism of corrosion by alkaline ash is still unknown, it is generally believed to stem from complex iron and/or aluminum alkali sulfates that form in the ash deposits. These complex salts become molten on tubes with surface temperatures above 1050 F, and a "molten salt" corrosion results. Corrosion is fairly rapid up to a temperature of 1300 F, the apparent upper limit for thermal stability of the complexes.

Corrosion in coal-fired boilers is not always associated with large accumulations of alkali-rich ash. Generalized corrosion has been experienced where the ash coating amounted to only mere flecks scattered over the tube surfaces. In some cases corrosion in these boilers has been related to borderline firing conditions, with reducing or near reducing gases in the region of the high-duty surfaces. In other cases sulfurous gases have been suspected of being the cause of corrosion, particularly where high alloys have been affected. Higher tube-metal temperatures again seem to be the common denominator.

Corrosion from chlorides in the gas or in the ash has not been given much attention in this country, although it is one of the forms of high-temperature corrosion which has affected boilers in Europe. In most U.S. coals, the chlorine level is considerably lower than that normally found in European coals. It has been demonstrated that very small amounts of chloride can trigger the corrosive action of alkali sulfate salts, and it is possible that the chloride content of some domestic coals could contribute to their corrosive nature.

The present state of knowledge concerning the causes of high-temperature corrosion in coal-fired boilers is not sufficient to establish direct corrective measures. There are probably several corrosion mechanisms that occur simultaneously. Since this is a relatively new problem that has become more prevalent with the use of higher steam temperatures and pressures, much research will be required to find a solution.

Some provisional corrective measures have been taken to reduce or arrest corrosion in specific boilers. Careful selection of coals with lower alkali content has, in some instances, been helpful. The use of basic additives (calcium and magnesium compounds) has occasionally been successful, although their exact role is not clearly defined. It is probable that they combine with SO_3 and thereby limit the amount of complex sulfates that can form on tube surfaces.

Fuel-Oil Ash

In recent years there has been a marked increase in the deposition of slag and ash on the heating surfaces of oil-fired boilers in both stationary and marine service. This may, in part, result from the higher ash content of residual oils from modern refineries. This condition has affected boiler operation in the following ways:

1. Deterioration of refractory furnace walls.
2. Slagging of high-temperature superheater surfaces with deposits not successfully removed by air or steam soot blowers.
3. High-temperature corrosion of steels, including high-chromium-content alloys, such as superheater tubes, spacers, and hangers.
4. Reduction of heat transfer through boiler, economizer, and air heater surfaces because of ash deposits.
5. Low-temperature corrosion of the cold sections of air heaters and duct work.

Of these difficulties, slagging of superheater surfaces has resulted in lower availability, increased draft loss, and decreased steam temperature. In some plants, monthly outages for cleaning have been required. Recently high-temperature corrosion of tubes and fixtures has resulted from increased steam and metal temperatures.

Since the early 1930's, the major factors responsible for the troublesome conditions noted above are the increased use of higher steam temperature, with the attendant increase in superheater metal temperature and the widespread use of poorer quality fuel oils. The B&W Co. has extensively investigated these problems.

There are two principal reasons for the deterioration in the quality of heavy fuel oils used for steam generation. First, the vanadium and sulfur contents

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of crude oil from certain new oil fields, particularly the foreign fields, are higher than those from the older domestic fields which supplied most of the demands in the past. Second, changes in refinery practice, including the extraction of larger quantities of top-quality products from the crude, have increased the concentration of undesirable ash in the heavy residual oil. Thus there is an increase in ash content as well as a change in ash quality.

Early records show oil-ash content ranging from 0.01 to 0.10%. Today, ash content may be as high as 0.2%, and 0.10–0.15% ash is common. Despite this relatively large increase, the ash content of present-day heavy fuel oil is still quite low—in the order of 1% of the amount commonly found in coal. Without the accompanying change in oil-ash composition, this increase would not be significant.

There is a great difference between the composition of the ash of present-day troublesome fuel oils and that of the earlier trouble-free fuel oils. Ash from the latter contained chiefly iron oxide, silica, calcium and magnesium compounds, and small amounts of other materials. The ash from troublesome fuel oils contains large quantities of alkali sulfates and, in many instances, considerable vanadium, which is more prevalent today and in large quantities. The source of the vanadium is the crude oil itself. Heavy fuel oils from Venezuelan crudes are particularly high in vanadium. Several other fields also yield crudes high in vanadium.

Typical analyses of oil ash causing the difficulties listed previously in items 1, 2, and 3 are given in columns 1 and 2 of Table 4. A typical ash analysis of a trouble-free oil is given in column 3. The analysis

TABLE 4

Ash Analyses of Some No. 6 Fuel Oils

Column No.*		1	2	3	4	5	6	7
Oil analysis, % by wt								
Ash		0.2	0.14	0.04	0.10	0.11	0.01	0.02
Sulfur		3.0	—	—	2.4	0.9	2.5	3.2
Ash analysis, % by wt								
Silica	as SiO_2	1.8	8.8	24.0	2.3	1.6	—	—
Iron	as Fe_2O_3	3.9	7.6	54.0	1.5	4.3	—	—
Aluminum	as Al_2O_3	3.0	4.1	5.7	0.1	0.1	—	—
Calcium	as CaO	6.0	9.5	3.0	0.1	2.4	—	—
Magnesium	as MgO	1.2	3.3	8.9	1.9	2.0	—	—
Sulfur	as SO_3	45.7	43.3	0.6	13.9	51.8	—	—
Nickel	as NiO	2.1	—	1.0	6.4	1.3	—	—
Vanadium	as V_2O_5	13.0	0.6	—	63.2	0.7	none	39.6
Alkalies	as Na_2O	20.5	23.5	0.6	12.4	30.5	—	—
Cu, Sn, Pb, etc., as oxides		—	—	—	—	0.3	—	—
Undetermined		—	—	—	—	2.9	—	—
Ignition loss @ 900 F, % by wt		—	—	—	2.5	—	—	—
Ash-fusion temperatures, F								
Reducing atm (ASTM)								
Initial deformation		1540	—	—	2880	1550	—	—
Softening		1560	—	—	> 2880	1850	—	—
Fluid		2430	—	—	> 2880	2560	—	—
Oxidizing atmosphere								
Initial deformation		†	—	—	< 1400	1590	—	—
Softening		†	—	—	1400	‡	—	—
Fluid		2720	—	—	1520	2800	—	—

*Col No. Fuel Oil
1, 2, 3 Source unknown
4 Venezuelan crude
5 East Texas crude
6 Arabian crude
7 Iranian crude

†Material started to drain out of cone at 1650 F, leaving skeleton cone which started shrinkage at 2250 F.

‡Material drained out of cone.

of an oil ash from a Venezuelan crude is given in column 4. Oil ash from east Texas oils, column 5, may be expected to be troublesome. Partial analyses of oil ash from an Arabian oil and an Iranian oil are given in columns 6 and 7, respectively.

The transition from the trouble-free oils through those containing alkali sulfates as the principal trouble agents to those containing both alkali sulfates and vanadium pentoxide is illustrated in Table 5 and graphically in Fig. 17. The data presented are an average of analyses of ash from heavy fuel oils.

TABLE 5

Averaged Analyses of Ash from Heavy Fuel Oil

Constituents		Fuel Oil Ash Analyses, % Trouble-free	Troublesome		
Ferric oxide	Fe_2O_3	54	8	6	1
Silica	SiO_2	24	9	5	4
Alumina	Al_2O_3	6	4	1	1
Lime	CaO	3	10	1	2
Magnesia	MgO	9	3	2	1
Vanadium pentoxide	V_2O_5	—	1	35	76
Alkali sulfates		1	67	37	8
Nickel oxide	NiO	—	—	—	6

In the past, the sulfur content of most heavy fuel oils burned in the U.S. was in the order of 1% or less. Today, the sulfur content can be as high as 3 to 3.5%, and 2% sulfur is common. Although this increase in sulfur content has not been as clearly related to the increase in slagging difficulties as has the change in ash quality, it is believed that the higher sulfur content contributes to slagging troubles. High sulfur content, however, has been definitely related to increased troubles with air heater fouling and corrosion.

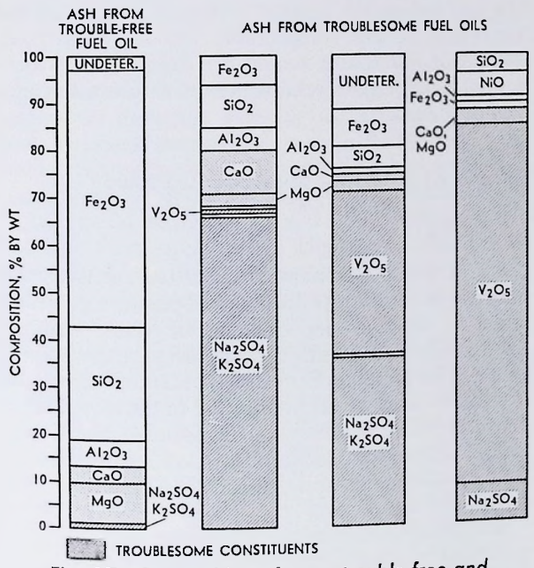


Fig. 17. *Composition of some trouble-free and troublesome ash from fuel oils*

Oil-Slag Formation and Deposits

It has been found that alkali sulfates and vanadium pentoxide are the principal constituents of troublesome superheater slags. Each of these compounds has a low melting temperature, and they react to form sodium vanadates. These sodium vanadates are formed by several combinations of sodium and vanadium, and some of the vanadate compounds have melting temperatures lower than either sodium sulfate or vanadium pentoxide. The other high-melting constituents of oil ash are trapped in the liquid portion of the slag and become a bonded deposit. Since the matrix of the deposit consists of relatively low-melting constituents, oil-ash deposits do not tend to become as massive as those of coal ash. A few examples of the melting temperatures of these constituents are given in Table 6.

TABLE 6

Melting Temperatures of Some Oil-Ash Constituents

Constituent	Melting Point in Air, F
V_2O_5	1240
$Na_2O \cdot 6V_2O_5$	1180
$Na_2O \cdot 3V_2O_5$	1040
$Na_2O \cdot V_2O_5$	1040
Na_2SO_4	1625
$CaSO_4$	2640
Fe_2O_3	2850
$MgSO_4$	Does not melt—begins to decompose @ 1634 F

In contrast to coal ash, melting temperatures of oil ash containing a major content of vanadium tend to be lower in an oxidizing atmosphere and higher in a reducing atmosphere. This is because V_2O_5 is reduced to V_2O_3 and V_2O_4, which melt at about 3500 F, whereas V_2O_5 melts at 1240 F.

External Corrosion

High-Temperature Corrosion

When superheater or reheater surfaces are covered with molten oil ash, corrosion from sodium-vanadium compounds can occur. A measurable corrosion rate can result over a wide range of metal and gas temperatures, depending upon oil-ash composition. Therefore it is impossible to single out a specific metal tem-

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perature at which corrosion begins. Fig. 18 shows the combined gas and metal temperature effects on corrosion for a specific fuel composition of 150 ppm vanadium, 70 ppm sodium, and 2.5% sulfur. As the vanadium concentration of the fuel oil varies, the amount of corrosion, compared to a 150-ppm V fuel, will vary according to the curve shown in Fig. 19.

In the corrosion process, the role of sodium in fuel oil is not as clear as that of vanadium. Varying the sodium content of fuel oil does not have as pronounced an effect on the corrosion rate. It does, however, affect the minimum temperature at which corrosion will begin because of the formation of the low-melting sodium vanadates that produce a liquid ash at lower temperatures.

At the present time there does not appear to be any alloy immune to oil-ash corrosion. In general, the higher the chromium content of the alloy, the more resistant it is to attack. This is one of the principal reasons for the use of the 18Cr-8Ni austenitic alloys for high-temperature superheater tubes. The higher alloy 25Cr-20Ni has been used as a protective cladding over other tubes, but it, too, does not afford complete protection against corrosion. The higher material cost must be justified by longer life, which is not always predictable.

Low-Temperature Corrosion

In oil-fired boilers the problem of low-temperature corrosion resulting from the formation and condensation of sulfuric acid from the flue gas is similar to that described on page 18-14.

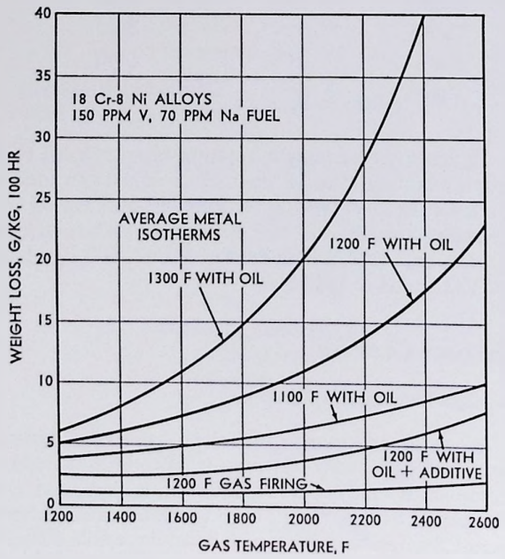


Fig. 18. *Effect of temperature on oil-ash corrosion*

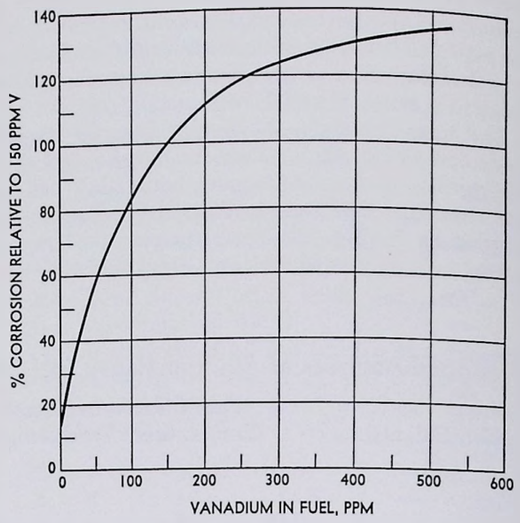


Fig. 19. *Effect of vanadium concentration on oil-ash corrosion*

Oil-fired boilers are more susceptible to low-temperature corrosion than are most coal-fired units for two reasons: 1) vanadium in the oil ash is a catalyst for the conversion of SO_2 to SO_3 (vanadium pentoxide is one of the catalysts used in the production of sulfuric acid) and 2) the smaller quantity of ash in the gas stream apparently is a factor contributing to the difference. Ash particles in the gas stream can absorb SO_3 and reduce the amount of free SO_3 vapor in the gas. Since oil has considerably less ash than coal, significant differences would be expected. Furthermore, the basicity of coal ash tends to neutralize any acid deposited.

Control of Ash Deposition and Corrosion

Cleaning

Air or steam soot blowers are often relatively ineffective in removing oil ash, and operators generally resort to out-of-service cleaning by water-washing. Fortunately most oil-ash deposits are partially soluble and can be removed in this way. Suitable means for disposal of wash water are needed in those regions of the unit where this method of periodic cleaning is used. To avoid weak-acid corrosion of metal parts, the unit should be promptly returned to service so that it will dry out.

In-service water-washing at reduced loads has been practiced to some extent on oil-fired units, using the hot high-pH boiler water in carefully regulated amounts either through hand lances or specially

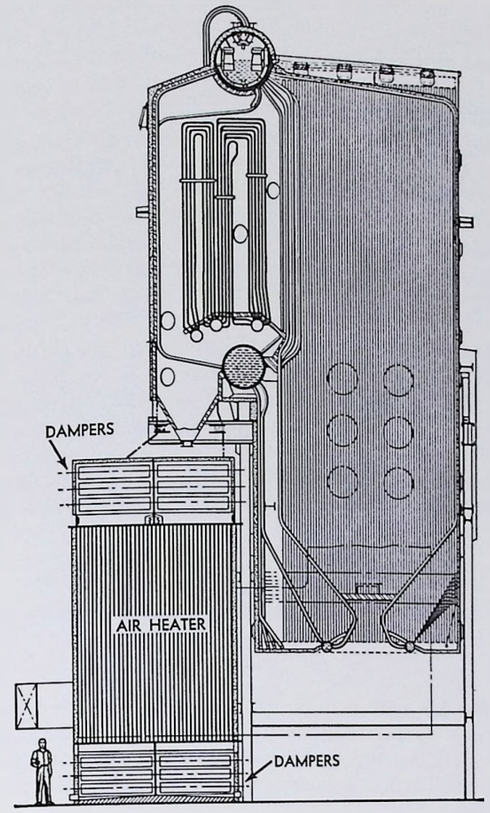


Fig. 20. *Sectionalized air heater for in-service water-washing*

equipped soot blowers. Under proper control, the method is generally safe for carbon-steel surfaces, such as boiler, economizer, and air heater tubes. In-service water-washing of carbon-steel superheaters is a somewhat questionable practice because of the higher metal temperature and consequent possible damage from quenching. High-alloy superheater tubes should never be water-washed in service, since these alloys are generally quite susceptible to thermal-shock damage (quench cracking).

The air heater arrangement illustrated in Fig. 20 has the ducts and dampers so designed that one half of the heater can be isolated with the boiler in service at reduced rating. Service water at room temperature is satisfactory for this operation, which can be performed in a few minutes with an ordinary hose and nozzle manually directed through the cleaning and access doors. Large quantities of water should be used so that any acid formed will be immediately diluted. The drains for final disposal of wash water are at the bottom of the air heater.

Fuel-Oil Additives

The practice of water-washing out of service and, to a limited extent, in service has been beneficial in overcoming some of the troubles experienced with present oil fuels. In addition, continued study of the problem has revealed another approach that is effective where the fuel-oil ash is most troublesome. In brief, the method involves adding to the fuel or furnace small amounts of materials that sufficiently change the character of the ash to permit its removal by steam or air soot blowers or air lances.

It has been found that additives are effective in reducing the troubles associated with superheater slagging, high-temperature ash corrosion, and low-temperature sulfuric acid corrosion. The most effective additives are alumina, dolomite, and magnesia. Kaolin is also a source of alumina. Analyses of typical superheater deposits from a troublesome fuel oil, before and after treating it with alumina or dolomite, and the results for a different oil with magnesia are shown in Table 7 and a bar graph in Fig. 21.

The reduction of slagging and high-temperature corrosion is basically the result of producing a high-melting-point ash deposit that is powdery or friable and easily removed by soot blowers or lances. When the ash is dry, corrosion is considerably reduced.

Low-temperature sulfuric acid corrosion is reduced by the formation of refractory sulfates by reaction with the SO_3 gas in the flue gas stream. By thus removing the SO_3 gas, the dew point of the flue gases is

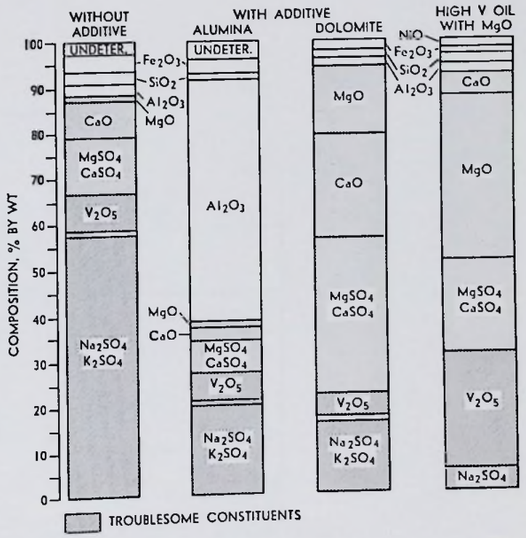


Fig. 21. *Effect of fuel-oil additives on composition of oil-ash deposit*

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TABLE 7

Fuel-Oil Additives—Effect on Oil Slag

Analysis of Typical Superheater Deposits Before and After Use of Additives

Constituents		Without Additive	With Alumina Additive	With Dolomite Additive	With Magnesia Additive*
Silica	SiO_2	2.8	1.1	2.0	2.0
Alumina	Al_2O_3	2.1	53.7	1.7	2.0
Titania	TiO_2	0.1	0.1	—	0.1
Ferric oxide	Fe_2O_3	1.8	3.2	1.6	1.0
Ferrous oxide	FeO	1.4	0.3	—	—
Phosphorus pentoxide	P_2O_5	0.037	0.06	0.1	—
Manganese oxide	MnO	—	—	—	—
Lime	CaO	7.8	2.4	22.6	6.4
Magnesia	MgO	1.8	1.0	15.2	35.0
Sulfur trioxide	SO_3†	43.8	18.6	44.8	20.0
Vanadium pentoxide	V_2O_5	8.6	6.6	4.9	26.0
Nickel oxide	NiO	2.9	1.3	2.0	2.0
Potassium oxide	K_2O	8.6	1.2	—	—
Sodium oxide	Na_2O	18.6	8.4	7.6	5.4
Ignition loss		0.2	0.4	—	—
Undetermined		—	1.7	—	—
Total		100.5	100.0	102.5	99.9
Physical appearance		Semifused, rocklike	Friable	Powdery	Powdery

*High-vanadium oil. †Combined with magnesia (MgO) and lime (CaO).

sufficiently reduced to protect the metal surfaces. The sulfate compounds formed are relatively dry and easily removed by the normal cleaning equipment.

In general, the amount of additive used should be about equal to the ash content of the fuel oil. In some instances, slightly different proportions may be required for best results, especially for high-temperature-corrosion reduction, in which it is generally accepted that the additive should be used in weight ratios of 2 or 3 to 1, based on the vanadium content of the oil.

Several methods have been successfully used to introduce the additive materials into the furnace. The one in general use consists of metering a controlled amount of an additive oil slurry into the burner supply line, as shown diagrammatically in Fig. 22. The additive material should be pulverized to 100% through a 325-mesh screen (44 microns) for good dispersion and minimum atomizer wear.

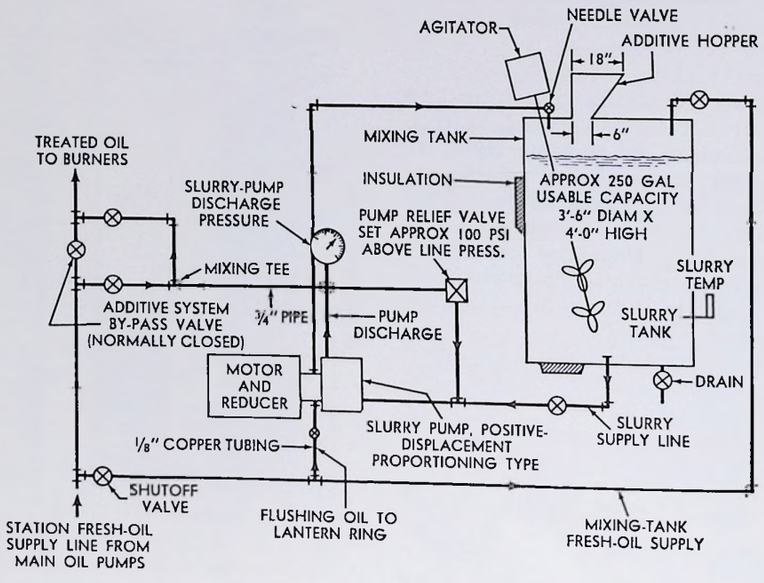


Fig. 22. *Fuel-oil-additive system (diagrammatic) for a 275,000 lb steam/hr boiler unit*

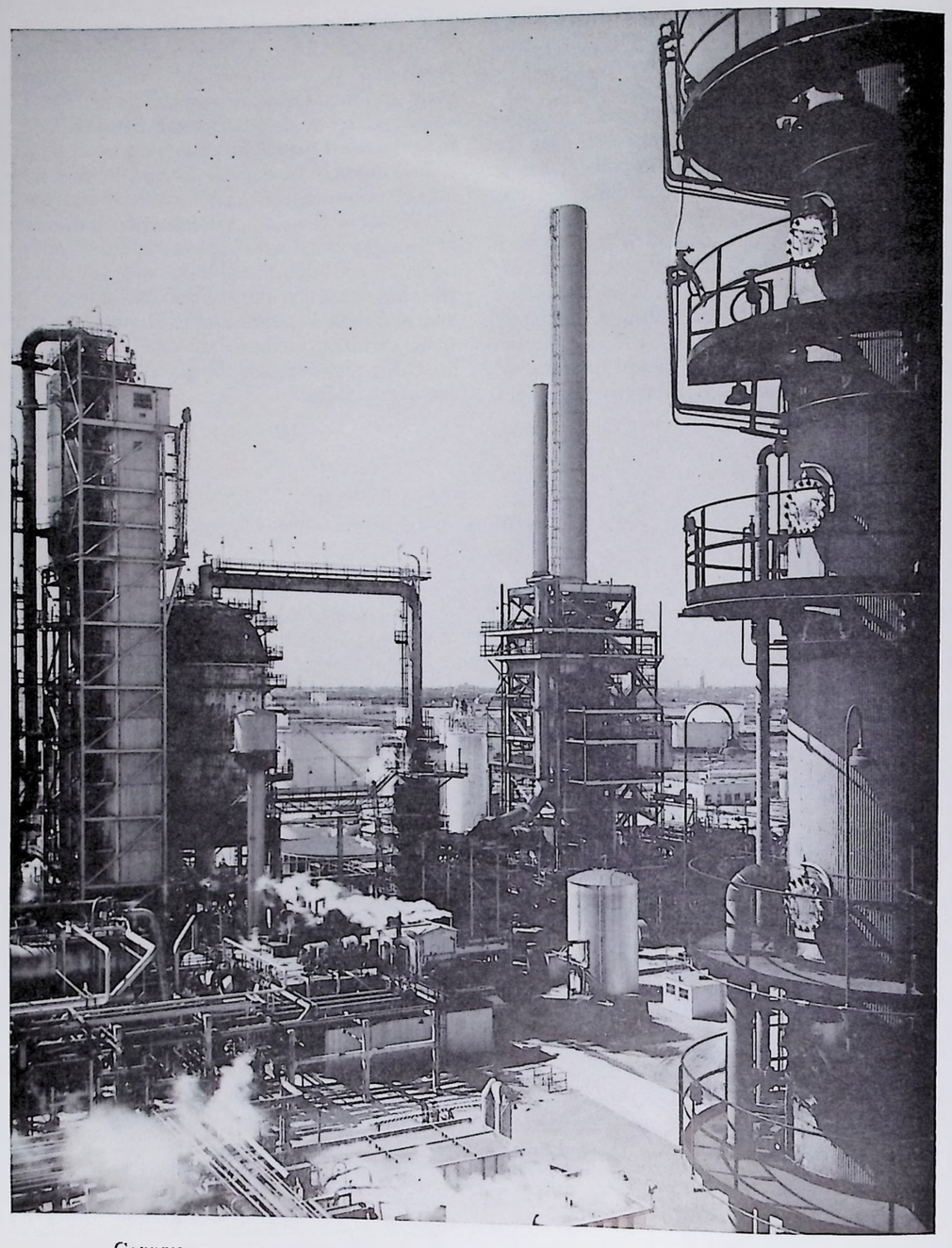
For a boiler fired by a high-pressure return-flow oil system, it has been found advantageous to introduce the additive powders by blowing them into the furnace at the desired locations. The powder has to be 100% through a 325-mesh screen for good dispersion.

A third and more recent method is to introduce the additive as a water slurry through specially adapted soot blowers or lances. This method offers the advantage of applying the additive in exactly the location desired, with a possible reduction in the quantity required. Some caution should be observed with this system to prevent possible thermal-shock (quench-cracking) damage to the hot tubes. The presence of chlorides in the water slurry, from either the water or the additive material, could possibly produce stress-corrosion cracking of austenitic tubing and should be considered.

The choice of the particular additive material depends upon its availability and cost to the individual plant and the method of application chosen. For example, alumina causes greater sprayer-plate wear than the other materials when used in an oil slurry.

The quantity of deposit formed is, of course, an important consideration for each individual unit from the aspect of cleaning. A comparison of the amounts of deposit formed with different additives shows that dolomite produces the greatest quantity because of its sulfating ability, alumina and kaolin form the least, and magnesia is intermediate. However, when adequate cleaning facilities are available, the deposits are easily removed, and the quantities formed should not be a problem.

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CARBON MONOXIDE, FORMERLY A WASTE GAS, IS THE MAIN FUEL FOR THE CO BOILER (CENTER) OF THIS MODERN CATALYTIC CRACKING PLANT

19

INDUSTRIAL BY-PRODUCT ENERGY SOURCES

THE progressive increase in the cost of fuel and the technical progress in the utilization of waste energy have led to specialized designs and applications of boilers. Many industries have their specific problems with waste-product recovery. For example, the oil industry has a tremendous amount of energy in the gas discharged from the catalytic regenerators; steel mills have their blast-furnace gas; the sugar industry has its cane refuse; the wood-and-pulp industry has sawdust, hogged fuel, bark, and process liquors (see Chapter 20) as wastes. All these waste products or residuals can be burned as fuels for the generation of steam. In addition, there are industries or processes that have large quantities of high-temperature gases from which the sensible heat may be abstracted for steam generation. Such gases are produced in copper reverberatory furnaces; annealing, forge, and billet-heating furnaces; open-hearth furnaces; and fired kilns of many types.

Gaseous By-Product Fuels

Carbon Monoxide

In the petroleum industry, the efficient operation of a fluid catalytic cracking unit produces gases rich in carbon monoxide. To reclaim the thermal energy represented by these gases, the fluid catalytic cracking unit shown in Fig. 1 was designed to include a pressure-fired "CO" boiler as an integral part. The arrangement is such that it can be operated independently or taken out of service without affecting the operation of the cracking unit.

The fluid catalytic cracking unit consists essentially of a reactor and a regenerator. In this process the finely divided catalyst, suspended in the vapors, flows continuously in a cycle from the reactor to the regenerator and back to the reactor. Gas-oil feed stock is injected into the hot regenerated-catalyst line just before it enters the reactor. The catalyst circulation

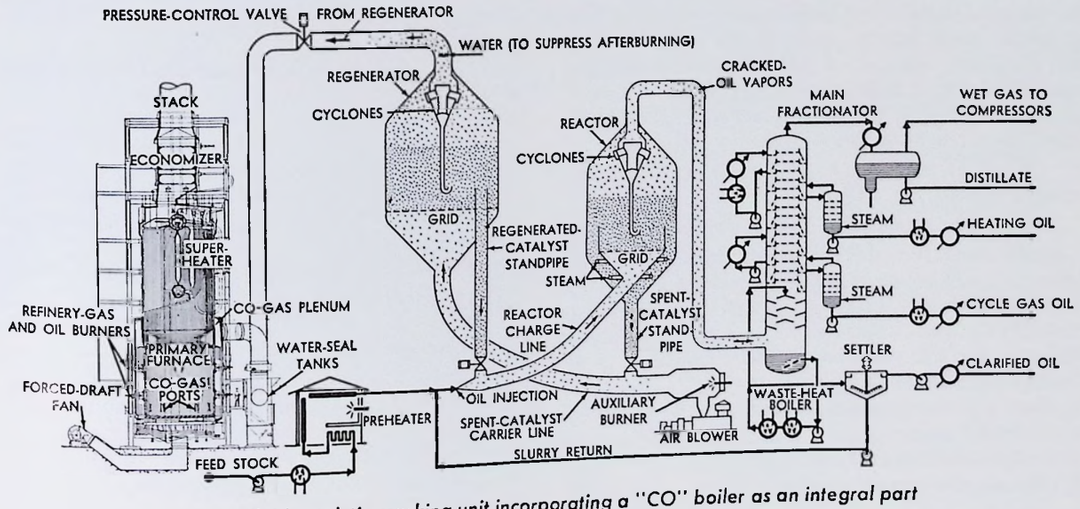


Fig. 1. *Fluid catalytic cracking unit incorporating a "CO" boiler as an integral part*

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rate is controlled to maintain the required reaction temperature of about 950 F. Hydrocarbon vapors leave the reactor through cyclone separators, which return the entrained catalyst to the reactor bed, and the cracked petroleum products are separated in the fractionator.

In the reactor the catalyst accumulates a carbonaceous deposit. The spent, or carbon-coated, catalyst is transported to the regenerator by injecting compressed air into the catalyst stream. Additional air is supplied directly to the regenerator to burn the carbon from the catalyst. The heat of combustion is absorbed by the regenerated catalyst, which, in turn, heats the oil feed stock to effect vaporization. The oil vapors and catalyst are then discharged into the reactor to begin the cracking and refining process.

The products of combustion from the burning of the carbon coating have a CO_2/CO ratio of nearly 2/1, with CO percentage ranging from 5 to 8. The gases are discharged from the regenerator through cyclone separators to remove as much of the entrained catalyst as possible, prior to entering the CO boiler for heat recovery.

The CO boiler (Fig. 2) was developed to recover the heat discharged from the catalytic regenerator —practically all the heat of the combustibles and approximately 50% of the sensible heat.

The gases leave the regenerator at approximately 1000 F, which, with a temperature drop of 500 F, provides 145 Btu/lb as sensible heat for steam generation. The combustible content of the gas stream is the result of the incomplete burning of the carbon at low temperature with, in most instances, a deficiency of air. The unburned combustibles consist primarily of carbon monoxide with some traces of entrained hydrocarbons. In catalytic cracking units it is desirable to burn the carbon to produce a maximum of CO instead of CO_2, since a cubic foot of air combines with twice the amount of carbon when CO is made. Thus, blower power is saved by burning the maximum amount of carbon with a minimum amount of air, and the catalyst temperature is held down to 1000 F by releasing a minimum amount of heat. These reactions are:

$$2C + O_2 = 2CO + 3{,}958 \text{ Btu/lb of carbon}$$

compared with

$$C + O_2 = CO_2 + 14{,}093 \text{ Btu/lb of carbon}$$

Supplementary fuel is always required in the operation of a CO boiler. It is necessary not only to raise the temperature of the CO gases to the ignition point but also to assure complete burning of the combustibles in the CO-gas stream. The following design criteria have been established:

1. The basic firing rate shall produce a temperature of 1800 F in the primary furnace, to provide safe and stable combustion of the fuels.
2. The combustion air supplied by the forced-draft fan shall be sufficient to furnish the oxygen for combustion of the supplementary fuel, the oxygen for combustion of the CO gases, and the excess air required to provide 2% excess oxygen leaving the unit.
3. The firing equipment shall be sized to permit supplementary firing for raising the temperature of the CO gases to 1450 F to start the burning of the combustibles in the CO gases.

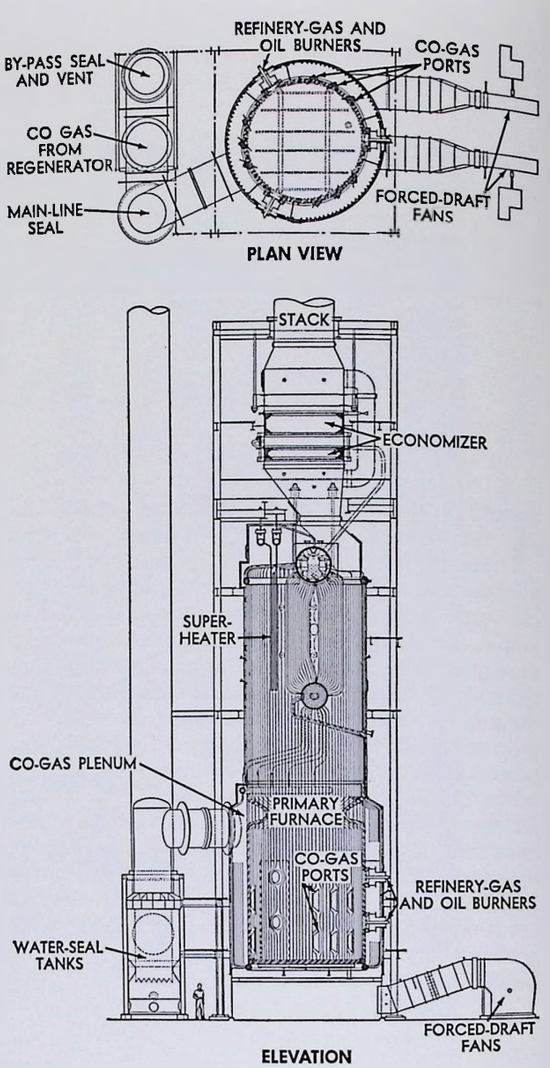


Fig. 2. *The "CO" boiler, a pressure-fired top-supported unit*

The furnaces, both primary and secondary, and the boiler tube bank are designed as a single integrated boiler unit supported at the top, with provision for downward expansion. The supplementary-fuel burners (Fig. 3) are positioned radially and the

CO-gas nozzles tangentially to the cylindrical furnace to effect thorough mixing and promote rapid and complete combustion. The fuel burners are arranged in a staggered pattern with respect to the CO-gas nozzles. The wall tubes are covered with refractory to aid in burning the CO gas with minimum supplementary fuel. The wall tubes also cool the refractory, thus protecting it when only supplementary fuel is fired.

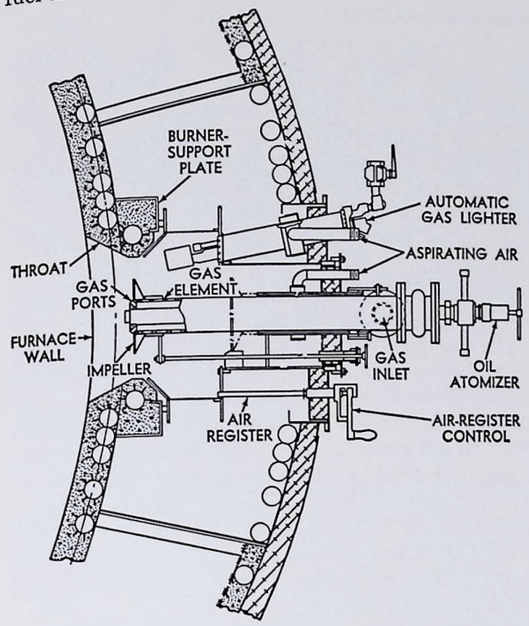


Fig. 3. *Center-fired gas and oil burners supply the supplementary fuel to "CO" boilers*

The CO-gas and combustion-air windboxes, or distribution chambers, are designed as an integral part of the furnace. Thus, a simple water-cooled arrangement for the high-temperature CO gases is provided to eliminate difficult and expensive differential-expansion and seal problems, especially important with toxic gases.

Normally the CO-boiler unit is designed for full-load operation when firing only supplementary fuel, so as to provide steam for starting the cracking unit. Operators have found that the CO boiler is different from the conventional waste-heat boiler in that steam is always available during not only normal operation but also start-up and shutdown. It is possible that a cracking unit may require more steam at start-up than under normal full-load operating conditions. It is not necessary to "boot strap" the cracking unit on the line when the CO boiler is used, nor is it necessary to "borrow" steam from other processes.

During this short period of operation with supplementary fuel, the combustion-air requirement is considerably higher than when utilizing CO gases. Consequently two fans are installed, each designed to supply one half of the maximum requirement. Both fans are used while operating the boiler at full capacity with supplementary firing only. Each fan is capable of supplying the combustion air for full-load operation with CO-gas firing and the necessary supplementary fuel; thus, only one fan is required for CO-boiler operation, and the second fan is a spare. Each fan is equipped with a pressure-interlock system that prevents the discharge of gas through the fan inlet in the event of fan failure. The system automatically closes the fan-discharge damper for a preset minimum pressure and admits purging steam to the duct.

The possible variations in the combustible and oxygen content of the CO gases, in the sensible heat of the CO gases, and in the amount of supplementary firing make it impractical to set up a fuel-air relationship as is done for conventionally fired units. Because of this variable relationship when firing multiple fuels simultaneously, it is necessary to determine directly the amount of excess oxygen leaving the unit. This may be determined intermittently by an Orsat, continuously by an oxygen recorder, or by a combination oxygen-combustible recorder.

Water-seal tanks (Fig. 4) are installed to provide for the independent operation of the CO boiler without interfering with the operation of the regenerator, so that the CO gases from the regenerator may be passed through the boiler or sent directly to the stack.

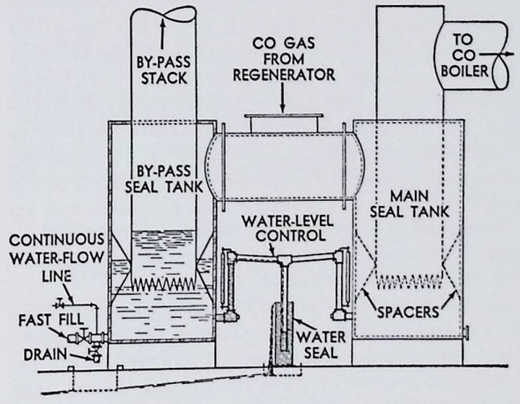


Fig. 4. *Water-seal tanks are the shutoff valves in the CO-gas lines*

Actually these water-seal tanks act as shutoff valves in the large gas lines. The flow of gas through any one of the seal tanks may be stopped by filling it with water to the level where the differential water pressure equals the CO-gas pressure, thus forcing

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the CO gas to flow through the other seal tanks. The CO-gas pressure at the seal is determined by the resistance to gas flow of the equipment downstream from the seal. To aid in establishing the water seal, a mechanical damper is installed ahead of the seal tank to throttle the gases while making the water seal. This mechanical damper is not a positive shutoff damper. Water-seal tanks are preferred to mechanical shutoff dampers because of the high gas temperature, the size of the CO ducts, and the necessity for a leakproof arrangement.

The operation of the CO boiler is coordinated with that of the catalytic cracker. Normally, the boiler will be required to supply steam for the operation of the catalytic cracking unit and will be started earlier, using supplementary fuel. However, the CO boiler should always be started using only the supplementary-fuel burners and by-passing the gases from the regenerator to the atmosphere. CO gases should not be introduced into the boiler until it is in operation, since they usually are at or below 1000 F and consequently tend to cool the furnace. They create a swirling action in the furnace and should therefore be brought in gradually to provide a smooth changeover. They ignite quite readily and burn with a nonluminous flame. As they are introduced to the boiler, steaming increases materially, making it necessary to cut back on the supplementary fuel and requiring a major downward adjustment of the combustion air. This readjustment in the air requirement is determined from the oxygen-recorder reading.

Since there are only slight variations in the operation of a catalytic cracking unit, the CO boiler is normally base loaded. The CO boiler handles all the gases from the regenerator regardless of the CO_2/CO ratio. A change in this ratio merely affects the quantity of supplementary fuel necessary to maintain the required furnace temperature of 1800 F. This temperature provides a reasonable operating margin for possible variation in the operation of the regenerator or the boiler. Stable operation can be maintained at furnace temperature as low as 1500 F, but the margin above the ignition temperature of the CO gas is considerably reduced.

The economics of the CO boiler depends upon the amount of available heat—both sensible heat and heat from combustibles—in the regenerator exhaust compared with an equivalent amount of heat from an alternate fuel. The amount of heat from the CO gases available for steam generation may be calculated by taking the sensible heat above an assumed boiler stack temperature plus all the heat from the combustibles. The additional steam generated in the CO boiler by the supplementary fuel is comparable with the steam generated in a conventional power boiler using the same fuel. Normally, the supplementary-fuel requirement will account for ¼ to ⅓ of the output, when the temperature of the entering CO gas is maintained at 1000 F.

The application described above has been confined to the oil-refining industry. There are areas in other industries where these principles could be used to advantage. An economic study will show the feasibility of burning waste gases as fuel for the generation of steam.

Blast-Furnace Gas

A gas containing about 25% by volume of CO is generated in the blast furnaces of steel mills. This valuable by-product fuel is called blast-furnace gas and is used principally for the generation of steam.

Although this gas can be burned in the condition in which it is discharged from the blast furnace, its high dust loading (5 to 7 grains/cu ft) would cause plugging of burner ports and excessive fouling of heat-absorbing surfaces. It is desirable, therefore, to reduce the amount of entrained dust through washing, followed by electrostatic precipitation.

The gas most difficult to burn is that which has been washed only. Since washing does not remove all the solids, those that remain mix with the water entrained in the gas. The mud thus formed adheres firmly to hot surfaces and is very difficult to remove. A gas with solids content reduced to 0.15 grain/cu ft would carry 14 lb of solids per hour to a burner of 650,000 cu ft/hr capacity (60 million Btu/hr). If only 1% of this amount (a very conservative estimate) collects in the burner itself, 3¾ lb of solids will accumulate in 24 hours.

A modern combination blast-furnace-gas, coke-oven-gas, and pulverized-coal burner that is admirably suited for such a gas is shown in Fig. 5. Note the large-area scroll entrance to the nozzle, where much of the foreign matter in the gas deposits. The scroll is cool, therefore the "mud" does not bake and can be removed readily through the quick-opening

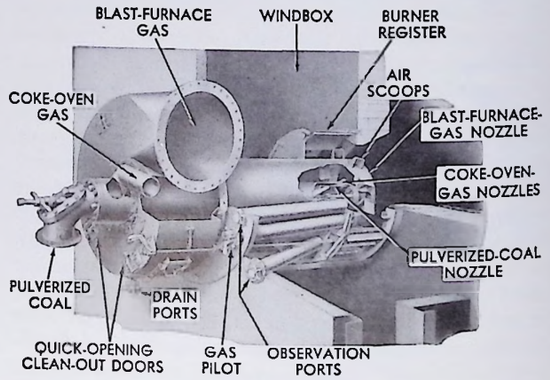


Fig. 5. A combination blast-furnace-gas, coke-oven-gas, and pulverized-coal burner

clean-out doors shown. The nozzle itself has a large area, and deposits that accumulate between normal boiler outages do not materially affect the burner capacity. No plugging or bridging is possible.

Several times a day the charge of iron, coke, and chemicals in a blast furnace will arch over. The collapsing of this arch is known as a furnace "slip." When a slip occurs, there is a resulting surge in gas pressure throughout the system, and, although momentary, it can extinguish the flames from the burners. Consequently, provision for immediate reignition is made in the design of blast-furnace-gas-fired boilers, in order to prevent serious explosions of furnace gas. Older units with refractory furnaces had the burners mounted in extension furnaces (Dutch ovens), in which small hand-fired grates were installed. The combination of hot fuel bed and incandescent brickwork served to reignite the burner after a blast-furnace slip. Modern boiler units use little or no refractory in order to minimize maintenance, and, to re-establish ignition after a slip, continuous-burning pilots are used. These pilots burn coke-oven gas, natural gas, or fuel oil.

Liquid By-Product Fuels

Pitch and Tar

The liquid and semiliquid residues from the distillation of petroleum and coal are known as pitch and tar. Most of these residues are suitable for use as boiler fuels. Some handle as easily and burn as readily as does kerosene, whereas others give considerable trouble. To determine whether a particular pitch or tar might be a suitable fuel for a given installation, the following items are important.

Moisture. If the fuel contains moisture, it must be well emulsified to avoid reaching the burner in slugs. If there is a brief break in the continuous flow of fuel to the burners, the fires will be extinguished. Upon re-establishing the fuel flow, a furnace explosion might occur if there is any delay in reigniting the burners. Consequently, a slug of water in the fuel supply can be disastrous if it extinguishes the flame briefly. Tars and pitches containing as much as 35% moisture may be burned in properly designed units.

Flash and Fire Points. Many liquid fuels are blends of two or more different liquids. One of these might have low flash and fire points, whereas the other might have high flash and fire points. Such a fuel usually burns with a bright flame at the burner, where the low-flash-point constituents are burning off; but beyond, where the components with the higher flash and fire points are burning, the flame is a dark yellow. Actually, if there is too little turbulence at the burner or if the burning products are quenched by passing too quickly from the active combustion zone, combustion is incomplete, and high unburned-combustible loss results. Consequently, while the flash temperature is useful for determining the possible hazard involved in storing the fuel, the fire point determines its suitability for firing in a boiler. Fuels with fire points as high as 600 F can be burned in properly designed equipment.

Viscosity. Practically all tars and pitches are burned in the same manner as fuel oil. They are reduced to a foglike dispersion in an atomizer located in a burner and then vaporized and burned. To produce the fine particles, the viscosity of the fuel must be correct—not over 180 Seconds Saybolt Universal (SSU) for most atomizers, although if favored by the burner-furnace arrangement, viscosities as high as 1000 SSU may be used.

Suspended Matter. Many of these fuels contain suspended matter. If they are delivered to the burners in this condition, there will be:

1. Abnormal fouling of the atomizers, requiring excessive cleaning.
2. Excessive rate of wear of burner parts.
3. Deposition of unburned carbon throughout the unit or objectionable stack emission.

Such fuels should therefore be passed through strainers before they are fed to the burners.

Compatibility. When some of these fuels come into contact with ordinary fuel oil, they combine to form liverlike substances. If this happens in tanks or piping, trouble results. The mixture cannot be pumped from the tanks, and plugging of the piping often requires complete dismantling for cleaning. Burner operation, too, is erratic and spasmodic. Therefore, before mixing large quantities of tar or pitch with fuel oil, sufficient laboratory tests should be made at both storage and pumping temperatures to determine the compatibility of the fuels.

Solid By-Product Fuels

Bagasse

Bagasse is the refuse from the milling of sugar cane. Physically it consists of matted cellulose fibers and fine particles, the percentage of each varying with the process. In the older plants the raw cane was ground by passing it in series through sets of grooved rolls, each set comprising a mill having finer grooves than the preceding one. Modern practice incorporates a shredder that cuts the cane with revolving knives prior to the tandem milling previously described. The end product has a higher percentage of fines and short fibers.

A typical analysis of bagasse is given on page 3-A3 of the Appendix. Bagasse generally contains about

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50% moisture and has a heat content of 3600 to 4200 Btu/lb as fired. It is used chiefly as a fuel to generate steam and power for the plant. Other by-product uses are for cellulose, for paper and paperboard manufacture, and for furfural production.

As a fuel for the production of steam, bagasse has been burned in several types of furnaces, the oldest being a Dutch oven with flat grates. Since it was difficult to distribute the bagasse evenly on the grates, the latter were subject to high maintenance costs from burning. A new type of furnace was therefore developed to burn the bagasse in a pile on a refractory hearth. Air was admitted to the pile around its circumference through tuyères. The most popular of these extension furnaces was the Cooke, but it also suffered from high maintenance costs because of excessive radiation and cleaning difficulties. To overcome these problems the Ward furnace was designed, as illustrated in Fig. 6.

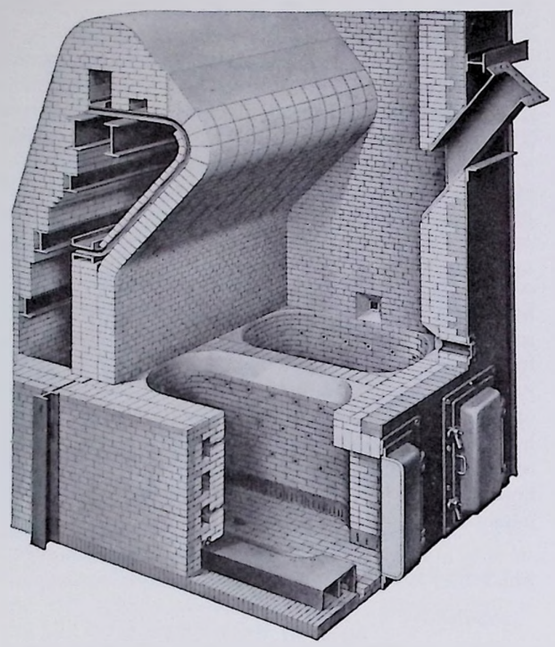


Fig. 6. *A Ward furnace for burning bagasse*

The Ward furnace has been very successfully used under sugar-mill boilers. It is easy to operate and maintain. Bagasse is gravity fed through chutes to the individual cells, where it burns from the surface of the pile with approximately 85% of the air which is injected into the sides of the pile adjacent to the hearth. This causes local incomplete combustion, but there is sufficient heat released partially to dry the raw fuel entering. Additional drying is accomplished by radiant heat reflected from the hot refractory to the cells. Combustion is completed in the secondary

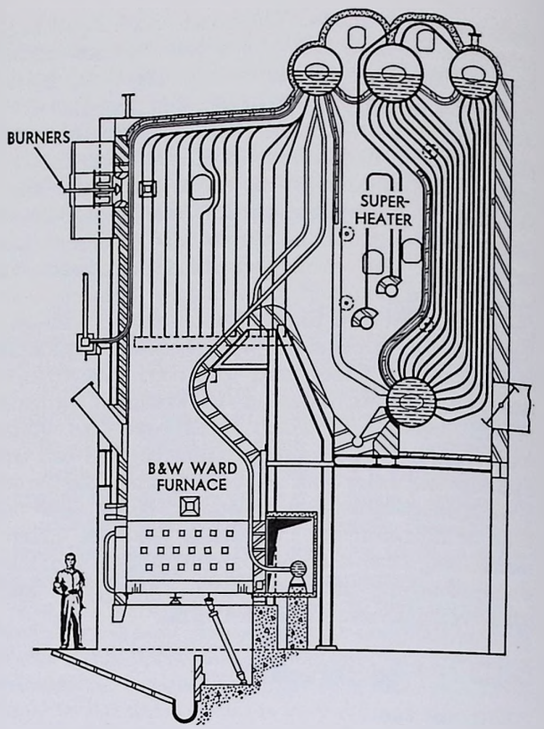


Fig. 7. *A low-draft-loss Stirling boiler with a water-cooled Ward furnace*

furnace above the arch. Ward furnaces are now equipped with dumping hearths, which permit the ashes to be removed while the unit is in operation.

Mechanical harvesting of sugar cane increases the amount of dirt in the bagasse to as much as 5 to 10%. To overcome the resultant slagging tendency of the ash, water-cooling is incorporated in the furnaces. Fig. 7 shows a modern unit with water-cooled furnace.

In the older mills, the drives for the milling equipment were large reciprocating steam engines, which used steam at a maximum of 150 psi and with a few degrees of superheat, exhausting at 15 psi to the boilinghouse steam supply. In more modern mills the drives are either turbines or electric motors with reducing gears. Both the turbines and the turbogenerators use steam at pressures of 400 to 600 psi and with temperatures up to 750 F.

Raw-sugar mills produce sufficient bagasse to meet all their steam requirements and in some cases an excess. Sugar mills that also refine usually generate from 80 to 90% of their steam requirements with bagasse, the remainder with supplementary fuel oil. Because of the high moisture content of the gas, the weight of the gaseous combustion products is about twice that from oil and one and one half times that from coal. This high gas weight causes excessive

draft loss and requires either extremely high stacks or fans to obtain the required steam capacity from the boilers. To alleviate these conditions a low-draft-loss Stirling boiler, as shown in Fig. 7, was designed. The vertical tube bank facilitates cleaning, and the special baffle arrangement usually permits natural-draft operation to the required peak capacity at thermal efficiencies in the order of 57%. In sugar mills with refineries, a higher efficiency is necessary to reduce the amount of supplementary oil required. A thermal efficiency of 65% may be obtained by the addition of an air heater and an induced-draft fan, as shown in Fig. 8.

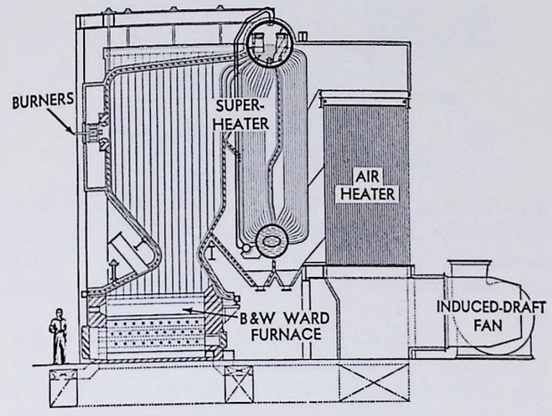


Fig. 8. *A Two-Drum Stirling boiler with air heater and water-cooled Ward furnace*

In recent years bagasse has been burned on stokers of the spreader type. This method of burning, however, requires bagasse with a high percentage of fines, a moisture content not over 50%, and a more experienced operating personnel. Because of such limitations, the Ward furnace is considered the most reliable, flexible, and simple method of burning bagasse.

Wood Refuse

The burning of wood as a fuel for steam generation is generally confined to those industries or operations where wood refuse is available. Average sawmill practice shows that, in the cutting of lumber, nearly half the total weight of a log appears as refuse—sawdust, chips, slab, and bark. The pulp-and-paper, furniture, and plywood industries are typical operations where such refuse appears in appreciable quantities.

All woods have substantially the same chemical analysis but may vary considerably in density and moisture content. Typical analyses of several kinds of wood are given on page 3-A3 of the Appendix.

Wood refuse available as a fuel may consist of large pieces, such as slabs, logs, and bark strips, and small pieces, such as sawdust and shavings. Furnaces for burning wood refuse are usually designed to handle chip size, in which case it becomes necessary to pass the large pieces through a hogger or chipper. Reducing the wood to chip size permits uniform continuous feeding, a more rapid burning of the small particles, and a more complete coverage of the grate.

Wood refuse from some processes contains 75% or more moisture. In order to obtain useful heat from the combustion of this type of fuel, it is necessary to reduce the moisture content to 60 or 65%. Mechanical presses are generally used for this purpose.

Wood-burning furnaces are not usually troubled with severe slagging conditions unless the wood is burned in combination with other fuels or unless the refuse comes from salt-water-borne logs. Ashes from two fuels often combine to form an ash of lower melting temperature than that of either ash taken separately. In some parts of the country, provision must be made for the removal of large amounts of sand and dirt which are dragged in with the logs. Such foreign material entrained in the combustion gases would cause the erosion of such surfaces as tubes, baffles, flues, and fans.

Grates for furnaces may be divided into air-cooled and water-cooled classes, or they may be a combination of the two. They should have about 30% free air space. For ash removal, grates may be of the rake-out, dumping, or moving type.

To design an efficient wood-burning furnace, the following steps in the combustion process must be considered: 1) drying, which proceeds rapidly on the surface but requires time to reach interior parts of large pieces, 2) distillation and burning of volatile matter, and 3) burning of fixed carbon. The temperature in the furnace must remain above the kindling point (between 750 and 1000 F), or the fire will go out. Furnace temperature depends upon the calorific value and the moisture content of the fuel, the weight of air flowing through the fuel bed, and the heat loss from flame and fuel bed by radiation. Furnaces designed for the burning of wood refuse fall into three types; pile, thin-bed, and cyclonic.

Pile Burning. The Dutch oven, or extension type of furnace with a flat grate, is the oldest design of furnace for burning wood refuse in a pile. Fig. 9 shows a modern version having water-cooled side walls, roof, and partial front wall. The hogged fuel is fed continually through a low roof and is permitted to pile up in a cone, the apex of which is directly under the feed hole in the roof. Most of the air for combustion enters through the grate around the edge of the pile and sweeps over the fuel, so that most of the burning takes place on the

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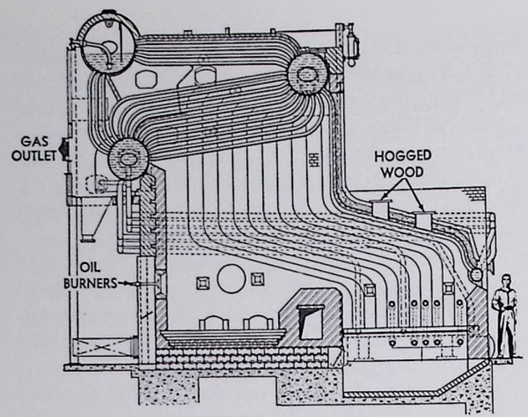


Fig. 9. *A water-cooled extension furnace for burning hogged wood under a Type H Stirling boiler*

surface of the pile. Some of the air is injected through the side walls toward the pile in order to obtain turbulence for completion of combustion.

The fuel supply cannot be closely regulated with this method of firing, and frequent manual adjustment is required for good operating results. Excess air at the boiler outlet is between 30 and 40%. Combustion rates of 125 lb per hr per sq ft of grate have been attained, burning wood having 45% moisture (as fired), using cold air. With hot air (300 to 350 F), the burning rate can be increased 25%. Above 45% moisture the combustion rate decreases rapidly. Ash is removed by rake-out or by dumping grates.

A small two-drum boiler unit designed to utilize

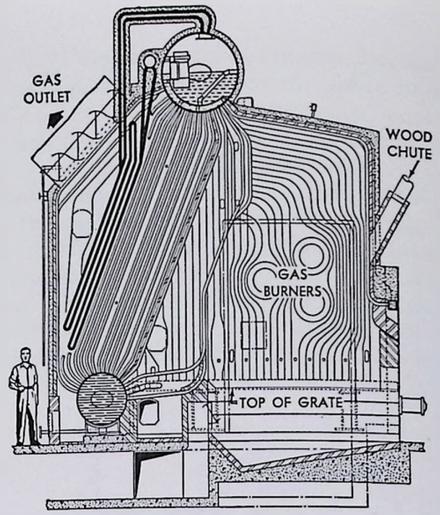


Fig. 10. *An Integral-Furnace boiler with the furnace serving as a Dutch oven*

the furnace as a Dutch oven is shown in Fig. 10. The wood drops by gravity into the furnace through circular chutes located in the upper front wall and falls into piles on top of dumping-type air-cooled grates.

A large two-drum single-pass boiler unit, in which the furnace acts as a Dutch oven, is illustrated by Fig. 11. In addition, provision has been made to burn oil in combination or alone. The furnace consists of two cells having water-cooled pinhole grates. Each cell is fed by its own air-swept chute through the

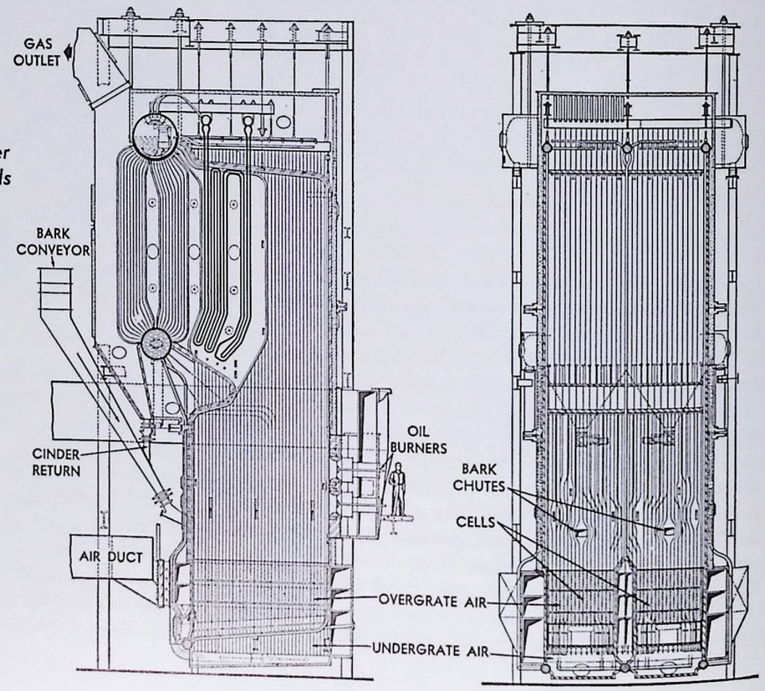


Fig. 11. *A Two-Drum Stirling boiler with water-cooled furnace and cells for hogged-fuel burning*

bridge wall. Most of the 600 F air for combustion is introduced through tuyères at three elevations around the four sides of each cell. Some undergrate air is required to aid ignition, particularly when fuel is first fed to the cell or when burning-down preparatory to removing ash. This unit is capable of burning hogged fuel having 60 to 65% moisture at a rate of 285 lb per hr per sq ft of grate with 30% excess air at the boiler outlet. Ash is raked out every third day.

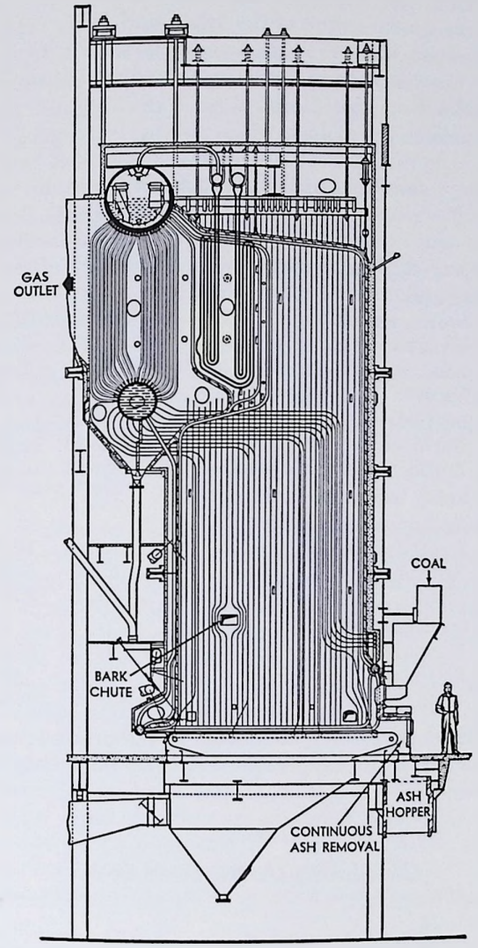


Fig. 12. *A Two-Drum Stirling boiler with spreader stoker for coal and/or wood firing*

Thin-Bed Burning. Fig. 12 illustrates a two-drum single-pass boiler equipped with a conventional spreader stoker for coal firing and single chutes through each side wall for wood firing. With this arrangement wood and coal can be burned separately or in combination. Introduction of the wood refuse well above the grate permits the smaller particles to dry out and burn in suspension, while the remainder of the fuel continues in flight to the grate, where combustion is completed. Most of the air for combustion goes through the grate, a small portion being injected at high pressure (30 in. of water) through tuyères in the front and rear walls to promote turbulence. This type of unit is designed to burn refuse containing 45 to 60% moisture at rates up to 1,000,000 Btu/hr, sq ft, when supplied with 35 to 70% excess air at 400 F. Ash is continuously discharged from the moving grate.

Another variation of a two-drum single-pass boiler design for wood burning in combination with oil, gas, or pulverized coal is shown by Fig. 13. The air-swept wood chutes are located across the front wall. With this arrangement most of the wood burns in suspension. Oil burners are located in the bridge wall as shown.

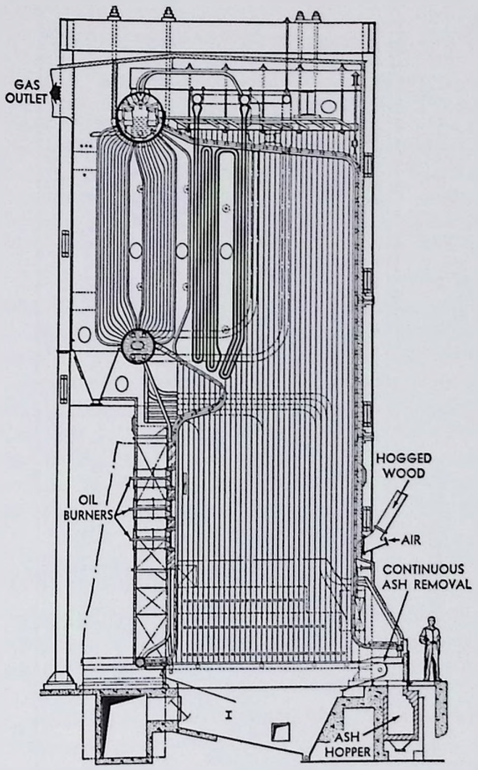


Fig. 13. *A Two-Drum Stirling boiler with water-cooled furnace designed to burn wood with either oil, gas, or pulverized coal*

Still another variation for a combination of wood and coal firing is that wherein the wood chute is located in the front wall directly above the coal flippers. The wood feed, consisting of unhogged refuse, passes through a rotating distributor that directs the flow sequentially through the chutes distributed across the furnace width. This permits the individual

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piles of wood partially to dry out before receiving the next feed fraction.

Cyclonic Burning. A furnace-boiler combination designed primarily for coal firing in a Cyclone Furnace is shown by Fig. 14. Provision has been made to burn some bark within the Cyclone Furnace in conjunction with coal. In order to accomplish this, however, it has been necessary to use double-screened hogged fuel for ease of handling and adequate control.

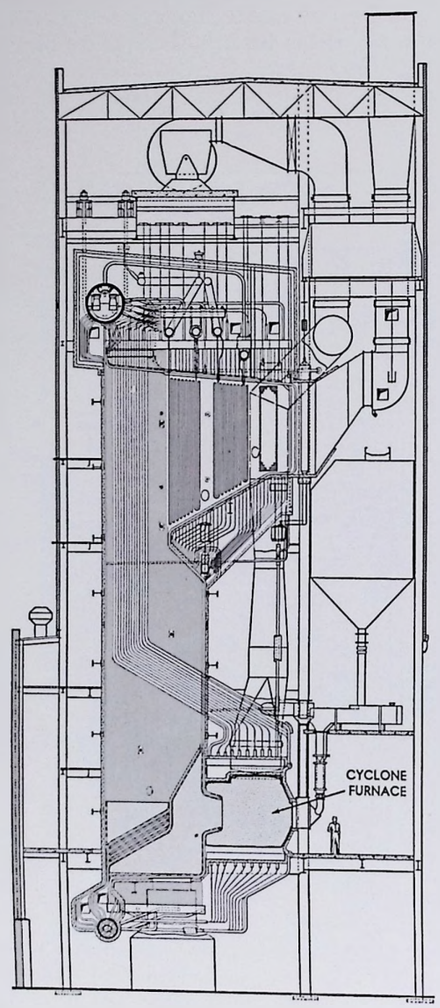


Fig. 14. *A Radiant boiler with Cyclone Furnaces adapted to burn hogged fuel with coal*

Coal Char

Coal char is a residue from the low-temperature (900-1000 F) carbonization of coal or lignite. The removal of gas and tar leaves a char which may be used as a fuel. Not until recently has there been any active interest in char as a boiler fuel.

Char varies in analysis, depending upon the source of the coal from which it is produced. In the carbonization process about half of the volatile matter is removed. This results in increased percentages of fixed carbon, ash, and sulfur, with a corresponding reduction in heating value of the char itself. The ash-softening temperature is quite close to that of the ash from the original coal, since the process does not change the nature of the ash. Grindability ranges from 41 to 150 (ASTM). A comparison of a typical char analysis with that of the original coal is shown in Table 1.

TABLE 1

Analyses of Coal and Coal Char

Fuel	Original Coal	Coal Char
Analyses (Dry Basis), % by wt		
Proximate		
Volatile matter	41.4	15.1
Fixed carbon	48.7	69.0
Ash	9.9	15.9
Sulfur	4.2	4.6
Heating value		
Btu/lb, dry	13,340	11,910
Btu/lb, moisture-and-ash free	14,810	14,160
Ash-fusion temperatures, F		
Reducing atmosphere		
Initial deformation	1960	1930
Softening	2100	2020
Fluid	2410	2500
Oxidizing atmosphere		
Initial deformation	2440	2360
Softening	2520	2490
Fluid	2670	2680

The char resulting from most of the present processes is of a sizing represented by the data of Table 2.

TABLE 2

Char Sizing (As-Received Basis)

Screen Size, No.	% Through
8	100
16	97.9
30	87.1
50	68.3
100	40.2
140	28.1
200	21.4

Apparently the volatile matter of *low* ignition temperature is driven off in the carbonization process, since char is more difficult to ignite than coal having the same percentage of volatile matter. Because of its sizing and high ignition temperature, most char is not well suited for burning on stokers. However, char which has been properly sized may be burned in a furnace resembling that used for anthracite. The two most successful methods for burning char are in suspension after pulverizing and in Cyclone Furnaces.

If char is pulverized to a fineness of at least 80% through 200 mesh, it can be burned in a dry-ash-removal furnace. The air for combustion must be preheated to 600 F. If downshot burners are used with the major part of the combustion air introduced through the front wall of the furnace, as is done for anthracite firing, no auxiliary fuel is required to maintain ignition. However, if burners are used where all the combustion air is introduced into the furnace through the burners, as in modern bituminous-coal-fired furnaces, auxiliary fuel in the form of oil, gas, or bituminous coal is required to stabilize ignition.

It is not necessary to have exceptionally fine raw fuel for Cyclone Furnace firing. Therefore many chars can be burned in these furnaces without pulverization or other conditioning. If the ash-softening temperature is above 2200 F (reducing), it may be necessary to include a flux to reduce the slag viscosity, or to fire oil, gas, or coal along with the char. Dust collectors are essential with both of these firing methods to eliminate objectionable stack discharge.

Waste Heat

Utilization of the heat in the exhaust gases from industrial process furnaces becomes more important as fuel costs increase. Boiler equipment, properly designed to absorb the heat in what was formerly waste gas, often will generate all the steam required to power the process. Also, where the waste gases carry some of the process material in suspension, suitable hoppers associated with the boiler equipment will collect a portion of the material, and the cooled gases leaving the boiler may be passed through precipitators for a major recovery of the remainder. Many types of boilers are necessary to meet the wide range of requirements in this field. Their design depends upon the chemical nature of the gases, their temperature, pressure, quantity, and cleanliness.

Heat Transfer from Waste Gases

The rate of heat transfer from the gas to the boiler water depends upon the temperature and specific heat of the gases, their velocity and direction of flow over the absorbing surfaces, and the surface cleanliness. To obtain the proper velocity of the gases over the surfaces, sufficient draft must be provided, either by a stack or a fan, to overcome the losses caused by the flow of gases through the unit, with adequate allowance for normal fouling of the heating surfaces. Temperatures of process gases are relatively low, as shown in Table 3; consequently the radiation component in heat transfer also is low. The tendency, therefore, is to design waste-heat boilers for much higher gas velocities than those for fuel-fired units, in order to compensate for these factors. However, high velocities with dust-laden gases should be avoided to prevent abrasion of tubes from sandblasting action. This is particularly critical where changes in direction of gas flow occur.

TABLE 3

Temperature of Waste-Heat Gases

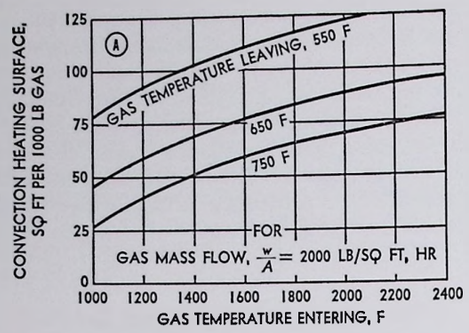
Source of Gas	Temp, F
Ammonia oxidation process	1350-1475
Annealing furnace	1100-2000
Cement kiln (dry process)	1150-1350
Cement kiln (wet process)	800-1100
Copper reverberatory furnace	2000-2500
Diesel engine exhaust	1000-1200
Forge and billet-heating furnaces	1700-2200
Garbage incinerator	1550-2000
Open-hearth steel furnace, air blown	1000-1300
Open-hearth steel furnace, oxygen blown	1300-2100
Basic oxygen furnace	3000-3500
Petroleum refinery	1000-1100
Sulfur ore processing	1600-1900
Zinc-fuming furnace	1800-2000

Diagrams A and B, Fig. 15, give an approximate measure of the convection heating surface required for usual conditions in waste-heat boiler practice. A water-cooled "vestibule" or furnace is a feature of many modern waste-heat boiler units, as shown by Figs. 20 and 21. The vestibule provides cooling of the gases to the temperature necessary for prevention of slagging before they enter the convection surface. The approximate amount of surface required for this purpose is given in Diagram C of Fig. 15.

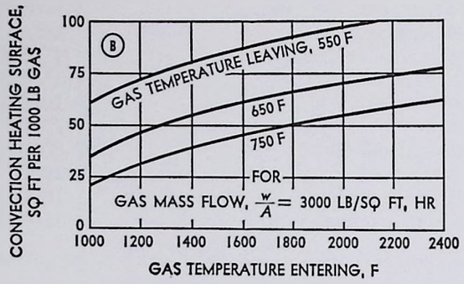
Types of Waste-Heat Boilers

The design of boiler for a particular requirement depends upon many factors, most of which vary not only from process to process but also within an industry. The cost of equipment, auxiliary power, and maintenance must be compared with the dollar value of the savings expected. The design of the boiler itself is subject, to a degree, to the cost of power at the plant. A small unit with closely spaced tubes will cost less but will require more fan power, because of high draft loss, than a larger, more expensive unit with lower draft loss. However, if power costs are low, the small inexpensive unit may be justified. Other factors to be considered are the available space, the location required for the proper flue connections, the corrosive nature of the gases, the effect of dust loading on erosion, and the process operation (pressure or suction).

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CONDITIONS FOR DIAGRAMS Ⓐ AND Ⓑ
TUBES, 2½ - IN. OD IN-LINE
TUBE SPACING, $l_\perp$ AND $l_\parallel$ = 5 IN.
SAT. TEMPERATURE, t'_s = 450 F
DRAFT LOSS, 0.2 IN. TO 0.4 IN. WATER



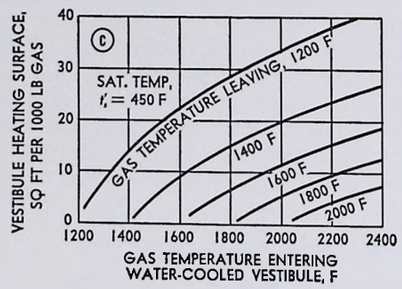


Fig. 15. *Approximate surface required in convection tube bank and vestibule for various entering and leaving waste-gas temperatures*

If the gases carry dust, attention must be given to tube spacing and provisions for the removal of dust dislodged from the heating surfaces. The tubes must be spaced reasonably close for good heat transfer and must still be far enough apart to prevent bridging of deposits. Quite often the boiler is arranged for wide tube spacing where the gases are hottest, with spacing reduced where the gases are cooler.

Sometimes the deposits that are carried into the boiler from the process can be removed by mechanical cleaners, whereas the nature of the deposits from other processes may call for periodic manual lancing to keep the boiler passes open. In either case, suitable hoppers should be provided to collect the removed deposits.

Gases from oil- or gas-fired process furnaces are relatively clean and, therefore, can be used in units with close tube spacing (one inch in the clear) without fear of bridging and plugging.

Relatively Clean Waste Gases

In the reclamation of heat from relatively clean gases leaving some industrial furnaces (such as billet-heating and air-blown open-hearth furnaces), where the dust loading does not exceed 0.5 grain per cu ft, the very compact design of gas-tube waste-heat boiler, illustrated in Fig. 16, is widely used. Since the boiler and flues are inherently tight, this unit is well suited for the use of gases under pressure or gases that should not be permitted to escape from the system except through the stack. The boiler has no refractory lining, and round flues may be used—an advantage with gases in pulsating flow, such as the exhaust from a Diesel engine when used as a waste gas to generate steam for heating or process. This boiler occupies but little space and, depending upon its shell diameter, may be used for steam pressures of 450 psi and entering gas temperatures as high as 2100 F.

In the circulating system of this boiler, water enters the steam-and-water drum and passes down through vertical circulating tubes to a header below the shell, then through the shell, and returns to the drum as a mixture of steam and water through the circulating tubes.

Feedwater enters the upper steam-and-water drum through a perforated pipe with the holes at the top. There it mixes with the boiler water, and dissolved gases, including traces of oxygen that may not have been eliminated by the treating system, are driven off with the steam before they come in contact with the heating surface.

This design of gas-tube boiler eliminates the leakage problems usually associated with boilers of this type for the following reasons:

1. Incoming feedwater mixes with the water in the steam-and-water drum and in the circulators to the main shell, raising its temperature close to saturation before it reaches the shell. Thus there is no temperature shock on the tube sheets or the shell.
2. The tubes themselves are the "staybolts," and, as they are all at saturation temperature, there are no expansion strains.
3. Circulation in the shell is positive, with separate downcomers and risers so located as to assure circulation throughout the shell.

The shell is completely filled with water at saturation temperature, and, since all tubes are submerged, they are at substantially the same temperature. Unequal expansion of metal parts is thus avoided, and this cause of troublesome leakage is removed. This is in contrast to designs having some tubes in the shell above the water level and some below. The difference in temperature between the submerged and the dry tubes will lead to leakage at the tube seats and at the joints between the tube sheets and the shell.

Gas flows through the tubes in a straight path. The tubes are kept clear of ordinary deposits by a rotating-arm soot blower, with nozzles directed toward the tube ends and tube sheet. For heavier incrustations, hand lances can be used through lance doors located in the vertical wall of the connecting flue.

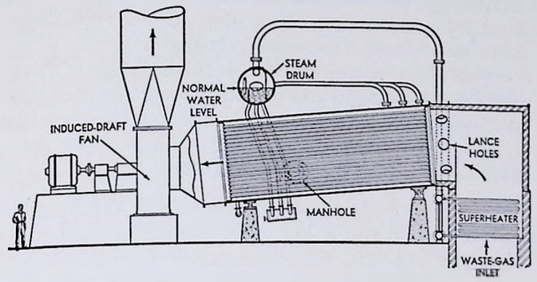


Fig. 16. *Gas-tube type of waste-heat boiler suitable for relatively clean waste gases*

If superheated steam is required, a superheater may be located in the flue to the boiler, as illustrated in Fig. 16. For this class of waste-heat-boiler service, with a maximum operating pressure of 450 psi and entering gas temperatures of 1000–2100 F, superheaters may be sized to provide steam temperatures as high as 750 F.

Waste-Heat Boilers for Open Hearths

Until recently, open-hearth operation used air for the oxidation and reduction necessary to convert iron to steel. To improve the open-hearth efficiency, this oxidizing air was preheated by alternately heating and cooling the checkers. To further improve the over-all efficiency of the open-hearth operation, waste-heat boilers comparable to the unit shown in Fig. 16 were used to recover the sensible heat of the gases for the generation of steam.

Oxygen blowing of the open hearth, in preference to air, greatly increases the production rate and materially reduces the cost of steel. Therefore, most open hearths of recent vintage are being changed to the oxygen-blown process.

While the new process of oxygen blowing greatly increases the steel-producing rate, it also increases the dust discharge. The use of oxygen requires less preheating of the combustion air which, in turn, reduces the cooling of the checkers. This results in a higher discharge gas temperature, from the previous 1300 F to a possible maximum of 2100 F. The combination of increased dust carry-over, higher gas temperature, and, in many cases, an increased total gas flow makes the gas-tube boiler unsuitable with respect to capacity and cleanability. Therefore, for most conversions it is desirable to replace the gas-tube boiler with a water-tube unit of higher capacity, designed to fit into the space previously occupied by the replaced unit. Furthermore, because of the increase in dust carry-over, the new boiler is usually arranged in conjunction with a new precipitator.

As a result of the increase in both gas weight and gas temperature, the steam output is materially increased. For many installations it is desirable to maintain the steam flow, during furnace-charging and -maintenance periods, by firing an auxiliary fuel, which requires a boiler furnace for its combustion. Thus the waste-heat boiler for oxygen-blown open hearths is a versatile unit that takes into account the space available, the amount of waste gas, steam capacity, cleanability, and supplementary-fuel firing. The steam capacity for a single unit serving a single open hearth may be as high as 150,000 lb/hr, Fig. 17.

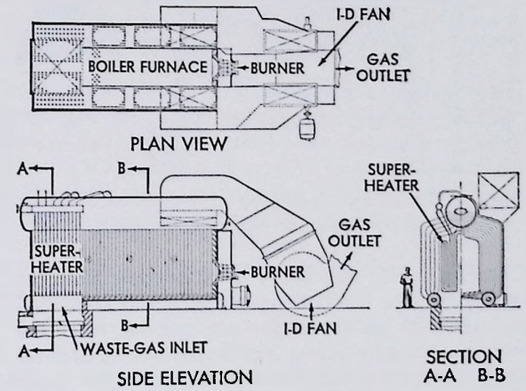


Fig. 17. *Waste-heat boiler for oxygen-blown open-hearth furnace*

Three-Drum Waste-Heat Boiler

A simple three-drum waste-heat boiler, designed to operate with dust-laden gases, is illustrated in Fig. 18. This type of waste-heat boiler is particularly well suited for use with waste gases of high solids content (up to 10 grains/cu ft) from cement kilns. Maximum precipitation of solids is assured by the horizontal flow of the gases through the vertical tube banks and by the effective arrangement of baffles. With this tube arrangement, hand lancing is possible from both sides. Every space in the full width of the boiler can be lanced. Hand lancing can also be done through the roof and above the lower drums, making all absorbing surfaces accessible with a short hand lance.

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With gases of high solids content it is frequently possible to reduce the amount of hand lancing by the use of long retractable soot blowers (see Chapter 20). These are located at one, or sometimes two, elevations along the depth of tube banks at gaps provided by the omission of a single row of tubes. Fixed-position rotary soot blowers are not very effective in the staggered tube arrangement of this unit.

To maintain optimum conditions of heat transfer without changing the direction of gas flow, the tubes in the rear sections of the boiler are closely spaced. Steel baffles and baffled sectional hoppers prevent the gases from by-passing the tubes by flowing below the tube bank. Since the baffles in both boiler and hoppers are made of smooth steel plates and are positioned vertically, no appreciable dust build-up is possible.

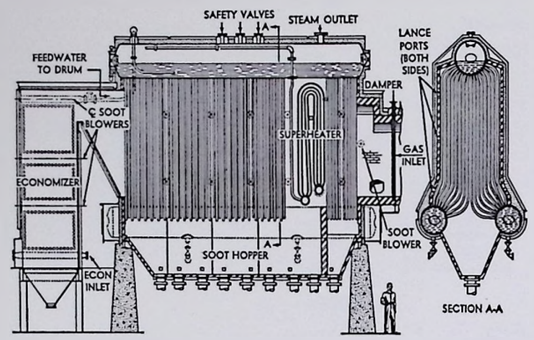


Fig. 18. *Three-drum bent-tube waste-heat boiler fitted with lance ports and soot blowers*

Circulation in this boiler is very simple, with the boiler tubes in the hot-gas end acting as risers and the tubes in the cooler-gas zones acting as downcomers or supply tubes. The boiler has a relatively long drum, in which steam separation takes place without the use of baffles. The steam is collected in a dry pipe located in the quiet cooler-gas end of the drum. Feedwater is thoroughly mixed with the boiler water rising into the steam-and-water drum at the hot-gas end.

Expansion and contraction of the drums and tubes have no effect on the steel casing, firebrick, or insulation; hence, the most common source of brickwork trouble is eliminated, and air infiltration is reduced to a minimum. All pressure parts rest upon supports below the lower drums.

The location of the superheater can be varied to secure the degree of superheat required. Further to increase heat absorption, an economizer can be installed in the flue leaving the boiler. The economizer is arranged for downward flow of gases to aid in the collection of solids.

Solids collected in the hoppers under the boiler and economizer can be readily removed while the boiler is in service. In a single boiler behind a cement kiln, from 20 to 40 tons of cement dust may be recovered each day from these hoppers.

Typical data and performance of the waste-heat boiler shown in Fig. 18 are given in Table 4.

TABLE 4

Performance of Three-Drum Unit (Fig. 18)

Waste Gas from Cement Kiln

Boiler heating surface, sq ft	12,000
Superheater surface, sq ft	523
Steam flow, lb/hr	43,000
Flue gas entering boiler, lb/hr	150,000
Gas temperature entering boiler, F	1,500
Gas temperature leaving boiler, F	438
Gas temperature leaving economizer, F	320
Steam pressure at superheater outlet, psi	200
Steam temperature at superheater outlet, F	480
Feedwater temperature entering economizer, F	212
Draft loss, boiler, superheater, and economizer, in. water	9.6

Waste-Heat Boiler for Ore Roaster

Another three-drum bent-tube boiler, arranged to reclaim heat in the waste gases from ore roasters, is shown in Fig. 19. Hoppers under the boiler gas passes are arranged to collect ore dust that is thrown out of suspension when the gases make the low-velocity turns around the gas baffles. Typical data and performance of the unit illustrated are given in Table 5.

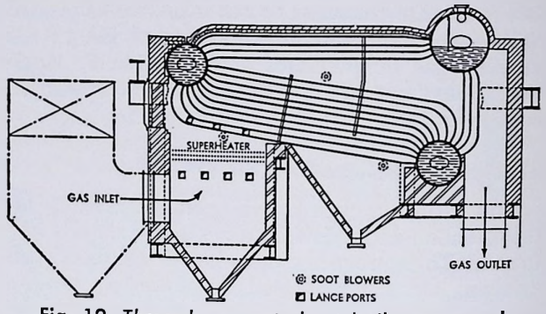


Fig. 19. *Three-drum waste-heat boiler arranged to use waste gases from ore roasters*

TABLE 5

Performance of Three-Drum Unit (Fig. 19)

Waste Gas from Ore Roasters

Boiler heating surface, sq ft	2,617
Superheater surface, sq ft	137
Steam flow, lb/hr	9,200
Flue gas entering boiler, lb/hr	52,000
Gas temperature entering boiler, F	1,435
Gas temperature leaving boiler, F	690
Steam pressure at superheater outlet, psi	450
Steam temperature at superheater outlet, F	720
Feedwater temperature entering boiler, F	340
Draft loss, boiler and superheater, in. water	2.0

High Solids—Relatively Low Temperature

Waste-heat boilers are also widely applied to some types of refining furnaces and to cement and dolomite kilns. The temperature of these gases rarely exceeds 1400 F, and the solids content is high—4 grains and over per cu ft. In many applications the boilers are quite large, so that they can serve as fuel-fired units when the process furnace is idle. The usual requirements for such units are maximum heating surface for a given floor area, vertically arranged heating surface to shed deposits readily, and a chamber ahead of the tube banks. For these applications this chamber serves as the combustion space for burning the direct-fired fuel. Two waste-heat boiler designs that admirably meet these requirements are illustrated by Figs. 20 and 21.

Smelter-Furnace Waste Gases

For smelter-furnace gases carrying solids in a semimolten or sticky form, the designs of waste-heat boilers illustrated by Figs. 20 and 21 are particularly suitable because of the cooling of the gas by radiation in the large water-cooled areas of the furnace ahead of the convection surfaces. The absence of baffles in the convection banks is a further advantage when carry-over dust is abrasive.

Zinc-Fuming-Furnace Gases

The reclamation of zinc from slag waste as a by-product of lead-smelting furnaces requires a fuming furnace, in which pulverized coal and air are injected into the molten slag. The zinc and lead are boiled out from the slag as a metal vapor and then reoxidized above the slag bed in an atmosphere free of slag and other impurities.

The products leaving the fuming furnace at a temperature of about 2200 F must be passed through a combustion chamber, where the zinc vapor is completely oxidized and then cooled to permit recovery of the zinc oxide. If this cooling takes place in a waste-heat boiler, a considerable portion of the heat in the gases can be recovered in steam for generating power. A waste-heat boiler unit for this service should be designed to reduce the temperature of the gases and entrained solids before the gases enter boiler and superheater convection surfaces, so that the solids can be readily removed as dust from these heat-absorbing surfaces.

Water-Cooled Furnace. A water-cooled-furnace waste-heat boiler unit of this type, designed to operate with gases from a fuming furnace, is illustrated by Fig. 20. Heat absorption by the water-cooled furnace walls and the baffle wall in the furnace reduces the temperature of the gases entering the superheater from approximately 2200 to 1400 F. Good separation of dust from the gases is obtained in the furnace by the 180-degree turn ahead of the

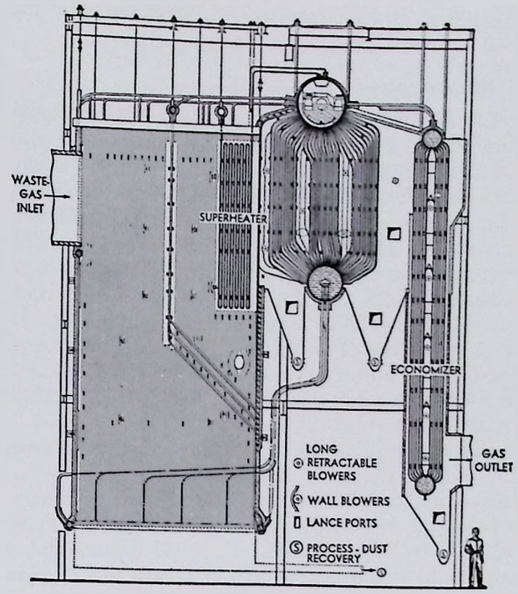


Fig. 20. *Waste-heat boiler unit suitable for high-temperature waste gases carrying sticky or semimolten particles*

superheater. The dust, thrown out of suspension, drops into the water-cooled hopper bottom of the furnace and then into a dust-collecting hopper, from which it is removed by a screw conveyor. Some dust accumulates on the walls of the furnace, from which it can be removed periodically by rotating blowers, using air or steam (see Chapter 20). These are distributed over the furnace wall areas, as shown in Fig. 20.

Boiler Heating Surface. Boiler heating surface, arranged between two drums, provides a single gas path over the surface with no baffles required. To reduce the gas temperature to desired limits, an economizer is included. The permissible final gas temperature will depend upon the sulfur content of the gases. The gas temperature should be maintained above the dew point to avoid corrosion of the flues and the dust-separating equipment between the boiler and the stack. Hoppers are arranged under the boiler, the economizer, and the long flue to the stack to collect the valuable zinc dust.

Automatic soot blowers of the long retractable type are fitted to remove the dust that accumulates on the tube banks of the boiler, superheater, and economizer. This type of soot blower has generally replaced other types for this service. By concentrating the operation through two or three nozzles, the soot blowing is more effective. Since the blower remains outside the setting except when in actual use, maintenance is less, and there is no additional surface to which dust can cling. Lance ports of the type

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illustrated in Chapter 20 are distributed over the wall areas for hand lancing when necessary.

Reverberatory-Furnace Gases

Copper ore concentrates are melted in reverberatory furnaces fired by pulverized coal, oil, or gas. The impurities float to the top of the melt and are skimmed off as slag. The copper, tapped in molten form, is transported to other furnaces for further refinement. The products of combustion leave the reverberatory furnace at about 2300 F, heavily contaminated with slag particles and solids, a portion of which may be in a pasty or semifused state. As large quantities of steam are required for the power used in the process, waste-heat boilers are installed to recover the heat in the products of combustion from the reverberatory furnaces.

In the operation of a copper reverberatory furnace, the draft at the outlet of the furnace must be maintained at about 0.06 in. of water, which is practically a balanced draft. The smelter operation is adversely affected by higher drafts. To avoid uncomfortable conditions in the boiler room, the stack effect within the setting must be reduced to a minimum. This is accomplished by so positioning the waste-heat boiler that the boiler passes and boiler gas outlet are about level with, or even above, the outlet of the reverberatory furnace (see Fig. 21). With such an arrangement there is no plus pressure to force sulfur-laden gases out through doors, around soot blowers, or through setting cracks in the walls.

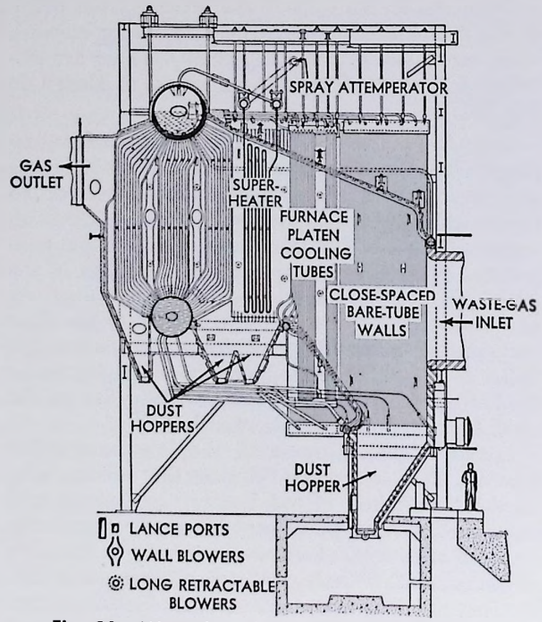


Fig. 21. *Waste-heat boiler unit specifically designed for the use of waste gases from copper reverberatory furnaces*

Water-Cooled Furnace. A waste-heat boiler unit with water-cooled furnace, designed for use with a copper reverberatory furnace, is illustrated by Fig. 21. The gases leaving the reverberatory furnace are highly contaminated with sticky slag particles, which will adhere tenaciously to boiler and superheater tubes. To cool these slag particles, a large water-cooled furnace area and spaced platens of heat-absorbing tubes are provided in the furnace.

Slag cannot bridge from one platen to the next, when these are spaced at 18-in. centers. Long retractable soot blowers are fitted at two elevations in the gap between the two banks of platens. Slag is removed periodically from the heat-absorbing surfaces by jets of steam or air from opposed nozzles near the ends of these soot blowers. Similar blowers are fitted, as indicated, for cleaning the tube banks of the superheater and the boiler. Wall blowers are provided for cleaning the furnace walls.

The heat absorption of the water-cooled furnace walls and the water-cooled platens reduces the temperature of the gases from 2300 F to approximately 1600 F before they enter the superheater. The "sticky" slag particles suspended in the gases are thus cooled to a dry ash that will not bridge the gap between the superheater tubes spaced on 6-in. centers. Boiler convection surface following the superheater has a tube spacing of 4 in. on centers. The deposits that adhere to superheater and boiler convection tubes can be removed periodically by the soot blowers, arranged as shown. Experience indicates that no additional hand lancing is required.

Dust and Slag Particles Reclaimed

The floor of the furnace is arranged as a hopper bottom to facilitate the collection of dust and slag particles. The boiler and the long flue to the stack are also equipped with hoppers, all for the purpose of collecting the copper-bearing particles of dust and ash, which are returned to the reverberatory furnace for reclamation of the copper.

Typical data and performance of this type of unit (Fig. 21) are given in Table 6.

TABLE 6

Performance of Two-Drum Unit (Fig. 21)

Waste Gas from Copper Reverberatory Furnace

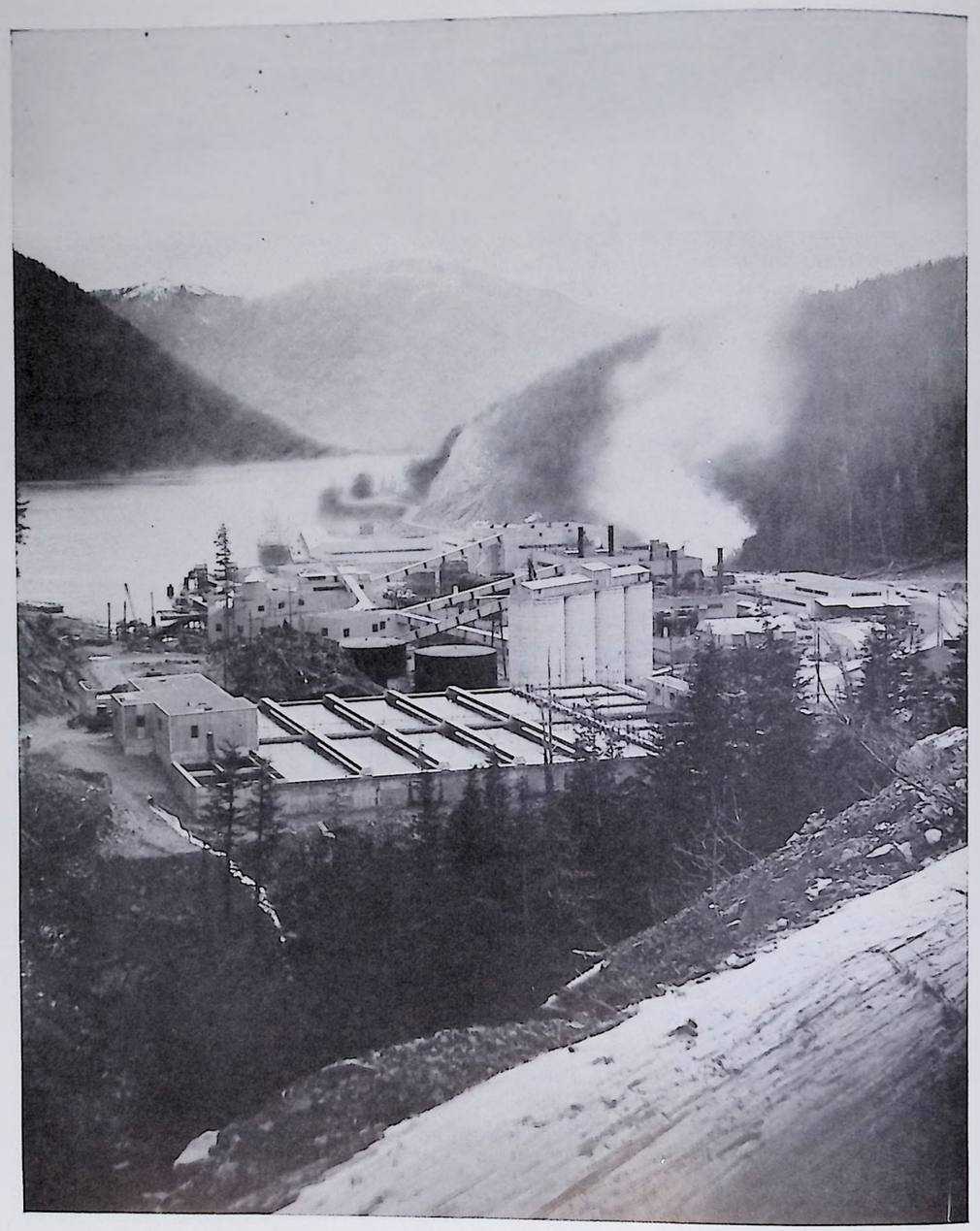
Boiler heating surface, sq ft	12,085
Superheater surface, sq ft	2,170
Steam flow, lb/hr	71,000
Flue gas entering boiler, lb/hr	166,000
Gas temperature leaving reverberatory furnace, F	2,269
Gas temperature entering superheater, F	1,687
Gas temperature leaving boiler, F	770
Steam pressure at superheater outlet, psi	475
Steam temperature at superheater outlet, F	775
Feedwater temperature entering boiler, F	350
Draft (static) at furnace bottom, in. water	0.1
Draft loss, boiler and superheater, in. water	0.2

It will be noted that the boiler surface is arranged for a single pass of gases. The draft loss of 0.2 in. water is exceptionally low and reflects the low gas velocity over the heating surface, which is desirable to avoid tube erosion. The high sulfur content of these gases makes the addition of an economizer to these boilers inadvisable, since lowering the gas temperature would cause corrosion in the long flues and dust separators between the boilers and the stack.

Waste-Heat Boilers for Special Conditions

Other types of waste-heat boilers to recover the heat from waste gases or fluids from industrial heating processes have been designed for special conditions of space, temperature, pressure, and draft. With all of the limitations known, waste-heat-recovery boilers can be designed to accommodate almost any set of conditions.

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CHEMICAL-RECOVERY UNITS POWER THIS ALASKAN PULP MILL

20

CHEMICAL AND HEAT RECOVERY IN THE PAPER INDUSTRY

THE pulp and paper industry is of far more than passing interest to the engineer involved in the generation of steam and power. Every type of pulp or paper produced requires large quantities of steam for drying of the sheet on the paper machine. In chemical pulping, steam is used for heating in the cooking process, during which the lignin is abstracted from the wood, leaving the cellulose pulp. Steam and its companion, electric power, are two of the basic "raw materials" necessary for the manufacture of pulp and paper. In 1960 the pulp industry in the United States alone used more than 30 billion kwhr of electricity, of which 18 billion kwhr was generated internally. Between 1954 and 1961 B&W installed steam-generating capacity in excess of 22,000,000 lb per hr in the paper and allied industries.

The past and predicted production of this industry and the annual per capita consumption of paper and paperboard products in the U.S. is illustrated by Fig. 1.

The following tabulation, compiled by *Pulp and Paper* and *Pulp and Paper International*, shows the per capita use in pounds of paper products in various countries of the world for the year 1961:

United States	439.5
Canada	280
Sweden	270
United Kingdom	235
Netherlands	228
Denmark	207
Switzerland	205
Australia	195
Norway	192
New Zealand	188
West Germany	179.5
Finland	172
Belgium	147.7

A large share of the steam and power consumed by this industry is generated by the combustion of waste fuels. The collection, combustion, and recovery of chemicals and heat from these waste fuels present many interesting and unique problems for the engineer.

It was not until the middle of the nineteenth century that wood was used as a raw material in making paper by the major basic pulping processes. These processes are: groundwood, alkaline pulping, and acid pulping. The first groundwood mill and the first chemical wood-pulp mills for both the alkaline and the acid processes were started in the period from 1852 to 1874.

In the past 100 years phenomenal progress has been made. The range of wood species used as raw material has required many technical modifications to the several processes. Refinements have been made in the techniques of forestry, logging, barking and chipping the logs, digesting and washing the pulp, supplementary purification of the pulp for chemical use, bleaching, and loading.

The virtual existence of modern man is dependent upon pulp and paper products. These products are necessary in business and in the home for magazines, books, newspapers, records, contracts, checks, money, stocks, reports, correspondence, advertising, packaging, drying, cleaning, sanitary use, insulation, and

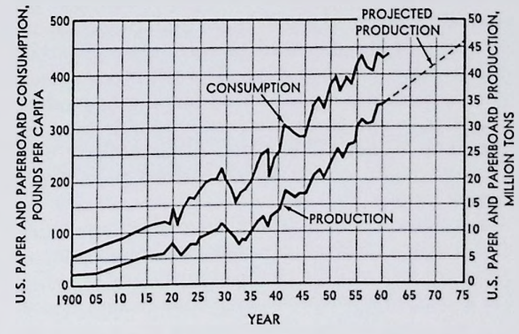


Fig. 1. *Production and consumption of paper and paperboard products in the United States*

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construction materials. In the production of such products as rayon, explosives, lacquers, and photographic film, wood pulp in a refined form is the basic raw material.

In 1961 about 40.5 million tons of pulp, paper, and paperboard were consumed in the United States. Of the 35.5 million tons of paper and paperboard products, 26 million tons were produced by the wood-pulp industry, the remainder coming from reclaimed waste paper and pulp imports. For this production, steam requirements varied within wide limits, from a minimum of 5,000 lb per ton to a maximum of perhaps 20,000 lb per ton, depending upon the grade and degree of bleaching and converting done at the mill.

Boilers to supply the heavy steam demands of this industry, when fired by the usual primary fuels (coal, oil, gas, wood refuse, or bark), are of conventional design, as covered in other chapters. Some departure from the conventional, involving special design of the furnace, is required to burn successfullly sand-impregnated or salt-water-borne bark and hogged wood. A large proportion of the steam requirements for the pulp mills, however, is produced in highly specialized chemical- and heat-recovery units.

Electric power requirements in the making of pulp and paper have risen faster than process steam requirements. This factor coupled with steadily rising fuel costs has led to higher pressures and temperatures and high efficiencies in boilers for pulp and paper mills. The increased value of steam has led to a demand for more reliable and efficient heat- and chemical-recovery units. Whereas steam was formerly regarded as a by-product of chemical recovery, today the heat value of liquor solids is a reliable source of steam for power generation and process.

Alkaline Process

The two types of alkaline pulping are the soda and the kraft processes. In the soda process the nonfibrous constituents of the wood are dissolved by digestion in a liquor that is essentially sodium hydroxide (NaOH), at a pressure of approximately 125 psi and a temperature of 350 F. In the kraft process about 20 to 30% of the chemical present in the digesting liquor is sodium sulfide, Na_2S. The sulfur compounds combine with the lignin during pulping to produce thiolignins, which are more readily soluble than the usual soda lignins. This allows faster cooking with less degradation of the fibers, thereby providing a stronger pulp.

Since the soda and kraft processes are quite similar, the discussion of alkaline pulping and application of a recovery unit will be confined to the kraft process, with subsequent comments (see *Soda Process Recovery Unit*, page 20-14) on differences in detail for the soda process.

Kraft Process Recovery Unit

The process flow diagram (Fig. 2) will serve to show the relationship and importance of the recovery unit in the chemical and heat balances of a typical pulp and paper mill. Following the flow diagram: the logs entering the mill are first debarked (1), reduced to chips (2), and then charged into a digester (3), where they are cooked under pressure in a steam-heated aqueous digestion solution of sodium hydroxide and sodium sulfide, known as "white liquor" or "cooking liquor." In the cooking operation the lignin binder, which holds together the cellulose fibers of the wood, is dissolved.

After cooking, the cellulose fibers, now called "pulp" or "brown stock," are separated from the spent cooking liquor in the pulp washers (4). The pulp then goes through several refining processes and finally to the paper machine.

The spent cooking liquor containing the lignin dissolved from the wood is called "black liquor." As the dilute or "weak" black liquor (about 15% dissolved solids) comes from the washers, it is first concentrated in multiple-effect evaporators (5) by the use of steam. The concentrated or "strong" black liquor then passes to the recovery unit, where it is further concentrated to "heavy" black liquor in a direct-contact evaporator (6) using the sensible heat of the flue gas.

The concentrated black liquor then goes to the mixing tank (7), where salt cake (sodium sulfate) is mixed with the liquor to make up the chemical losses in the system. Salt cake recovered from the boiler hoppers and from the dust or fume collector following the direct-contact evaporator is also returned to the liquor cycle at this stage, either by conveying the salt cake to the mixing tank or by pumping the black liquor to the collection hoppers. The "heavy" black liquor, with its salt cake burden, is heated to lower its viscosity and is then pumped to the recovery furnace (8). In the recovery unit, the heavy black liquor is sprayed upon the furnace walls for dehydration prior to final combustion of the dried char on the furnace hearth.

Chemical and Heat Recovery

In the recovery furnace, heat is obtained from the combustion of the organic liquor constituents (dissolved from the wood), and, equally important, the inorganic constituents (sodium salts) in the liquor are recovered as molten ash or smelt. Another essential function of the recovery is the reduction of the sodium sulfate content of the black liquor to sodium sulfide.

The smelt, composed largely of sodium carbonate, Na_2CO_3, and sodium sulfide, Na_2S, is tapped from

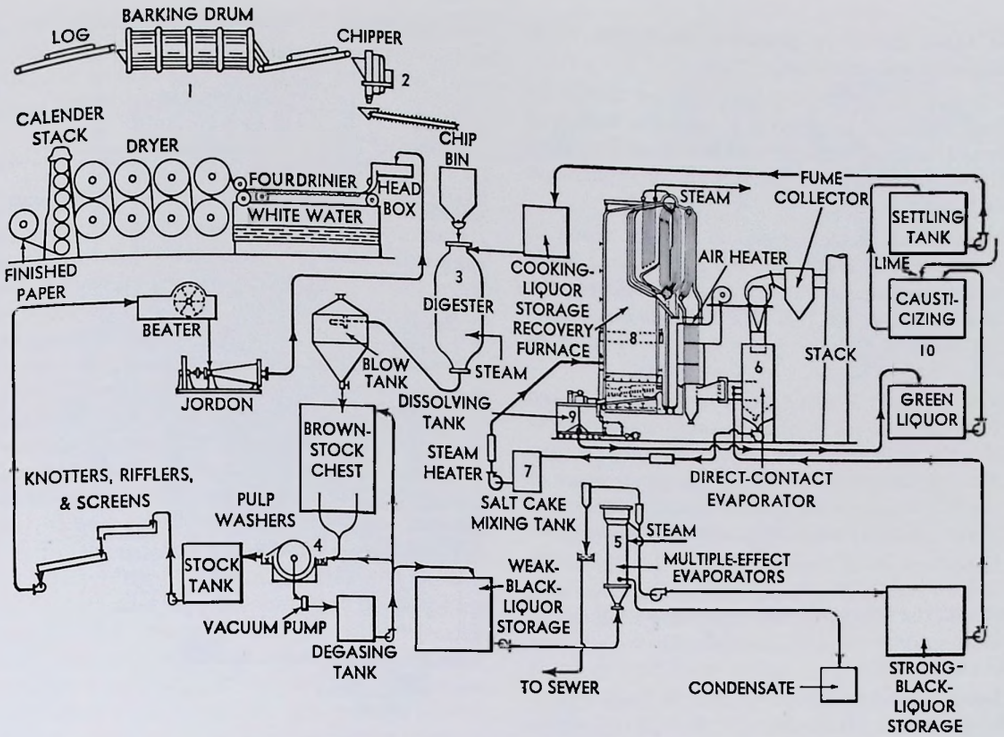


Fig. 2. *Process flow diagram of a typical pulp and paper mill*

the furnace and dissolved in water in the dissolving tank (9) to form "green liquor." The green liquor is then subjected to a causticizing treatment (10) with lime, $Ca(OH)_2$, to convert the sodium carbonate to sodium hydroxide. In this step, the sodium sulfide remains unchanged. The liquor, now known as "white liquor," is then ready for re-use as cooking liquor in the digester. The calcium carbonate sludge from the white liquor in the causticizing operation can be reburned in a lime kiln, with loss of CO_2, to yield calcium oxide, which is re-used in causticizing green liquor to white liquor.

Reclaiming and Concentrating Black Liquor

The maximum quantity of waste liquor solids from the digesters, washers, multiple-effect evaporators, and finally from the direct-contact evaporator should be in solution at optimum concentration at each stage of the process, and as much of the heat from the combustion of the liquor should be utilized as is economically practical. Under normal conditions, the "strong" black liquor comes from the multiple-effect evaporators at a solids concentration of 40 to 50%. It is then ready for further concentration in the direct-contact evaporator. The concentration to "heavy liquor" in the direct-contact evaporator, using hot flue gas, is limited to 67% approximately because of difficulty in pumping a too viscous fluid, and over-concentration must accordingly be avoided.

Salient Features of a Kraft Recovery Unit

The functions, objectives, and requirements of the kraft recovery unit and auxiliaries may be listed as follows:

1. The black liquor is dehydrated, and the organic constituents are burned.
2. The ash (inorganic constituents) from the incineration of the char is subjected to active reducing conditions to convert its sodium sulfate and other sodium-sulfur-oxygen compounds to sodium sulfide.
3. The ash is recovered in molten form with as little contamination as possible.
4. The furnace floor must be tight against leakage of the molten smelt. The molten sodium compounds are dangerous to handle.
5. The furnace floor must provide positive drainage of smelt to the smelt spout. Under certain conditions and proportions, smelt and water react violently and with explosive force; hence, no pools

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of smelt should be present to react with water inadvertently admitted.

6. Furnace and char-bed temperatures should be kept as low as practicable to minimize fouling of heat-absorbing surfaces and loss of sodium from the furnace through "boiling off" or sublimation of sodium salts.
7. Furnace gas velocities should be kept as low as practical (500 to 900 fpm is a good range) in the char-bed and liquor-spray zones to minimize the loss of sodium and fuel value through mechanical entrainment and the attendant fouling of heat-absorbing surfaces.
8. Water-cooled furnace walls and widely spaced slag-screen tubes must have adequate surface to absorb sufficient heat from the furnace gas, through radiation, to reduce the temperature of the entrained ash, so that it will not be "sticky" or "tacky" as it enters closely spaced tube sections.
9. Laborious hand lancing should be eliminated, first, by minimizing the amount of carry-over and, second, by arranging the boiler-unit heating surfaces so that automatic soot blowers are effective in cleaning.
10. The maximum possible quantity of steam should be generated for process use. (Usual practice is to generate the steam in the boiler unit at a pressure and temperature sufficiently above the process pressure that the electrical requirements of the mill may be produced as by-product power in back-pressure turbines.)
11. The black liquor from the multiple-effect evaporators is concentrated, by using the heat in flue gas in a direct-contact evaporator, to the values desired at the furnace spray nozzle.
12. The flue gas temperature from a unit which has a cascade or cyclone evaporator is limited to a minimum of 300 F to prevent corrosion in the electrostatic precipitator. A unit equipped with a venturi scrubber requires no precipitator and permits an exit gas temperature of 180 F.
13. Ease of control and operation must be provided.
14. Reliability and continuity of service of the recovery unit and its auxiliaries are essential.

The modern recovery unit, illustrated by Fig. 3, is designed to provide these features. The important components of this design are discussed below.

Characteristics of a B&W Recovery Unit

Spray Nozzle and Oscillator

As previously stated, the black liquor is sprayed onto the walls of the furnace, where it is dehydrated

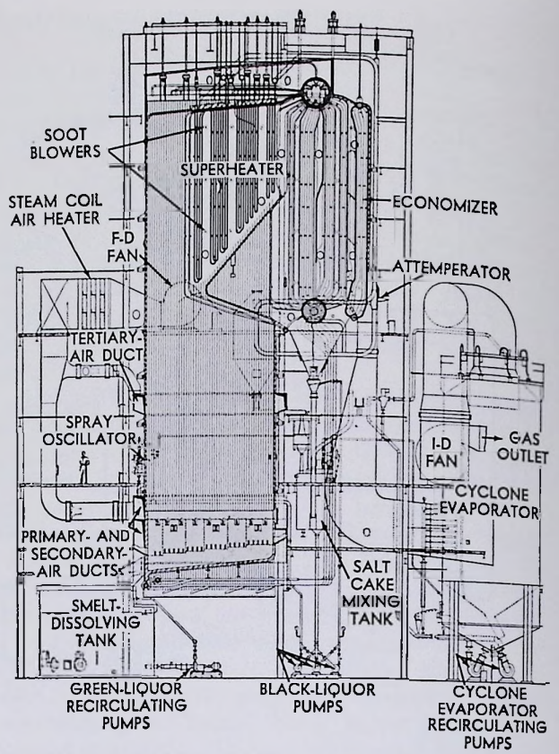


Fig. 3. *Modern boiler unit for chemical and heat recovery from black liquor*

and then falls to the char bed below. The spray nozzle, Fig. 4, located in the center of the front wall, emits a flat or sheet spray and is continuously both rotated and oscillated to cover a wide band of the side and rear walls. The degree of oscillation is adjustable to cover a greater or smaller area of wall surface, as required to compensate for variation in the

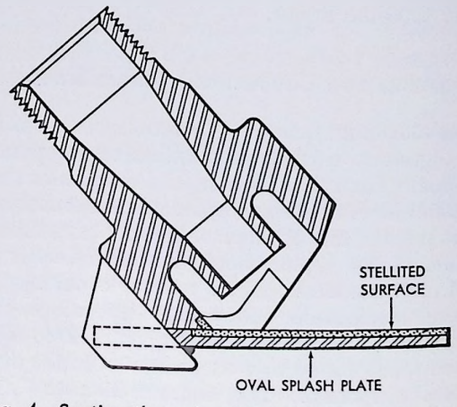


Fig. 4. *Section through a black-liquor spray nozzle*

solids concentration of the atomized liquor and, consequently, the time needed for dehydration. The spray-nozzle actuating mechanism is driven by a fractional horsepower motor.

Floor and Wall Construction of Furnace

The lower half of the furnace, Fig. 3, is actually a chemical retort and is operated as such. Incomplete combustion of the char in the porous bed supplies carbon monoxide and incandescent carbon, which act as reducing agents to convert the sulfate in the smelt to sulfide. All of the smelt filters through the char bed to the furnace floor or hearth, from which it is tapped. To withstand the erosive and penetrating characteristics of the smelt, the furnace walls and the floor must be so constructed as to assure tightness.

For the floor construction, illustrated by Fig. 5, plastic-chrome-ore refractory is packed solidly around studded water-cooled tubes. Its thickness over the tubes is no greater than that which can be effectively cooled. Positive cooling of the abutting flat studs at the center line of the tubes provides the necessary medium to chill the smelt and prevent penetration beyond that point. As further assurance against leakage, a plate membrane is seal welded to the tubes behind the flat studs to provide a watertight barrier.

Refractory-faced water-cooled stud-tube construction is used for all four walls of the furnace in the retort zone. Furnace walls above this zone are bare tubes with the space between tubes closed by flat studs. This construction, shown by Fig. 6, presents to the furnace a fully water-cooled metallic surface. It is upon this surface, at approximately the center third of the furnace height, that the liquor from the oscillating spray nozzle is deposited.

An outstanding feature of the B&W recovery unit is the absence of expansion joints to compensate for expansion of the furnace. This construction contributes to the reduction of air infiltration and thus helps

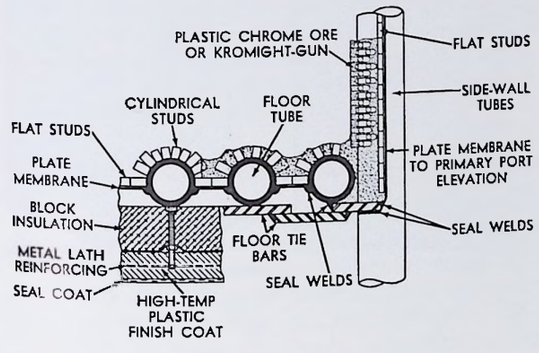


Fig. 5. *Section through recovery-furnace floor and side wall*

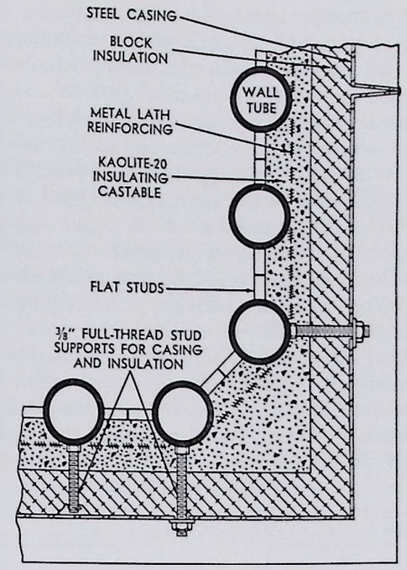


Fig. 6. *Furnace wall construction above retort zone*

to obtain maximum steam generation. The modern recovery unit (Fig. 3), including furnace, superheater, boiler, and economizer, is hung from overhead steel. The wall tubes of the furnace, boiler, and economizer support the monolithic refractory, the block insulation, and the steel casing, so that these parts move with the tubes as they expand, thus eliminating the need for special expansion seals. Furnace buckstays are attached directly to, and expand with, the tubes. Buckstays fitted completely around the furnace and boiler at several elevations are joined together at the corners to provide structural stability. The complete assembly of wall tubes, refractory, insulation, buckstays, and casing is free to expand and is, therefore, quite independent of the supporting steel.

Combustion Air to Furnace

Combustion air is admitted to the furnace through three sets of air ports, designated from the hearth or bottom of the furnace upward (see Fig. 3) as primary, secondary, and tertiary. The primary-air ports are located at an elevation of 36 in. above the hearth and extend around the four walls of the furnace. The secondary-air ports are just over the char bed. Tertiary-air ports are located in two opposite walls just above the upper limit of the spray zone.

The primary-air ports, located in every second space between furnace wall tubes, are proportioned to admit about 60% of the total air required, at a low static pressure for low velocities and minimum penetration of the bed. Dampers are arranged to control groups of three to five individual primary-air ports.

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Approximately one half of the remaining air required is admitted at each of the secondary- and tertiary-air zones. Ports in each zone are individually dampered to permit control of air pressure. The high pressure on large secondary ports enables complete penetration across the furnace and assures mixing of the air with the volatile gases arising from the char bed. Combustion at the secondary-air level creates a high-temperature zone below the liquor spray and aids in drying the liquor in flight from the spray nozzle to the furnace walls. Secondary air also effectively sizes the char bed by providing heat and turbulence across the top of the bed.

Further turbulence is created by the admission of the tertiary air to assure complete combustion of any unburned gases rising from the secondary zone and any volatiles driven from the sprayed liquor during its flight across the furnace. Tertiary-air mass penetration also provides a uniform furnace exit-gas temperature.

Combustion-Air Control

The control of combustion air is essentially manual, with some assistance from monitoring instruments. The operator establishes the liquor flow to the recovery unit to be consistent with pulping-process requirements. Total-air flow is selected in proportion to liquor flow, and the ratio of primary-air to total-air flow is established to provide stable furnace-bed conditions. Such a control system is shown in Fig. 7.

Recently, to insure the maintenance of optimum furnace firing conditions, a number of installations have been made in which the total-air requirement is determined by liquor flow, with fine adjustment provided by continuous analysis of combustion products.

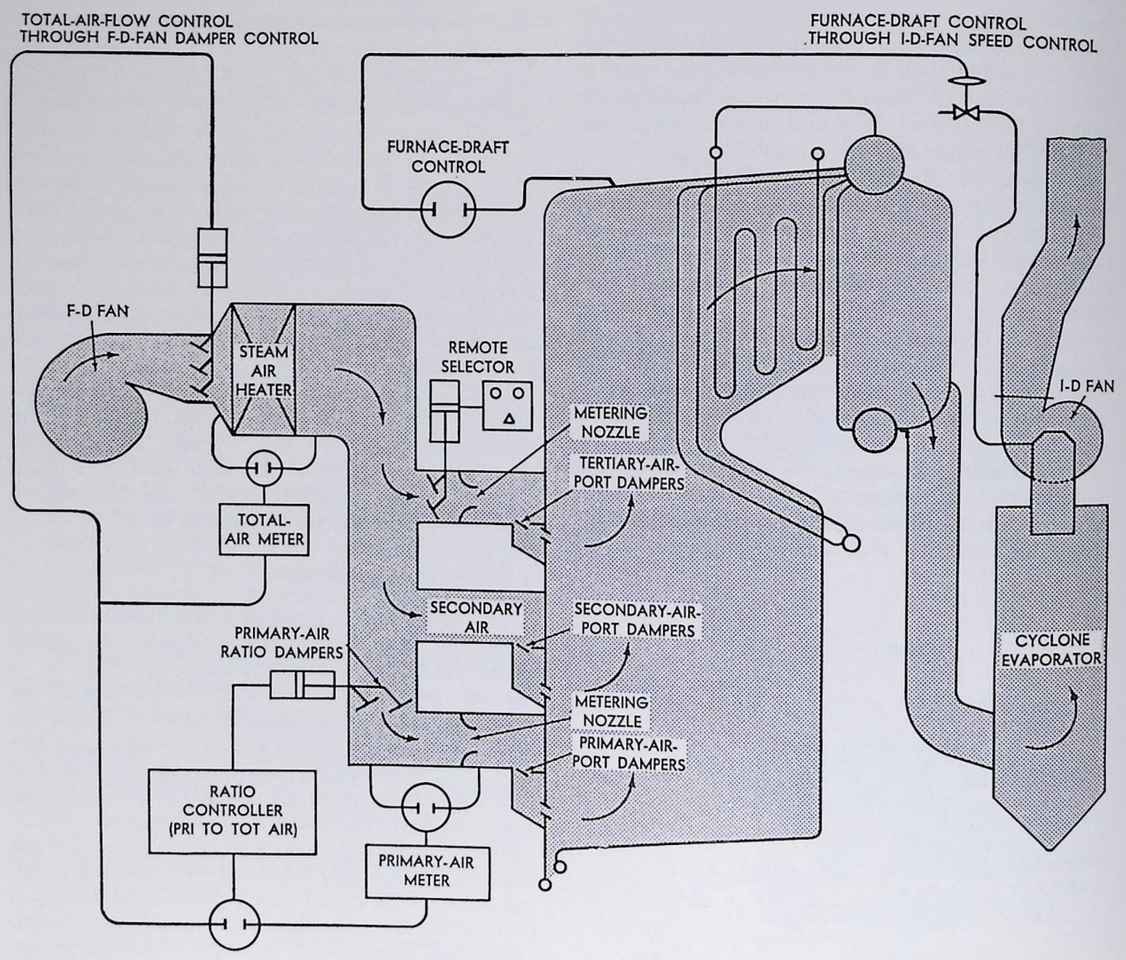


Fig. 7. *Schematic arrangement of combustion-air control system*

The oxygen and combustibles content of flue gas at the boiler outlet is an accurate measure of combustion efficiency. Using such measurements for automatic adjustment of air flow to the unit enables the operator to gain maximum benefit from the combustion of waste liquors.

Control of Gas Temperature

The arrangement of upper furnace and superheater, shown in Fig. 3, is most important in the performance of a recovery unit. Combustion is completed in the tertiary-air zone, as a result of turbulence and air admitted at this point. Above this zone there is adequate gas travel in the water-cooled furnace to cool the gas to a temperature that will permit satisfactory removal of ash and fume from convection surfaces by soot blowers.

The nose baffle shields most of the superheater from the radiant heat of the active burning zone of the furnace and causes the gas to enter the screen and superheater at a uniform temperature and velocity. An eddy above and behind the tip of the nose causes the gas to recirculate within the superheater tube bank and thus provides cooling for the portion of the superheater exposed to the radiant heat of the furnace.

The superheater is arranged for parallel flow of gas and steam. Saturated steam enters the front tubes of the superheater in contact with the hot gas and flows through successive loops, so that the final tube with the hottest steam is in contact with cooler gas. There is a dual advantage with this arrangement: first, cooling of the gas is most rapid at the front of the superheater, where the need for cooling of the ash is greatest; second, the parallel-flow arrangement results in a lower average tube-metal temperature. This is particularly desirable in recovery units designed for steam temperatures above 700 F. Because of the highly corrosive nature of the smelt, metal temperatures of carbon-steel tubes in contact with fluid smelt should not exceed 850 F. In zones where there is no fluid smelt, the metal temperatures may be higher. Careful coordination of superheater metal temperatures, flue gas temperatures, and alloys for superheater tubes permits final steam temperatures up to 900 F.

Maximum Gas Temperatures

Design requirements of the B&W recovery unit are based on a maximum gas temperature of 1450 F (MHVT basis) entering the more closely spaced tubes. As with other slag-forming fuels, an important objective is to reduce the temperature of the ash particles in suspension below their sticky or tacky temperature before they enter a closely spaced tube bank. This is accomplished by providing a combination of radiant-heat-absorption surface in the open furnace and convection surface in the widely spaced screen and superheater tubes.

Superheater Characteristics

For a nominal total steam temperature of 700 to 750 F, depending upon pressure and feedwater temperature, a six-loop superheater with forward and intermediate cavities for soot blowers is installed following the boiler screen. The superheater is larger than that required with other fuels because of the low temperature of the entering gas, the need for rather wide spacing, and the parallel-flow arrangement. In many units wide bends are used for the first, second, and even third loops of the superheater, and the adjacent elements are staggered, simulating screen construction, to reduce to 1450 F the temperature of the gas entering the remaining more closely spaced tubes. For higher steam temperatures, additional loops are added to the wide-spaced section and the number of screen rows is reduced. The screen and superheater surface arrangements are such that they can be effectively cleaned by type IK soot blowers (see Figs. 3, 8, and 9).

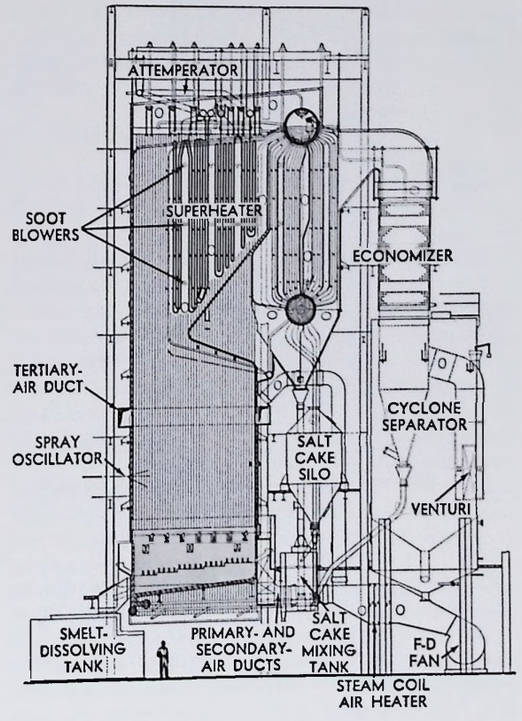


Fig. 8. *Black-liquor recovery unit arranged with economizer and venturi evaporator-scrubber*

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Recognition of the need for careful consideration of gas temperatures at various points in the screen and superheater has led to use of the nose-baffle construction and tertiary air to provide *uniform* temperatures and velocities across the entire screen entrance.

Test Traverse Results. The results of a study of actual temperatures and velocities ahead of the superheater in a 275-ton per day recovery unit, similar to that in Fig. 9, are shown in Fig. 10. During this test the unit was operated at a rating of 273 tons per day, firing liquor at a concentration of 67% solids at the furnace nozzle. The values shown are the averages determined with multiple-shield high-velocity thermocouples and pitot tubes at several points across the width of the unit and at several elevations. For the traverse shown, the average MHVT temperature entering the superheater is 1465 F (note recirculation indicated by lowest arrow). Other traverses for different ratings and liquor-firing concentrations have shown similar temperature and velocity patterns.

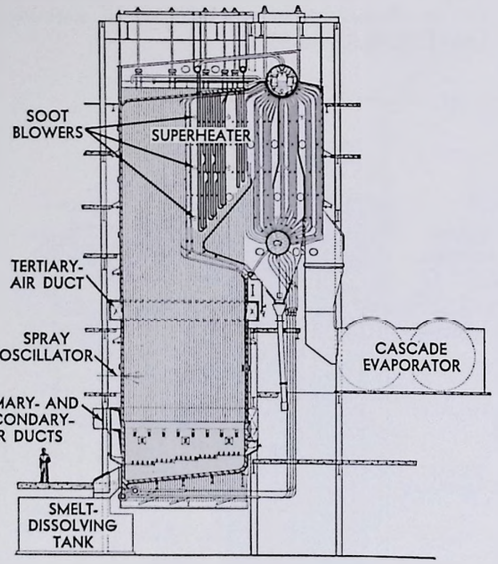


Fig. 9. *Black-liquor recovery unit arranged with cascade evaporator*

In addition to creating a favorable pattern of gas flow, the nose baffle shields the bottom of the superheater loops from the radiant heat of the furnace. This shielding reduces localized heat input to the tubes, resultant tube-metal temperatures, and slag deposition. Shielding is of special importance at the outlet loops of the superheater, where the steam temperature is highest.

Temperature Leaving Superheater. The temperature of the gas leaving the superheater has been reduced to 1100-1300 F (MHVT values), which is well below the temperature at which ash particles in the gas are fluid. Deposits formed below these temperatures are readily blown from the tubes and may be satisfactorily collected in hoppers beneath the boiler for return to the chemical cycle.

When Superheat is Not Required for Power. Where the generation of electric power is not required or economically justified, the customary superheaters normally used in recovery units are omitted. Under these conditions the heat absorption by the furnace walls and the standard boiler screen is insufficient to reduce the gas temperature to a satisfactory point for entry to the close-spaced tubes of the boiler section. In such cases the standard unit can be modified by bending the first several rows of the boiler section forward into the superheater cavity. By this arrangement, the gas temperature is reduced in the widely spaced tubes within the superheater cavity and above the superheater floor to a level suitable for entry to the remaining closely spaced boiler section. When steam at 175 to 200 psi is supplied from a recovery unit for process only, 100 F superheat can be used to advantage in reducing condensation loss in the distribution lines.

Rated Capacity of a Recovery Unit

As the rate of liquor fired and completely burned in a furnace of a given-size recovery unit increases, the gas temperatures in the furnace and entering the screen increase. The optimum capacity of the furnace of a modern recovery unit has been exceeded when the maximum temperature of the gas entering the screen is so high that fouling of the screen and superheater cannot be controlled with sequential mechanical soot-blowing equipment. It may also be said that the peak capacity for that furnace has been exceeded when the gas temperature reaches a level where the fouling of the screen and superheater cannot be controlled with a moderate amount of hand lancing in addition to soot blowing.

The capacity of a pulp mill is based on the tons of pulp produced per 24 hr. The object of a recovery unit is to reclaim the chemicals used and to generate steam by burning the liquor resulting from the production of pulp. The capacity of a recovery unit, within the limitations noted above, should be based on its ability to burn completely, in 24 hr, the dry solids contained in the liquor recovered from the pulp produced in 24 hr. Since the proper measure of the capacity of a recovery unit is the heat released in the furnace, B&W has established as a unit of capacity a heat input of 19,800,000 Btu in 24 hr. This unit,

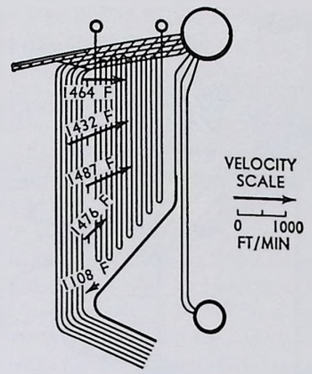


Fig. 10. *Typical gas velocities and temperatures entering superheater for a 275-ton recovery unit*

known as the B&W-Btu Ton, corresponds to the heat input from 3000 lb of solids having a heating value of 6600 Btu per lb, which are fair averages for the liquor from a ton of kraft pulp.

Nominal Size of a Recovery Unit

The nominal size of a B&W recovery unit for the conditions at a particular mill, where the solids contained in the liquor from an average ton of pulp and the heating value per lb of solids differ from the standard values used in the unit of capacity, will be greater or less in B&W-Btu Tons. The nominal size of a recovery unit for a particular mill will, therefore, be:

$$\text{Nominal Size} = \frac{A \times B \times C}{19{,}800{,}000}, \text{ in B\&W-Btu Tons}$$

where:

A = lb of solids recovered from each ton of pulp
B = Btu heating value per lb of solids
C = pulp output of mill in tons per 24 hr

Thus, for a mill producing 200 tons of pulp per 24 hr, where the cook yields 3400 lb solids in the liquor from a ton of pulp at a heating value of 6700 Btu per lb solids, the nominal size of the recovery unit will be $3400 \times 6700 \times 200/19{,}800{,}000 = 230$ B&W-Btu Tons.

A B&W recovery unit of nominal capacity as defined above has a margin for overload, with attendant comparable steam production. Higher overloads are not precluded, but, if carried much beyond 15 to 20%, the steam production will not keep pace with the liquor feed and will ultimately decrease because of the heat required to evaporate the water in the liquor and to smelt and reduce the chemical ash.

Boiler Section of Recovery Unit

The boiler section of a modern recovery unit is of two-drum vertical-tube design. Figs. 8 and 9 show two-pass and three-pass boiler gas baffling, respectively. When used, the gas baffles are arranged to provide generous flow areas at the gas turns where crossflow occurs, and adequate provision is made for the inclusion of soot blowers. Cyclone steam separators (see Chapter 9) are normally furnished in the steam drum.

Economizer

When used, the economizer is either the horizontal continuous-tube type (see Fig. 8) or the two-drum vertical-tube type (see Fig. 3). The former lends itself to high feedwater temperatures of 350 to 450 F, normal with high-pressure (1200 psi) application, and the latter to low feedwater temperatures. Feedwater entering the inlet header flows upward through the economizer tubes and through multiple tubular connections to the steam-and-water drum of the boiler. An economizer is used to take advantage of the greater temperature head existing between the gas and feedwater than that between the gas and saturated boiler water. The feedwater must be completely deaerated, and, to minimize external corrosion, the feed temperature must not be less than about 260 F (see Chapter 11). This is mandatory because of the high moisture and sulfur contents of recovery-unit flue gases. Spacing of the economizer tubes is designed for ease of cleaning with mechanical soot blowers.

Gas Air Heater

The gas air heater, a very efficient means of absorbing "low-head" heat, is a component of recovery units when practical amounts of boiler and economizer surface cannot reduce the gas temperature to a level suitable for use in the direct-contact evaporator. It is also furnished when feedwater conditions are below the standards set for safe economizer use. When liquor from the multiple-effect evaporators is delivered at a concentration of 50% solids or higher, the heat balance of the recovery unit will frequently require a gas air heater. The trend toward higher steam pressures in pulp-mill operation has increased justification of the gas air heater, since the temperatures of the exit gas from the boiler-economizer combination increases as the saturated steam and feedwater temperatures increase.

In the gas air heater, steel tubes are expanded into steel tube sheets. Gas flows inside and air passes around the outside of the tubes. The air flows parallel to the gas in multiple horizontal passes, with steel plate baffles directing the air flow. The gas air heater is cleaned with specially designed soot blowers or by shot cleaning (see below). The parallel-flow arrangement protects the tubes against corrosion by assuring

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that tube-metal temperatures are above the dew point of the gas.

Soot Blowing

Prior to 1940, soot blowers were generally ineffective in recovery units. Application was difficult, maintenance was high, and the results were not satisfactory. Consequently, hand lancing was a necessary part of the recovery-unit operation. Even after mechanical soot blowers of improved design became available for cleaning the rear boiler passes and the economizer, cleaning of the screen, superheater, and front boiler passes still required the constant attendance of hand lancers.

Increasing labor costs and the trend toward larger units forced the development of automatic mechanical cleaning devices to cope with the tenacious deposits that form in the hot-gas regions of the recovery unit. Even with the most effective known cleaning equipment, surfaces must be arranged with due regard to gas temperatures, gas velocities, and tube spacing, in order to achieve a "self-cleaning" recovery unit. First application of modern-type soot blowers in a kraft recovery unit was made in 1946. There has been rapid progress since that time, and at present, when a recovery unit is operated at or near its rated capacity, no hand lancing is required to keep the gas passages open. As load is increased on a unit, however, mechanical entrainment of ash and sublimation of sodium compounds increase and invariably lead to cleaning problems. In addition to excessive quantities of ash in the flue gas, velocities and temperatures at all points in the unit are increased, and ash deposits become more difficult to remove.

Great care is taken in the design of all recovery-unit heating surfaces to assure that such surfaces are arranged for soot blower cleaning. Cavities are left at optimum locations in screen, superheater, boiler, and economizer banks to provide for insertion of soot blowers. Gas temperatures are calculated to make certain that velocities and particular tube patterns are compatible with good cleaning characteristics. Even building layout is to a considerable extent dictated by the soot blowers. To take advantage of the superior cleaning ability and the desirability of standardization, retractable blowers, as required in the superheater, are generally used throughout recovery units. When not in actual use, they retract outside the setting to prevent damage or corrosion by the hot gases. Thus, building clearances and platform elevations are designed to avoid interferences with the retracted blowers. Soot blowers are arranged for automatic sequential operation, controlled from a panel at the main recovery-unit panel board.

In spite of the great dependence upon mechanical soot blowers, lance doors (Fig. 11) are provided at

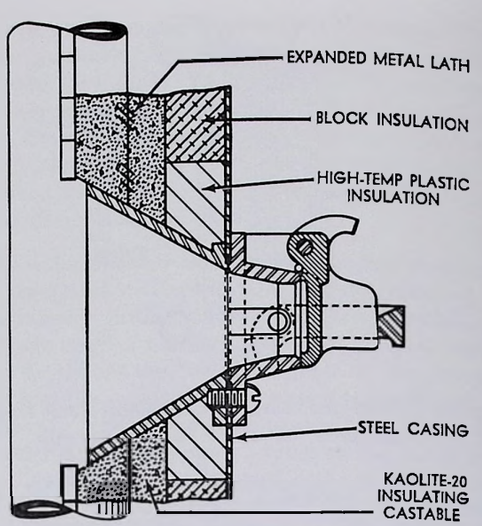


Fig. 11. *Lance-port and wall construction*

strategic locations to allow for hand lancing when required at high overloads or when soot blowers may be out for repairs.

Both air and steam have been used as the cleaning media in hand lances and soot blowers for recovery units. For hand lances, air is generally preferred from both safety and comfort standpoints. In soot blowers, however, steam is more commonly used. Air, being more dense than steam, is theoretically a better cleaning medium. In order to make air competitive, compressors are generally sized to deliver less air than would be the case with steam. Cleaning effectiveness then is about equal. However, limited compressor size is a serious disadvantage when upset furnace conditions or high overloads make it necessary to clean heating surfaces more often than is normal. Under these circumstances, the virtually inexhaustible supply of cleaning medium in the steam drum may become most important. Another factor in the almost universal use of steam for soot blowing is that economics generally rules out the use of air for the retractable soot blowers in the hard-to-clean areas of the unit. Such blowers require large quantities of 500 psi air, so that a costly compressor plant is required. As a result, steam soot blowers are almost universally used in modern recovery-unit installations.

Optimum Heat Balance

The design of the economizer and air heater is subject to a practical limitation because of the high moisture and sulfur contents in the products of combustion from kraft black liquor. To prevent rapid corrosion and failure as a result of condensation and formation of dilute sulfurous acid, the metal temperature of ordinary steel in contact with the gas must not be below the dew point of the gas. This means a) that the temperature of the feedwater to the economizer should be above the dew point of the gas, and b) that the relative heat transfer rates on the gas and air sides of the air heater tubes should be controlled to keep the tube-metal temperature above the dew point of the gas.

The heat in the flue gas between a practical exit temperature from an economizer (650 F) or air heater (450 F) and the optimum temperature leaving the direct-contact evaporator (300 F for cyclone and cascade evaporators and 180 F for a venturi scrubber) is used in direct-contact evaporation for further concentration of the black liquor. Heat represented by the temperature difference between these points is used for the evaporation of moisture from the liquor after it leaves the multiple-effect evaporators and before it is fired in the furnace. Normally, therefore, a recovery unit can be designed with a heat balance in which all the heat is effectively used.

If the heat available in the gas from the economizer or air heater is greater than that required to evaporate the liquor from the multiple-effect concentration to the firing concentration, the excess must be wasted in some manner. This can be done a) through a higher gas temperature to the fume collector, b) by reducing the temperature of the gas through dilution with air, c) by adding water to the direct-contact evaporator to replace the excess evaporation, or d) by reducing evaporation in the multiple-effect evaporators (least objectionable in the loss of thermal efficiency). However, the last method is still undesirable, since multiple-effect evaporation is the most efficient way to remove moisture from the liquor. Thermal efficiency of the recovery unit is at a maximum when the difference between the total heat from combustion of the liquor leaving the multiple-effect evaporators and the total heat in the flue gas entering the fume collector is absorbed in steam or in useful work in processing the chemical ash, without resort to any of the foregoing wasteful procedures. Obviously, it does not matter whether heat is used for final evaporation of the liquor in the furnace or in the direct-contact evaporator. If, however, a low exit gas temperature is attained by diluting the liquor in the direct-contact evaporator, then the exit temperature is no longer a measure of efficiency.

Where the maximum concentration of the black liquor from the multiple-effect evaporators is limited to a rather low figure because of the characteristics of the liquor or other considerations peculiar to a particular mill, the necessary gas temperature for direct-contact evaporation to a concentration suitable for burning in the furnace may be obtained by using only a boiler or boiler-economizer combination. Steam at a pressure of 125 to 150 psi is then used in a steam coil heater to preheat combustion air. In using this steam there is little direct loss in thermal efficiency, since the preheated air becomes an additional source of heat input to the system, resulting in the generation of a correspondingly greater quantity of steam.

Direct-Contact Evaporation

In the direct-contact evaporator the liquor and gas are brought into intimate physical contact, and there is a mass transfer of water vapor from the liquor to the gas across the liquor-gas interface. While there is a decrease in the temperature of the gas in the evaporator, the combined heat of the gas and the evaporated water leaving is nearly the same as the total heat of the gas entering the evaporator. Any difference may be accounted for by radiation loss from, and air leakage into, the evaporator plus the heat given up or absorbed by the remaining liquor.

In the design of the direct-contact evaporator, adequate surface must be provided for the desired heat and mass transfer. If liquor is splashed or sprayed on hot metal, the quantity must be sufficient for continuous washing. The liquor must be agitated to prevent local overconcentration. A further reason for continuous agitation or movement of the liquor is its acidification by the absorption of CO_2, which decreases the solubility of the dissolved solids. Because of the decreased solubility and the increased concentration, there is a tendency for sludge to precipitate. Gas velocity and flow pattern in the evaporator must be such that liquor entrainment in the exit gas is minimized.

The direct-contact evaporator used in the modern recovery unit is of the cascade (Fig. 9), cyclone (Figs. 3 and 12), or venturi (Figs. 8 and 13) type.

Cascade Evaporator. In the modern cascade evaporator (Fig. 9) there are from one to four wheels in varying parallel and series configurations. The arrangement and the amount of total contact surface required depend upon the weight and temperature of the gas and the weight of water to be evaporated. Each wheel, made of two circular side plates connected by tubes, is mounted on a heavy shaft, which is rotated at low speed. The wheel is partially submerged in a vat of liquor, so that the tubes alternately dip into the liquor and emerge to present a wetted surface to the gas stream flowing through the wheel above the liquor level. The liquor is constantly agitated by the paddle-wheel action of the rotating banks of tubes. External flow channels are provided for return circulation of the liquor as it is moved toward one end of the vat by the rotation of the wheel. Each wheel is driven by a constant-speed motor through a gear reducer and chain drive. Because of the high starting load, an hydraulic coupling is generally used

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between the motor and reducer. The concentrated liquor is withdrawn through a screen in a flow box mounted at the side of the evaporator. The screen is equipped with a moving arm for scraping lumps, caused by precipitation or localized liquor drying, into a separate chamber, where they are redissolved.

Cyclone Evaporator. The cyclone evaporator (Fig. 12) is a vertical cylindrical vessel with a conical bottom. The gas is admitted through a tangential inlet near the bottom. It flows in a whirling helical path to the top and leaves the cylinder through a concentric re-entrant outlet. Black liquor is sprayed across the gas inlet. The liquor droplets mix intimately with the high-velocity gas and are then thrown by centrifugal force to the cylinder wall. Recirculated liquor flowing down the wall of the cylinder carries the droplets and any dust or fume thrown from the gas to the conical bottom, out through the drain, and into a sump tank. Sufficient liquor from the sump tank is recirculated to the nozzles at the top of the evaporator to keep the interior wall continuously wet, preventing ash collection or localized drying on this surface. By adjusting the quantity of spray liquor at the inlet, it is possible to control the evaporation to some extent, or, conversely, to control gas temperature drop.

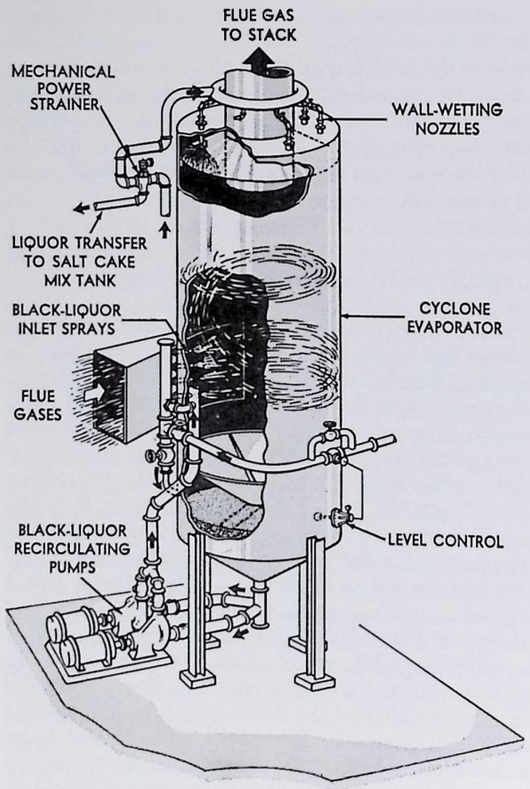


Fig. 12. *Cyclone evaporator*

Sodium Fume Particle Recovery

When char is burned in the furnace, an appreciable quantity of sodium is sublimed. As the combustion gases cool, the sodium vapor condenses and is entrained in the gas stream as a very finely dispersed fume. Much of this fume is reclaimed by deposition on the heating surfaces of the recovery unit and on the wetted surfaces of the direct-contact evaporator. The remainder, in suspension in the flue gas as very fine solid particles, passes from the unit. Although some of these particles are as large as 10 microns, collection of the fume from the waste gas must generally be considered in terms of submicron size. The quantity of sodium, largely as sodium sulfate, carried out of the unit in the flue gas depends upon the service conditions and care in operation, with probable outside limits of 90 to 275 lb and a normal average of about 150 lb per ton of pulp. This is not only a costly chemical loss, which must be replaced with make-up sodium sulfate (salt cake), but it is also a nuisance as an air pollutant. Most alkaline pulping mills are now collecting this fume from the flue gas.

Electrostatic Precipitator

The first equipment successfully used to collect the fume was the electrostatic precipitator. Many installations of this type are now in use, operating at collection efficiencies between 85 and 95%. The equipment operates on the principle that a particle, suspended in a gas subjected to a sufficiently strong electrostatic field, becomes charged electrically and tends to migrate toward an oppositely charged collection electrode to which it will adhere. The collection electrode is mechanically rapped periodically to shake the particles loose to drop into a hopper or vat of liquor.

To energize the precipitator, potentials of 50,000 to 100,000 volts are used and are adjusted so that a bluish glow may be seen at the discharge electrode. Potentials must be low enough to avoid sparking between discharging and collecting electrodes. Usually the electrical supply is provided by a separate unit, which steps up the voltage of the alternating current and then rectifies it to an interrupted but unidirectional current.

The gas temperature, moisture content, velocity, and time of retention within the area subject to the electrical charge are important. Care should be taken that the gas temperature entering a precipitator is not less than 300 F, for otherwise local cold spots result in condensation, fouling, and rapid corrosion.

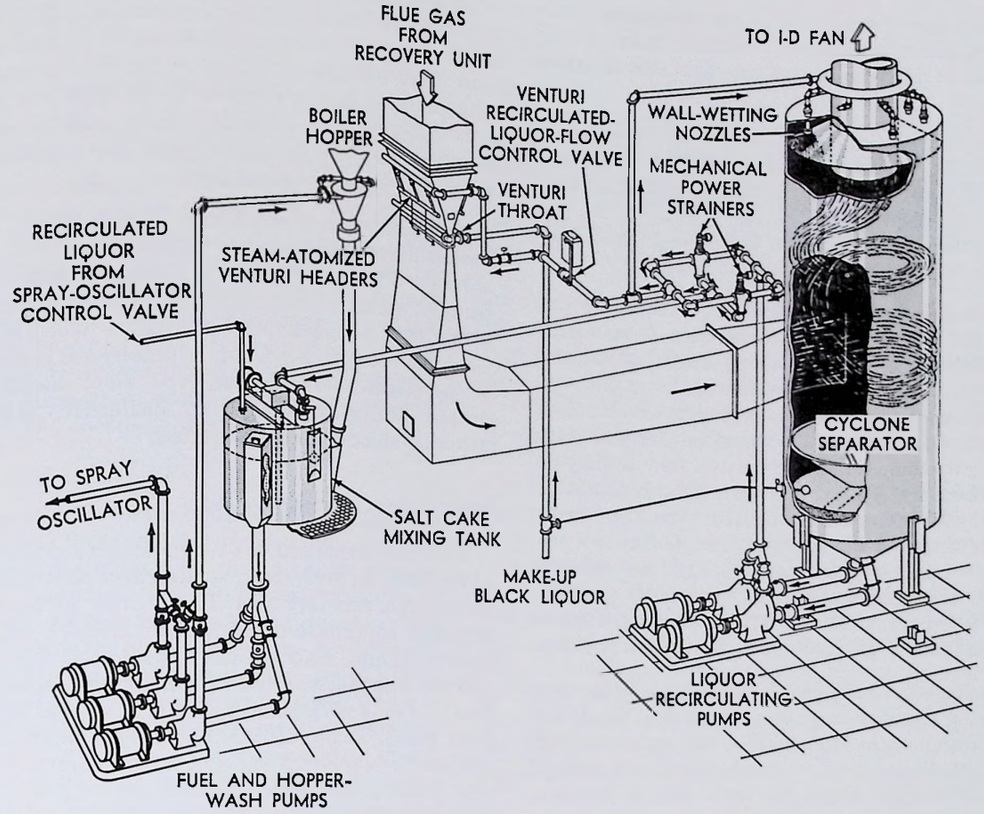


Fig. 13. *Schematic arrangement showing location of venturi evaporator-scrubber*

Venturi Evaporator-Scrubber

The venturi evaporator-scrubber (see Fig. 13) is a device which both collects the fume from the waste gas and also provides contact of the gases with the black liquor for concentration by evaporation. This scrubber depends upon the collision of fume particles with liquid droplets. In practice, the gas is accelerated in a venturi section, and liquid is admitted through steam-atomizing nozzles at the throat in a plane normal to the flow of the gas. The liquid is finely atomized by the steam and high-velocity gas, and intimate mixing results. The fume or dust particles collide with and adhere to the liquid droplets. The mixture of gas and liquid droplets then goes to a cyclone separator, where the droplets with their fume particles are centrifugally removed from the gas stream and drain to the sump tank.

Using black liquor as the scrubbing medium, the venturi scrubber replaces both the direct-contact evaporator and the fume collector. In this application, the liquor is concentrated by cooling the gas to about 180 F. The liquor is recirculated from the cyclone-separator sump to the venturi-throat nozzles at a concentration of 60 to 65% solids. Liquor is fed from this system to the furnace for firing, as with the direct-contact evaporators previously described, and make-up liquor from the multiple-effect evaporators is added to the system at the venturi throat to compensate for the liquor fired.

The venturi scrubber is a simple piece of equipment, easy to operate and control. Fume collection efficiencies of 90 to 95% are continuously maintained. A recovery unit equipped with a venturi scrubber achieves a higher thermal efficiency, since its terminal gas temperature is about 120 F lower than that permissible with an electrostatic precipitator. When liquor is the scrubbing medium, there is good absorption of SO_2 from the flue gas.

Auxiliaries for Recovery Unit

Successful operation of a recovery unit depends to a large extent upon the performance of its auxiliaries. All the smaller components must be designed and selected with the same care and attention to

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reliability and continuity of service as the main components. The auxiliaries, preferably and usually supplied as a part of the recovery unit, are described below:

Auxiliary Oil or Gas Burners. These burners, for curing the furnace refractory, preheating at start-up, and emergency use in case of unstable furnace conditions, are designed for quick and easy operation, are provided with safety devices for flame-failure protection, and are interlocked for maximum safety.

Meters and Controls. Automatic controls are provided for boiler water level, furnace draft, direct-contact-evaporator liquor level, black-liquor firing temperature, and the combustion-air flows. Flue gas is continually analyzed by an oxygen-combustibles analyzer. Instruments are provided for recording steam flow, total-air flow, primary-air flow, tertiary-air flow, boiler water level, gas temperatures at various points in the system, air preheat temperature, steam temperature, and black-liquor flow. Gages are provided to indicate draft losses through the different sections of the recovery unit, primary-air pressure, secondary-air pressure, tertiary-air pressure, furnace draft, induced-draft-fan suction, and steam pressure.

Fans. Because of the possible imbalance of the rotor caused by deposits from fume-laden gas, the design and selection of the induced-draft fan are particularly important. Speed must be conservative, and straight or nearly straight blades are preferable. A variable-speed drive is often used in the form of a steam turbine, if the mill steam balance warrants, or an hydraulic or magnetic coupling.

Smelt-Dissolving Tank. This tank should be of heavy construction and should be equipped for good agitation. One or more side-entering agitators are generally used. The dissolving tank must be vented with a vertical stack of sufficient size to carry off the vapor formed in the tank. Green liquor is recirculated to shatter-nozzles, to disperse the stream of molten smelt before it enters the liquor in the tank. Steam jets are also provided, to help break up the stream of molten smelt and to reduce the noise resulting from the chilling of the hot smelt by the cooler liquor.

Mixing Tank and Salt Cake Feeder. Careful attention to detail is required in the design of this tank and feeder, so that make-up salt cake may be fed at a regular rate and be well mixed with the black liquor. The mixing tank includes a mechanically scraped screen, to assure that all material which passes to the fuel pumps is of small enough size to pass readily through the burner nozzle.

Pumps for Moving the Liquor. Pumps are very important components of the recovery unit. All liquors encountered in the recovery cycle are to some extent corrosive. As concentrations increase above 65% solids, black liquor becomes more viscous at a rapid rate, and, after salt cake is added at the mixing tank, it also becomes highly abrasive. Centrifugal pumps of special design are used in this service, and due attention must be given to suction heads, liquor concentrations, and liquor temperatures.

Black-Liquor Piping and Heaters. Piping, fittings, and heaters must be designed to withstand corrosion and abrasion. Piping sizes must represent a compromise between pressure drops with highly viscous fluids and a tendency for sludge to precipitate when liquor velocities are low. Black-liquor heaters are generally of the direct-steam-injection type, since scaling on surface heaters renders them ineffective, thereby requiring standby or spare heaters.

Soda Process Recovery Unit

With few exceptions, the requirements and features of the kraft recovery unit, as described above, apply to the soda recovery unit. In the soda process the digestion liquor is sodium hydroxide, NaOH, without sodium sulfide, Na_2S, which characterizes the kraft process. The make-up chemical is sodium carbonate, Na_2CO_3 (soda ash), instead of sodium sulfate, Na_2SO_4 (salt cake). Since sulfur is not present in the soda process, there is no reduction of sulfur in the recovery-unit furnace. Therefore, chemical make-up is added to the recovered green liquor rather than to the black liquor ahead of the furnace (as in the kraft process). Consequently, the salt cake feeder and mixing tank are not required. The dust and fume collected from the boiler hoppers and the fume collector are in the form of soda ash, and it can be added directly to the green liquor after the furnace rather than to the black liquor before burning.

In the furnace, the soda liquor does not form a suitable reactive char for burning in a bed, and it is, therefore, finely atomized and sprayed into the furnace. The fine spray dehydrates in flight, and combustion takes place largely in suspension. Combustion air is admitted through ports around the furnace periphery, as in the kraft operation, and the hottest part of the furnace is just above the hearth. The ash, sodium carbonate, collects in molten form on the hearth and is tapped through the smelt spout or spouts.

Soda-liquor burning is subject to the same sublimation of chemical and mechanical carry-over of particles as in the kraft process, but the handling of hopper dust and recovered chemical from the fume collector is somewhat different, since it is not necessary to return the material to the furnace. Physically, the design of the recovery unit is the same for soda as for kraft liquors except that the oscillating spray

nozzle is replaced by two or more steam-atomizing soda-liquor burners.

Acid-Sulfite Process

The acid-sulfite process for wood-pulp manufacture differs from the alkaline processes in that an acid liquor is used in the cooking of wood. Chips are prepared in the same fashion as for the alkaline processes and are then cooked in a digester, under steam pressure, to dissolve the lignin binder that holds the cellulose fibers together. Older sulfite mills use a solution of the bisulfite of calcium or calcium-magnesium, prepared from limestone or dolomite. In recent years, however, increasing interest has been paid to magnesium, ammonia, and sodium as bases for the bisulfite cooking liquor.

Bisulfite Acid

In the preparation of the bisulfite acid for the sulfite process, SO_2 is first produced by burning either elemental sulfur or pyrites. The quantity of air used for burning is controlled to minimize the formation of SO_3. The SO_2 is cooled and passes to the absorption towers. The number of towers may be two or more. The towers are operated in series, and cool SO_2 is introduced at the bottom of the first tower where it passes upward through the packing countercurrent to the flow of strong acid. The major portion of SO_2 is absorbed in this first or strong-acid tower. The remaining SO_2 passes out the top of the strong-acid tower and is introduced into the bottom of the second, or weak-acid, tower where it again passes upward through packing countercurrent to the downward flow of weak acid or water.

The raw acid withdrawn from the base of the first, or strong-acid, tower is further fortified with free SO_2 relieved from the digesters to accumulators. Raw acid from the absorption towers contains about 4% total SO_2 by weight, of which 1.6-1.8% of the total weight of the solution is combined SO_2. Corresponding percentages of SO_2 in the cooking liquor range from 5.5 to 6.5 total, with approximately 1% combined SO_2.

For the calcium-base digestion liquor, the base itself, relatively pure limestone, is used as packing in the absorption towers in the acid-making system. For other bases, dolomitic limestone, magnesium, ammonia, or sodium as a slurry or solution is used. This slurry or solution passes downward in the tower over chemically inert packing material, and the SO_2 passes upward to form a bisulfite of the base with an excess of sulfurous acid.

Cooking Procedure

After charging the digester with chips, the cooking acid is added, either hot or cold. The digester is brought up to pressure either directly or by recirculating the liquor through an external heat exchanger where it is heated and the liquor returned to the digester. Because of the release of SO_2 as the temperature increases, the excess gas must be released to permit raising the temperature to correspond to the pressure within the digester. Cooking time normally varies from 7 to 10 hr, and for most of the cooking the pressure and temperature are approximately 90 psi and 300 F, respectively.

At the completion of the cook, the gas and vapor are relieved from the top of the digester, and the pressure is reduced to 30-40 psi. At this pressure all the contents are discharged to the blowpit through the blow valve at the bottom of the digester. The blowpit is an enclosed chamber approximately 2½ times the volume of a digester. It is equipped with a false bottom of perforated stainless steel plate and stainless steel target, against which the chips from the digester are blown. A vent, or vomit stack, is provided to relieve the gas and vapor released from the digester charge as it enters the blowpit. The liquor is drained from the pulp through the perforated false bottom, and the pit is then flooded several times with warm water which also drains through the perforated plate to wash the pulp. The waste liquor, commonly called red liquor, and the wash water go to the sewer unless the red liquor is to be burned. The pulp is hosed from the blowpit through side doors and, after refining, passes to the paper machine.

Sulfite Waste Liquor

The total pulp production by the three major pulping processes in the U.S. is divided approximately as follows: the kraft process 70%, the sulfite process 25%, and the soda process 5%. Reclamation of chemicals and the production of steam from waste liquor are well established in the kraft and soda processes. Corresponding progress in recovery of sulfite waste liquor has lagged, however, for technological and economic reasons. During the past 15 yr, pressure for finding a better means for the utilization and disposal of waste sulfite liquor has increased, both to improve the overall economy of the pulping process and to permit compliance with stringent government regulations regarding stream pollution. At current fuel costs in the U.S., recovery of heat values alone from the waste liquors is considered economically marginal. Much effort has gone into development of processes to salvage chemicals for re-use in cooking or to obtain salable by-product compounds, such as alcohol, yeast, vanillin, oxalic acid, adhesives, and plastics.

The problem of sulfite-waste-liquor disposal is of considerable magnitude, since the weight of organic solids in the waste liquor will roughly equal the weight of cellulose pulp produced. As individual sul-

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fite pulp mills have production rates from 50 to 600 tons of pulp per day, it can be seen that large quantities of waste liquor in dilute concentration are also produced. The markets for by-product chemical compounds are inadequate to provide more than isolated relief from the problem, and the only universally applicable solution thus far suggested is the use of the sulfite waste liquor as a fuel in the mill where it is produced.

Sulfite-Waste-Liquor Collection and Evaporation

The optimum arrangement of the equipment for burning sulfite waste liquor will vary somewhat, depending upon the base of the liquor and whether or not chemicals are to be recovered. Collecting the red liquor at the highest possible concentration, in order to reduce the amount of evaporation necessary in preparing the liquor for burning, is of the utmost importance, as the following will illustrate: Based on a 90% recovery, a pulp mill of 100 tons/day capacity would produce about 9,000 lb of red liquor solids/hr. The quantity and concentration of weak red liquor collected and the quantity of water to be evaporated to reach a solids concentration of 50% are given in Table 1. It will be noted that more than twice the quantity of water must be evaporated from liquor collected at 8% solids than from liquor collected at 14% solids, which requires a much larger, more costly evaporator installation and doubles the steam consumption for evaporation.

TABLE 1

Water Evaporation for 100-Ton Mill

Concentration of Red Liquor as Collected, % Solids	Weak Red Liquor Collected, lb/hr	Water Evaporated for 50% Solids, lb/hr
8	112,500	94,500
10	90,000	72,000
12	75,000	57,000
14	64,280	46,280
16	56,250	38,250

To effect the 90% recovery of the red liquor at a concentration dictated by sound economics, the industry has adopted the multistage vacuum washers. Economics of recovery-system operation demands a liquor feed to multiple-effect evaporators of at least 10% solids.

The red liquor leaving the multistage vacuum washers at 10-15% solids must be further concentrated by evaporation of water in multiple-effect evaporators. Development work during the past 10 yr has led to a method for evaporating bisulfite waste liquor in the continuous uninterrupted manner essential to a recovery and burning process. For burning, the red liquor should be concentrated to 60% solids. Liquor concentrations above 50% are desirable.

Sulfite-Waste-Liquor Recovery

The principal difference in burning the red liquor of the four major bases is in the physical properties of the base itself after combustion. When calcium-base red liquor is burned, the calcium is present largely as calcium sulfate entrained in the flue gas as finely divided dust or fly-ash. When magnesium-base red liquor is burned, the magnesium is present in the flue gas as a fine magnesium oxide ash. When ammonia-base red liquor is burned, the ammonia disassociates to nitrogen and hydrogen, the latter burning to water vapor, thereby destroying the base. Compared with the combustion of the calcium-base liquor, this is an advantage, as the calcium sulfate fly-ash must be collected to avoid air pollution, and some means of disposal must be provided. Of the three mentioned, magnesia-base ash is the only one which can be collected and recombined to produce a new raw acid.

It is possible to burn sodium-base liquor either alone or in combination with kraft black liquor in a recovery unit of the kraft type. The base or ash of the sodium-base red liquor would, in this case, be recovered as a smelt of sodium sulfide with some sodium carbonate, which would be suitable as make-up chemical for an associated or nearby kraft mill. The regenerative cycle for sodium bisulfite pulping requires a complex secondary recovery unit, using secondary reactions to convert the recovered sodium sulfide and sodium carbonate to sodium bisulfite.

Since sodium-base liquor must be burned in a kraft-type recovery unit, its burning and recovery principles will not be discussed in this section. Of the four common bases discussed, then, the magnesium-base recovery system, where the MgO ash is collected, slaked with water, and used to absorb SO_2 from the flue gas to produce magnesium sulfite raw acid, can be seen to be the simplest and most economical system.

Burning Sulfite Liquor

While red liquor at concentrations of about 45% solids has been successfully burned in refractory furnaces, higher concentrations are recommended to assure stable combustion. In general 50% solids is considered minimum. The liquor must be sprayed into the furnace in finely atomized form, so that it will quickly absorb radiant heat to complete the evaporation of water from the droplets. Radiation in the furnace may be from hot brickwork or from the flame of another fuel fired simultaneously. When burning red liquor alone, large areas of uncooled brickwork are desirable in that part of the furnace where the liquor

spray is introduced. Turbulence of the atomized-liquor-and-air mixture is essential for rapid and complete combustion. Preheated combustion air is necessary to effect furnace stability.

For the magnesium bisulfite process, a high combustion efficiency is required to minimize unburned carbon in the recovered ash. A high degree of turbulence is obtained by using two sets of burners located opposite each other in the side walls of a refractory furnace. The atomized liquor is dried in passing from the spray nozzles into the furnace, and a high turbulence is developed when the two opposing streams of air and liquor particles meet. This design of furnace is also suitable and efficient for burning calcium- or ammonia-base red liquor.

Magnesium-Base Pulping and Recovery Process

Commercial experience has shown that chemicals and heat may be efficiently and easily recovered when magnesium is used as the cooking-liquor base. In the combustion of the waste liquor, the magnesium sulfur compounds are disassociated to MgO and SO_2, and both are reclaimed and used directly in making fresh cooking acid. Heat from burning the liquor is absorbed by boiler surface before recovery of the chemicals from the flue gases. High recovery efficiencies for both heat and chemicals make the magnesium process commercially profitable, particularly in large mills. Furthermore, the elimination of stream pollution is an important additional advantage obtainable with the use of magnesium-base cooking.

Industrial interest in the improved pulp from a wide variety of wood species has stimulated the development of advanced pulping techniques using magnesia base. The magnesia bisulfite (Magnafite) pulping process is a simplification of the magnesium acid bisulfite process, without the presence of "true free" SO_2. The combined and free SO_2 (by TAPPI definition) are equivalent. The Magnafite process produces sulfite pulps of increased yield and higher strength, from both coniferous and deciduous wood species, with a much shorter cooking time than those produced by the acid sulfite process. The single-stage Magnafite process was followed by the development of multi-stage magnesium-base cooking processes. It is now possible to produce a wide spectrum of pulp grades

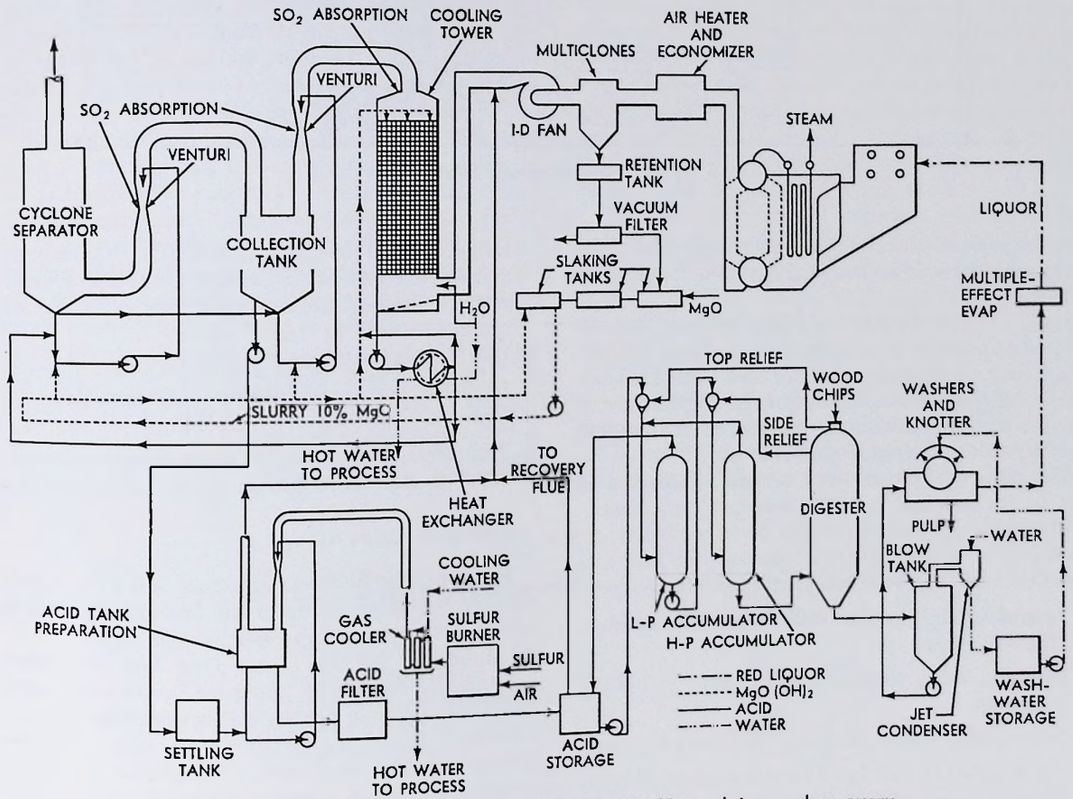


Fig. 14. *Process flow diagram for magnesium bisulfite pulping and recovery*

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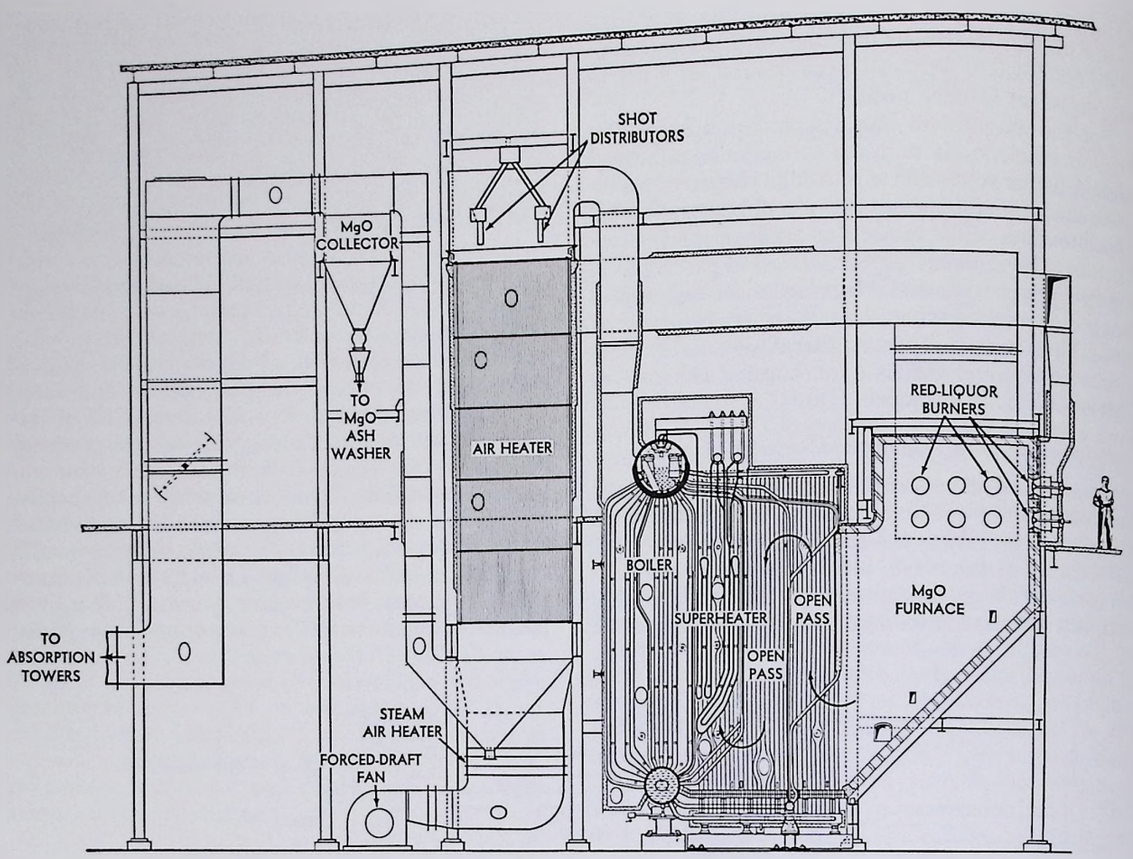


Fig. 15. *B&W MgO chemical- and heat-recovery unit*

from high-alpha dissolving pulp to high-yield semi-chemical grades from both hardwood and softwood species.

Early interest in magnesia-base pulping was restricted exclusively to benefits of the recovery process. The improved pulping processes now available combined with the commercially proven MgO recovery process have intensified the interest of the pulping industry in MgO pulping and recovery.

The classic MgO recovery process, illustrated in Fig. 14, is applicable to all MgO pulping processes.

Certain design features, briefly described below, are necessary to obtain optimum efficiency of heat and chemical recovery from this process.

1. A combustion chamber with large areas of high-alumina refractory.
2. Opposed firing to give highly turbulent furnace conditions.
3. Internal-mix steam-atomizing stationary burners.
4. Open passes to cool by radiant-heat absorption the furnace gases rapidly through the temperature range at which the conversion of SO_2 to SO_3 is high, and also to cool them to a temperature below which any sodium and potassium impurities in the ash are in a "sticky" or "tacky" state.
5. Wide spacing of boiler and superheater tubes to facilitate cleaning with automatic retractable soot-blowers.
6. Preheated air at a temperature of approximately 700 F to assist combustion.

Magnesium Oxide Ash

Following the heat-recovery unit, it is necessary to separate the MgO ash from the flue gas. The ash is collected in cyclone-type mechanical separators located beyond the air heater (see Fig. 15). After collection of the ash, to minimize evaporator-scaling and boiler-cleaning problems it is desirable to eliminate the sodium and potassium salts which enter the system with the wood and, as impurities, with the make-up chemicals. These salts are readily soluble in water and may be washed from the MgO slurry on a vacuum type of filter. The collected MgO ash is transported to the filter by a combination screw conveyor and hydraulic flume.

Magnesium Oxide Slurry

The washed MgO cake, free of contaminating salts, is then mixed with the make-up MgO and pumped as a slurry to the MgO slurry tank. The flue gas, relatively free of ash, passes directly to cooling towers on its way to the absorption system (Fig. 14).

Magnesium oxide slurry, at a concentration of 10% solids, is pumped from the slurry tank to the absorption system, where it is added at the suctions of the recirculating pumps to insure complete mixing with the absorbing liquor.

Sulfur Dioxide Absorption

Furnace flue gases, of 1 to 2% SO_2 by volume, flow from the recovery boiler and air heater to the mechanical ash collector and thence to the packed cooling tower, where they are cooled to approximately 100 F by recirculated acid water that condenses the water vapor in the flue gases and collects the small amount of residual MgO ash not collected in the mechanical separator. The recirculating coolant also absorbs a portion of the SO_2 to form a dilute bisulfite acid. Cooling of recirculated weak acid is accomplished through heat exchangers to provide heated process water for pulp bleaching and washing.

Subsequent to the cooling tower, the combustion gases containing SO_2 pass concurrent with a flow of circulating acid in each of the venturi throats, where the circulating acid collects additional sulfur dioxide in the production of the magnesium bisulfite acid. The number of venturi stages required in series is dependent upon the concentration of the product, magnesium bisulfite raw acid, and the over-all SO_2 recovery efficiency desired. A typical two-stage absorption system is shown in Fig. 14. Absorption of sulfur dioxide, to obtain an over-all SO_2 recovery efficiency of 95% for the entire absorption system, can be obtained with the two-stage venturi system for acid concentrations characteristic of the magnesium acid sulfite process.

A portion of the slightly acid circulating coolant water in the cooling tower is removed at a rate relative to the system flow of acid to the mill and fed to the final or weak-stage venturi scrubber as make-up for the venturi stages. The acid flows between the successive venturi stages countercurrent to the flue gas flow to yield an acid of the required raw-acid composition in the collection tank of the first, or high-concentration, venturi.

Magnesium hydroxide slurry is added at a controlled rate to the circulating acid in the venturis for maximum recovery of sulfur dioxide. This rate of magnesium hydroxide addition is controlled by pH meters, which maintain a concentration of 0.5-0.7% magnesium sulfite (referred to as monosulfite) in the acid sprayed into the venturi throats. Absorption is effected by the monosulfite, which has a high affinity for SO_2. The maximum sulfur recovery before the flue gases leave the stack is assured by maintaining the optimum magnesium sulfite concentration in the acid in each venturi.

The rate of magnesium hydroxide addition to the first venturi is controlled to obtain an acid as near to true bisulfite as possible. Theoretically, the absorption in the first venturi stage can be controlled to react a sufficient quantity of sulfur dioxide with the magnesium hydroxide to form the bisulfite with none or only a small quantity of the sulfite salt. However, it is necessary to maintain an excess of magnesium sulfite in the acid at this point to obtain maximum over-all SO_2 recovery. The final raw acid contains an excess 0.2-0.3% sulfite, which, together with the magnesium bisulfite, is in solution at a pH of 4 to 5.

The raw acid from the collection tank of the first venturi flows to a settling tank, where insoluble materials, such as calcium silicate and inerts picked up from the wood and make-up magnesia, settle out and are periodically flushed to the sewer.

Economics

An analysis of the cost of chemicals per ton of air-dry pulp produced with various bases is given in Table 2. This analysis indicates the range of chemical savings possible in addition to the value of the steam gener-

TABLE 2

Relative Cost of Base Chemical and Sulfur per Ton of Air-Dry Pulp

	Base	Base, lb	Base, $	Sulfur, lb	Sulfur, $	Total Make-up, $
Without Recovery	Calcium carbonate	350	0.74	270	4.38	5.12
	Magnesium oxide (acid bisulfite)	140	2.80	270	4.38	7.18
	Anhydrous ammonia	119	3.57	270	4.38	7.95
	Sodium carbonate	370	3.70	270	4.38	8.08
With Recovery	Magnesium oxide (acid bisulfite)	20	0.80	50	0.81	1.61

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ated, when the base chemical and the SO_2 are recovered by burning the red liquor in a recovery unit for the magnesium bisulfite process, as described above. The savings in chemicals alone, when comparing magnesium base with recovery to the relatively inexpensive calcium base, is $3.51 per ton of pulp. When comparing magnesium base with the more expensive bases, the savings are greater.

Neutral Sulfite Semichemical Process

Semichemical Pulping

"Semichemical" covers pulping processes in which the wood chips are partially cooked and the balance of the pulping is carried out by mechanical means. The aim of the process is to produce pulp at yields approaching those for groundwood but still resembling chemical pulp in its properties. The process was developed to produce high-yield low-cost pulp from wood species not normally suitable for pulping by the kraft or sulfite processes.

Hard woods, such as oak, gum, alder, chestnut, aspen, and maple, have been used to produce semichemical pulp. The cooking liquor may be alkaline or acid, as previously discussed for full chemical pulping. However, the term "semichemical pulp" as commonly used generally denotes a pulp produced with a solution of sodium sulfite buffered with sodium carbonate to a near-neutral pH, thus, the term "NSSC," denoting *Neutral Sulfite Semi Chemical* pulping.

Semichemical pulping has been under development for almost thirty years. Early attempts to develop a chemical recovery system utilizing NSSC waste liquor were not successful. As a result of the absence of chemical recovery, advancement of this pulping process has been retarded.

The small amount of heat that can be recovered from NSSC waste liquor per ton of pulp produced limits the economic return and thus has retarded development of recovery processes. As a result of the high pulp yield, the organic and total solids in NSSC waste liquor are low—1200 to 2100 lb/pulp ton compared with about 3000 lb/pulp ton in soda and kraft processes. Also the heating value of the solids is relatively low—4300 to 5700 Btu/lb dry solids—because of the high chemical-to-wood ratio used in the NSSC cook and because of the small amount of wood substances in the waste liquor. The heat in the waste-liquor solids from the manufacture of one ton of NSSC pulp is between 5,500,000 and 12,000,000 Btu, whereas the kraft process averages 19,800,000 Btu/pulp ton. Variations in the heat in the solids produced per ton of pulp depend upon the cooking procedures, type of wood used, yield, and grade of pulp produced. Thus, while NSSC waste liquor can be burned in a conventional kraft recovery unit, the size of unit required for a given pulp-production capacity is generally from one third to one half the size required for the same kraft pulp output.

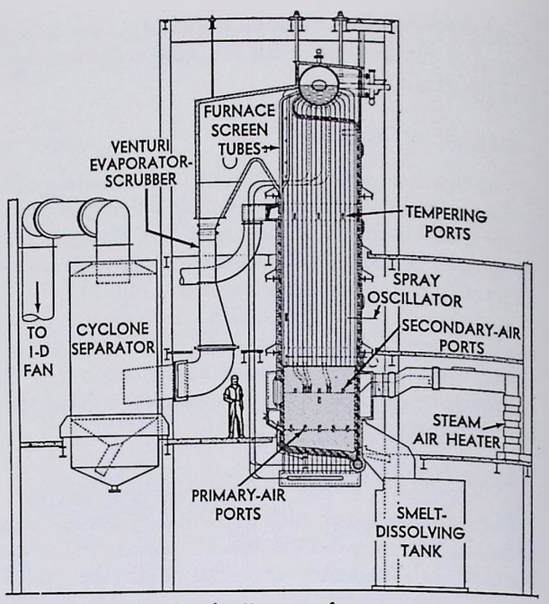


Fig. 16. *"Smelter" type of recovery unit*

The "smelter" type of recovery unit, designed for economical burning of small quantities of black liquor, is shown in Fig. 16. This type of unit is unconventional in appearance, for it does not include the usual boiler section and other heat traps. The flue gas leaving the furnace passes directly to the venturi evaporator-scrubber. This unit sacrifices some steam flow in favor of lowering first cost, reduces operating labor, and eliminates the problem of cleaning heating surfaces.

Flue gases at 900-1000 F enter the venturi to evaporate liquor from an initial concentration of 28-30% solids to a burning concentration of about 60% solids. The use of the smelter in the neutral sulfite recovery system is incidental, but it is well suited to this application because of the small amount of heat in the waste liquor. Such smelters may also be applied to small kraft installations.

Neutral Sulfite Recovery Process

The Mead Corporation developed a process for the regeneration of the chemicals recovered from burning NSSC waste liquor for re-use in cooking liquor.

The recovery process consists essentially of two operations. The first stage includes the furnace, boiler heat traps, and contact evaporator. The venturi evaporator-scrubber is included not only to effect reduction of flue gas temperature and evaporation of liquor but also to collect the sodium compounds from

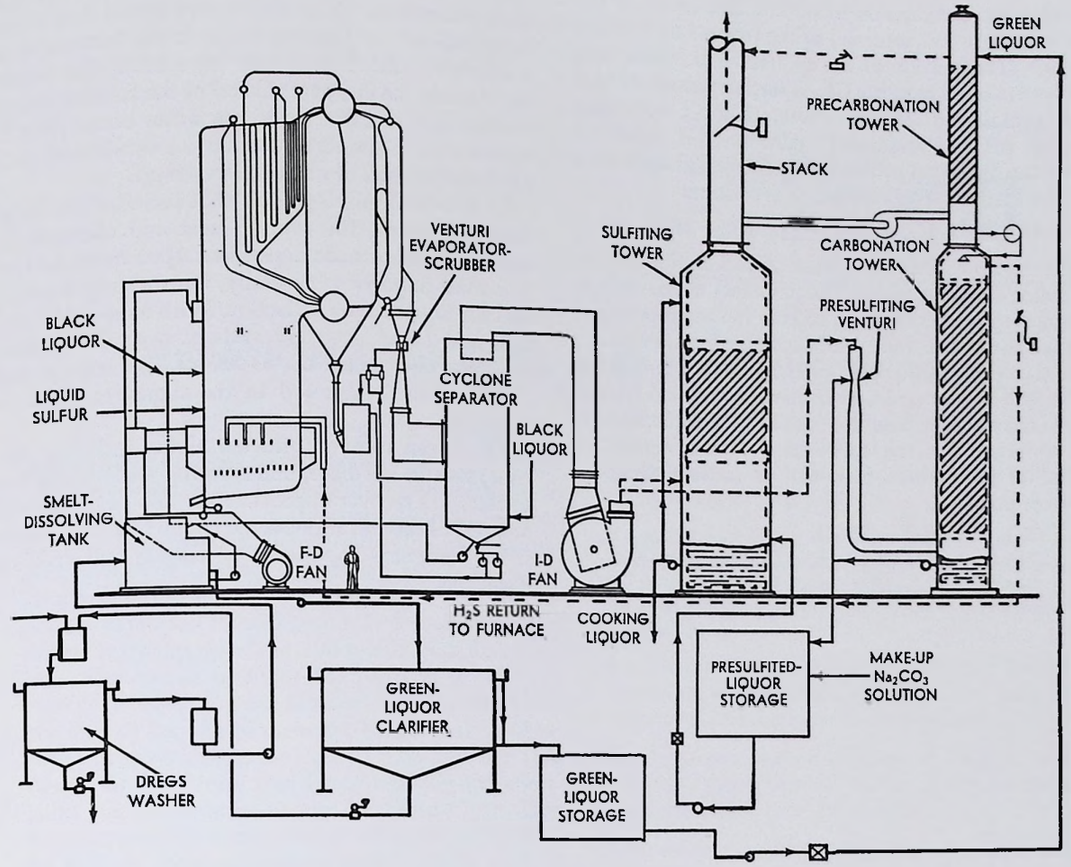


Fig. 17. *Schematic diagram of the Mead recovery process*

the flue gases and thus eliminate inert-salts contamination of the cooking liquor produced in the second stage. The latter consists of a group of packed towers in which a water solution of the sodium sulfide and sodium carbonate smelt, obtained from the recovery furnace, is first carbonated to convert Na_2S to Na_2CO_3 and then sulfited to convert Na_2CO_3 to Na_2SO_3. Boiler flue gas is used for both carbonating and sulfiting, and the H_2S driven from the liquor during carbonation is returned to the furnace for burning.

Fig. 17 shows a schematic diagram of the Mead recovery process. Flue gas from a conventional kraft recovery unit flows through the venturi evaporator-scrubber and cyclone separator to the secondary absorption system. Part of the flue gas from the I-D fan goes through the presulfiting venturi, the rest through the sulfiting tower. The latter, being rich in CO_2, is partially diverted to the precarbonation tower, the remainder passing up the stack. The flue gas from the presulfiting venturi passes through the carbonation tower and, being rich in H_2S, is returned to the furnace to be burned to SO_2.

On the liquor side of this system, smelt from the furnace flows into a conventional dissolving tank and is dissolved to form green liquor of approximately 50% sulfidity. Sulfidity is defined as the percentage ratio of the weight of sodium sulfide to the sum of the weights of sodium sulfide, sodium hydroxide, and sodium carbonate all expressed as Na_2O. The green liquor then passes to a green-liquor clarifying system and thence to the precarbonation tower, where it flows countercurrent to the CO_2-rich flue gas and reacts as follows:

$$2Na_2S + CO_2 + H_2O \rightarrow 2NaHS + Na_2CO_3$$

From the precarbonation tower the green liquor passes through the carbonation system, where the precarbonated liquor again reacts with CO_2-rich gas as follows:

$$2NaHS + CO_2 + H_2O \rightarrow Na_2CO_3 + H_2S$$

Most of the sulfite sulfur is stripped from the liquor, leaving a carbonate-bicarbonate solution. The gas

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leaving the carbonation tower contains about 5% H_2S by volume and is returned to the furnace for burning to SO_2. The portion of the sodium hydrosulfide that has not reacted with the CO_2 is further passed through the presulfiting venturi where, under a controlled higher pH, the remaining hydrosulfide is stabilized, with the hydrogen sulfide being stripped and returned to the furnace. The reaction is as follows:

$$2NaHS + SO_2 + H_2O \rightarrow Na_2SO_3 + 2H_2S$$

This step prevents the release of H_2S in the ensuing sulfiting-stage reactions. The net effect of the carbonating step is to convert all sodium in the green liquor to the carbonate and bicarbonate forms and to liberate the sulfur in the form of H_2S. The H_2S-containing flue gas is then returned to the furnace and burned to SO_2. In the sulfiting towers the SO_2 of the flue gas is absorbed by the carbonated liquor, converting some of the sodium carbonate to sodium sulfite, as shown below:

1. $SO_2 + H_2O \rightarrow H_2SO_3$
2. $H_2SO_3 + 2Na_2CO_3 \rightarrow Na_2SO_3 + 2NaHCO_3$
3. $H_2SO_3 + 2NaHCO_3 \rightarrow Na_2SO_3 + 2H_2O + 2CO_2$

Sodium is replenished in the system by adding sodium carbonate to the carbonated liquor, and sulfur is replenished by burning sulfur in the furnace or in a separate sulfur burner and introducing the resulting SO_2 into the flue gases ahead of the sulfiting tower. Introduction of salt cake to the waste liquor prior to incineration in the furnace could be considered as an alternate method of chemical make-up.

In an NSSC pulping cycle that includes the Mead recovery system, the over-all heat and chemical recovery efficiencies are dependent upon many factors. The most important, naturally, are the pulp-washing and the liquor-solids collection. With a modern NSSC mill and Mead recovery system, over-all chemical recovery efficiencies of 80-90% of the sodium and 75-85% of the sulfur fed to the digesters could be expected.

The Mead recovery process alone, including furnace, venturi scrubber, and sulfiting and carbonating towers, will recover approximately 96% of the sodium and 96% of the sulfur delivered to the recovery system in the form of evaporated waste liquor and make-up chemicals.

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OPERATION OF BOILERS AND RELATED EQUIPMENT

To obtain the best results from a boiler unit of any given design, an active interest must be taken in its performance. It is necessary to learn what the unit is capable of doing under the most intelligent operation. Control equipment, which will do at all times and under adverse conditions what the operators alone are unable to do, is a most important factor in the approach to optimum results in operation. The almost unbelievable progress in the generation of steam during the last 40 years was made possible, in very large measure, by the makers of instruments and automatic controls. Performance has been constantly improved by eliminating the uncertainty and inaccuracy of manual control in making the many sensitive and delicate adjustments required for the safe operation of a modern boiler unit. It would, of course, be virtually impossible to operate a modern unit by hand alone.

The gain in standards of personal safety, elimination of drudgery, and saving in fuel made possible by this auxiliary equipment has been worth many times its cost. It has also exerted a powerful influence in changing the dingy uninteresting boiler room of four decades ago to the pleasant well-appointed modern steam plant which is attracting a constantly improving caliber of operating personnel. It has been a vital factor in leading the manufacturers of steam-generating and steam-using equipment to improvements in size and in function unimaginable with hand operation and with underdeveloped instruments.

Good boiler operation to obtain low maintenance, only scheduled outages, and reasonable supervision may be summarized as follows:

The most efficient use of the fuel, or the most Btu output in steam for the least Btu input in fuel.

No external or internal corrosion or erosion of any of the pressure parts.

No deposits of scale on the water-heating and steam-generating contact surfaces.

A minimum of sludge or other foreign matter on the water-heating and steam-generating surfaces.

No deposits of any type on the steam-superheating surfaces.

A minimum amount of slag, ash, or other foreign material deposited on the gas-contact surfaces or discharged into the outside atmosphere.

No carry-over with the steam of foreign material, which can be harmful to the equipment or to the process which uses the steam.

A minimum amount of physical work and an increasing amount of productive work for the operators.

Constantly increasing safety for the operators.

Maximum protection for the equipment.

Indifferent results from inadequate equipment cannot fairly be charged to operation, but a wholesome dissatisfaction on the part of the operators can be made the starting point for bringing about improvements in equipment necessary to obtain good results.

The two important prerequisites to good operation are knowledge of the equipment and interest in it, and one usually augments the other. A real interest in boiler-room equipment will soon develop knowledge and understanding. A better knowledge and understanding of the equipment will develop an interest in its performance and in its operating records. Fundamentally, of course, there would be a tendency to lose interest in the records of perfect operation. The operator's real interest in records is in their measure of the degree of perfection he has attained, so that he can tell what he is not carrying out as well as he might or as well as others have been able to do under similar conditions.

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The operating instruments, supplemented by records or recordings of their indications, needed by an operator vary over a wide range, depending upon the size and type of equipment. With some exceptions, obvious in specific instances, the information which he wants to obtain from the instruments and other sources may be listed in the approximate order of their importance, as follows:

1. Water level.
2. Visual observation of the fire.
3. Pressure of steam and of feedwater.
4. Temperatures of superheated steam leaving the major subdivisions of the superheater and of the reheater.
5. Drafts or pressures of gases and air entering and leaving the principal sections of the equipment traversed.
6. Boiler-water conditions, including carry-over.
7. Operation or nonoperation of feed pumps, fans, fuel-preparation and fuel-burning equipment.
8. Relationship of actual combustion air passing through the furnace to that actually needed, as indicated by flue gas analyses, oxygen determinations, or other means.
9. Temperatures of water, gas, fuel, and air entering and leaving the principal sections of the equipment traversed.
10. Feedwater flow.
11. Fuel flow.
12. Steam flow.
13. Knowledge of what operating functions are interlocked, so that they cannot be performed in an improper sequence.
14. Knowledge of what important operating functions are being performed automatically.

Automatic control is virtually inseparable from instruments. The list of functions which should be controlled automatically varies—somewhat as the list of instruments varies—with the size of the plant, the proficiency of the operators, the type of equipment, and, therefore, with the savings which can be effected. In general, the use of automatic control can be justified in the regulation of all the operating functions that do not require an operator's judgment and that cannot be performed at less total expense by an operator.

The functions which are to be controlled automatically cannot with safety be limited to some arbitrary number. The selection should cover groups of related operations. For instance, if the coal output from a pulverizer is controlled automatically but the total combustion air is not, the fire may be put out by abnormally high excess air and be followed by an increase in fuel feed to the fireless furnace. In this example, both functions or neither of the functions should be automatically controlled. Systems that permit the operator to disconnect one function normally under automatic control should incorporate safeguards to prevent the dangers inherent in temporary partial-hand partial-automatic control.

The grouping of the instruments and the automatic control on the instrument and control panel, as well as the location of the panel in relation to the boiler, should be given careful consideration.

Grouping of instruments and controls for closely related functions is important not only in time saved for the operators but because a clear over-all view of the situation, particularly in an emergency, may impel corrective action which otherwise might not be taken.

Location of the instrument and control panel should be determined from a mental dress rehearsal of the entire plant operation. Bearing in mind that the fullest measure of operator performance will depend upon the fewest number of steps required in walking to the panel, its location should be selected after taking into account the anticipated number and qualifications of operators, their preferred location at the time each operation is performed, and the location of the equipment that will be in operation. Some panels are designed for one or more related auxiliaries, some for one complete boiler unit, some for two or more boiler units, and others as master panels for entire generating stations.

In the operation of power-plant equipment, gradualness in shifting operations is a primary requisite. If, for instance, an emergency arises, such as a lapse in good combustion which produces too rich a mixture of combustible in the furnace with a profusion of smoke, it is much more dangerous to shut down and thereby permit the furnace atmosphere to be diluted through the explosive range than to "hold fast" and gradually increase the air for combustion to a normal safe excess. The handling of situations involving low water and other temperature-stress conditions should follow the same principle of gradualness.

When a new boiler has been completed, a certain procedure should be followed in placing the boiler in operation. The steps in this procedure are: (a) "drying-out" the refractory setting; (b) "boiling-out" with chemically treated water to remove any oil or other deposits accumulated in the pressure parts during fabrication and erection; (c) "initial operation" at moderate loads; and (d) "placing on line" in normal high-duty operation. All of these steps may be taken separately, but it is common practice to combine (a) and (b), and recently in many cases (c) has been combined with (b). Chemical cleaning

(see page 21-31 ff) is sometimes used as a supplement to, or as a substitute for, boiling-out.

Upon completion of major repairs or alterations of old units, some of the procedural steps noted are usually required. Those which should be taken as proper prerequisites in again placing the unit in service will depend upon the extent of the repairs or alterations and are easily recognized.

The following rules, covering the procedure outlined above, are offered for the general guidance of the operator and may be modified to comply with the requirements and preferences at different plants or to meet the demands of the future.

Drying-Out, Boiling-Out, and Initial Operation

General

By consultation with the erector of a new boiler or the repair supervisor for an old boiler and by his own inspection, the operator must verify the completeness and fitness of the unit for operation. This will establish that the unit has been thoroughly cleaned internally by mechanical means and that it has been hydrostatically tested with all attendant adjustments as outlined under *Principles and Methods of Installation*, Chapter 25.

The steam-drum manhole plates should be removed and the drum internals checked for tightness and inspected, to determine that the boiler is in satisfactory condition for boiling-out. No part or section should be closed until the man responsible for the inspection is certain that all passages to which the part permits access are free and clear of all foreign material except that which can be removed by boiling-out. All parts should be closed immediately after this approval is obtained. The gaskets used for this closure should be those intended for subsequent regular service.

Auxiliary apparatus pertaining to the firing or feeding of the boiler should be tried out as fully as possible, to make certain that it is in proper condition. This procedure should include all fans and dampers and the setting and trial of all interlocks. If an automatic combustion-control system is used, the various regulators should be set for remote hand control and checked to determine that all can function as intended. It is good practice to blow out the superheater and all steam piping when the boiler is first brought up to pressure, to make sure that no foreign material gets into prime movers or process equipment. The furnace and gas passages should be clean and free of all combustible matter.

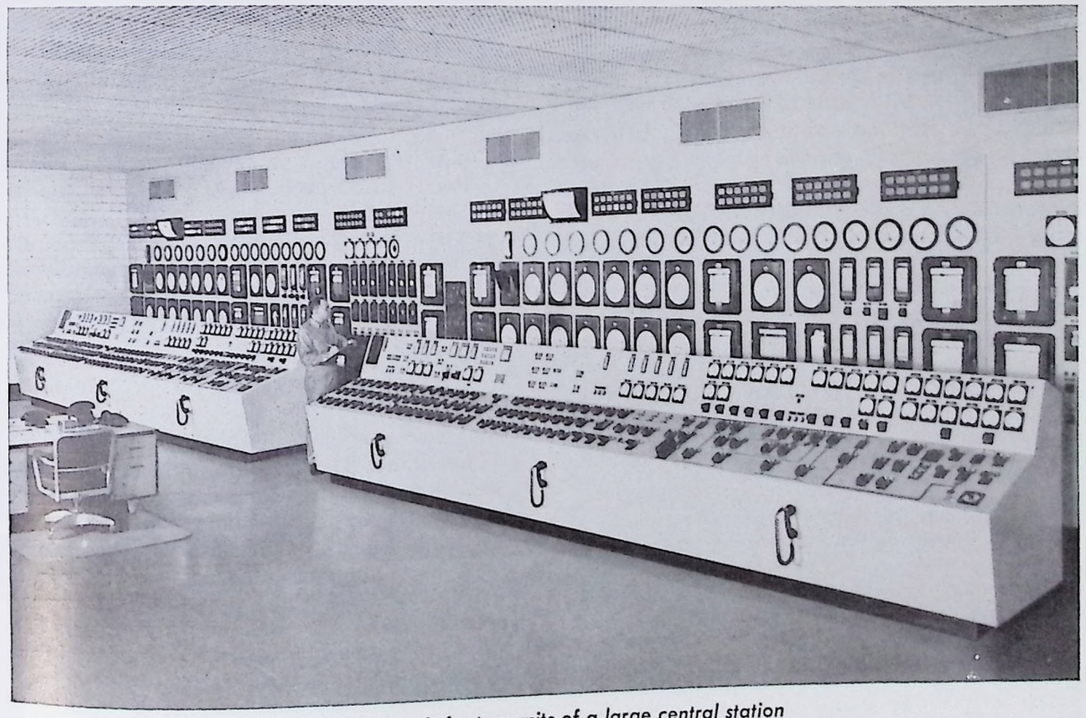


Fig. 1. *Controls for two units of a large central station*

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Valves, Water-Level Gages, and Steam Gage

After the boiler is closed, the blowoff valves, water-column and water-glass drains, and gage cocks should be examined for proper operating condition and closed; the vent valves, the water-column shutoff valves, the gage-glass shutoff valves, and the valves to the steam pressure gage should be opened; and the economizer and boiler should be filled. The water level showing in the glass should be checked by opening and closing the gage-glass drain and noting the movement in the glass. The reasons for any tendency to sluggish movement should be thoroughly investigated, and the difficulty should be corrected.

Water

The unit should be filled through the feed connections to the normal working level with water of the same quality as that of the feedwater to be used for regular service.

To avoid damaging temperature strains and excessive localized heating or chilling of the drum, the difference between the water and the drum temperatures should be not more than 50 F. In most cases in regular service, the temperature of the boiler drum will be between 100 and 200 F, and a source of 150 F feedwater should be provided. Feedwater of this temperature can be diluted with cold water before feeding, when the drum is at a lower temperature than 100 F, such as may be the case in starting-up a new boiler or on other infrequent occasions. If hot water is used to fill the economizer and boiler, extreme care must be taken to feed slowly and vent sufficiently, so that no pressure is built up by the attendant flashing into steam.

Care should be taken to use only properly treated and deaerated water, so that optimum conditions will be in force from the start in limiting the formation of scale, corrosion, and other difficulties caused by improperly conditioned water.

To avoid excessively high levels when the water swells on firing the unit, the water level should be adjusted (except for Type H boilers noted below) so that only 1 in. shows in the gage. For B&W Type H boilers, the water level should be raised to the center line of the gage glass to keep the temperature of the top rows of steam-and-water circulators within safe limits during the starting-up period. When raising pressure on the boiler, it is also important to establish a steam flow through these circulators by opening the drum vent and superheater drains. This procedure applies to starting-up from cold or bringing the unit up from bank.

To prevent water from being carried over into the superheater and the steam line, with the danger of undue temperature strains in the superheater and damage to the steam-driven equipment, extreme care must be taken to avoid an abnormally high water level. As precautionary measures, the water level should never be allowed to rise to the top of the glass during the starting-up period, and the boiler should be blown down to one-half glass shortly before line pressure is reached.

Chemicals. The chemicals used for boiling-out are completely dissolved in water before they enter the unit. The dissolved chemicals are pumped into the boiler through the chemical-feed connection or, as a second choice, through the boiler vent without passing through a dry pipe, dry pan, or scrubber. For small units lacking a pumping system for this purpose, the chemical solution may be poured through the upper-drum manholes after bringing the water level up to about half way between the bottom of the drum and the manhole.

With all the water in the unit at the concentration noted below, a good cleaning job can be done with any one of the four following combinations of chemicals:

1. Trisodium phosphate ($Na_3PO_4 \cdot 12H_2O$)	2500 ppm
Caustic soda (NaOH)	2500 ppm
2. Trisodium phosphate ($Na_3PO_4 \cdot 12H_2O$)	5000 ppm
Caustic soda (NaOH)	500 ppm
3. Trisodium phosphate ($Na_3PO_4 \cdot 12H_2O$)	1500 ppm
Caustic soda (NaOH)	1500 ppm
Soda ash (Na_2CO_3)	1500 ppm
4. Caustic soda (NaOH)	3000 ppm
Soda ash (Na_2CO_3)	3000 ppm

In addition to the above combinations of chemicals, some operators use sodium sulfate and sodium sulfite, respectively, as precautionary measures to prevent caustic embrittlement and oxygen corrosion. However, since the unit is not sufficiently susceptible to attack from either source because of the relatively short duration of the boiling-out period, the additional treatment is hardly justified.

Drying-Out the Setting

The refractories in the setting of the unit must be dried out with a slow fire for a period ranging from three days to one week, depending upon the size of the setting, the thickness of the walls, and the conditions under which the refractories were installed. The fire should be sufficient to hold the water in the boiler at such a temperature that a light vapor is apparent at the open vent on the boiler drum. Water in the boiler should be carried at the normal level, and the superheater drains should be wide open.

Block-covered and stud-tube floors of slag-tap furnaces are covered with chrome ore carefully applied to a thickness of 3 in. or more. This in turn should be covered with not less than 3 in. of dry granulated slag or stoker ash, free of tramp iron. The slag or stoker-ash covering is provided chiefly as a protection for the chrome ore surface against traffic during erection and premature surface drying.

When boiling-out procedure is combined with the process of drying-out the refractory setting, the fires should be operated for two or three days at an extremely low rate, to warm the setting gradually, and then increased to the minimum that will hold the pressure required for boiling-out. The rate of firing must be so regulated that the gas temperature entering the superheater does not exceed 900 F.

For small low-pressure jobs it is often convenient to fire with wood thrown on the grates or furnace bottom. For higher-pressure and larger installations it is impracticable to fire with other than the combustion equipment and fuel to be used in regular service.

The fuel-burning equipment should be placed in service in accordance with instructions issued by the manufacturer for the type of system installed and with rigid observance of all the recognized rules of safety applying to the fuel used. Of special importance, when burning gaseous, liquid, or pulverized fuels, are the requisites to purge the setting before lighting or relighting the first burner and to use a torch for lighting the individual burners as they are placed in service.

Precautions. To summarize, the following important precautions should be taken before lighting the fire:

1. Remove all structural timber from the furnace and from the gas passages. A glowing ember may ignite explosive gases, which can accumulate if the combustion equipment is not operated with care while firing at low rates during boiling-out.
2. Make connections so that all blowoff and drain valves can be used effectively and without danger to the personnel.
3. See that all dampers are in good working order and so set on the shafts that their positions are correctly indicated.
4. Make sure that the gage glasses and other means of indicating the water level are in good working order.
5. Make sure that the fuel-burning equipment, temporary or otherwise, is in such shape that it can be operated in strict accordance with all recognized rules of safety applying to the particular fuel used.
6. Make sure that the unit is filled with water and chemicals in accordance with the foregoing instructions.
7. Before lighting any fires, completely purge the setting of all combustible gas by circulating air through the setting for at least 5 minutes at the rate of approximately one fourth of the requirement for maximum capacity of the unit. For units with dampered superheat control, both the superheater and gas by-pass dampers should be wide open during the purging.

Firing

After the fires have been lighted, it is important that they be held continuously at the minimum rates suitable for drying-out and boiling-out, respectively. To avoid puffs or perhaps a serious explosion, it is especially important to maintain stable combustion conditions. Should the fire be accidentally or purposely extinguished, it is absolutely necessary to purge the setting for at least 5 minutes before relighting.

With oil firing at very low rates in furnaces designed for high capacities, vapor leaving the furnace may condense in the colder parts of the setting. To lessen the possibility of the occurrence of this condition, steam or air atomizers should be used, or temporary, relatively small, refractory furnaces should be built around the oil burners in the regular furnace to aid in supporting ignition and in completing the combustion of the oil before the gases reach the tube surfaces.

When gas firing is used, it is important to make sure that the burners are operated at the gas pressures required to maintain stable ignition. Every precaution should be taken to insure the maintenance of the required gas-pressure regulation or control to absolutely prevent even temporary lapses in stable ignition.

It is of critical importance to make sure that all instruments and control equipment are in good working order before the unit is in operation. Instrument accuracy should be checked as completely as possible. All circuits and signals—electrical, air, hydraulic, or other types—should be verified. Each of the different pieces of equipment controlled from these circuits should be started up or tried out doubly to confirm that each of the controls is correctly installed. There have been many instances of serious accidents traceable directly to the failure to delay initial operation until the necessary instruments and control equipment were in proper operating condition.

As a final warning, there is no substitute for a good fireman. Operation of the firing equipment should be entrusted only to experienced operators who are fully familiar with the type of system in use and who are keenly aware of the hazards involved in firing at the low rates required. All too often, the firing during drying-out and boiling-out is looked upon simply as a routine to be endured and completed with minimum inconvenience, and

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consequently the new equipment is frequently handled by inexperienced operators or men pressed into service from the construction force. If such a course is pursued, not only does the job of drying-out and boiling-out, so important to subsequent successful performance, suffer but the safety of personnel and equipment is seriously endangered.

Pressure and Duration of Boiling-Out

The firing rate should be gradually increased to raise the pressure in the unit slowly to the boiling-out pressure. The recommended pressure for boiling-out is 50% of the normal operating pressure.

Duration of the boiling-out period will vary with the amount of cleaning required. However, experience indicates that a period of 48 hr at the recommended pressure and chemical concentration is usually sufficient to clean a unit. A clear appearance of the blowdown at the end of this time will serve as a guide for ending the process, but the subsequent internal inspection is the final step in determining how thoroughly the boiler has been cleaned and whether it will be necessary to repeat the process.

Protection of Heat Transfer Components

Economizers. For economizers with recirculating connections (Fig. 4), the valves are kept open during boiling-out and while pressure is being raised preparatory to initial operation. When such units are on the line and have taken water for approximately 5 minutes, these recirculating valves should be closed. Before feeding water, if there is a possibility of steam forming in the economizer inlet header while the pressure is being raised, the header vent should be kept open until it is certain that the accumulated steam has been blown out. Economizers of this type, designed for the high-gas-temperature zones, will automatically induce circulation of the boiler water. In economizers without recirculating connections, suitable for low-gas-temperature zones, the only flow during boiling-out is from economizer blowdown and the water fed to make up for boiler blowdown. In this type, part of the chemical solution for boiling-out should be introduced into the feed line before the economizer, to bring the concentration up to that recommended for the water in the boiler proper.

Superheaters. Before lighting the fire, all superheater drains should be opened wide. For units with dampered steam-temperature control, the superheater dampers should be closed, except for a slight opening to avoid gas pockets, while the dampers over the by-pass section should be wide open.

During the boiling-out procedure, superheaters should not be flooded; in fact, the object is to keep them clear of water at all times during the various phases of the starting-up procedure insofar as drainability permits, so that the superheater tube metal will be cooled by some flow of steam through the tubes.

For nondrainable units, it may be necessary to open the superheater dampers to provide the heat required to boil any condensate out of the tubes. It may also be necessary to reduce the temperature and increase the over-all firing time in order to boil out the water trapped in the tubes. Obviously, there will be no steam flow through a tube partially filled with water until the water has been cleared, and those portions of the tube not in contact with the water will be subjected to excessive temperatures and attendant danger unless the entering gas temperature is held to safe limits to boil the water out of superheater elements.

Thermocouples installed in the outlet legs of the superheater tubes, where they pass into the unheated vestibule, will read saturation temperature until the tubes are cleared and there is a flow of steam through them. The temperature of these outlet legs rises sharply, immediately after this flow is established, to superheated steam temperature which, under normal rate of starting-up, will run 75 F or more above saturation. Experience indicates that a thermocouple on each outside tube and one on a tube near the center are sufficient, for the usual installations, to judge the clearing of water from the entire superheater. It may be necessary, for large nondrainable superheaters with a relatively large number of loops in series, to continue the starting-up firing rates after the boiler is up to line pressure and delivering steam.

Throughout the starting-up period, including drying-out, boiling-out, and bringing the unit on the line, the firing should be regulated to keep the temperatures of the superheater tubes within safe limits for the tube metal and to avoid high stresses from abnormal temperature differences between one part and another. The tube-metal temperatures are normally kept below the temperature usually reached at the maximum design capacity by bleeding steam through the superheater-outlet-header drains, bypassing gas around the superheater on units with dampered superheat control (for superheaters of the drainable type), and regulating the time of bringing the boiler up to line pressure.

A direct method for determining the time is to use thermocouples to measure the superheater tube temperatures and gradually increase the rate of firing, always keeping the temperature within safe limits. An approximate method is to regulate the firing to limit the temperature of the gas entering the superheater to 900 F for steel tubes and 950 F for

alloy tubes. When it is impossible to regulate the firing to the minimum rates required, firing should be in intermittent periods of 10- to 15-minute duration at 5- to 10-minute intervals. To avoid momentarily exceeding the gas temperatures just noted, it may be necessary to shorten the duration of, and to lengthen the intervals between, firing periods.

For both boiling-out and afterward, whenever the unit is being brought up to pressure for equipment operation, the vent on the steam drum should be closed when the boiler pressure has risen to 25 psi. At this time also, all superheater drains except those in the outlet section of the outlet header should be throttled sufficiently to reduce the amount of steam waste, yet left open enough to keep the header clear of condensate. However, periodic wide-open blowing of all superheater drains is assurance against stoppage of a continuously partially open valve. The superheater outlet drains should normally be left wide open until the boiler is on the line, although it may be necessary to throttle these drains to maintain the desired rate of pressure rise without exceeding the permissible firing range.

The time required to place a boiler in service depends upon its size and type. Some small low-pressure boilers with superheaters may be placed in service, with comfortable margins of safety, within 1 to 1½ hours. See *Protection of Drums*, below, for large high-pressure units with superheaters.

To serve as a schedule for the operators in arranging a starting-up program, test measurements should be carefully translated into terms of available operating indices, such as time as a function of steam pressure, firing-equipment speeds, gas temperatures normally indicated, air pressures, drafts, and valve openings and closures.

Reheaters. Since they are not connected with any supply of steam from the boiler, reheaters, which are in effect superheaters interposed between the outlet of a high-pressure turbine and the inlet of a low-pressure turbine, require special consideration during boiling-out. In some cases a temporary connection is made from the boiler drum or the superheater outlet, so that some steam may be blown through the reheater drains. If this is done, the reheater should be handled in the same way as the superheater. In some of the latest installations, however, the reheaters have been left isolated during boiling-out and have been given no attention except that the entering gas temperature has been limited to 900 F.

Before lighting the fire, all reheater drains should be opened wide, and care should be taken to see that the reheater is drained as completely as its design will permit. For units with dampered steam-temperature control, the reheater dampers should be closed except for a slight opening to avoid gas pockets, while the dampers over the by-pass section should be wide open. If the reheater is nondrainable, the water or condensate that has accumulated in the tubes must be boiled out during the firing period, and it may be necessary to open the reheater dampers to provide the heat required for this purpose. In boiling-out a nondrainable reheater, the procedure outlined above for nondrainable superheaters should be followed except that,

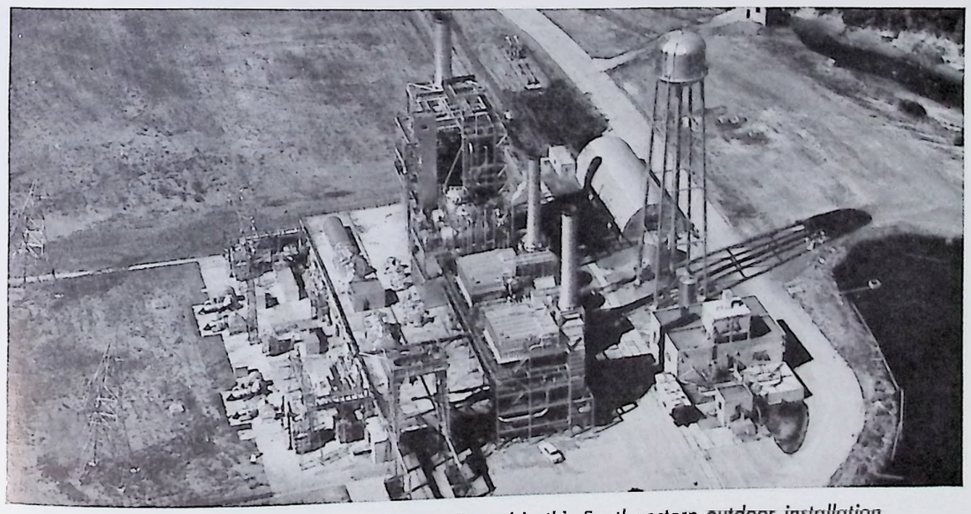


Fig. 2. *Savings in building costs are effected in this Southwestern outdoor installation*

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instead of thermocouples on only three tubes, they should be located every few feet across the reheater as necessary to obtain representative coverage.

Reheater drains discharging to the condenser should remain open until a load equivalent to 5 to 10% of the full rated steam flow of the boiler has been reached. Reheater drains discharging to the atmosphere should remain open until it is time to start the vacuum pumps of the condenser, when they should be closed to prevent air from being drawn into the condensate system.

During the pressure-raising period, before there is steam flow through the reheater, the firing rate must be controlled, as noted above for superheaters, to limit gas temperatures, with the additional limitation that the temperature of the gas entering the reheater must not exceed 900 F until steam flow through the reheater has been established. Usually, the restrictions on firing rate necessary to protect the superheater will provide adequate protection for the reheater. A critically important difference between the superheater and the reheater is the time of establishing normal steam flow. In the superheater, normal steam flow starts with the unit up to pressure and delivering steam through the superheater to the line. In the reheater, normal steam flow may correspond approximately to that of the superheater or it may be completely unrelated. The starting-up gas-temperature limitation for reheaters must be adhered to, therefore, until normal steam flow has been established through the reheater itself.

Protection of Drums

In the majority of installations, the time required to place a boiler in service is limited to the time necessary for the protection of the superheater and the reheater against overheating. However, with higher steam pressures and thicker boiler drums, the time for both starting-up and shutting-down may be determined by the time required to limit the thermal stresses in the drums.

To comply with the limiting temperature differences determined by simple analyses of the thermal stresses, various rules have been formulated in the past, based on generally accepted operating practice. One rule widely used limited both the starting-up and the shutting-down rates to a change in saturation temperature not to exceed 100 F per hr, in the belief that this rate would limit the differential temperature of the drum-shell metal to not over an assigned value of 100 F. Recent investigations indicate that these rules are generally too conservative and that heating and cooling rates may be much greater than the old empirical value of 100 F without exceeding safe limits for the drum-metal thermal stresses.

Based on stress analyses and supporting field data, it is now recommended that the principal criterion for allowable rates of heating and cooling should be the relation between the temperature drop, $\triangle t_{to}$, through the drum wall at any location and the temperature difference, $\triangle t_{TB}$, between temperatures of the outside surface of the drum at top and bottom. The limiting values in this relationship are given in Chart $\triangle t$ for drum-shell thicknesses up to 6½ in. and for drum diameters of 60 in. and above.

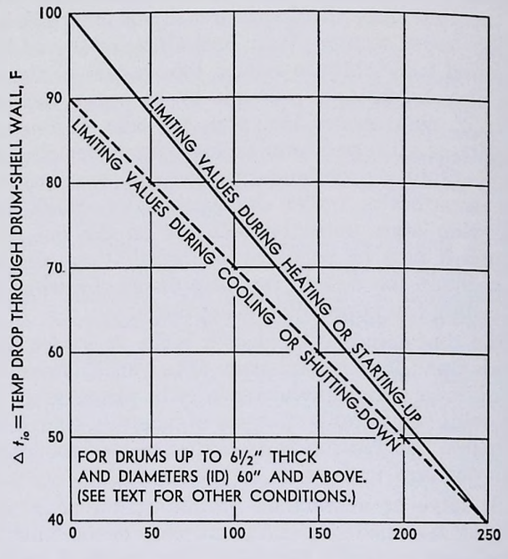


Chart $\triangle t$. Recommended guide for control of thermal stresses in drums during starting-up and shutting-down

To determine the temperature differences during the periods of starting-up and shutting-down, it is necessary to establish continuously and accurately the outside and inside surface temperatures of the drum shell.

Temperature of the outside of the drum shell is best determined by thermocouples (see Chapter 6). At least six thermocouples should be installed outside at the following locations:

(a) At top center of drum.
(b) About 24 in. from (a), measured circumferentially on the same circumference.
(c) At bottom center of drum.
(d) On a boiler or circulating tube about 6 in. from drum shell.
(e) At top of drum near one end, outside the zone of baffles and scrubbers.
(f) At bottom of drum, same circumference as (e).

Temperature of the inside surface of the drum can be established as follows:

The inside surface temperature of the portion of the drum shell covered by water may be taken as the temperature of the water, and this may be considered to be the saturation temperature corresponding to the pressure as long as the pressure still shows on the steam gage. Before the pressure shows on the gage (heating, during start-up) or after pressure has dropped to zero (cooling, during shutdown), the temperature of the water is estimated from the temperature indicated by thermocouple (d) and from the temperature and the amount of any feedwater added.

During starting-up with pressure rising (pressures above zero gage), the inside surface temperature of the portion of the drum shell not covered by water may be taken as saturation temperature corresponding to the pressure. On the other hand, during shutting-down with pressure decreasing, the inside surface temperature of the portion of the drum shell not covered by water may be assumed to be equal to the temperature of the outside surface, since for this condition the temperature gradient through the drum shell is considered negligible because of the insulating effect of a relatively low-conductivity steam film in contact with the inner surface.

In starting-up before pressure begins to show on the gage and during shutting-down after pressure has dropped to zero, the temperature of the inside drum-shell surface not covered by water may be taken, in either case, as that of the outside surface.

For all values of $\triangle t_{to}$ and $\triangle t_{TB}$, established simultaneously in each case during start-up or shutdown, that fall on or below the respective curves in Chart $\triangle t$, the drum thermal stresses are considered to be within safe limits. By control of the rates of operation to come close to but not above the respective curves, large boiler units may be started up and shut down in much less time than was formerly considered possible. Many high-pressure units are now brought up to 1000 psi in approximately 1 hr, and continued practice has shown no ill effects. Field experience has actually indicated a more uniform expansion of pressure parts, since higher initial firing rates help to promote circulation in all circuits.

By assigning an arbitrary conservative limit of 100 F for the value of $\triangle t_{TB}$, Chart $\triangle t$ may also be used to approximate the maximum allowable value of $\triangle t_{to}$ for drum diameters less than 60 in. Extensive calculations are required to establish more exact values of the temperature differentials for smaller drums and for all drums thicker than 6½ in.

In cold boiler rooms or for outdoor units, the drum shell above the water level remains cool until steam is generated, and adherence to even the maximum permissible values of the top curve in Chart $\triangle t$ will rightly limit the rate of start-up. However, the start-up can be accelerated by completely filling the drum with warm boiler water, taking care to prevent spilling over into the superheater by the use of a high-level gage glass or a vent with a visible overflow connection, or both. Thermocouple (b) can also be used as an indication of water level, since there is a sharp change in temperature as the warm water approaches the top of the drum. If the water is not hot enough to warm the top of the drum, it should be heated to the desired temperature by firing the boiler under normal water-level conditions. Heating the boiler water to 175 F is usually sufficient to warm the top of the drum to approximately 150 F and so comply with the temperature differential requirements of Chart $\triangle t$ during the subsequent start-up.

During the filling of the drum beyond the range of the operating gage glass and during the draining of the drum back to operating level, the fires should be extinguished.

As previously noted, to prevent harmful thermal stresses from local temperature differences, it is always good practice in filling boilers to keep the temperature of the entering water at not more than 50 F above or below the temperature of the drums.

Accelerated Shutdown. The most effective way to cool down large boilers uniformly and quickly is gradually to reduce the pressure to the minimum value permitted by plant conditions as the load is reduced. When firing is stopped, all pressure parts will then be at a uniform reduced temperature. If this procedure is not possible, the fires should be extinguished to allow the pressure to drop naturally. The forced-draft fans can also be used to speed the cooling down of large boilers. Whichever method is used, the temperature differences should not exceed the limiting values of the cooling curve in Chart $\triangle t$.

The simplest way to keep the cooling-down rates within permissible values is to allow the pressure to drop until the temperature differential reaches the limit set in Chart $\triangle t$. Then, with the fires extinguished, the water level should be raised to the top of the drum, avoiding overflow into the superheater. In this condition, with both the feedwater and the fires shut off, the boiler will continue to cool down. When temperature differentials again reach the permissible limits in Chart $\triangle t$, feedwater should be injected to make up for the small amount of shrinkage from cooling. During the shutting-down period, the drum should be vented to remove any steam which tends to collect at the top of the drum.

Checking Steam Gage

As soon as steam is heard blowing from the vent, the steam gage connection should be checked, to see that it is clear, by closing the shutoff cock and opening the drain valve. As soon as the line is clear,

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the drain valve should be closed and the gage cock opened. Action of the gage needle should be watched, as the pressure increases, to make sure that the gage itself is functioning properly.

If the steam gage has not been set with a dead-weight tester, it should be reset to allow for the water leg in the gage line, using a test gage in parallel. The setting should be made at a pressure approximately equal to that at which the boiler is normally operated.

Nonreturn Valve

During the early stages of the starting-up period, after boiling-out, the drain on the boiler side of the main steam stop valve should be opened and the nonreturn valve stem raised slightly, so that the steam piping to the main steam stop valve may be gradually warmed with increasing boiler pressure. If the nonreturn valve is equipped with a by-pass line, the by-pass may be used to warm up the piping.

Blowing-Down

Each piece of equipment should be blown down at least once every 8 hr during boiling-out. The total amount of water blown down should be the equivalent of a difference in level of one-half gage glass, divided equally between all main blowdowns. If necessary, water should be added between the blowing of individual valves to hold the level within safe limits. At the end of the blowdown, the water should be brought up to the normal working level and replenished with chemicals to the recommended concentration.

Also, as previously suggested, all drain or bleed valves, such as superheater drains that are being carried only partially open, should be opened wide during the blowdown periods for a short blow.

Procedure after Boiling-Out

General. After completing the boiling-out it has been customary to cool the boiler unit slowly, to drain, flush out, remove all foreign materials, and prepare the equipment for its initial operation.

In many instances this procedure has been superseded by combining initial operation at moderate loads with boiling-out. In this method, boiling-out is completed in the normal way. Then, instead of cooling down the boiler, the concentration of solids in the boiler water is reduced to that desired in regular operation by blowing-down and by feeding properly treated feedwater. Boiler pressure is then raised to full operating pressure, the safety valves are set, and the unit is put into normal operation at moderate steam output. The unit is operated at this relatively low steam output for a week or ten days and is then cooled and shut down for cleaning and preparation for normal high-duty operation.

Safety Valves. The settings of all safety valves should be checked, and adjusted if necessary, immediately after reaching full operating pressure while operating at moderate steam output subsequent to boiling-out. In general, the superheater safety valves should be set to open, under any load conditions, before the safety valves on the boiler drum. For units of high rating, this may be qualified to the extent of setting the superheater safety valves so that the pressure drop through the superheater will cause a valve on the boiler drum to open first at a rating high enough so that, in spite of its unloading capacity, sufficient steam will flow through the superheater to protect the elements from overheating. Such provision should be made only after a careful study of the temperatures involved.

Any safety valve requiring attention that cannot be given while the unit is hot and under pressure can be taken care of during the subsequent final preparation for regular service. Following any work of this nature on safety valves, a careful check must be made to be sure that all gags have been removed and that the valves have been resealed.

Each safety valve should be checked, before taking the boiler off the line for an outage and after the boiler is put back on the line following an outage, either by raising the steam pressure until the safety valve blows or by gently raising the valve from its seat by hand.

Gage Glasses. During the initial period of operation it is particularly important that the gage glass be clean and the water level clearly visible. If the gage glass is found to be dirty during the inspection following the boiling-out, the fittings should be disassembled, cleaned, and repacked. Connections between the gage glass and the drum and all gage fittings and water-column drains should be confirmed free and clear. In the case of high-pressure gage glasses, if the cleaning requires peeling off so many sheets of mica that the remaining bundle is less than standard thickness, the mica bundle should be entirely replaced.

In general, water columns should be blown down once every 8 hr by very gradual wide-open blowing, and the manner in which the water leaves and returns in the gage glass should be observed. If the action of the water in the glass is sluggish, the gage glass should be blown sufficiently to clear the obstruction. Because of the high maintenance cost in glass, mica, and fittings, full wide-open gage-glass blowing is considered unnecessary unless the operator believes that it is required by the action of the water in the gage glass. For high-pressure boilers with flat-glass gage glasses, maintenance considerations may make it desirable to rely on the fluctuations in indicated water level, without frequent blowing-down of the water column, to show

whether or not the water-column and gage-glass connections are clear.

Gage glasses should be kept clean and unscratched. Leakage at the gage-glass connections will cause rapid deterioration and breakage. Leakage at the water column or its drain connections will cause a false level in the gage glass. The glass should be well illuminated under all conditions.

When the high-pressure flat-type gage glass is dismantled, particular attention should be given to the mica sheets. As long as these are not allowed to wear through, the glass should remain serviceable, but the mica bundle should be renewed if necessary to maintain standard thickness. A hooded light, located near the base at the rear and shining upward at an angle to the surface of the water, or various special lighting arrangements are used to illuminate this type of gage glass. The water level will appear as a bright spot in a darkened column. There must be sufficient indirect lighting at the glass to show, at all times, the full length of the glass and the location of the water level.

The difference in refractive index between water and vapor (or air) is used extensively to indicate a strong contrasting color distinction between the water- and the vapor-filled portions of a gage glass. In this bicolor gage, colored glass strips, properly positioned between the source of illumination and the gage-glass windows which act as prisms, show the water-filled portions as a distinct green. The vapor portion shows as a distinct red. Thus, an empty gage glass appears red from top to bottom, and a full glass appears green. The refractive principle is used to show contrasting shades of black and white in other gage-glass assemblies.

Gasketed Joints. For handhole or manhole fittings with seats inside, the internal pressure helps to compress the gaskets and hold the joints tight. Because of high temperature, it is impractical and even dangerous to pull up the fittings with the boiler under full steam pressure. Gasketed joints of this type in new boilers are pulled up during erection under the hydrostatic test pressure of 1¼ times design pressure. The recommended practice after erection, where handholes or manholes have been opened or disturbed, is to renew the gaskets and pull up the nuts with the wrench provided for the purpose, under a hydrostatic pressure a few per cent less than normal full steam-operating pressure.

Internal pressure tends to open the joints of bolted safety-valve flanges, pipe flanges in drain lines, etc. In a new boiler, joints of this type are pulled up before and during the hydrostatic test. They should be retightened immediately after the first raising and dropping of steam pressure, when the pressure is zero but while the unit is still hot. To obtain the desired temperature, it may be necessary to blow out drain lines while the unit is first up to full steam pressure, then shut the drains and pull up the joints while the line is still hot.

Operating Company's Responsibility

The boiling-out, handling of gage glasses, pulling-up of gasketed joints, checking of the setting of safety valves, and any other preparations for regular operation are usually considered the responsibility of the operating company.

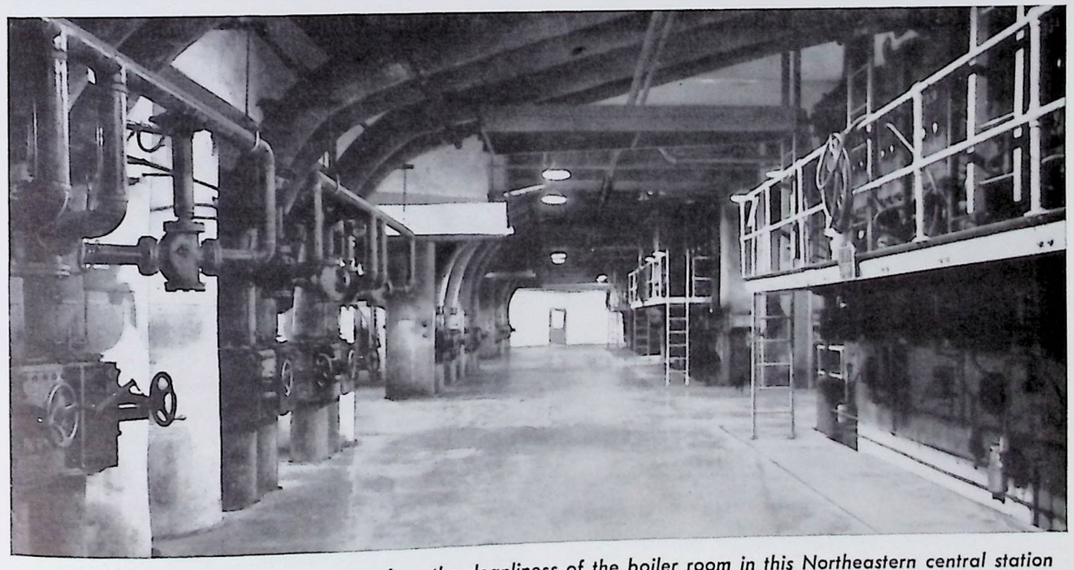


Fig. 3. *Good housekeeping is evident from the cleanliness of the boiler room in this Northeastern central station*

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Placing Boiler Unit on Line

In placing a boiler unit on the line and in normal high-duty service, additional instructions, covering items not previously mentioned or fully covered, follow:

Boiler and Superheater

When the boiler pressure has risen almost to line pressure, the by-pass around the main steam stop valve should be opened to equalize pressures and temperatures in the piping; the main steam stop valve should then be opened. The drains on the inlet superheater header should be opened wide as an added precaution to remove all condensate. As the boiler reaches line pressure and is actually steaming, the nonreturn-valve stem should be slowly raised to full-open position. This procedure will prevent the possibility of a surge in the flow of steam, which would result from the sudden opening to full capacity of a slightly sticking nonreturn valve were the boiler cut in wholly automatically. When two nonreturn valves are used in parallel on a unit equipped with a double-outlet superheater, it is important to make sure that both valves are open wide and functioning, immediately after the boiler pressure reaches line pressure and before the fire is increased to bring up the load. Overheating part of the superheater may result from neglect of this precaution.

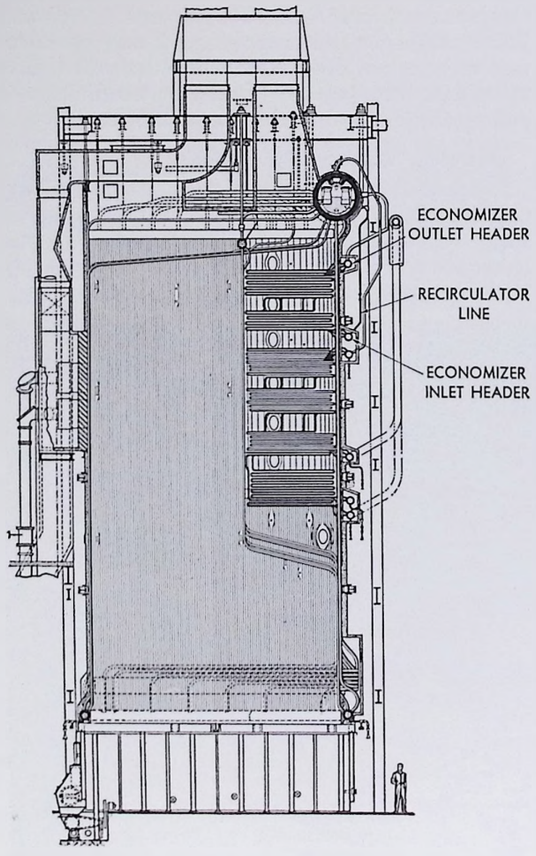


Fig. 4. *Radiant boiler with upflow economizer. Note recirculator connection used in starting-up*

After the boiler is on the line, all superheater drain valves may be closed. The superheater drains should be handled in the same general way when the boiler is being brought on the line after having been banked. When a superheater is being drained, care must be taken to avoid back pressure in blowdown lines to which the drains are connected. The drains cannot function under back pressure, and the resultant hammer may damage the superheater and boiler.

On units provided with superheat-control dampers, the gas dampers over the superheater should be slowly opened, and the by-pass dampers should then be regulated to give the desired steam temperature.

During periods of normal operation, protection of the superheater depends upon the flow through it of sufficient steam at proper temperature. The temperature is the important variable, since the quantity is usually fixed at the total steam output of the unit.

In low-steam-temperature units it is common practice not to provide special means for the control of superheat but to depend upon adjustment in firing conditions to maintain adequately uniform steam temperature. Excess air has a profound influence on such control, but care must be exercised in its use. As superheats approach the higher limits, it is mandatory to provide special means, such as attemperators or gas-control dampers, to maintain within close limits the temperature of the steam that cools the superheater tubes.

Economizer

Instructions noted above (page 21-6) should be followed in handling the economizer.

Reheater

The reheater will go into normal operation in tandem with the superheater if, in bringing the boiler unit up to line pressure, the steam path is open from the high-pressure throttle through the high-pressure turbine, the intercepting valve, and the low-pressure turbine to the condenser.

If the intercepting valve is closed while the boiler unit is brought up to line pressure and the reheater is isolated from the normal steam flow, it is very important to hold the firing rate within the starting-up range until the steam flow through the reheater is properly related to the firing rate.

During periods of normal operation, the reheater is cooled by the steam that is directed through it as a function of certain valve openings or arrangement in the steam piping between the high-steam-pressure section of the station or the high-pressure-turbine exhaust and the low-steam-pressure section of the station or the low-pressure-turbine inlet. When steam is flowing through the reheater, it will be protected against overheating if the final steam temperature, as regulated by attemperator, gas by-pass dampers, firing rate, burner adjustment, etc., does not exceed design limits. Control of temperatures of superheated and reheated steam must be coordinated, particularly during periods of load change, since these temperatures are both affected by operating variables, such as feedwater temperature, firing rate, and furnace cleanliness. Turbine-operating considerations may require reduced steam pressure during periods following start-up or preceding shutdown.

In an isolated single-boiler single-turbine station, the steam flow during normal operation through the reheater is related to the firing rate almost as dependently as that through the high-pressure superheater, and the important variable of steam temperature is controlled similarly to that of the high-pressure steam.

In stations where each boiler-turbine unit is not isolated and where the primary steam passing through the high-pressure superheater is mixed with that from other boilers, the amount of steam which is directed through the reheater can become completely unrelated to the firing rate. In such plants it is imperative to provide, in addition to the controls for reheat steam temperature, safeguards against subnormal steam flow through the reheater.

Attemperator

When the boiler is being brought up to pressure and put on the line, and before passing steam through any attemperator piping which has been out of service, the drain valves should be opened sufficiently to make sure that all condensate is removed. To avoid wasting steam needlessly, the drain valves should not be left open. The spray nozzle of a spray attemperator should be inspected occasionally, to make sure that it has not become enlarged by water erosion.

Gasketed Joints

The procedure applying to gasketed joints, as outlined above, should be followed if it is found necessary to renew gaskets or to tighten these joints.

Water Level

Normal water level in the boiler gage glass is at its mid-point. In case of heavy concentration and high rating, the water should be at the lowest level consistent with the assurance of constant feed and the conditions for emergency feed supply. The boiler should be fed only through connections provided or approved by the manufacturer.

Gage Glasses and Water Columns

The procedure applying to gage glasses, as outlined above, should be followed as may be necessary.

Safety Valves

Before a new unit is considered ready for operation in regular service, the instructions noted above should be followed in handling all safety valves.

Blowing-Down

Many plants use a chemical analysis of the water from the boiler as a guide to determine the amount and frequency of blowing-down. Where such analyses are not made, the boiler should be blown down at least once every 24 hr, and the amount of blowdown will depend upon the class of feedwater and the amount of steam generated.

In the ordinary sense, economizers, water-cooled furnace walls, and stoker side-wall or bridge-wall water boxes should never be blown down while the unit is in active steaming service. Blowdown valves for this type of equipment serve only as drain valves and should be padlocked while the boiler is in service.

Intermittent blowing-down of the boilers should be done only at reduced or moderate rates of steaming, and preferably when the fires are banked. The blowdown valves or any drain valve that permits hot water or steam under pressure to flow into a relatively cold line should be opened slowly and gradually. For instance, in opening a drain line of say ½-in. wall thickness from a 1200-psi unit, at least 1 minute should be allowed from the time the hot medium starts coming through until the line is opened fully for the complete blow. Proportionately less time is required for lower pressures and thicknesses. Completion of the desired amount of the blowdown is determined by noting the water level in the gage glass.

Unless otherwise specified by the valve manufacturer, the arrangements of the blowoff valves and the sequence of operation should be:

For blowoff cock and cast-iron valve: Arrange the valve in the outside position, with the pressure over the seat. The valve should be opened first and closed last.

For tandem steel valves: Arrange with the pressure under the seat of the inner valve and over the seat of the outer valve. The inner valve should be opened first and closed last.

Use of continuous blowdown has increased because of the disadvantages of blowing-down in-

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termittently, such as variations in concentration and the inadvisability of blowing-down large quantities of water while the boiler is operating at high duty. In using the continuous method, a small-size connection with a suitable regulating valve or orifice is so located that blowing-down can be done without adversely affecting boiler circulation. A small quantity of water is discharged continuously, and the quantity can be easily adjusted as called for by routine chemical analyses. Continuous blowing-down of furnace walls and stoker side walls or bridge-wall water boxes is permissible and occasionally advisable, provided the quantity is not great enough to interfere with water circulation. This procedure, however, should be adopted only with the approval of the boiler manufacturer.

Soot Blowing

Cleanliness, internally and externally, is a very important factor in maintaining the efficiency and capacity of a boiler unit. Systematic cleaning should be included as a regular feature in the operation of all steam plants. Internal cleaning is done periodically when the boiler unit is taken out of service for routine inspection and cleaning. The outer surface of the tubes should be blown free of soot at regular intervals during operation of the unit, using soot blowers or hand lances.

The amount and frequency of cleaning the external heating surfaces will depend upon the type of operation and the fuel. Before blowing soot, the supply lines to the blowing elements or the hand lances should be thoroughly drained of water. Soot blowers should be operated only while the burners are in service at a sufficiently high rate of combustion to insure a low oxygen content in the flue gases and to prevent the fires from going out. When using the soot blowers, it is best to operate the boiler between half load and full load. In operating at low load the furnace will be cooler, and some unburned combustible from the furnace may be deposited on tubes or may collect in pockets. Agitation of these combustibles during soot blowing at low load may result in an explosion. It is, therefore, important that all areas of the gas passes be thoroughly swept with flue gas, thus purging any combustible gas from pockets.

Before starting the soot blowers, the furnace draft should be increased just enough, without an increase in excess air, to prevent a positive pressure in the furnace when the blowers are in operation. During soot blowing, to maintain the proper relation between air flow and fuel, both should be controlled by hand. The unit should never be operated with hand control for fuel and automatic control for air, or vice versa, since fuel and air will not change together and loss of ignition may result from a sudden or wide variation in the automatic control.

Soot blowing should not be attempted when the boiler is out of service until all sources of hot materials, in either the furnace or setting, which might cause an explosion by igniting the dust blown into suspension, have been eliminated. Under this condition (boiler cold), air should be used for soot blowing. There is a possibility of external corrosion of the pressure parts if steam is used.

When permanent soot blowers are installed, care must be taken to maintain the proper positions of the nozzles in relation to the boiler tubes. If the nozzles become displaced so that there is a direct impingement of steam on the tubes, erosion of the metal will result. Also, in operating a soot blower or steam lance, to avoid erosion no portion of the heating surface should be blown steadily for too long a period.

Boiler or superheater surfaces adjacent to furnaces fired with slag-forming fuels should be cleaned at intervals, using slag-cleaning equipment or blowers installed for the purpose. The frequency of cleaning should be as required for commercial efficiency, as indicated by draft loss and exit gas temperatures.

Furnace-Floor Covering

If combination pulverized-fuel, oil, and gas-fired furnaces with block-covered floors are to be operated for extended periods on either oil or gas alone, the floor blocks should be protected with a layer of refractory tile or ash, to eliminate such block maintenance as may result from high-duty operation with no covering whatsoever on the blocks. When plant conditions necessitate a shift from pulverized fuel to oil or gas without shutting down, it is generally considered advisable to omit cleaning the floor before the shift is made, so as to leave a thin coating of ash for block protection. Short periods of operation with oil or gas alone do not justify the trouble or inconvenience of providing protection for floor blocks.

Abnormal Operation

Low Water

In the case of "low water" in a boiler, from whatever cause, the primary objective is to extinguish the fire in the quickest possible manner. Should the water in the glass drop from sight, any decision to keep the boiler in operation, even for a brief period, should be made by the person who is in responsible charge and who is fully aware of the conditions existing at the time. In most cases it is exceedingly dangerous to feed water into a hot boiler that has suffered from low water.

Boiler-Tube Failures

It is well recognized that a boiler drum empty of water yet under steam pressure, as would be the

case immediately following a tube rupture, will be subject to serious temperature stresses if the drum metal is quenched by the flow of relatively cold feedwater in direct contact with it. All drums, thick or thin, may be expected to withstand a certain number of such temperature shocks without developing incipient cracks, but it is impossible to predict accurately how many shocks of whatever violence may be withstood before actual rupture of the metal will occur. In the protection of a boiler unit, safety of the drum should be the primary consideration of the operator. The amount of damage to the drum, resulting from the relatively low heat capacity and low heat-emission capacity of any slag remaining in the furnace or setting, is not serious compared with the damage that can be caused by feeding water into a hot boiler.

The above will apply to boilers fired by pulverized fuel, oil, or gas, as well as any other type of fuel which permits stopping the combustion almost instantly.

In units served by stokers or any other type of fuel-burning equipment where the combustion cannot be stopped almost instantly, severe damage to the furnace tubes and boiler structure may result if the flow of feedwater is not continued for some time after a tube rupture.

Accordingly, in case of a tube failure of sufficient magnitude to require immediate removal of the boiler from service, the procedure outlined below for the two distinct classes is recommended:

1. For units fired by pulverized fuel, oil, or gas:
   - (a) Shut off the fire.
   - (b) Shut down forced-draft fans and primary-air fans.
   - (c) Shut off completely any output of steam from the boiler. It is of the utmost importance that this action be taken as quickly as possible in order to prevent a sudden drop in pressure with the corresponding sudden drop in temperature of the water within the boiler.
   - (d) Shut off the supply of feedwater to the boiler.
   - (e) Adjust the flow of air through the unit to the minimum permissible consistent with safety from water or steam flowing into the boiler room. This will prevent unnecessary and uneven chilling of the pressure parts.
   - (f) After the pressure has dropped to zero, the subsequent or continued cooling should follow the procedure of a normal scheduled shutdown. The boiler should not be emptied until the furnace has cooled to a temperature at which one can enter and remain in the furnace.
   - (g) After the unit has cooled, a complete inspection should be made to determine the damage. Special care should be taken in the examination of the drums, and the extent of any incipient cracks found should be accurately determined. Before the unit is returned to service, all traces of any such cracks in the drums should be eliminated, and the drums as well as other pressure parts, after necessary repairs, should be hydrostatically tested and approved by the proper authorities.

     Shutting off the feedwater after dropping the load, as recommended in (d) above, is for the protection of the drums against temperature shocks caused by the introduction of relatively cold feedwater. By the time the water level has receded to the point where the furnace tubes are dry, there will not be enough heat from slag or refractory, it is believed, to do any damage to the pressure parts.
2. For stoker-fired units or those with fuel-burning equipment where the combustion cannot be stopped almost instantly:
   - (a) Shut off the supply of fuel and the supply of combustion air. Also, use whatever means are available to effectively smother the fire without danger of subsequent explosion.
   - (b) Follow the procedure outlined in (c) of the recommendations listed under 1. above for units fired by pulverized fuel, oil, or gas.
   - (c) Try to maintain normal water level in the damaged unit by adjusting the supply of feedwater to the maximum allowable without encroaching on the amount required by the other boilers. Shut off the supply of feedwater entirely just as soon as possible after combustion is reduced to the point where there is no longer any danger of seriously overheating the pressure parts. This should be possible within 15 minutes following the rupture.
   - (d) Follow the procedure outlined in (e), (f), and (g) of the recommendations listed under 1. above for units fired by pulverized fuel, oil, or gas.

Where a tube leak is not of such magnitude that immediate removal of the unit from service is necessary but will be required in due course, the procedure outlined in 1. and 2., above, for emergency cases should be followed as soon as other phases of plant operation can be adjusted to anticipate the outage.

In the recommended procedure outlined above, no reference is made to operating pressure or drum

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thickness. While this procedure was developed particularly for the protection of thick drums, it is also applicable to low-pressure thin-drum units and therefore should be used in all cases, superseding older methods. However, before attempting to modernize the procedure for existing low-pressure units, careful consideration should be given to the status of personnel, equipment, and operating conditions. Such a survey may indicate that the possible gain in adopting this procedure for low-pressure units may be negligible compared with the potential loss from confusion in disrupting an old established routine.

Emergency Shutdown—General Considerations

In the era of small low-pressure boiler units, it was the custom during an emergency to simply shut the unit down. If the emergency was of short duration, the unit could be restored to service almost immediately, and the loss occasioned by the brief shutdown was ordinarily small.

With modern high-pressure high-temperature reheat units, a complete shutdown may be very costly, not only because of the length of time necessary to restore to service all the equipment involved but also because of the dangerous confusion which may result from deciding too quickly whether or not a complete shutdown is necessary.

If, for instance, the electrical load is lost suddenly from a reheat turbine unit and it is impossible to determine immediately the expected duration of the lapse, it is of great advantage to have controls which will cut back the firing rate to a predetermined safe minimum for the reheater tubes, exhausting the steam generated through the boiler and superheater safety valves. With the superheater protected partially by the reduced firing rate and partially by the flow of steam through the superheater safety valves, and with the reheater protected simply by the low firing rate, the necessary auxiliaries can be held in service until a proper evaluation of the emergency can be made.

Shutting-Down, Inspection, and Cleaning

Shutdown

Boiler units should be taken out of service at regular intervals for internal inspection, cleaning, and repair.

In shutting-down the fuel-burning, fuel-preparation, and draft equipment, while taking a boiler out of service, the instructions for the type of firing system used should be followed. To avoid possible damage from overheating or an explosion during or subsequent to shutting-down, careful attention should be given, before and after shutdown, to the condition of the fuel equipment, particularly with reference to temperature and cleanliness.

After the firing equipment and fans are out of service, the dampers, including the superheater and superheater by-pass dampers when fitted, should be closed to permit the unit to cool slowly and uniformly. When feed to the boiler is no longer required and the nonreturn steam valve has closed, the main steam stop valve should be closed, and, for economizers with recirculating connections, the recirculator valve should be opened. The pressure should be allowed to drop naturally, without open vents or other intentional means of taking steam from the unit to hasten the lowering of the steam pressure. The superheater drains, however, should be opened sufficiently to keep all condensate out of headers. To avoid brickwork difficulties and unnecessary stresses in the pressure parts, cooling of the furnace should not be hastened by allowing large quantities of cool air to pass through the setting. After the furnace brickwork has lost its color, the use of a moderate amount of induced or natural draft is permissible.

To prevent a vacuum from condensation of steam within the boiler, the steam drum vent valve should be opened when the steam pressure has dropped to about 25 psi. A vacuum within the unit may cause future leakage of gasketed joints.

Since shutting-down large high-pressure boilers by the foregoing procedure takes such a long time, it is desirable, in many instances, to adopt other methods, such as thermocouples distributed over the pressure parts of the unit, to establish the maximum allowable rate of cooling. Specific procedures for alternative methods to accelerate shutdown are outlined above under *Protection of Drums.*

The boiler should not be emptied until the furnace has cooled to a temperature at which one can enter and remain in the furnace. If the heating surfaces are to be cleaned externally, the operation should be carried out as completely as possible before the fire has died out and should be finished by means of air lances and scrapers after the unit has cooled off.

Boilers equipped with reheaters are shut down in much the same manner as outlined above. The principal additional precaution made necessary by the reheater is that the reheater drains should not be opened until after the intercepting valve on the low-pressure turbine has been closed, so that air will not be drawn into the condensate system.

With the use of the reheat cycle in its high-temperature form, it may be desirable, in shutting down, to control the rates of temperature drop in the different parts of the turbine and boiler equipment. Exact procedures cannot properly be given as applicable to the many different plants. However, it is appropriate to suggest that shutting-down the high-pressure turbine, low-pressure turbine, superheater, and reheaters be coordinated with the dropping of pressure in the boiler proper, so that the rates of

change in temperatures of the heavy boiler sections, even though their absolute values are lower, may be coordinated with the rates of change of the higher temperatures in the steam system.

Inspection and Cleaning—Internal

The interval between outages for routine inspection and cleaning is determined by experience with the conditions of operation, water, and fuel.

Prior to entering the boiler for inspection or cleaning, particular care must be taken to guard against steam or hot water entering the unit through open drains or blowdown valves connected with similar parts of other units under pressure at the time, or through the careless opening of any steam or feed valve to the unit. Serious and sometimes fatal cases of scalding have resulted from neglect of these precautions.

So that the problem of water treatment may receive proper and immediate attention, the pressure parts should first be examined internally for signs of scale, oil corrosion, or other potential sources of trouble directly attributable to the condition of the water. Before the unit is returned to service, the deposits should be removed by air- or water-driven turbine cleaners or by chemical cleaning.

During the internal inspection, the physical condition should also be noted to make sure that all parts and appurtenances, such as feed connections, steam separators, safety valves, drains, blowdowns, and other fittings for taking steam or water to, through, and from the complete unit, are in place as designed, and that they can function as intended. Particular care should be taken to see that the connections to the water column, gage glass, and steam pressure gage are in good condition and unobstructed, and that these fittings are in a satisfactory condition for the purpose intended.

Inspection and Cleaning—External

Extreme care must be taken, when entering the setting of a coal-fired unit, to avoid injury from heavy slag accumulations falling from the walls or roof. If possible, such accumulations should be removed before anyone enters the setting.

The external surfaces should be inspected for signs of overheating, leakage, corrosion, wastage, and any other distress which will impair the safety of the pressure parts. The inspection should also cover the condition of all supports and hangers, gas baffles, refractories, and safety appliances. Repairs should be made to insure safety as well as continuity of service.

To supplement the cleaning done while the unit was in operation, such cleaning as is necessary and to the extent permitted by access should be done during the outage by brushing, scraping, air-lancing, water-washing, steaming, or any other means required by the conditions. Since the cleaning of slag from refractories will inevitably tear away the surface to a damaging extent, only as little cleaning of refractories or refractory-faced surfaces as is compatible with safety should be attempted with the unit out of service.

When water or steam is used for washing external heating surface, the best time for this operation is just before the unit is put back in service. The heating-up of the unit will thoroughly dry the water out of all crevices, pockets, porous refractories, or other places where it might cause corro-

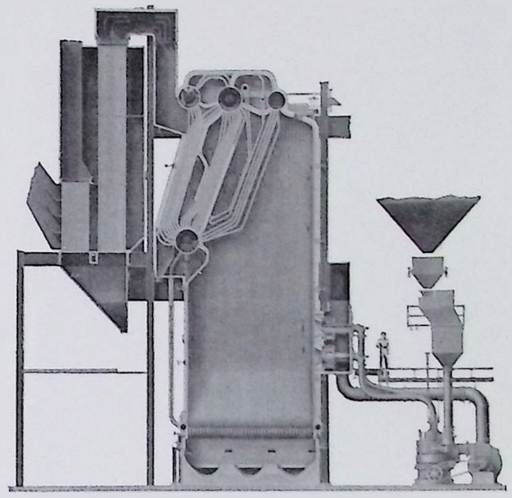


Fig. 5. *Stirling boiler, built in capacities to 500,000 lb of steam per hr, for pressures to 1050 psi and steam temperatures to 950 F*

sion. If water or steam washing is done at some other time during an outage, the unit should be immediately fired sufficiently to dry it thoroughly. Where the sulfur content of the fuel is high, it is advisable, before washing, to neutralize the acid condition of the soot by wetting down the surfaces exposed to the gases with an alkali solution.

The inspection should also include the fuel-burning equipment, especially those parts which are not accessible when the unit is in service. Repairs or replacements should be made, so that the equipment will function properly and thus reduce the possibility of interruptions during subsequent operation.

A check of all soot blowers should be a regular maintenance item during scheduled boiler outages. Tubes in zones of soot blowers should be inspected closely for any signs of metal loss caused by fly-ash or steam-cutting. Direct impingement of steam on tubes may result from the shifting of the soot blower

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element or from growth of blower elements, which may move the nozzles out of register with the spaces between the tubes. High soot blower pressure may cause polishing of tubes, especially on coal-fired jobs where the fly-ash is highly abrasive. In this case, the blowing pressure should be reduced. The pressure

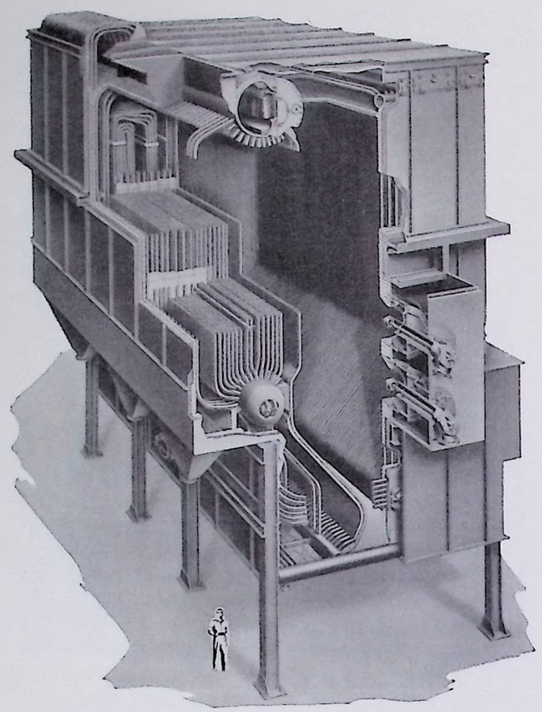


Fig. 6. Integral-Furnace boiler (may also be top supported), built in capacities to 350,000 lb of steam per hr, for pressures to 1050 psi and steam temperatures to 910 F

can be reduced either by adjusting the orifice or by installing a smaller orifice plate, depending upon the type of head on the soot blower unit. In hot locations the steam nozzles of the elements may shrink; they should therefore be checked for correct inside diameter and enlarged if necessary.

Special Inspection—State or Insurance

In addition to routine operating inspections, a thorough inspection concerned with the safety of the unit should be made yearly at the time of the visit of the insurance inspector or state inspector. This inspection should include a most careful search for any evidence of internal and external corrosion, leakage of seams, leakage of expanded, screwed, or welded joints, and overheating. The condition of the structural supports should also be established at this time. It is well to bear in mind that the parts which are most likely to be slighted, because of soot accumulation or difficulty of access, may be the very parts in which trouble will develop, and it may be necessary to remove small sections of brickwork or casing to make the inspection complete.

Idle Boilers

To minimize the possibility of corrosion, boilers to be held out of service must be carefully prepared for the idle period and closely watched during the outage.

Protection of Internal Surfaces—Dry Storage

When it is known that a boiler is to be idle for a considerable length of time and that a brief period will be allowed for preparation to return it to service, the dry-storage method is recommended. In this method the unit is emptied, thoroughly cleaned internally and externally, dried, and then closed up tight to exclude both moisture and air. Trays of lime, silica gel, or other moisture absorbent may be placed in the drums to draw off the moisture in the air trapped by the closing-up of the boiler. To insure against overflow of corrosive liquid after the moisture has been absorbed, the pans should not be more than three-quarters full of the dry absorbent. Particular care must be taken to prevent water, steam, or air leakage into the unit. Periodic inspections should be made to insure that no corrosive action is taking place, and at these times the absorbent should be replenished as required.

The same general method of preparing the boiler for out-of-service periods should be followed for the superheater, economizer, and reheater sections.

Protection of Internal Surfaces—Wet Storage

Where boilers go into standby service but must be available for immediate operation, they should be steamed in service, so that boiler water conditions may be stabilized and oxygen bubbles cleared from the boiler surfaces. The boiler should be brought down in rating slowly, and the water level should be raised as high in the glass as is consistent with safe operation while still unloading some steam to the line. The hydrate alkalinity should be built up to a minimum of 400 ppm. Sodium sulfite in the amount of 100 ppm will aid in preventing oxygen corrosion.

When the pressure, as indicated by the boiler vent, has practically disappeared but before there is a vacuum as a result of cooling off, water should be slowly fed to the boiler until it is completely full and the superheater is filled by the spill-over. This make-up water must be deaerated. The drain on the boiler side of the main steam stop valve should be wide open, so that any water getting past the nonreturn valve will not accumulate on the boiler

side of the stop valve. Filling should be continued until there is a pressure of at least 10 psi against the nonreturn valve. Close watch should be kept to avoid building any undue pressure against the nonreturn valve from leaking feedwater valves.

Boiler connections should be checked for leakage, and frequent boiler-water samples should be taken and analyzed. If the analysis indicates that the hydrate alkalinity has dropped below 250 ppm, the water in the drum should be lowered to the normal operating level, chemicals should be injected to bring the hydrate alkalinity back to 400 ppm, the boiler should be steamed sufficiently to circulate the added chemicals, and then the process of wet storage should be completed as previously described.

Even with the above precautions, the possibility of corrosion must not be underestimated, and no boiler should be held in wet storage without emptying for periodic inspection.

Protection of Idle Superheaters and Reheaters

Superheaters during out-of-service periods, except when the complete wet-storage method is used, should always be kept dry and closed from contact with the air. Reheaters should be taken care of in the same manner as superheaters except that the wet-storage method is never used. *No unit should be wet-stored when there is any probability that the temperature will drop to the freezing point.*

Protection of External Surfaces

The external surfaces of all pressure parts and other metallic surfaces should be completely cleaned of all ash and soot deposits. Other ash, cinder, or fuel accumulations, such as on the grates, furnace floors, shelf baffles, and in the cinder hoppers, should be removed.

The settings should then be closed and kept closed, but periodic inspection should be made regularly to guard against sweating and corrosion of the external surfaces of the pressure parts, particularly when the wet-storage method is used. In this connection it may be found necessary to use coke stoves, or similar heating devices, at convenient points to keep all metal surfaces above the dew point, particularly during protracted spells of damp weather accompanied by rising temperature.

Treatment of Water for Steam Boilers

FEEDWATER and boiler-water treatment is considered generally as an undertaking for the chemist. The problems involved are of course predominantly chemical, but their most effective solutions are so interrelated with safety, mechanical design, power plant operation, and cost that the best efforts of the operating and the design engineers also are essential.

Early Development

Prior to 1900, boiler units were small, spares were a recognized necessity, the knowledge of water treatment was meager, and it was normal practice to operate for a few weeks and then shut down for mechanical cleaning by men easily trained for the work.

As greater duty was imposed upon boilers, the formation of hard scale and heavy sludge caused sudden failures, fatal accidents, forced shutdowns, and far too costly maintenance. Thus it became apparent that this early procedure was inadequate.

Operation with Treatment

Between 1900 and 1920, the use of external treating plants became fairly well established as the best means of handling feedwaters high in scale-forming and sludge-forming materials. When the total sludge, including that resulting from the treatment, was considered insufficient to justify an external treating plant, internal treatment, principally with soda ash, was used.

Ordinarily, the high limits of all the constituents in the water were controlled by intermittent blowdown to avoid priming.

Until about 1918, dissolved oxygen was considered in the more general sense of air in the feedwater. With the idea of heating the feedwater close enough to the boiling point to drive off most of the air, feed pipes were usually arranged to discharge above the normal water level. Usually the protection afforded the metal surfaces by deposits of sludge or scale rendered the remaining air harmless. However, corrosion often occurred at or near the feed inlets because of feeding unheated water.

Near the end of the period of water treatment preceding 1920, surface condensers, because of their superiority in most instances, had replaced jet condensers. Evaporators for make-up had come into prominence, and, for central station work, the feedwater-treating problem narrowed down to one of treating the make-up to the evaporators and of conditioning the boiler water so that it would render nonscale-forming the small amount of condenser leakage and evaporator carry-over entering the system.

The clean boiler surfaces resulting from surface condensate and evaporated make-up became very vulnerable to attack from air not released from the

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feedwater in the steam space of the boiler. At about the same time, the steel tube economizer made its appearance and very rapidly forced the development of deaerators, designed at first to remove air, then most of the dissolved oxygen, and finally all of the dissolved oxygen from the feedwater.

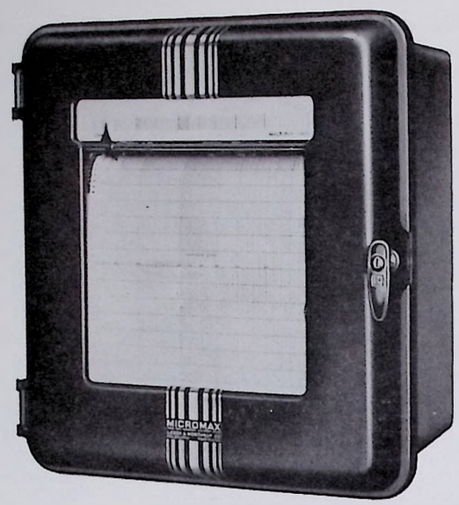


Fig. 7. *Strip-chart recorder for a continuous record of dissolved oxygen and pH values of boiler water*

Caustic Embrittlement—Early Periods

As early as 1895 there was evidence of the peculiar type of boiler-drum cracking predominantly intergranular, later termed caustic embrittlement. Between then and 1912 an increasing number of boilers developed this type of cracking. In 1912 a boiler exploded at Bloomington, Ill., and cracking developed in the boiler at the University of Illinois. At that time Isaac Harter, Sr., presented to S. W. Parr of the University of Illinois the results of all the research work which he had carried out for The Babcock & Wilcox Co.° This research proved that the drum plate failures were not from faulty material, as generally believed, but were caused by the use of waters high in caustic and low in other materials, particularly sodium sulfate. The attention of all boiler manufacturers, boiler users, and feedwater-treatment specialists was focused on this difficulty by the research work carried on by S. W. Parr and later by F. G. Straub. However, it was not until the middle twenties that the majority of those interested was convinced that the cause of the trouble was not faulty material and poor workmanship.

° University of Illinois Bulletin 94, January 1917, by S. W. Parr.

It was noted during the early investigations that this embrittlement of boiler plate was probably caused by hydrogen, but it was called caustic embrittlement because in boiler operation it was always associated with waters high in caustic. The research also showed that the cracking developed only in locations subjected to stresses approaching the yield point and in the presence of caustic concentrations far beyond those in the main part of the boiler water. High localized stresses were obviously present in every riveted joint, and the customary calking of the seams, on the outside only, permitted boiler water to enter between the plates of the joints, while extremely small leaks through the calking edges allowed an abnormal concentration of boiler-water salts within the joints.

In 1923 B&W inaugurated inside calking to prevent the entrance and concentration of boiler water within the joints. This attempt to build immunity against caustic embrittlement into the boiler structure proved successful and was particularly appropriate at that time, because 530 psi was then being projected and the established preventive of maintaining certain sulfate-alkalinity ratios† by water treatment was based on experience at pressures not much above 250 psi. Notwithstanding the adoption of inside calking, the old sulfate-alkalinity ratios, which had proved so effective at low pressures, were continued as an additional safeguard and were extrapolated from their necessary minima of 1 to 1 for pressures up to 150 psi, and 2 to 1 for pressures of 150 to 250, to a new ratio of 3 to 1 for pressures above 250 psi.

The Company does not know of any case of caustic embrittlement failure in any completely inside calked water-tube boiler drum nor, where the sulfate-alkalinity ratios have been maintained, in any outside-calked water-tube boiler drum.

The acceptance of welded drums in 1930 carried forward the success of the inside-calked drums by completely eliminating not only the possibility of caustic concentrations at the points of high stress but also the high stresses themselves.

In the late thirties a peculiar form of tube failure from intercrystalline cracking again raised the question of caustic embrittlement, but, since the newer problem involves several more recently developed factors, it will be referred to later.

Summary of Developments Prior to 1923

By the early twenties the feed and boiler-water situation in central stations had developed into the general pattern of surface condensers, evaporators for make-up, treated water for evaporators, feedwater heaters which eliminated much of the dissolved oxygen, boiler waters treated with sodium carbonate to maintain positive alkalinities, and the maintenance of sulfate-alkalinity ratios equal at least to those noted above. Boilers were shut down for cleaning the internal surfaces of sludge at reasonable periods governed concurrently by the need for making repairs to the equipment.

† This term evolved from the formerly used expression "sodium sulfate to equivalent sodium carbonate ratio."

Boilers operating at high duty with high percentages of make-up containing more than 34 to 85 ppm (2 to 5 grains per gal by the term used until the late twenties) of sludge-forming materials were equipped with external treatment plants. Boilers operating at low duty and boilers operating at high duty with high percentages of make-up having total sludge-forming materials of less than 34 to 85 ppm were taken care of by internal treatment to maintain 255 to 340 ppm (15 to 20 grains per gal) of alkalinity together with the sulfate-alkalinity ratios already given.

Oil was kept out of the feedwater except in those infrequent instances where the operators found from experience that, because of some good fortune in the combination of feedwater quality, oil composition or blend, and limited heat inputs to the highest-duty tubes, they were able to avoid the ill effects.

From Soda Ash to Sodium Phosphate

The results of the comprehensive research by R. E. Hall° on the use of the various sodium phosphates instead of sodium carbonate for treating boiler water, started in the latter part of 1922 while the first 530 psi boiler units were being designed and extending to 1926, had a profound effect on the entire course of feed and boiler-water treatment. Within a very short time soda ash was almost completely superseded by the sodium phosphates in the treatment of boiler water. The two outstanding advantages of the phosphate treatment are: 1) the calcium phosphate precipitated as sludge is much less soluble and much more flocculent than the carbonate, and 2) the phosphate radical does not break down as the carbonate does, and therefore all of it is available for the treatment of the hard scale-forming constitutents. With the phosphate treatment the alkalinities, which had been considered so important in the prevention of corrosion, were maintained by the addition of caustic rather than by the hydrolysis of sodium carbonate to caustic.

In 1938 Whirl and Purcell† extended the use of phosphate to maintain alkalinity, generally, without the addition of caustic. With reference to alkalinity, an optimum relationship was recognized between phosphate concentration (PO_4 in ppm) and pH value, where there is no free caustic other than that produced by ionization of the sodium phosphate —the hydroxyl ions being more abundant and therefore overbalancing the hydrogen ions. Plotting these optimum values as a curve (see Fig. 8), they worked out a control which permitted adherence to this relationship. If the pH value rose above the curve, indicating the presence of free caustic, it was decreased by adding either phosphoric acid or one of the sodium hydrogen phosphates, depending upon whether at the same time the phosphate concentration was too high or too low. On the other hand, if the pH value dropped below the curve, it was increased by adding caustic or trisodium phosphate, or a combination of them, depending upon whether at the same time the phosphate concentration was too high or too low.

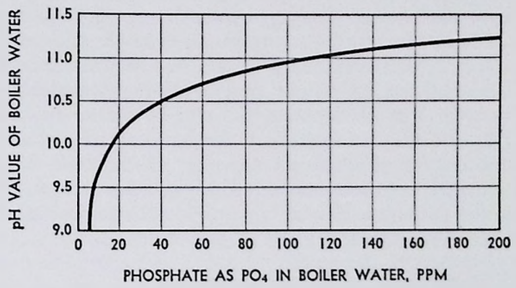


Fig. 8. *pH-phosphate relationship maintained for "coordinated phosphate" treatment of boiler water*

This treatment, known as the "coordinated phosphate" method, was developed primarily to produce a boiler water which would not be embrittling, thereby preventing further trouble in existing riveted drums and also providing insurance against this difficulty in newer units.

In 1942 relationships were developed for the use of potassium phosphate instead of sodium phosphate. The purpose was to establish the use of a reagent of greater solubility at the higher temperatures and one which did not have the inverse solubility characteristics of some of the sodium salts at these temperatures. Potassium salts are more expensive than the corresponding sodium compounds, and while the results have been successful in each of their limited number of applications, the greater expense has not yet justified their widespread use.

Modern Development for Natural Circulation

Boiler Design versus Water Treatment

In the late thirties, with the sharp increase in demand for more power at greater efficiency, there

° Carnegie Institute of Technology Bulletin 24, published in 1927.

† Proceedings of the Engineers' Society of Western Pennsylvania, Civil Section, November 1942.

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was a rapid development in high-duty, high-pressure, high-capacity, and high-temperature boilers. The fuller use of stage heating resulted in increased pickup of copper and iron by the pure high-temperature condensate passing through the feed lines and feedwater-heating systems. Increased difficulties from tube overheating and corrosion led to greater activity in boiler-water treatment and control. This also gave further emphasis to the need of close collaboration between the boiler designer and the water-treatment specialist, so that the limiting conditions under which each must work to correct these difficulties, by providing adequate circulation and water treatment, would be thoroughly understood and appreciated.

Responsibility of Boiler Manufacturer

The boiler manufacturer's responsibility is so to design and build the boiler unit that, with the water level showing in the gage glass, there will be adequate circulation during operation to insure all heat-absorbing surfaces being wetted on the water side. Under these conditions and with the water-side surfaces free of deposits, the rise in temperatures, $\triangle t$, of these surfaces above the temperature of saturation for the steam-and-water mixture will be equal to the heat transferred through the tube wall to the mixture in Btu/sq ft, hr divided by the boiling-water film conductance in Btu/sq ft, hr, F.

Responsibility of Operating Company

The responsibility of the operating company and its feedwater advisors is so to treat the feed and boiler waters as to prevent the deposition of solids on the water side of the heat-absorbing surfaces and also to prevent internal corrosion within the unit. The effect of a thin layer of solids in elevating tube-metal temperature is discussed in Chapter 29 and illustrated in Fig. 16 of that chapter.

If there is a uniform accumulation of flocculent sludge so disposed that it points to no weak spot in design or in treatment, then sludge should be looked upon as a measure of feedwater quality and as a limitation of operating periods between cleaning.

Coordinated Responsibility

If the boiler unit is relatively free of sludge, except in some one or more isolated spots which cause starving of certain tubes, then the implication is a discontinuity in the rate of water flow which fosters the build-up, and the problem is primarily one of design—notwithstanding that by altering the chemistry of the water treatment the design weakness may be overcome at less cost.

In instances where the feasibility of a projected installation depends upon the cost of making the water supply suitable for use in a boiler, the coordinated efforts of the chemist and the boiler designer are essential for best results. These instances almost invariably call for closely evaluated limits on what constitutes suitable water for given designs of boilers and what rearrangement of boiler designs or performances might be made to adapt them to one of the several possible choices of water treatment. If it is compulsory to use a high percentage of sludge-forming make-up, it is wise to be conservative in the choice of the boiler and the intensity of heat input.

In the subsequent practical solutions of the problems which arise in pursuing these projects, it is most important for each of the two coordinators to review very critically his previous conclusions, to determine—in the light of the newly developed information—where readjustment of conditions under his jurisdiction might currently be made or what the final limits are to correction by readjustment. By a fair assessment of the respective contributions of the engineer and the chemist, the usual result will be a coordinated answer to the problems involved, many times accompanied by an advance in the art.

Application of Responsibilities

Boiling-Water Film Coefficient. In the era of low pressures, the metal temperatures with clean tubes were so low that the boiling-water film coefficient was considered unimportant. In performance calculations a boiling-water film coefficient of 2000 Btu per sq ft, hr, F was commonly used, on the assumption that it was very conservative and with the knowledge that the adoption of this figure resulted in no harm. As pressures increased, it became necessary to use a coefficient nearer its true value to obtain the actual temperatures of the thicker-walled tubes. As late as 1949 the usual practice was to use a figure of 5000, knowing that it was conservative. In the late thirties and early forties it appeared necessary to establish a more accurate boiling-water film transfer coefficient for varying amounts of circulation, so that the responsibilities of design and water treatment, respectively, could be truly evaluated.

Research was carried out in 1943 and 1944 on an existing 1400 psi boiler of 500,000 lb of steam per hr capacity. Five special tubes were installed, connected to the furnace side-wall header and thereby cooled by natural circulation. They were so arranged that entering velocities in the different tubes varied from 1.4 to 12.0 fps. The lb of water per lb of steam leaving the low-velocity tube was 4. Heat inputs ranged between the approximate extremes of 80,000 and 180,000 Btu per sq ft, hr. Thermocouples were installed to determine both the gas-contact surface temperature and the steam-water-mixture contact surface temperature. These tests showed that the coefficient is above 20,000, which means that for a heat absorption of 80,000 Btu per sq ft, hr, the inside water-contact surface will be only 4 F above the steam-and-water mixture flowing past it. As a comparison, the temperature drop through the tube wall of a 3¼-in. tube, ¼ in. thick, is approximately 70 F at the same rate of heat absorption.

Circulation. Whether or not the circulation is adequate can often be determined by inspection without the assistance of measured tube temperatures. When, for instance, in floor tubes heated on the top, the circulation is insufficient to keep the top part of the tube wetted and cooled by contact with the steam-and-water mixture and the upper part is in contact with steam alone, a distinct waterline indicates this deficiency. If the water is *completely* depleted from an area, the resulting difficulty is usually indicated by water-level markings (see Fig. 9). Inclined tubes of sinuous-header boilers show the same type marks if the circulation is inadequate. Low-duty tubes, connected promiscuously to drums which receive discharge also from high-duty tubes, can show this same characteristic when the tubes are inclined from the vertical only a small amount. Even vertical tubes usually furnish their own evidence of guilt if the through flow of water is inadequate. There are, of course, exceptions which require the proof of measured metal temperatures, and means are available to obtain them.

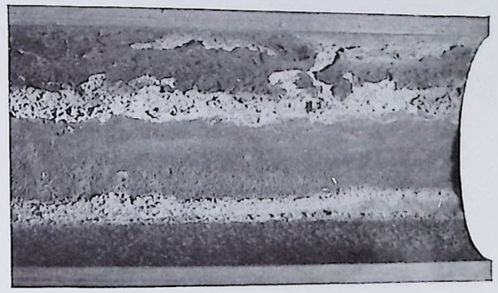


Fig. 9. *Water-level markings, "ghost lines." The central band indicates a complete depletion of water in top interior half of tube*

The one defect in circulation which can well elude visual diagnosis is the almost complete depletion of water in a steam-water mixture in contact with a tube. An "almost" complete depletion, if water conditions are correct, will ordinarily leave no deposits and no characteristic evidence that the cause of overheating the tube is inadequate circulation. Fortunately these cases are extremely rare, as the circulation rate at which overheating is possible is well under 1 lb of water per lb of steam—far below even the most daring design values.

When a clean and unobstructed tube fails from overheating with the water level showing in the glass, the boiler design should be questioned. When the failure is associated with deposits of scale on the water-contact face of the tube, the water treatment should be suspected. Nevertheless, in the latter class of failures, the charge of "poor circulation" is sometimes raised, and the finding of soluble sodium salts in the scale lends an apparent plausibility to these charges through the inaccurate use of the terms "evaporation to dryness" and "film boiling." In these allegations, however, no attempt is made to determine what the circulation rates are, nor is it possible to indicate what increased circulation rates are desirable or how even an absurdly high increase in the circulation rates could change the temperature of the tube surfaces appreciably.

Evaporation to Dryness. If a boiler or water-wall tube exposed, for instance, to high gas temperature in a modern high-duty furnace will function satisfactorily for extended periods when free of water-side deposits, it is safe to assume that the temperature of the inside wall approximates the saturation temperatures of the steam-and-water mixture within it. If a deposit, even a few thousandths of an inch thick, is allowed to form on the inside surface, the temperature of the metal under the deposit will rise perceptibly. Unless the deposit is 100% impervious, water will find its way into it or through it to the tube. This seepage water will have truly zero circulation, will be completely evaporated by the elevated temperature of the scale or of the tube, and will therefore leave in the scale, on the tube side of it, or even on the tube itself, all the solid materials it carried including, of course, the soluble salts. This is evaporation to dryness.

Film Boiling. If the tube metal under the scale is heated to the point where its differential above saturation is many times the clean-tube differential and the scale subsequently becomes dislodged during operation before the tube cools, it is possible, under some combination of high heat input and high tube-to-saturation temperature differential, for the exposed portion of the tube momentarily to resist rewetting, and its heat is transferred to the water through a film of superheated steam. This is film boiling.

Film boiling has an analogy in drops of water spluttering on a hot flatiron without appreciably cooling it and without wetting it. It cannot be produced in a high-pressure steam boiler unless the contact of the tube with the boiling water within it undergoes some lapse, caused either by artificial insulation such as scale or by an almost complete depletion of the water in the steam-and-water mixture in contact with the tube. If scale is the cause, the remedy is adequate water treatment. If water depletion is responsible, the remedy is increased water circulation.

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Sludge. Prior to World War II, the generally accepted attitude with reference to sludge was: to reduce to a minimum the amount actually and potentially entering the boiler; to condition what did enter to give the most favorable type of precipitate; to allow the precipitate to settle some place, preferably at points of very low heat input; to remove that which settled near the blowdown opening; and to ignore the indefinite remainder until the next boiler shutdown.

During the war an apparently new difficulty was experienced with sludge. With essentially identical waters, water treatment, and boiler designs, tube overheating occurred in some installations from sludge baked so hard on the water contact surfaces that it appeared almost like hard calcium sulfate scale, while other installations developed a fluffy precipitate which caused no trouble. The circulation in each case was known to be more than adequate, and the essentially identical waters were treated in exactly the same manner.

It was finally established that with some waters, if sodium phosphate is injected directly into the incoming feedwater under relatively low pH conditions, magnesium is precipitated within the boiler as magnesium phosphate, and a sticky sludge forms. On the other hand, if sodium phosphate is put into the boiler water at a point far enough from the feedwater inlet so that the treatment actually becomes part of the relatively high pH boiler water and if the incoming feedwater is then treated with this high-phosphate-bearing high-pH boiler water, the magnesium will precipitate as hydroxide, and a nonsticky sludge is produced.

This experience resulted in the eventual adoption of standards for feedwater and for chemical feed connections, which automatically permit treatment of the incoming feedwater with boiler water properly fortified elsewhere in the boiler with treating chemical.

Flame Impingement. The heat input to the boiler or water-wall tube at the point of flame impingement may be higher than that in other locations of the same general temperature zone where there is no flame impingement. The difference in heat input under the two conditions is much less than ordinarily suspected.

When tubes containing internal deposits fail from overheating in the flame-impingement zone and do not fail in the nonimpingement areas, the difference in heat inputs is naturally considered to be of critical importance, and much effort is sometimes expended in attempts to eliminate impingement. However, if no correction is made to the water conditions which permit this scale to form, these efforts are largely ineffective. Furthermore, water conditions which permit deposits to form in flame-impingement zones will permit them to form also in other zones, reducing their margin of safety so that some otherwise minor variation in operating conditions will cause delayed failures of either the same general nature or of a more subtle corrosive variety.

While the direct impingement of flame on boiler or water-wall tubes should be avoided or corrected to prevent unnecessarily increasing the duty on these parts, it has come to be recognized by most operators of modern high-duty boilers that the issue of how much scale can be tolerated has changed to one of whether scale of *any* thickness exists in the tubes.

Naturally, all tube failures from overheating can be avoided if the application of heat is sufficiently reduced, but this, of course, cannot be considered as a solution of the difficulty when it is caused by waterside scale, since it would unnecessarily restrict progress in future design and reduce the capacity and efficiency of boiler units already in operation. All parties concerned, therefore, agree that deposits which limit heat input must be prevented by proper water treatment.

Chemical Cleaning. During initial operation, large amounts of mill scale and rust are dislodged from the condensate and feedwater systems. This material accumulates in the boiler. During the same period, upsets in the water conditions may be experienced, which can cause deposit formation on the internal heating surface of the boiler. It is therefore recommended that the load on a high-pressure boiler be limited to 75% of full load during initial operation and that the unit be removed from service for chemical cleaning within a period of 3 to 6 weeks after start-up. By this procedure and with proper control of water treatment, the boiler is placed in optimum condition for full-load extended trouble-free operation.

Despite the most up-to-date control of water treatment, the majority of operating experiences reveals that deposits gradually form on the waterside of the heating surfaces. Such deposits must be periodically removed before they have reached sufficient thickness to cause any damage. Chemical cleaning for the removal of these deposits from boilers of modern design has proved effective, safe, convenient, and economical. Chemical cleaning is a refinement of the procedure followed several decades ago, when it was customary to turbine thick scale from low-pressure low-duty boilers.

Insufficient Circulation. Much indecision prevailed in the many cases of furnace floor tubes which corroded internally on their upper portions receiving heat from the furnace. These tubes showed stripes or water-level markings revealing clearly that the water-and-steam velocity was below the point at which the water separated from the mixture.

The upper parts of the tubes were therefore cooled by a slow-moving layer of steam, which became heated by amounts varying from a few to as many as 600 degrees above saturation temperature.

If the tubes failed in short periods from direct overheating and blistering with little or no corrosion, the cause was rightfully considered to be poor circulation, and the boiler designer was called upon to increase the quantity or velocity of water to the point where the entire surfaces of the tubes were wetted and cooled.

If slow corrosion occurred, there was a tendency improperly to consider the difficulty to be one of water treatment. The chemists responded wholeheartedly in these cases to a problem which was one fundamentally of boiler design and by decreasing the caustic soda in the boiler water were able, in the cases of slight attack, to arrest the corrosive action. This chemical study and accomplishment led to a movement to change the long-existing belief in high alkalinity and reduce it to a minimum. That part of the endeavor which brought forth the coordinated phosphate control was obviously a step forward in a new direction. On the other hand, a misguided component of this work, which proceeded toward generally reduced alkalinities or pH values with the more generally prevailing conception of caustic and phosphate treatment, caused a great deal of trouble and controversy, because the long established treating relationships had been compromised for the sole purpose of neutralizing a difficulty caused by an inadequate rate of water flow through the tubes.

Intergranular Corrosion of Tubes. In the late thirties a new type of tube failure developed in high-pressure boilers. Patches of very hard iron oxide, formed on the water-contact faces, were found to be partly embedded in the tube metal. Beneath the hard iron oxide, the unoxidized metal was seriously interwoven with intercrystalline cracks. Between the cracked area and the gas-contact face of the tube, the metal was sound.

Samples checked at the Company's research laboratory revealed copper in the intercrystalline cracks, the amount found being greatest nearer the water-contact face of the tube and decreasing to practically zero adjacent to the good outside layer of steel. Whether the copper played any part in the attack or simply followed into the cracks afterward has not been established.

This tube deterioration resulted in several failures, which took the form of pieces breaking loose, without any tube swelling or physical evidence of overheating. The attack occurred in a number of different plants in different types of boilers, but it was confined in all cases to the two general locations of high heat input and of expanded ends of tubes.

Understandably, both boiler design and water treatment were in question. In some cases boiler circulation was increased without any measurable decrease in the trouble. Emulating the attempted cure for attack in floor tubes with poor circulation, alkalinities were dropped, resulting, very promptly, in greater attack. Just as impressively, the attack lessened when the alkalinity was increased. In another case where the dissolved oxygen in the feed was appreciable, the operator claimed that increasing the pH value resulted in more attack.

Many of the cases were cleared up entirely by complete and continuous elimination of dissolved oxygen in the water and by maintaining pH values of between 11.0 and 11.5. Others were arrested by periodic acid cleaning of the units.

The exact diagnosis of this intergranular attack has not been made. Nevertheless the difficulty can be avoided, perhaps by methods newly developed and preferable to the old, but surely by the steadfast adherence to the proved principles of high pH values, zero dissolved-oxygen values, and clean boilers.

Silica Carry-Over with Steam. Of carry-over materials, silica is the most objectionable impurity in boiler water. Most other carry-over materials can be removed from the turbine by washing, but silica usually requires cleaning by mechanical or other special means. Furthermore, because of the increased solubility of silica in steam with increase in pressure (see Fig. 10),* the quantity of silica in the steam leaving the boiler drum increases with pressure for a given quantity in the boiler water. Therefore, with the demand for improved steam

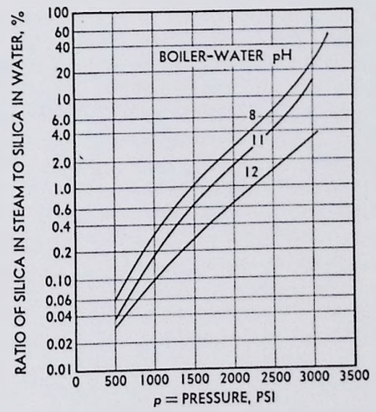


Fig. 10. *Effect of pressure and boiler-water pH on the solubility of silica in steam*

* From B&W research paper, *Selective Silica Carry-Over in Steam,* Transactions ASME, vol 78, May 1956.

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purity, it is necessary to reduce the quantity of silica in the boiler water in order to have as little silica carry-over as possible.

In low-make-up installations, increased boiler blowdown provides the necessary reduction in boiler-water silica to afford a large measure of relief. Recent developments in external-treating plants, which enable economical reduction of silica, have been a boon to operators of high-make-up plants.

A means for silica reduction by internal treatment, which has been moderately successful in a few cases, is the precipitation of silica by magnesia within the boiler under very carefully controlled conditions. Magnesia, either present naturally or injected as an additive, precipitates the silica as flocculent magnesium silicate if the sodium phosphate residual is kept very low to prevent the magnesia from forming magnesium phosphate.

In some cases, using a practical method developed by water-treating consultants, it has been possible to maintain this delicate balance. The operator is able, day in and day out, to keep the excess sodium phosphate at 1 ppm, high enough to prevent the formation of calcium silicate scale and low enough to allow the silica to precipitate with magnesia as magnesium silicate, leaving a residual silica of 3 to 6 ppm in solution. It must be emphasized that the control of the phosphate residual is critical, since complete depletion of phosphate, unnoticed through analytical errors, can lead to the formation of hard calcium silicate scale and tube failures. In general this method of silica control is not applicable at pressures in excess of 1000 psi, in view of the low silica values which must be maintained in the boiler water.

For some low-pressure installations, where the deposit of silica in the turbine is not a problem, this process has been reversed by adding silica to the boiler water to precipitate magnesia as flocculent magnesium silicate. This produces more turbidity than magnesium phosphate or hydroxide, and it is possible to remove the precipitated particles with the continuous blowdown, thus preventing troublesome sludge accumulation in the boiler.

A recent development which promises to be an important contribution to the prevention of siliceous turbine deposits is the steam washer. The principle of the steam washer involves the intimate contact of the steam with condensate or feedwater of acceptable quality in a countercurrent arrangement.

Early steam-washer arrangements suffered from limitations in either capacity or washing efficiency. Extensive research in the field and laboratory has produced a washer capable of reducing the silica at least 50%. This means that the silica in the steam leaving the washer is less than one half of the silica in the steam to the washer. A description of the washer and of its operation is given in Chapter 9, page 12.

Feedwater Conditions—Summary

The broad and general trend in feedwater treatment has been to produce the purest water possible at a cost that can be justified by the conditions at the particular plant to be served, taking into account all of the factors involved. There has been a very active development in external-treatment plants not only in the improvement of the original types of water softeners but also in the application of newer methods of treatment.

The fundamental purpose of feedwater treatment is to remove from the water, outside the boiler, by one means or another and within the cost limitations noted, the maximum amount of foreign matter which is considered potentially detrimental to the pressure parts. Among the impurities to be removed are: scale and sludge-forming materials, soluble salts, oil, dissolved or releasable gases, and all other materials foreign to the water itself which might damage the boiler unit or carry over with the steam.

However, it is now generally the aim in feedwater treatment to make the water suitable for boiler use contingent upon supplementary treatment within the boiler.

Boiler-Water Conditions—Summary

The outstanding development in boiler-water treatment during the late twenties and early thirties, as noted previously, was the almost universal replacement of sodium carbonate with one or more of the various sodium phosphates as water-treating agents for the prevention of scale. With this important exception, the improvements in boiler-water conditioning have been chiefly in the refinement of control and in administration. The steps taken to prevent difficulties from the most common and most fundamental actions or objectionable impurities of the boiler water are summarized below:

Corrosion

Alkalinity. In the early periods of water treatment, alkalinities were considered in terms of grains per gallon. They are now generally thought of and measured in terms of pH values. In using the free-caustic treatment, the most common successful range of pH values is from 11 to 11.5. Those using the controlled phosphate treatment without free caustic maintain pH values of approximately 10.5 to 11.2. Some consultants advocate the same pH range as the preferred practice when using caustic soda.

Dissolved Oxygen. The maximum permissible dissolved oxygen in the water entering the boiler proper dropped from a value of several cc per liter before

1920, through the temporarily recognized figure of 0.2 cc per liter, down to the present final requirement of zero. Mechanical deaeration alone will sometimes produce a zero condition, but in many cases even where this is so, sodium sulfite is maintained not only as a safeguard against lapses in operation but also as a means of detecting such lapses.

It has usually been impossible to maintain large excesses of sodium sulfite in boilers operating in the higher-pressure ranges. Decomposition of sodium sulfite at the higher pressures and concentrations leads to the liberation of sulfur dioxide and hydrogen sulfide in the steam. Accordingly, sodium sulfite concentrations in boiler water are limited to about 10 ppm at 1400 psi and 3 ppm at 1800 psi.

Hard Scale

For treatment within the boiler of the residual hard-scale-forming materials, such as calcium sulfate and calcium silicate, sodium phosphate has almost completely superseded sodium carbonate.

The phosphate residuals maintained in various plants cover quite a range. Probably good average values, expressed as PO_4, are 20 to 40 ppm at pressures up to 1500 psi and 5 to 20 ppm for pressures above this. Compared with sodium phosphate it is known that potassium phosphate has superior solubility characteristics at higher temperatures.

The advantage which this could theoretically provide both in the boiler and in the turbine suggests the substitution of potassium salts for the corresponding sodium salts as the internal-treatment chemicals, particularly for the higher pressures. The number of installations employing such treatment has been too few for the results to be conclusive.

Sludge

The limiting values of allowable sludge for central station work have undergone little change because, at the end of the earlier period of water treatment, surface condensers and treated evaporated make-up reduced sludge to an almost negligible quantity. With continuing improvement in condensers and evaporators, the problem of sludge in central station boilers has been practically eliminated. While this improvement was important in further increasing the purity of boiler water, no sudden or outstanding change occurred.

In the early thirties, for industrial plants the older allowable limit of sludge-forming materials permissible in boiler feedwaters shifted from 85 ppm to 34 ppm, and a few years later this lesser limit was reduced to a very few ppm. The improved methods of water treatment, the increasing objection to boiler cleaning, and the use of higher steam pressures for industrial work led to the conclusion that any sludge-forming materials are bad and should be kept to an absolute minimum by the best methods of external treatment available.

Even with the very few ppm of total sludge-forming materials entering high-pressure industrial boilers, in some instances it has been found necessary to condition the sludge with organic sludge-dispersal agents to minimize its adherence to boiler surfaces and to permit its removal through the continuous blowdown during operation. There are very successful installations of boilers for pressures up to 1400 psi using 100% treated make-up for feed. Where 100% make-up plants for high-pressure and increased heat-input duty are justified by economics, it is now possible to provide a satisfactorily treated water.

Silica

The amount of silica allowable in the boiler water received increasing attention primarily because of turbine deposits. Since the activity of silica is to some extent a function of other materials in the water, its limits are related to these materials. The important or controlling relationships have not been too clearly developed, but the low limits generally sought in central station work are now less than 5 ppm, and in some of the highest-pressure jobs less than 1 ppm.

Total Solids

The quantity of total solids in a boiler water has always had its primary limit in priming or carry-over. For every combination of water and boiler design there is some concentration at which the carry-over rises rapidly. It is compulsory to stay below this critical limit. In addition, good practice calls for keeping the total solids just as low as economically possible because, even though the value is below the so-called priming limit, the carry-over ordinarily is some function of the total solids in the water.

The ABMA has established certain arbitrary values for the quantity of total solids in boiler water, which are intended to give the operator all the latitude needed and at the same time set a limit to the boiler manufacturer's responsibility. The ABMA limits range from 3500 ppm for pressures below 300 psi to 1000 ppm for pressures above 1000 psi.

Control of Treatment

The most important factor in water treatment is the rigid and constant control of the chemical limits established as necessary and desirable. The finest system of water treatment becomes dangerously ineffective if the established limits are not maintained vigilantly and consistently. Unsuspected lapses have caused great difficulty as well as great confusion in establishing the causes of the difficulty. The best system is that which reduces these lapses to a minimum or one which is laid out intentionally to make

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it possible to detect, even by intermittent records, what has taken place between them.

Continuous records for dissolved oxygen are of particular importance in systems not using deaerators under positive pressure, where it is possible for air to enter the feedwater circuit through a leak, when the heater-steam bleed points shift from positive to negative pressure.

The continuous blowdown has been almost universally adopted as indispensable to precise control.

Some companies have operated boilers without any water treatment whatever. One organization associated with a number of inland plants has been able to operate at pressures up to 900 psi without using any water treatment. Where operation without water treatment has been a success, a perfectly tight feedwater system has been maintained to exclude practically all foreign materials and any access of dissolved oxygen to the feed and boiler water. Experience has shown, however, that even a very small lapse in keeping the system tight against dissolved oxygen will cause far greater damage than would be expected with a highly alkaline water. Notwithstanding the potential and critical danger from leakage into such a system, the effort to produce and maintain perfect purity is a bold step in the constant endeavor throughout boiler history to obtain better quality water for making steam.

The treatment of water for steam boilers has been made a major endeavor by reliable manufacturers of water-treating equipment and by competent water-treating consultants. They are better prepared than the boiler manufacturers to assist the operating companies in the handling of water-treatment projects and water-treating difficulties. The Babcock & Wilcox Company does not seek opportunities to enter this field, but it does intend to continue with whatever analyses and research work it deems necessary to enable its engineers to bring about improvements in the design, manufacture, and operation of its products.

Some Operating Suggestions

The purpose of feedwater and boiler-water treatment is to keep the internal surfaces of the boiler clean—not to keep out of trouble with a dirty boiler. Treating feedwater will not correct circulation deficiencies which cause tube failures.

Oxygen-free feedwater is of utmost importance. Feedwater to the boiler should be deaerated to contain less than 0.005 cc per liter, which is a usual guarantee of deaerator manufacturers. Actual deaerator performance is frequently even better—0.002 cc or 0.003 cc per liter is not unusual.

Chemicals should enter the boiler drum to mix with the high-pH boiler water, and the entering feedwater should mix with the chemical-replenished boiler water. Continuous blowdowns should be located to blow down boiler water which has passed through the boiler circuits and therefore is high in impurities.

Evaporator carry-over is responsible for many feedwater problems, and the importance of pretreatment ahead of the evaporator is now generally recognized. Make-up from the evaporator should be checked by conductivity equipment. This is just as important as determining any condenser leakage.

Silica should be kept low in high-pressure boilers to prevent (a) silica scale deposits in boiler tubes and (b) carry-over of volatilized silica in the steam to the turbine, where it is deposited as silica scale on the turbine blades. Volatilization of silica increases with temperature, particularly above 500 F, and the danger of silica passing over to the turbine increases as the boiler pressure increases. The ratio of silica in the steam to silica in the boiler water increases with increase in saturation temperature or boiler pressure and decreases as the pH of the boiler water increases (see Fig. 10).

The silica should be kept below 5 ppm for 900 to 1400 psi boilers, not over 3 ppm for pressures between 1400 and 2000 psi, and not over 1 ppm for pressures above 2000 psi.

While high-pressure boilers are being operated with as much as 100% make-up and concentrations in the order of 1500 ppm, most high-pressure boilers operate with only 1 or 2% make-up and concentrations considerably below 500 ppm, where silica content is controlled by per cent of blowdown.

In addition to its effect in reducing the ratio of silica in the steam to silica in the boiler water, a high pH, in the order of 11, will reduce the corrosive effect when small amounts of oxygen are present in the water. The pH value of the feedwater should preferably be maintained at between 8.5 and 9.0.

To protect against corrosion from oxygen that may not have been completely removed, sulfite should be fed to all boilers up to 1800 psi. As boiler pressure increases, the danger of sulfite breakdown increases, and for this reason it is pointless to attempt to maintain a sulfite concentration higher than 10 ppm for 1400 psi boilers or 3 ppm for 1800 psi.

Ammonia and volatile amines are being used to increase the pH of the condensate or feedwater, thereby reducing iron pickup in the preboiler system without increasing its total solids. The few parts per million of ammonia required for this alkalinity control is in contrast to some earlier experiences, where relatively heavy concentrations of ammonia (in the order of 100 ppm, originating from sewage or vegetable matter) picked up copper from heaters and other components of the system.

Demineralization of feedwater by the use of cation and anion exchangers, which, if carried to the extreme, can produce water with less than 1 ppm of solids, is now widely recognized as an efficient method of treatment. This method of getting pure make-up and pure feedwater is being used in many instances in place of evaporators.

Lignins are sometimes used in boilers where a high per cent of make-up is required. They aid in keeping the sludge in suspension, so that it may be removed more readily through the continuous blowdown.

Under normal conditions adequate cleaning of a new boiler is possible by a thorough boiling-out. However, some users believe that acid cleaning of a new boiler, followed by a good boiling-out, will give a more uniform oxide protective coating on the tubes. In service, to keep the internal surfaces clean, acid cleaning is recommended. After acid cleaning, it is always necessary to boil-out the boiler, preferably at half the operating pressure. Some operating boilers require frequent acid cleaning, others every 3 or 4 years, the period depending upon the amount of deposits which collect on the tube surfaces. To make the acid cleaning effective where there is oil or organic matter in the deposits, it is necessary to preboil-out and to follow with another boiling-out after the acid cleaning. Iron oxide barnacles from internal corrosion have been successfully removed by a preboil-out, using 25 ppm of sodium sulfite followed by the usual acid cleaning and final boiling-out (see section on *Chemical Cleaning*).

A brief outline of recommendations for the treatment of feedwater and boiler water for high-pressure boilers from 900 to 2500 psi follows:

1. All water going to the boiler should be free of dissolved oxygen.
2. For boilers of 900 to 1800 psi, sulfite should be fed in gradually reduced amounts, according to pressure, from 10 ppm for 900 psi to 3 ppm for 1800 psi.
3. The pH of the feedwater going to the boiler should be maintained at between 8.5 and 9.
4. The pH of the boiler water should be kept between 10.8 and 11.5.
5. Phosphates in the boiler water should be maintained from 15 to 20 ppm for 900 psi with gradual reduction for intervening pressures to 5 to 10 ppm for 2500 psi.
6. Silica in the boiler water should be not over 5 ppm at 900 psi with gradual reduction to not over 1 ppm at 2500 psi.
7. There should be supervisory chemical control of feedwater and boiler water during the day shift, 7 days a week, and supplementary control by operators each night shift.
8. A constant watch should be maintained for condenser leakage and evaporator carry-over by the use of conductivity instruments.

The Present and the Future

Throughout the history of water treatment for steam boilers, alkalinity is the golden thread in the pattern of all successful results. Different type waters, different boiler operating conditions, and different methods of treatment have all permitted some latitude in alkalinity limits, but the existence of alkali in controlled quantities has always been necessary, even when it has been present naturally and unobtrusively with unwanted impurities of waters not artificially treated.

Because of the importance of alkalinity, it has been common practice to provide excess or reserve quantities. In the trend toward perfect purity of boiler water, this excess or reserve is now under scrutiny with the intent of reassessing the amount of excess necessary. The "coordinated phosphate control" system of treatment has been successful in eliminating all excess alkalinity caused by what is termed "free caustic." A rigid control is of course essential. Whether the goal of maximum purity will be served by this system or by the more common systems will be determined by the total chemical used in one as compared with the other, when both are given equal attention in development and control.

The advocates of no-treatment-at-all appear to have made the greatest advance in the direction of purity as well as of low alkalinity. In some of these instances, 10 ppm of total solids have been maintained in the boiler water with pH values of less than 10. However, the value of reserve alkalinity is clearly demonstrated when, in using this system, serious corrosion occurs because of a small inward leak of an active impurity such as dissolved oxygen.

With purity as the goal in water treatment, it would be well, in the light of past experience, to maintain a wholesome respect for excess alkalinity and to reduce it only after reliable assurance that this decrease cannot lead to a difficulty far greater than the saving realized by the increase in purity.

Much attention is now devoted to the solubility at higher temperatures of the various materials found in boiler waters. Reliable data are meager, and the results of the search for accurate information about the solubility of materials alone, as well as in relation to each other, should be of great value. It would be well, in studying these solubilities at higher pressures as well as temperatures, to make certain that the laboratory boilers do not have the circulation defects and the poorly proportioned compartments that have in some instances produced unreliable data because of the critically different concentration in the zone of sampling and in the zone of generation.

Progress toward greater purity will always be accompanied by the discovery of unexpected or pre-

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viously dormant characteristics of the materials which remain and which perhaps establish an entirely new relationship. Water-treating chemistry research is now approaching the gentler sloping part of the curve representing improvements corresponding to the effort applied. Much effort is necessary to produce small gains. Any claim of a startling improvement which is not the result of a corresponding effort in research and experiment should be received with skepticism. With the purer water, the safeguards against attack will have to be observed with even more vigilance. The use of these safeguards will probably offset part of the gains in purity, but it will be wise always to consider that the ultimate goal is too far away to be reached by any simple improvement.

Steam purity is very close to perfection. The small fraction of 1 ppm of foreign material which remains is responsible for a disproportionate amount of troublesome turbine deposit. This residual impurity is being attacked in two ways. One is to remove it, while the other is to counteract it by additions of other material to the boiler water or to the steam. Since the demand for greater purity is not likely to be less exacting in the future than in the past and also since absolute perfection has not been achieved dependably without chemical help in related fields, such as in the performance of the deaerator, it seems likely that the anticipated improvement in steam purity will be accomplished with the help of some chemical means. Furthermore, the increased purity of the condensate formed from the purer steam can be expected to cause more active dissolution of the condensate return system, unless some chemical escort is provided.

Once-Through Boilers

The gain in cycle efficiency which can be realized by increasing the operating pressure eventually resulted in the utilization of pressures above the so-called critical pressure. Since the two phases, steam and water, do not coexist above the critical pressure, natural circulation cannot be employed, and the boiler design must be of the forced-flow type. Also, since there are no phases to separate, a drum and steam-separating equipment are unnecessary. Since there is no recirculation of a more highly concentrated liquid within the boiler, there is no boiler water as we conventionally regard it, and, instead, the feedwater and boiler water are identical.

The once-through principle has also been applied below the critical pressure, and the water requirements for both cases are substantially the same.

Suitability of the feedwater for a once-through boiler requires a more specific definition of the limits of the individual constituents than is required for a natural-circulation boiler. In order to prevent reduction of availability resulting from improper water conditions, the feedwater must be continuously maintained at the highest purity obtainable with the best practices in modern low-make-up central stations. To accomplish this, contamination of the feedwater from all sources must be avoided, and any chemicals added for treatment of the feedwater should be completely volatile in the steam. Make-up must be of the highest purity attainable with well-designed and properly operated modern feedwater-purification equipment.

To avoid contamination of the feedwater from sources exterior to the cycle, condenser leakage must be guarded against by proper condenser design and by provision for sensitive detection of leakage. Contamination of miscellaneous returns must also be avoided.

Contamination from sources within the cycle by the products of corrosion of the materials in the cycle must also be minimized. This can be accomplished by maintaining oxygen and carbon dioxide at very low values and by maintaining the feedwater pH within limits which will provide satisfactory protection to the materials in the cycle. This level of pH usually lies within the range of 8.5 to 9.5 and can be obtained by the addition of an alkaline volatile material, such as ammonia or morpholine.

A modern deaerator will reduce dissolved oxygen in feedwater to less than 0.005 cc per liter (0.007 ppm). To achieve even lower values and to provide against upsets in deaerator operation, supplementary chemical deaeration employing hydrazine can be used. Hydrazine, like ammonia and morpholine, is a completely volatile material and does not contribute to the dissolved solids content of the feedwater.

Since the quantities of certain constituents, such as iron and copper, in the feedwater may vary from point to point in the feedwater cycle, the locations at which the desired limits apply must be specific and consistent. The limits which follow apply to the feedwater entering the boiler and hold for all once-through boilers regardless of pressure.

Recommended Limits
Feedwater Entering Boiler

Total dissolved solids	0.10 ppm max
Suspended solids	zero
Hardness	zero
Free caustic	zero
Dissolved oxygen	zero
Carbon dioxide	minimum, preferably zero
Total silica (as SiO_2)	0.01 ppm max
Total iron (as Fe)	0.01 ppm max
Total copper (as Cu)	0.005 ppm max

pH value — Adjust to obtain 0.01 ppm iron maximum. This will normally require a pH value within the range of 8.5 to 9.5, measured at 77 F.

Chemical Cleaning of Internal Heating Surfaces

Chemical cleaning of internal heating surfaces was first proposed where it was difficult or impossible to clean by other methods. Because of early experience with improper technique and limited knowledge of cleaning solvents, this method was at first subject to certain reservations. With progressive experience and development of materials and procedures, much of the earlier uncertainty concerning chemical cleaning has been resolved, and the process is now widely used with complete confidence as the quickest, cheapest, and most efficient method of cleaning these surfaces in boilers of all sizes. It is, however, extremely important to use a procedure of known reliability under careful control.

The procedure outlined below is recommended with the understanding that the user must depend upon competent chemical supervision of his own choice, supplemented by consultants on boiler-water and scale problems. There are specialist companies equipped to provide complete chemical-cleaning service.

Advantages of Chemical Cleaning

Compared with the removal of scale and deposits from internal heating surfaces by mechanical means, the more important advantages of chemical cleaning may be listed as follows:

1. Outage time is reduced. Usually a boiler can be cleaned in less than 36 hours. More than 48 hours are seldom required.
2. Extensive dismantling of the unit is not required. The solvents can be introduced and discharged through existing connections. The only dismantling necessary is the removal of drum-manhole covers and a few header-handhole closures for inspection.
3. The total cost is less.
4. Inaccessible areas, such as small tubes, short-radius tube bends, and irregular surfaces, are readily cleaned, since the cleaning medium has access to all locations reached by steam or water. The result is a better cleaning job. The more thorough removal of scale will reduce the rate of subsequent formation of deposits.
5. New equipment can be designed with complete flexibility and simplicity for best economy, efficiency, and over-all cost, since no compromise is required to provide accessibility for cleaning.

General Procedures and Methods

In general, four steps are required in a complete chemical-cleaning process for a boiler:

First. The heating surfaces are washed with an acid solvent containing a proper inhibitor to completely dissolve or to partially dissolve and disintegrate the deposits.

Second. Clean water is used to flush out loose deposits, solvent adhering to the surface, and soluble iron salts. Any corrosive or explosive gases that may have formed in the unit are displaced. The unit is flushed to a positive overflow from the vents.

Third. The unit is boiled-out with soda ash solution to eliminate completely or to neutralize the effect of any acid remaining and to release hydrogen from the metal.

Fourth. The unit is flushed with clean water as a final rinse to remove any remaining loose deposits.

The two generally accepted methods in chemical cleaning are: (a) continuous circulation of the solvent through the parts to be cleaned, and (b) filling the unit with the solvent and allowing it to soak for a prescribed length of time.

Circulation. In the circulation method, after filling the unit the solvent is recirculated until cleaning is completed. Samples of the return solvent are tested periodically during the cleaning. Cleaning is considered complete when the acid strength of the returned solvent reaches equilibrium, indicating that no further reaction with the deposits is taking place. Compared with soaking, since a solvent of lower strength may be used and the solvent is drained from the unit as soon as the reaction is complete, the possibility of damage to the surface cleaned is less in using the circulation method. Also, the cost of chemicals will be somewhat less because of the control testing of solvent strength.

The circulation method is particularly suitable for cleaning forced-flow boilers, superheaters, and economizers with positive liquid flow paths to insure circulation of the solvent through all parts of the unit. A complete cleaning job cannot be anticipated by this method unless the solvent reaches and passes through every circuit of the unit.

Cleaning by the circulation method is aided mechanically by the constant change of solvent which flows over the deposits on the surface. This feature is of particular advantage when foaming may occur, as with deposits of a calcareous nature. The continuously circulating solvent also carries disintegrated undissolved materials through the unit. This is especially helpful in preventing clogging of small tubes.

Soaking. In cleaning by the soaking method, after filling with the solvent the unit is allowed to soak for a period of 4 to 8 hr depending upon deposit conditions. To insure complete removal of deposits, the acid strength of the solvent must be somewhat greater than that required by the actual conditions, since, unlike the circulation method, control testing during the course of the cleaning is not conclusive, because samples of solvent drawn from convenient locations

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may not truly represent conditions in all parts of the unit.

The soaking method is preferable for cleaning units where definite liquid distribution to all circuits by the circulating method is not possible without the use of many chemical inlet connections or where circulation through all circuits at an appreciable rate cannot be assured except by using a circulating pump of impractical size. These conditions may exist in large natural-circulation units with complex furnace-wall cooling systems.

Advantages of the soaking method compared with the circulating method are: simplicity of piping connections, assurance that all parts are reached by a solvent of adequate acid strength, and less chemical supervision during the cleaning operation.

Solvents, Deposits, and Inhibitors

Solvents. For the removal of boiler deposits, various solvents have been suggested or tried, including many acids and a number of alkaline compounds.

In general, the use of alkaline compounds requires considerable time and temperature, and, while the attack on steel is nonexistent or negligible, the surface will not be as clean as with the acid solvents.

On the basis of cost, availability, suitable inhibitors, and rapid removal of deposits, extensive experience indicates that hydrochloric and sulfuric acids are the most practical cleaning agents. Of these two, hydrochloric acid costs less, is more effective in dissolving a greater number of the usual constituents of boiler deposits, and is more readily available and inhibited. In contrast to the reactions of sulfuric acid, which produce relatively insoluble sulfates, the reactions of hydrochloric acid form readily soluble chlorides.

Pressure parts of alloy steels, particularly those high in chromium, should not be cleaned with acid solvents without competent chemical advice on the type of acid and inhibitor and the conditions of application in each specific case.

For deposits of certain characteristics, the solvent may require additional reagents, such as intensifiers to react with certain constituents or wetting reagents to promote solvent penetration of the deposit. The need for these auxiliary chemicals should be decided by the chemist controlling the specific cleaning operation.

Deposits. Scale deposits formed on the internal heating surfaces of a boiler unit come from the water. In general, the constituents belong to one or more of the following groups: oxides of iron; carbonates, phosphates, and sulfates of calcium and magnesium; silica and silicates. The deposits may also contain various amounts of oil, which are sometimes appreciable in boilers on return to service.

For the most efficient and economical scale removal, the deposit should be analyzed, and that acid should be selected for the solvent that will most effectively react with a deposit of this type. The deposit is usually a mixture of several compounds, and the acid that will best dissolve the different major constituents is generally selected as the solvent, since the dissolution of the soluble components will free any insoluble compounds present, which can then be removed by flushing.

To avoid possible redeposition of other iron compounds, hydrochloric acid is usually preferred in removing iron oxide, even though sulfuric acid is theoretically the better solvent. Hydrochloric acid is more effective than sulfuric acid in dissolving calcium and magnesium salts. Neither has any effect on calcium sulfate, although scales containing up to 75% of this compound have been successfully removed by dissolving the soluble constituents.

No universal recommendation can be made for silica or silicates, because some silicates react more readily with hydrochloric acid, some with sulfuric acid, and still others with a combination of strong alkalies and heat. A preboil for several hours with a 5 to 10% soda ash solution, before acid cleaning, has been found effective in changing the silica scale to a more acid-soluble form. In some instances, to remove silica scale hydrofluoric acid has been included in the solvent. Except under expert chemical advice and supervision, this procedure is not recommended.

Small amounts of oil in the deposit are removed when the general deposit is dissolved and disintegrated. For higher percentages of oil contamination, it may be necessary to use a wetting agent in the solvent to promote penetration of the deposit. If the deposit is predominantly oil or grease, boiling-out with alkaline compounds must precede the acid cleaning. In new or repaired boilers with water-soluble lubricants used in tube expanding, boiling-out with alkaline compounds may be omitted.

Inhibitors. The following equations represent the reactions of acids (in this instance hydrochloric) with constituents of boiler deposits:

$$Fe_3O_4 + 8HCl = 2FeCl_2 + FeCl_3 + 4H_2O$$
$$CaCO_3 + 2HCl = CaCl_2 + H_2O + CO_2$$

At the same time, however, the acids will also react with and waste away the metal of the boiler, as represented by the equation:

$$Fe + 2HCl = FeCl_2 + H_2$$

unless means are provided to prevent or retard this reaction without, of course, affecting the attack on the deposits.

For this purpose arsenic compounds, barium salts, starch, quinolin, pyridine, or other products have been added to the acid solvent as inhibitors, to prevent or materially reduce this reaction with the metal. A number of commercial inhibitors are sold under trade names, and still others are made and used by companies furnishing complete acid-cleaning service including the prepared solvents. If the cleaning is done by the operator's own organization rather than by a company specializing in this work, the use of commercial inhibitors offered by reputable chemical concerns is recommended.

The effectiveness of acids in attacking boiler deposits as well as steel increases rapidly with rise in temperature. However, the inhibiting effect decreases as the temperature rises, and at a certain temperature, likely to vary for each inhibitor, the inhibitor may be decomposed into compounds devoid of inhibiting properties. It should also be noted that not all inhibitors are effective with all acids.

Important points in the use of inhibited acid solvent may be summarized as follows: Selection of the acid and auxiliary chemicals should depend upon the type of deposit to be removed. The inhibitor must be suitable for the acid selected. The temperature of the solvent must not exceed that at which the effectiveness of the inhibitor will be reduced or destroyed. The limiting temperatures for the use of the inhibitor should be obtained from the maker.

Determination of Solvent Conditions

Deposit Samples. The preferred type of deposit sample is a small section of tube with adhering deposit. In any case a sample of the deposit, down to the bare metal, representing the worst condition in the unit, should be obtained. Selection of the acid and auxiliary chemicals is then based on an analysis of the sample by chemical, X-ray diffraction, or other methods. After selection of the acid for the solvent and the inhibitor suitable for this acid, it is necessary, to insure complete removal of the deposits, to determine the acid strength of the solvent, the solvent temperature, and the length of time required for the cleaning process.

Acid Strength. Technically, the acid strength would be in proportion to the amount of deposit in the unit to be cleaned. However, since the determination of the amount of deposit cannot be much more than a rough approximation and the general average of conditions observed falls within a fairly narrow range, the acid strength of the solvent usually is set satisfactorily by experience. It is therefore recommended that the acid strength of the solvent, as fed to the unit, should be:

Three per cent when the unit is cleaned by the continuous-circulation method, and

Five per cent when the unit is cleaned by the soaking method.

Commercial hydrochloric and sulfuric acids are sold in two grades in carboys of either 13 or 12.7 gal. The principal specifications for each grade and the dilution required for a 3 and a 5% solution, respectively, are noted below:

Commercial Acid	Hydrocloric		Sulfuric	
Degrees Baumé	18	20	60	66
Specific gravity	1.142	1.160	1.706	1.835
Acid strength, %	28.0	31.5	78.0	93.0
Carboy contains, gal	13	13	13	13
Carboy contains, gal	12.7	12.7	12.7	12.7
For a 3% solution, gal H_2O/gal acid	9.5	11.0	42.7	55.1
For a 5% solution, gal H_2O/gal acid	5.3	6.2	24.9	32.3

Solvent Temperature. To aid in the acid attack on the deposits, the temperature of the solvent should be as high as possible without seriously reducing the effectiveness of the inhibitor. Laboratory tests show that most commercial inhibitors break down or lose their effectiveness entirely above 160 F. Experience indicates that, as the temperature rises above 140 F, the corrosion rate increases. It is therefore recommended that the temperature be limited to 140 F, unless a higher temperature can be tolerated in a particular unit and with an inhibitor safe at the higher temperature.

If a higher temperature is used, care should be taken to inspect the unit thoroughly after the first cleaning and after *each subsequent* cleaning. In this way the actual corrosion rate can be determined, and it can be decided either to continue cleaning at this temperature, if the rate is not damaging, or to reduce the temperature for subsequent cleanings. In making up the solvent to the required strength, it should be heated by warm water to 140 F or to the temperature decided upon after all conditions are known. *The temperature of the solvent should not be raised by firing the unit.*

Cleaning Time. When cleaning by the circulation method, completion of the process is determined by analyzing samples of the returned solvent for acid strength and noting when the acidity reaches equilibrium.

In using the soaking method, the length of time the solvent is to be retained in the unit should be decided upon beforehand. While the type and quantity of the deposit and laboratory determination of the reaction rate of the solvent on samples of the deposit should apply in setting the length of time, a practical guide in determining the solvent retention time for the first few cleanings, using the acid strength noted above, is given as follows:

a) For moderate thickness of relatively soft sludges or for conditions that would otherwise require turbining at not less than yearly intervals, the

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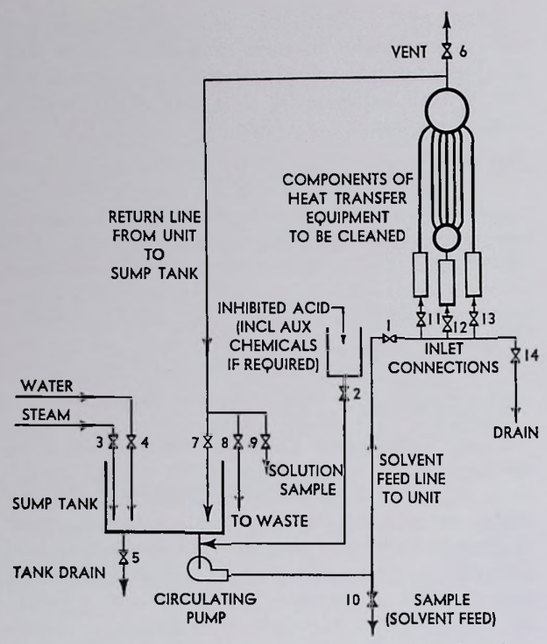


Fig. 11. *Typical arrangement of connections in using the circulating method in chemical cleaning of boilers*

solvent retention time should be a minimum of 4 hours.

b) For thin coatings ("eggshell thickness") of relatively hard scale formation, the solvent retention time should be 6 hours.

c) For deposits such as might result from a severe upset in the feedwater-conditioning system, the solvent retention time should be a maximum of 8 hours.

The periods noted are for the actual retention of the solvent in the unit. The surfaces are, of course, also wholly and partially subjected to the solvent during the time of filling and draining. In establishing permanent schedules, the periods recommended should be extended or shortened on the basis of the results achieved for specific conditions during the first few cleanings, as indicated by: visual inspection; test turbining of selected tubes, before and after cleaning, in areas typical of the worst deposit conditions; or examination of sample sections of tubing cut out of the unit.

Preparation for Cleaning

Heat Transfer Equipment. All parts not to be cleaned should be isolated from the rest of the unit. To do this, connections to these parts may be broken and blanked or plugged with wood or rubber. Where arrangements permit, parts of the unit can be isolated by completely filling with water.

Bronze or brass parts should be removed or replaced temporarily with steel. All valves should be steel or steel alloy. Gage and meter connections should be closed or removed from the unit.

All parts of the unit not otherwise protected by blanking off or by flooding with water should be covered by the inhibited solvent. Vents should be provided wherever vapors might be pocketed, since acid vapors from the cleaning solution do not retain the inhibitor.

Cleaning Equipment. The cleaning equipment should be connected as shown in Fig. 11 if the continuous-circulation method is used, or as shown in Fig. 12 if the soaking method is used. Continuous circulation requires sufficient inlet connections to insure distribution and to promote circulation through all circuits. It also requires a return line to the sump tank from the highest point of the unit or from the end of a continuous-tube heat transfer apparatus. The soaking method does not require a return line, but means should be provided, such as a rubber hose, to carry the spill-over from the vent to a safe location, and the pump discharge should be connected to the lowermost inlet to the unit. Depending upon the volume of the liquid to be handled, the pump discharge and the return or vent spill-over lines, as the case may be, are usually of 1½-in. to 3-in. pipe size.

The filling or circulating pump should not be fitted with any bronze or brass parts, and it should have the capacity to deliver, in not over 2 hours at 100 psi, a volume of liquid equal to the content of the vessel to be cleaned. For convenience in handling

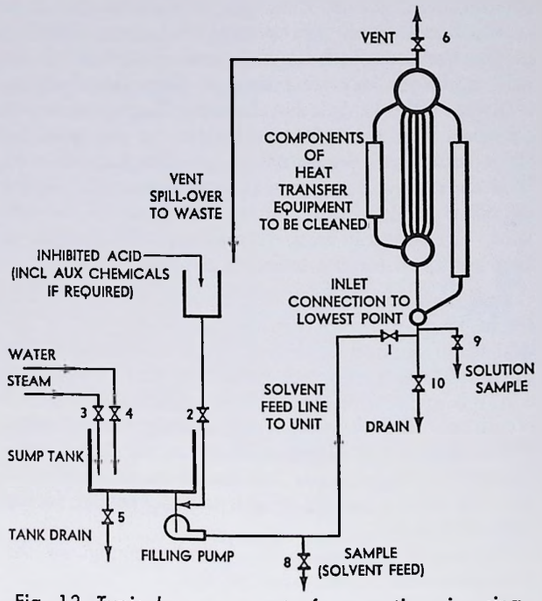


Fig. 12. *Typical arrangement of connections in using the soaking method in chemical cleaning of boilers*

and mixing the solution, the sump tank should be large enough to hold about 5% of the total volume of liquid required to fill completely the vessel to be cleaned. In practice, the capacity should be at least 300 gallons. An adequate supply of clean water and steam for heating the solvent to the desired temperature should be provided. Provision should be made for adding the inhibited acid to the suction side of the filling or circulating pump, as described later in the procedure to be followed for each method.

Cleaning Solutions. To determine the amount of solvent required, an estimate should be made of the content of the vessel to be cleaned, adding 2% to allow for wastage and other contingencies. Commercial acid suitable for the deposit to be removed should be procured in the quantity required, when mixed with water, to make a solvent of the amount and acid strength desired.

An inhibitor suitable for the acid to be used should be procured. In obtaining the manufacturer's recommendation for the amount of inhibitor to be used, care should be taken to make certain that this recommendation is based on full knowledge of the acid strength of the solvent and what it is to be used for, to avoid any possibility of a higher corrosion rate than can be tolerated when cleaning a boiler. The amount of inhibitor specified for each carboy of acid should be completely dissolved or suspended in about 1 gal of acid removed from the carboy, taking care not to let the solution heat. After pouring the mixture of inhibitor and acid back into the carboy, the contents should be thoroughly mixed.

In preparation for the neutralizing process, commercial soda ash should be provided to make up a solution sufficient to fill the unit completely in the proportion of 10 lb of soda ash to 100 gal of water. This will produce a solution strength of about 1%.

If the operator retains an organization specializing in acid cleaning of heat transfer equipment, the latter ordinarily determines and supplies the type of cleaning solvent and handling equipment best suited for the conditions.

Circulating-Method Procedure

To reduce to a minimum the time required for cleaning, the preparations indicated in Fig. 11 and as noted above should be carried out as far as possible before the unit is taken off the line. The unit should be removed from service, allowed to cool in the normal manner to about 150 F, and then drained.

Referring to Fig. 11, clean water, 4, with a hydrogen ion value preferably no higher than pH 7, is admitted to the sump tank and heated with steam, 3, to 140 F or to the temperature decided upon. The circulating pump is started, the unit filled, and the water circulated through the system, using the return line, 7, to the sump tank until the return levels off at the desired temperature. Inhibited acid, 2, is then added to the circulating warm water at the pump suction until a sample of the solvent feed, 10, indicates that the solvent has reached an acid strength of 3%, after which the process is carried on in accordance with the schedule indicated in Fig. 13.

The addition of inhibited acid to the solvent is continued during the first period until the acid strength of the returned solvent, sampled at 9, reaches a maximum, as at A in Fig. 13. This sample should also be tested for iron content. Should this test show a rapid rise or an increase of iron content to 1%, which indicates an excessive attack on steel, the solvent should be immediately drained from the unit. While the iron content is not a positive check of the corrosion rate, since it is affected by the amount of iron oxide in the unit, it seems best to use it as a guide. As the volume of the liquid increases and the level in the sump tank rises because of the addition of acid, it may be necessary from time to time, during

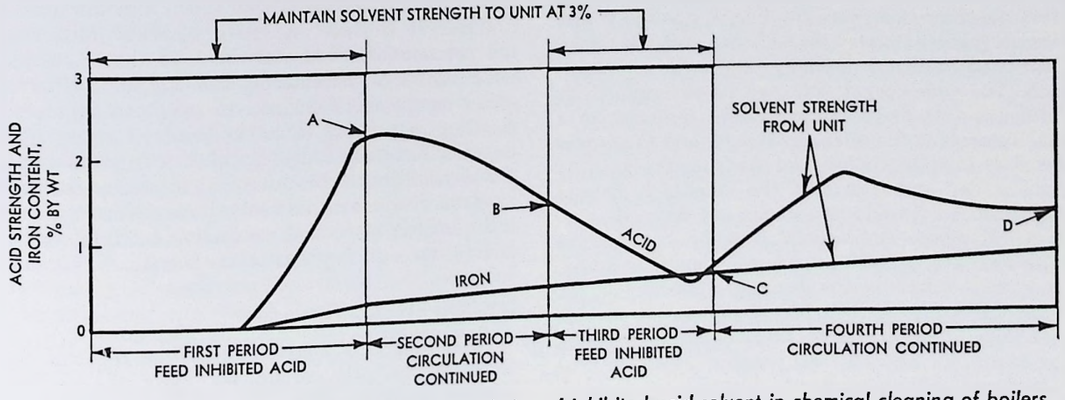


Fig. 13. *Approximate schedule for continuous circulation of inhibited-acid solvent in chemical cleaning of boilers*

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this and subsequent operations, to close the return line, 7, and open the waste line, 8. The vent, 6, should be opened now and then to release any acid vapors. Occasionally, for short periods during the circulation of the acid solution and the subsequent circulation of other solutions, the full flow should be circulated alternately through each of the circuits controlled by valves 11, 12, and 13 in Fig. 11. This is to make sure that the solution passes through all circuits.

When the sample tests of the acid strength of the returned solvent indicate the maximum concentration desired, as at A in Fig. 13, addition of the acid should be discontinued, but circulation of the solvent through the system should be continued. The acid strength and iron content of the returned solvent, 9, should be checked periodically at about 10-min intervals. It will be helpful, throughout the course of the solvent application, to plot the actual solvent conditions, as indicated in Fig. 13.

If the tests indicate a reduction in acid strength of the returned solvent exceeding ¾ of 1% and point to a continued sharp decline, as at B in Fig. 13, and there is no evidence of a corresponding sharp increase in iron content or that it exceeds 1%, the solvent should be fortified by adding inhibited acid to bring the acid strength of the solvent, 10, to 3%. The addition of acid should be discontinued when the acid strength of the returned solvent starts to increase, as at C in Fig. 13. Circulation of the solvent through the system should be continued until the acid strength of the returned solvent levels off, as at D in Fig. 13, indicating that the reaction of the solvent with the boiler deposits has been completed.

No additional heat, in any way, should be applied during the acid-cleaning operation.

When the reaction is completed, as indicated by the acid strength of the returned solvent reaching equilibrium, the unit should be flushed by closing the return line, 7, and feeding warm clean water, 4, to the sump tank, with the line, 8, open to waste. Flushing heated water through the unit should be continued until the discharge shows no acid reaction. The presence of acid can be determined by adding a few drops of methyl red indicator to a small sample of the discharged water and to samples from all sampling points and vents on the unit. As long as any acid is present, the samples so treated will turn red. In addition to the test for acid reaction and before any attempt is made to neutralize with soda ash, a pinch of soda ash should be added to a small sample of the discharged flushing water. This will cause a greenish precipitate if soluble iron salts are still present, and the flushing should be continued. The soda ash solution should not be introduced until repeated tests show that the greenish precipitate is no longer present.

The reason for making sure that soluble iron salts are completely flushed out of the unit is that any soda ash in excess of the requirement for the neutralization of free acid will react with the iron in solution to form gelatinous solids which tend to adhere to metal surfaces.

When the above conditions have been met, the return line, 7, to the sump tank should be opened and the line, 8, to waste should be closed. Circulation of water through the unit should be continued, with the temperature increased to as near 212 F as possible, and soda ash should be added as described below.

To neutralize completely any trace of acid, the surfaces should be washed with a solution in the proportion of 10 lb of soda ash to 100 gal of water by gradually adding enough soda ash to the water in the sump tank to fill the unit with a solution of this strength. The solution should be circulated through the unit until the return shows a strong alkaline reaction, as indicated by a sample turning red on the addition of a few drops of phenolphthalein. The same test should be made with samples from all sampling points and vents, to make sure that the soda ash solution has reached all areas previously reached by the acid.

After the circulation is discontinued and the soda ash solution is lowered to the normal operating level, the unit should be fired to raise the steam pressure and to hold it for 2 hours at 50 to 100 psi. This will aid in cleaning the surfaces and will also drive off hydrogen from the metal. The unit should be vented several times during this period to get rid of the hydrogen liberated. When the pressure has fallen to zero, the line, 8, to waste should be opened, the return line, 7, to the sump tank should be closed, and clean hot water should be flushed through the unit for about 1 hour.

Following completion of the foregoing procedure, the circulating system is shut down and the unit is drained. It is then inspected, carefully observing the precautions noted elsewhere, to determine the effectiveness of the cleaning and any indications of attack on the metal surface. In some cases the acid cleaning may bring down undissolved heavy iron deposits that may lodge in tubes with little slope. These tubes should be cleared, as these deposits may result in overheating and tube failure when the unit is fired. After inspection, on finding conditions satisfactory, the unit is prepared for service.

Soaking-Method Procedure

To reduce to a minimum the time required for cleaning, the preparations for using the soaking method, as indicated in Fig. 12 and as previously noted, should be carried out as far as possible before the unit is taken off the line.

The unit should be removed from service, allowed to cool to about 150 F in the normal manner, and then drained. If the unit has not been in service and is cold, it should be filled and refilled with water of not more than 50 F above the boiler temperature for each successive filling until the boiler temperature is 150 F or slightly higher, or the boiler may be fired to this temperature and then drained.

Referring to Fig. 12, clean water, 4, with a hydrogen ion value preferably no higher than pH 7, is admitted to the sump tank. The filling pump is started, and the heating steam, 3, is adjusted to heat the water to the unit to 140 F or to the temperature decided upon for the specific conditions. Inhibited acid, 2, is added to the water entering the pump, the rate of acid feed being adjusted to bring the solution to an acid strength of 5% at the sampling point, 8. The unit is then filled to a positive overflow in the vent spill-over to waste. During the filling operation, frequent checks should be made and conditions adjusted so that the solvent fed to the unit is held at an acid strength of 5% and the maximum temperature desired is maintained (see Fig. 12).

When the unit is full of solvent under the conditions noted, the filling pump is stopped, the shutoff valve, 1, in the filling line is closed, and the vent, 6, is left open. The unit should then be allowed to soak for a period of time suitable for the conditions, as previously outlined. During this period, samples, taken at 10- to 30-min intervals from chemical feed, continuous blowdown, water column or other suitable connection to upper drum, and from any convenient connection to lower drum or headers, 9, should be tested for acid strength and iron content. If the iron content increases to 1%, the unit should be immediately drained for the reason noted above in using the circulating method. The tests for acid strength will indicate the time required.

After having held the solvent in the unit for the prescribed length of time, the unit should be completely drained by opening the drain valve, 10. During this test it is advisable to draw samples of the discharged solvent at frequent intervals. Later analysis of these samples for acid strength and iron content, in conjunction with the original analysis of the deposits, serves as a basis for estimating the quantity of deposits removed and as a guide for maintaining or modifying the solvent conditions for future cleaning operations.

When the solvent has been drained from the unit, the drain line, 10, is closed, the filling-line valve, 1, is opened, the pump is started, and the unit is filled with warm clean water until it overflows through the vent spill-over to waste. Then the unit is drained, and the flushing operation is repeated until there is no acid reaction from the discharged water.

Before any attempt is made to neutralize with soda ash solution, care must be taken to make certain that all soluble iron salts have been flushed out of the unit. This condition should be determined by the tests described above for the circulating method. Flushing should be repeated until there are no indications of soluble iron salts in the discharge water.

When the flushing operation is completed, the surfaces should be neutralized by washing with a soda ash solution. Water is admitted to the sump tank, soda ash is added in the proportion previously noted, and the unit is completely filled with the solution. The water level is then lowered to normal, and the unit is fired and the pressure held at 50 to 100 psi for 2 hours, as outlined above for the circulating method. The neutralizing operation is followed with another flushing, using clean warm water. The unit should then be drained and inspected, following the procedure outlined above for the circulating method and observing the precautions noted below. After inspection, on finding conditions satisfactory, the unit is prepared for service.

Precautions and Measure of Results

Cleaning must not be considered a substitute for proper water treatment. The rigid control maintained in well-operated plants to achieve the best possible water treatment should not be relaxed because of the relative ease of cleaning by the chemical method. Chemical cleaning ordinarily is not necessary more often than once a year. Intervals between cleanings should be extended or reduced as conditions warrant. This is essential, since each time a unit is chemically cleaned the metal surfaces are depacified or stripped of their protective oxide coating, and, in addition, a slight amount of parent metal is dissolved because none of the known inhibitors is completely effective.

In handling acids, the recognized precautions of using goggles, rubber gloves, and rubber aprons should be observed. Acid should be poured slowly into water; *never pour water into acid.*

Acid cleaning removes mill scale and products of oxidation. A thorough visual inspection should be made after cleaning to determine whether any parts need replacement.

Undrainable pressure parts, such as pendent-type superheater tube elements, require special attention to make certain that they are cleared of all acid solvent and loose deposits. Usually each individual element must be separately flushed through a tightly fitting hose connection, using a similar arrangement at the discharge end to avoid spilling the flushing discharge over into the other elements. A mixture of compressed air and water as the flushing medium

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has also been used effectively.

The solvent should be introduced through connections located at the lowest points of the equipment to be cleaned (see Figs. 11 and 12) and should rise gradually in all components of the unit without cascading or undue turbulence.

In riveted joints the solvent may enter seams, and damage can occur without being detected. Therefore, chemical cleaning of riveted drums is usually avoided. This applies particularly to drums calked on the outside.

The possibility of hydrogen gas being generated during the acid cleaning of internal heating surfaces should never be overlooked. The quantity of hydrogen released is a measure of the ineffectiveness of the inhibitor. A portion of the hydrogen will be absorbed by the metal, while the remainder will form part of the atmosphere in the vessel. The portion absorbed by the metal will be liberated gradually, the time required decreasing as the temperature of the metal increases. While the metal wastage indicated by its generation may be inconsequential, the hydrogen is a source of two potentially dangerous conditions—embrittlement of the steel and an explosive atmosphere in the vessel.

On the completion of acid cleaning, because of the possibility of hydrogen embrittlement, the steel should not be cold-worked (tubes rolled, flanges tightened, seams calked, etc.) in any manner for at least four or five days until the pressure boiling-out has been completed or the boiler has been brought up to steam pressure in the normal manner.

In opening a unit for inspection following an acid cleaning, the possibility of an explosion from the presence of free hydrogen in the atmosphere within the unit must be recognized. There have been explosions of hydrogen atmospheres in boilers after acid cleaning, and fatal injuries have resulted. At atmospheric pressure and ordinary temperatures, the values indicated for the inflammability of a mixture of hydrogen in air lie in the range from 6.2% as the lower limit to 71.4% as the upper limit. The ignition temperature of hydrogen in air, at atmospheric pressure, is between 1076 and 1094 F.

It is imperative, therefore, because of the possibility of an explosion, to observe the following precautions:

After an acid cleaning, no one should enter the unit or get in the way of any openings to the pressure parts unless the atmosphere within has been cleared of explosive gases. Until this has been done, open flames, flashlights, any other lighting equipment, or anything that might produce a spark of any kind must not be used within or near the openings of pressure parts.

The following steps may be taken to clear the atmosphere of explosive gases within the pressure parts:

Following an acid cleaning, apart from other reasons the unit should be thoroughly flushed with warm water to a positive overflow from the topmost vent, to make certain that all the atmosphere within has been displaced. To warm the unit and thus accelerate the liberation of hydrogen absorbed in the metal, the temperature of the water should be held as nearly as possible at 212 F by light firing or other means.

After the unit has been opened, air eductors, placed at locations such as open drum manholes, should be used to circulate air through the unit for the removal of hydrogen being liberated from the metal. This procedure should be continued until analysis by a Burrell or other suitable instrument shows that the air within is free of explosive gases.

Measure of Results. Determination of the type and approximate amount of different kinds of scale and impurities removed from a unit at a cleaning will be possible if, at periodic intervals during the operation, the spent cleaning solvent and the water solid matter ejected are properly sampled and analyzed and a record is kept of the total amount involved. This information will serve as a valuable guide for further boiler-water treatment and the determination of the required frequency of cleaning.

The Practical Significance of pH Values

THE term "pH" is used to express the degree of acidity or alkalinity of a solution. Acidity is commonly associated with a sour taste, while alkaline or basic materials impart a somewhat bitter taste.

When substances such as acids, bases, or salts are dissolved in water, ions are formed in the solution. These ions consist of electrically charged atoms or groups of atoms which result from the partial or complete dissociation of the solute molecules. The formation of the ions is termed "ionization," and the extent to which it occurs can be expressed by an "ionization constant," which has a fixed value for a given solution at a specified concentration and temperature.

Though many different ions may be formed in solution, the ones which establish whether the solution is acid or alkaline are the hydrogen (H^+) and the hydroxyl (OH^-) ions, respectively. When the hydrogen ions exceed the hydroxyl ions in number, the solution is acid. When they are equal, it is neutral. Alkalinity or basicity denotes that the hydroxyl ions exceed the hydrogen ions.

Acids and alkalies vary in the degree to which they form ions in solution. Those which almost completely ionize are called strong acids or bases. Hydrochloric acid and sodium hydroxide are examples of this type. Other acids and bases ionize to only a small degree and are, therefore, called weak acids or bases. Acetic acid and ammonium hydroxide are typical examples.

Even pure water dissociates to a minute degree into hydrogen and hydroxyl ions. It has been found that pure water at 25 C always contains 1.0×10^{-7} gram equivalents per liter of hydrogen ions and, since pure water is neutral, an equal concentration of hydroxyl ions. The product of these concentrations is a constant equal to 1.0×10^{-14} at 25 C. Thus, if the hydrogen ion concentration of a solution were 1.0×10^{-5} gram equivalents per liter, the hydroxyl ion concentration would be equal to 1.0×10^{-9} gram equivalents per liter. Since, for this case, the hydrogen ion concentration is greater than the hydroxyl ion concentration, the solution is acidic.

In order to express hydrogen ion concentrations more conveniently than by the use of decimals or by negative exponents, the term "pH" was adopted. Expressed mathematically, pH is equal to the negative logarithm to the base 10 of the hydrogen ion concentration. This is expressed as:

Hydrogen ion concentration $= 1 \times 10^{-x}$ where "x" equals the pH. A neutral solution, since it contains 1×10^{-7} gram equivalents of hydrogen ion per liter, has a pH value of 7. Since acid solutions contain more than 1×10^{-7} gram equivalents of hydrogen ion per liter, the pH values of these solutions are less than 7. Conversely, alkaline solutions have pH values greater than 7.

The commonly accepted range within which pH values are expressed covers the scale of 0 to 14. The extremes of this scale correspond to 10^0 and 10^{-14} gram equivalents of hydrogen ion per liter. Acidities or alkalinities greater than those represented by pH values of 0 and 14, respectively, are more conveniently expressed as per cent concentrations.

The following table is helpful in understanding the relation between hydrogen ion concentration and pH.

Hydrogen Ion Concentration, gram equivalents per liter			pH
1.0		10^{0}	0
0.1		10^{-1}	1
0.01		10^{-2}	2
0.001	ACID	10^{-3}	3
0.0001		10^{-4}	4
0.00001		10^{-5}	5
0.000001		10^{-6}	6
0.0000001	NEUTRAL	10^{-7}	7
0.00000001		10^{-8}	8
0.000000001		10^{-9}	9
0.0000000001		10^{-10}	10
0.00000000001	ALKALINE	10^{-11}	11
0.000000000001		10^{-12}	12
0.0000000000001		10^{-13}	13
0.00000000000001		10^{-14}	14

Examples of acid solutions encountered in everyday life are lemon juice with a pH value of approximately 2.0 and carbonated soft drinks with a pH of about 6.5. On the other hand, a solution of 1 teaspoon of baking soda in a cup of water would be alkaline and have a pH value of nearly 10.

Since the pH value of a solution varies with temperature, it is the usual practice to measure and report all pH values at 25 C. Notwithstanding the scarcity of accurate values at elevated temperatures, such data as are available indicate that pH decreases with increase in temperature, the neutral point decreasing in like manner.

The pH value may be estimated conveniently by the use of specially prepared papers containing dyes which vary in color according to the pH. For feedwater and boiler-water pH determinations, where a higher degree of accuracy is required, it is usual practice to employ an instrument which measures the potential difference established between two suitable electrodes and indicates this reading on a scale graduated in pH units.

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BIRTHPLACE OF THE CROLOYS

22

METALLURGY AND PROPERTIES OF MATERIALS

METALLURGY is concerned with the extraction of metals from ores and with the combination, treatment, and processing of metals into useful engineering materials. As an art, metallurgy has been of great importance to the progress of civilization for at least 6000 years, beginning with the production of the first metal. As a science, metallurgy is only about 60 years old. Important technological developments in modern industry have stimulated its rapid progress and application since the early 1920's. During this period, new theories were advanced, new practices were introduced, and new and better alloys were developed. Without these advances in the theory and practice of metallurgy, the present high standards of efficiency, economy, and reliability of operation of steam-generating equipment would not have been possible.

Main Fields of Metallurgy

Metallurgy is divided into two main fields: the ferrous, which covers iron and iron-base alloys; and the nonferrous, covering metals and alloys other than iron or iron-base. In accordance with the closeness of its association with other sciences in the study of metals, metallurgy also may be divided into the fields of 1) process or chemical metallurgy and 2) physical and mechanical metallurgy or metallography. Process metallurgy includes the numerous mining and refining processes, such as reduction of ores, refining of metals, and the manufacture of metals and alloys. Physical metallurgy is devoted to the study of the constitution and structure of metals and alloys, including all the factors and processes that control the physical characteristics and mechanical properties.

The principal pressure parts of a power boiler—the drums, headers, and tubes—are made from ferrous alloys. The same is true for most of the other parts of the boiler and its closely related auxiliaries, such as coal pulverizers, fuel-conveying systems, soot blowers, structural supports, economizers, air heaters, casings, ducts, stokers, baffles, hangers, bolting, steam piping, flanges, fittings, and valves. Nonferrous metals are used only to a minor extent in boiler installations; in auxiliary equipment, such as water heaters, deaerators, and attemperators; in the condensing system beyond the turbines; and in valves and instrumentation. Consequently, the following discussion is confined to ferrous metals—their production, fabrication, thermal treatment, and application in steam boiler equipment.

It is the purpose of this chapter to present the fundamental metallurgical concepts and practices and to outline the metallurgical requirements for the selection of materials for boiler construction.

Ores of Iron

Iron was first known in ancient civilization, probably 5000 years ago. Nevertheless, its great usefulness was not fully realized until the characteristics of the metal itself and its effects when combined with alloying elements were established by the application of the science of metallurgy.

Although iron is one of the most common of metals, metallic iron is very rarely found in nature. It usually occurs in the form of mineral oxides (Fe_2O_3 or Fe_3O_4), and as such it comprises about 6% of the earth's crust. The principal ores of iron, in the order of abundance and importance in this country, are: hematite (Fe_2O_3), magnetite (Fe_3O_4), limonite ($2Fe_2O_3 \cdot 3H_2O$), and sideride ($FeCO_3$).

Reserves of high-grade iron ores in the United States have been depleted so rapidly that lower-grade ores and imported ores are being smelted in increasing amounts. The iron minerals in the lower-grade ores are concentrated to yield a product of greater iron content for smelting. The beneficiation of taconite is one example of the concentration (usually by magnetic methods) of the iron minerals in large deposits of low-grade ores in the Lake Superior region.

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However, it has been found expedient for blast furnaces in the United States to smelt also foreign ores, so that the reserves of domestic high-grade ores will not be so rapidly depleted.

Substantial amounts of iron ore from the following countries are smelted in American furnaces: Brazil, Canada, Chile, Liberia, Mexico, Peru, Sweden, and Venezuela. Ore is also imported from Angola, British West Africa, Cuba, Dominican Republic, Iran, Italy, Norway, Philippines, Surinam, and the United Kingdom. Much of the iron ore consumed in this country at the present time is the red oxide of iron, hematite. It is found chiefly in the Lake Superior region, in the states of Minnesota, Michigan, and Wisconsin. In a part of this region the ore is economically extracted by open-pit mining. In locations where the ore is too far below the surface, shafts are sunk and underground mining is employed.

Iron ores are classified on the basis of their metallic iron content and the percentage of undesirable impurities, known as "gangue." The most objectionable of these impurities is phosphorus, because of its harmful effects in steel and, especially, since it cannot be removed in the reduction of the ore. An iron ore containing phosphorus not in excess of 1 part to 1000 parts of iron is classed as a "Bessemer" ore. An ore which contains phosphorus in excess of the 1:1000 ratio is classed as a "non-Bessemer" or "basic" ore.

Ores may be shipped as mined or subjected to concentration processes, known as "beneficiation," prior to shipment. Beneficiation of ores includes drying to reduce the water content, roasting to reduce the sulfur content, and washing to remove some of the gangue materials. The iron ore, either as mined or as improved by beneficiation, is shipped by rail and boat to the blast furnaces, where it is smelted.

Manufacture of Iron

The first step in the production of iron and steel is the reduction of the iron ore with coke and limestone in the iron blast furnace. In this process, "reduction" is the removal of the oxygen from the ore, leaving a mixture of iron and carbon and small amounts of other elements as impurities. The coke is the reducing agent and source of heat, and the limestone ($CaCO_3$) is the flux to promote fusion, which, in the molten state, combines with impurities of the ore, floats to the top, and can be removed as slag. A blast furnace is a vertical steel-shell structure 90 to 100 ft high lined from bottom to top with fireclay brick. Carbon block linings are used in some furnaces. The principal components of the blast-furnace plant are illustrated in Fig. 1. Its product is pig iron, which is an impure form of iron containing about 4% carbon and varying amounts of silicon, manganese, phosphorus, and sulfur.

The pig iron tapped from the blast furnace may be used directly as hot metal in certain steelmaking processes, or it may be taken to pig-casting machines and cast into standard-size pigs. It may also be taken to a mixer, where iron from several blast furnaces is mixed to produce more uniform or average analyses.

Cast Iron

Liquid pig iron tapped from the blast furnace is sometimes used directly for metal castings. More often, however, the iron is remelted in a cupola, air furnace, or electric furnace, to adjust its composition before it is used for castings. Pig iron that has been so remelted is known as cast iron, a general term applicable to a group of iron-carbon alloys containing carbon in excess of 2% (usually less than 4.5%) and appreciable amounts of silicon. Compared with steel, cast iron is decidedly inferior in malleability, strength, toughness, and ductility. On the other hand, cast iron has better fluidity, can be cast satisfactorily into complicated shapes, and is less costly than steel. The most important types of cast iron are the white and the gray cast irons.

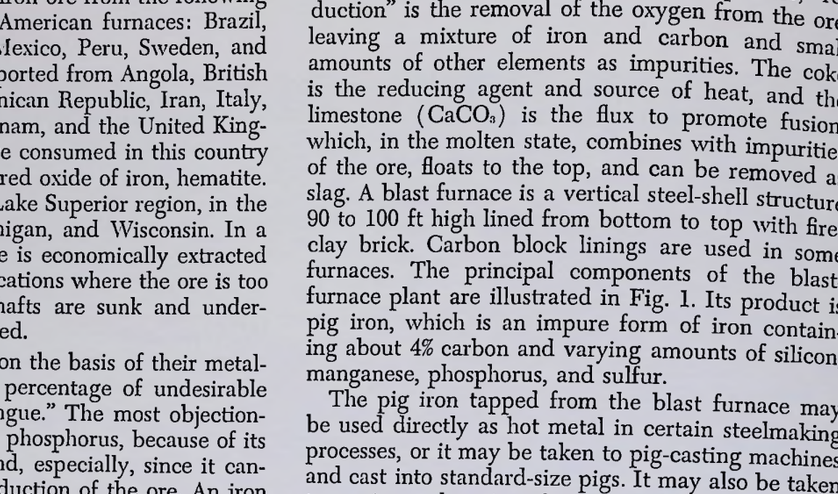


Fig. 1. *Schematic outline—external features of a blast furnace and its components*

White Cast Iron

White cast iron is so known because of the silvery luster of its fracture. In this alloy, the carbon is present in combined form as iron carbide (Fe_3C), known metallographically as "cementite." This carbide is chiefly responsible for the hardness, brittleness, and poor machinability of white cast iron. These remarks also apply to chilled iron, which differs from white cast iron only in the method of manufacture. Chilled iron is cast against metal chills that cause rapid cooling at the adjacent areas, thus promoting the formation of cementite. Consequently, a white or mottled structure is obtained, which is characterized by high resistance to wear and abrasion. Elverite, a type of chilled-iron casting developed by B&W for use in pulverizers, cement and clinker grinders, and in other wear-resistant parts, has long been noted for its uniformity and high quality.

Gray Cast Iron

Gray iron is by far the most widely used of cast metals. In this alloy, the carbon is predominantly in the free state, in the form of graphite flakes that form a multitude of notches and discontinuities in the iron matrix. The appearance of the fracture of this iron is "gray" since the graphite flakes are exposed. The strength of gray iron depends upon the size of the graphite crystals and the amount of cementite formed



Fig. 2. *White cast iron—typical microstructure (250x) reduced about ⅔. Note iron carbide "Fe_3C" (light areas) and tertiary eutectic "Steadite" (mottled areas)*

TABLE 1

General Engineering Grades of Gray Iron—Mechanical and Physical Properties

Class	30	35	40	45	50	60	70	80
Tensile strength, psi (min)	30,000	35,000	40,000	45,000	50,000	60,000	70,000	80,000
Compressive strength, psi	105,000	115,000	125,000	135,000	150,000	175,000	200,000	225,000
Torsional strength, psi	40,000	45,000	54,000	60,000	67,000	76,000	85,000	90,000
Modulus of elasticity°, psi × 10^{-6}	14	15	16	17	18	19	20	21
Torsional modulus, psi × 10^{-6}	—	—	5.5	6.6	7.0	8.0	8.1	—
Impact strength, Izod AB (1.2-in. diam unnotched), ft-lb	23	25	31	36	65	75	120+	120+
Brinell hardness	180	200	220	240	240	260	280	300
Endurance limit, psi:								
Smooth	15,500	17,500	19,500	21,500	25,500	27,500	29,500	31,500
Notched	(13,500)	(15,500)	17,500	(19,500)	21,500	23,500	25,500	27,500
Damping capacity	Excellent	Excellent	Excellent	Good	Good	Good	Fair	Fair
Machinability	Excellent	Excellent	Excellent	Good	Good	Fair	Fair	Fair
Wear resistance	Fair	Fair	Good	Good	Excellent	Excellent	Excellent	Excellent
Pressure tightness	Fair	Fair	Good	Good	Excellent	Excellent	Excellent	Excellent
Specific gravity:								
g per cm^3	7.02	7.13	7.25	7.37	7.43	7.48	7.51	7.54
lb per cu in.	0.254	0.258	0.262	0.267	0.269	0.270	0.272	0.273
Thermal coefficient of linear expansion: (50-200 F)	6.5-6.7	—	6.6-6.8	6.4-6.4	—	—	—	—
(50-500 F)	6.9-7.2	—	7.1-7.3	6.8-7.0	—	—	—	—
(50-800 F) (in. per in., F) × 10^{-6}	7.4-7.6	—	7.4-7.6	7.0-7.2	—	—	—	—
Magnetic properties	Mag	Mag	Mag	Mag	Mag	Mag	Mag	Mag
Pattern shrinkage, in. per ft	⅛-⅛	⅛	⅛	⅛	⅛	⅛-3/16	⅛-3/16	⅛-3/16
Coef of friction (against steel)	—	—	—	(0.19)	(0.195)	(0.20)	—	—

°At 25% of tensile strength. NOTE: Values in ( ) estimated.

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together with the graphite. The strength of the iron increases as the graphite crystal size decreases and the amount of cementite increases. Gray cast iron is easily machinable because the graphite carbon acts as a lubricant for the cutting tool and also provides discontinuities which break the chips as they are formed. Modern gray iron having a wide range of tensile strength, from 20,000-30,000 psi to 90,000 psi, can be made by suitable alloying with nickel, chromium, molybdenum, vanadium, and copper. Mechanical properties of several engineering grades (ASTM classes) of gray cast iron are given in Table 1.

Malleable cast iron is white cast iron that has been heat-treated to change its combined carbon (cementite) into free or "temper" carbon (nodules of graphite). The iron becomes malleable because in this condition the carbon no longer forms planes of weakness.

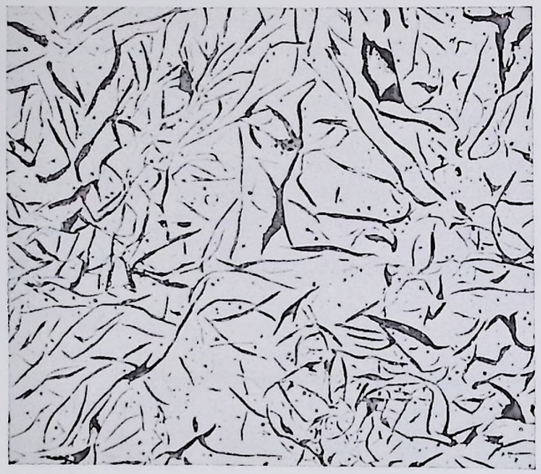


Fig. 3. *Gray cast iron—typical microstructure (50x) reduced about ⅔. Note graphite flakes (dark areas)*

Ductile Cast Iron

The most recent addition to the cast-iron family is the so-called "ductile cast iron." It is a high-carbon magnesium-treated ferrous product containing graphite in the form of spheroids or impacted particles. Ductile cast iron is similar to gray cast iron in melting point, fluidity, and machinability but possesses superior mechanical properties. This alloy is especially suitable for pressure castings, and by special procedures (casting against a chill) it is possible to obtain a carbide-containing abrasion-resistant surface with an interior of good ductility.

The structure of several types of cast iron used in B&W equipment is illustrated in Figs. 2 and 3.

Cast iron was used to a considerable extent in early steam boilers for both tubes and headers. At present, this material is no longer used in the pressure parts of modern power boilers but is utilized in related equipment, such as stoker parts, Bailey metal furnace-wall blocks, grinding rings of coal pulverizers, etc. Typical B&W iron castings are shown in Fig. 4.

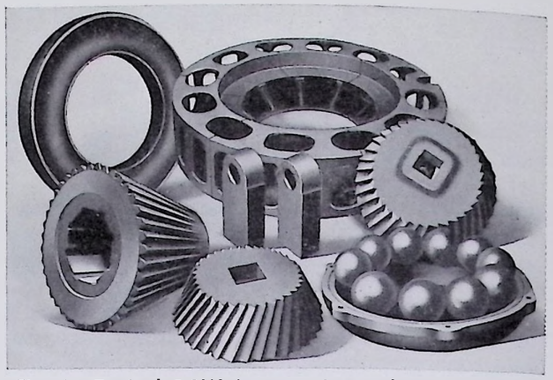


Fig. 4. *Typical B&W iron castings of crusher and pulverizer parts in Elverite. Balls are steel (SAE-52100)*

Manufacture of Steel

Steel may be simply defined as an alloy of iron and carbon which contains not over 2.0% carbon and is cast in an initially malleable mass. This is by no means a complete definition of steel, but for all practical purposes it is adequate. Since the early 1920's, a great variety of steels has been developed for widespread use. All of these steels are basically iron-carbon alloys, but they differ appreciably in carbon content and, in many instances, also in the amount of other alloying elements present. In this way several classes of steels have been produced, with a wide range of physical characteristics and mechanical properties suitable for the specific requirements of many different industrial applications.

All modern processes in the manufacture of steel start with pig iron. Pig iron is transformed into steel by the oxidation of the impurities with air, oxygen, or iron oxide, since these impurities, as compared with iron, have greater affinity for oxygen.

Pig iron may be refined by oxidation alone("acid" process) or by oxidation in conjunction with a strong basic slag ("basic" process). Carbon, silicon, and manganese are removed by both processes, but phosphorus and some of the sulfur in the pig iron can be removed only in the basic process. When phosphorus is oxidized to a phosphate, it will stay persistently with the iron if the slag is strongly acidic (high in silica) and therefore will not be removed. However, if in addition to oxidizing conditions there is also an abundance of a basic constituent (such as calcium or magnesium) in the slag, then the latter will hold the phosphate in combination and prevent it from being again reduced and seized by the metallic iron. The removal of sulfur is similarly facilitated by oxidizing conditions and basic slag.

The principal methods of manufacturing steel for subsequent rolling or forging are: basic open-hearth, basic electric, basic oxygen, and acid Bessemer. Steel for castings can be made by the foregoing methods and also by the acid electric and acid open-hearth methods.

The furnace charge is molten pig iron in the Bessemer process, steel scrap and pig iron in the open-hearth processes, and selected steel scrap in the electric-furnace process. Each of these requires raw materials of particular composition and produces a steel somewhat different in chemical analysis and physical characteristics.

Bessemer Process

Purification of molten pig iron by the oxidation of some of the impurities is the principle of steelmaking in the Bessemer process. Air is blown through the charge until the silicon, manganese, and carbon are oxidized. The resulting product is known as blown metal, and the necessary additions, in order to meet steel purification requirements, are made in the ladle. Bessemer steel is characterized by its low carbon and high phosphorus and sulfur content and a considerable amount of nonmetallic inclusions and nitrogen. The major tonnage of steel produced by this process goes into the production of free-cutting screw steels, sheet metal, wire, bolts, nuts, reinforcing bars, and tin plate. Bessemer steel is not used in high-pressure boiler parts, although killed* Bessemer steel is now employed to a considerable extent in the manufacture of butt-welded or continuous-welded pipe for ordinary uses.

Open-Hearth Process

a) Basic

By far the greatest tonnage of steel in this country (about 90%) is produced by the basic open-hearth process. The tremendous importance of this method is because of the fact that a uniformly good-quality steel can still be produced, even though the furnace charge (pig iron, scrap, iron ore, and limestone) is varied widely to suit economic conditions. For instance, scrap may be at a premium. The ability to control oxidation of carbon in the bath permits use of large quantities of molten pig iron directly from the blast furnace. Some steel plants have used upwards of 70% hot metal in the charge, although about 50% is nearer the average practice. In the open-hearth it is also possible to complete the refinement of partially refined hot metal from the Bessemer converter. Such practice is called duplexing.

The basic open-hearth process is carried out in a large regenerative furnace lined with basic refractories (magnesite, dolomite). Open-hearth furnaces of recent construction have a capacity of at least 200 tons. A diagrammatic section through a typical open-hearth furnace is shown by Fig. 5.

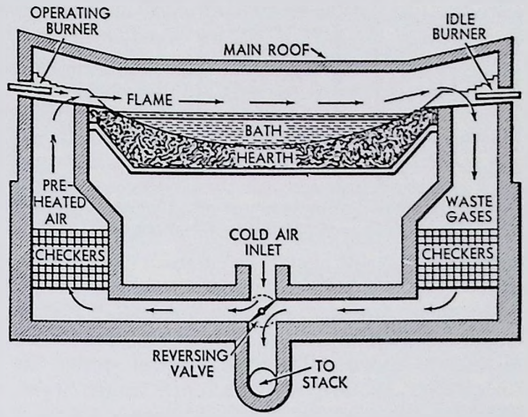


Fig. 5. *Diagrammatic cross section, open-hearth furnace*

In addition to various types of carbon steel, a considerable quantity of certain grades of alloy steel is produced by the open-hearth process, notably in SAE or AISI types, for engineering uses. In alloy-steel boiler material, carbon-molybdenum steel for boiler plate and superheater tubes or piping, and the low chromium-molybdenum alloy tube materials up to about 1¼% chromium can be made by the basic open-hearth process. It is usually preferable, however, to make the chromium-containing tube grades by the electric-furnace process.

b) Acid

The acid open-hearth is essentially a melting process, and, as in all acid processes, the removal of phosphorus and sulfur is impossible. Consequently, it is necessary to use a charge lower in these impurities to produce satisfactory steel. The smaller heats usually made in the acid process and the higher cost restrict the use of acid open-hearth steel to castings and special products.

Acid open-hearth steels are cleaner than average basic open-hearth steels. Transverse mechanical properties are superior to basic open-hearth steels, particularly in forgings. Forged seamless steam drums of acid open-hearth steel were used in B&W equipment prior to adoption of the fusion-welding process for drum construction. A very small tonnage of rimmed steel (see Fig. 6) is made by this process; most of the production is killed carbon and alloy steels.

*A steel is said to be killed when it has been degassed (mainly deoxidized) prior to pouring, so that the liquid metal lies quiet in the mold without the presence of gas to form porosity in the ingot.

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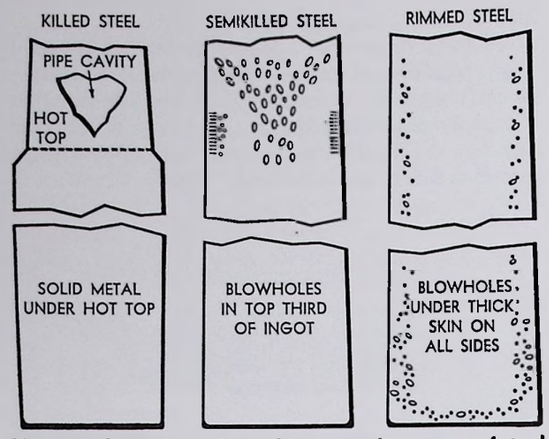


Fig. 6. *Schematic sections of ingots—three types of steel*

Basic-Oxygen Process

Experiments in steelmaking begun during 1941 in Germany culminated in the commercial production of steel at Linz and Donawitz, Austria, during the early 1950's and later in Canada, the United States, and other countries. Many modifications of the process have been developed.

In the United States the basic-oxygen process operates in a basic-lined cylindrical furnace with a solid bottom and open top. The furnace resembles a Bessemer converter, since it is tilted to receive molten iron and restored to an upright position during the refining of the iron by a jet of high-purity oxygen, which enters the furnace through the top. The refined steel is considered the equivalent of basic open-hearth steel.

Electric-Furnace Process

Most high-grade alloy steels are manufactured in the electric furnace. For these alloys, the electric-furnace process offers the following important advantages over other steelmaking processes:

1. Any desired temperature can be obtained because the control of temperature conditions is definite with close limits.
2. Either oxidizing, reducing, or neutral conditions may be maintained in the furnace at will. This permits the addition of alloying elements to the bath with assurance of complete deoxidation in the furnace.
3. The bath is not subject to contamination by the source of heat.
4. Sulfur, solid nonmetallic impurities, and occluded gases are significantly eliminated.

Good Control of Composition

The higher temperature generated in the electric furnace increases the fluidity of the metal, permits rapid reaction in removing sulfur and phosphorus and permits recovery of oxidizable alloying elements from the slag. The control of quality and composition is, therefore, better than in the open-hearth process, but the cost is somewhat higher. An electric-furnace heat ranges from 10 to 100 tons as compared with 200 to 550 tons in the open-hearth. Steels of intermediate chromium-molybdenum composition, such as those used for superheaters, are generally made in the electric furnace. The basic electric-furnace process is always used for production of stainless, high-alloy, and tool steels. The acid process is used essentially for melting and finds application chiefly in steel foundries.

The rotary-piercing and the extrusion processes, employed in the manufacture of seamless tubing and pipe, are perhaps the most rigorous hot-forming operations in the entire metalworking industry. They require, therefore, the highest quality steel, not only to withstand these rigorous forming operations but also to obtain a satisfactory product suitable for the severe conditions encountered in high-temperature service.

Quality Alloy Steel for Tubes—B&W Practice

To insure a uniformly high quality in alloy-steel grades for tube production, an electric alloy-steel plant, to produce high-temperature and corrosion-resistant steels now commonly used by various industries for steam generation, oil refining, chemical processes, and other kindred uses, was erected in 1942 and integrated with the B&W Tubular Products Division's seamless-tube mills at Beaver Falls, Pa.

This plant originally consisted of two 25-ton basic-lined arc furnaces, a battery of soaking pits, an 8-ton steam hammer, bloom-heating facilities, and a 24-in. bar mill for the production of seamless tube-rounds. After some ten years of operation, it became necessary to expand these facilities in this same location by the addition of one 50-ton and one 75-ton basic-lined arc furnace, additional soaking pits, and a 1500-ton steam-hydraulic forging press with manipulator. At the same time, the heating facilities of the bar mill were increased, and some additional building area was added to take care of the bloom and bar-conditioning operations. By 1957 the demand for alloy steel for the Tubular Products Division became so large that a new plant was designed in a new location, at Koppel, Pa. Late in 1959 this new facility was put into operation. It consisted of one 100-ton basic-lined arc furnace, a battery of large soaking pits, and a 40-in. blooming mill that served not only for the production of blooms but also for the rolling of the larger-diameter rounds. This new plant is about 5 miles from the original steel plant and is operated in conjunction with it exclusively for the production of alloy steel tube-rounds for the Tubular Products Division of the Company. Fig. 7 shows a schematic drawing of an arc furnace typical of the type in operation in these facilities.

Practically the entire range of ferrous alloys has been melted in these furnaces, from plain low-carbon

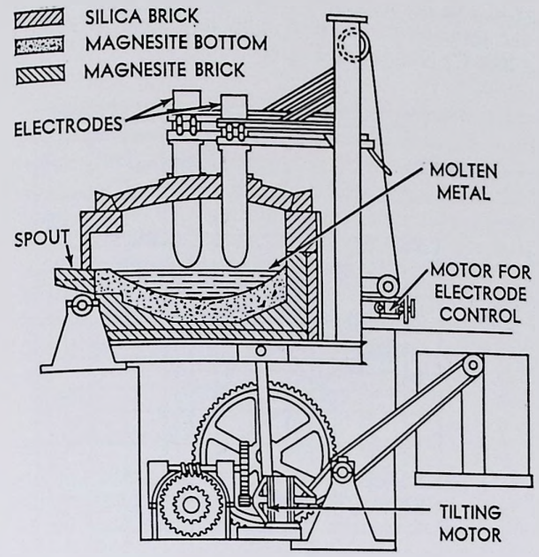


Fig. 7. *Electric furnace (50-ton capacity) used in the production of Croloy steels*

steel to 25% chromium-20% nickel stainless alloy, all with double-slag[a] practice and under rigid technical control. Melt procedures are set up for each grade, and no deviation is permitted after standard procedures have been established. This is important in providing uniform piercing qualities and uniform performance in service. The electric-furnace melt for high-quality alloy tube steel call for thorough degasification of the melt irrespective of composition.

Samples Quickly Analyzed

Only killed (fully deoxidized) steels are made in the plant. Alloying elements are recovered from the second or finishing slag formed on the completion of the oxidizing period. Samples of the molten steel are taken from the bath and cast into small ingots, which are rushed to the laboratory, about a mile away, by means of a pneumatic tube system. These samples are rapidly analyzed in the laboratory by chemical methods and spectrography. The results of the analyses are quickly reported to the melter and guide him in making the necessary alloy additions to the bath to bring the steel to the specified composition. To insure high quality, two or more samples are necessary in the course of making the steel.

Ingots are teemed into about 20-in.-square molds. The ingot weights are calculated for each specific billet multiple to be produced against the tubing order. Permanent hot-top equipment, a refractory-lined short section fitted over or into the ingot mold, is used with ample cropping to insure freedom from piping.

[a] In double-slag practice, the slag is drawn off twice during the heat. After the first removal of slag, fresh fluxing reagents are added, the heat is continued, and the newly formed slag is drawn off before tapping.

Special Heating Cycles

To prevent thermal shock or stress-cracking, special heating cycles are maintained for ingots of alloy steel, particularly for the hardenable grades. Each alloy has a certain thermal cycle which must be adhered to in heating ingots for blooming and for bar-mill rolling. To remove all surface defects, blooms are subjected to careful conditioning before bar rolling, with such additional conditioning of bars as may be required to remove any remaining defects. Conditioning is an important operation to remove such surface defects as checks, slivers, etc. It is done by grinding or scarfing, in which the surface where defects appear is removed to sound metal. Superficial scale and loose foreign matter are removed by air jets and hand manipulation. All stainless grades are turned in the round before being sent to the tube plant.

To make a satisfactory seamless tube by the rotary piercing process, the billet must have a dense outer surface, free from longitudinal defects or radial planes of weakness. Furthermore, it must have good ductility over a broad enough range of high temperature to permit piercing of hollows of commercial length. Adequate allowance should be made for any temperature build-up from friction during the deformation in piercing. The billet must also be free of pipe or center porosity and other defects that might open up on the inside of the bar in the process.

The Steel Ingot

Molten metal tapped from the steelmaking furnaces is poured from ladles into metal molds and allowed to solidify preparatory to forging and rolling. Such a casting is called an ingot.

All steel used in the production of wrought-steel products, such as plate, tubing, and bars for boilers or any other uses, is cast into ingot molds of suitable size and shape. Proper ingot practice is essential to insure a good quality product, free from breaks, internal defects, or excessive segregation.

Ingots are cast in many sizes and shapes, established by the research and experience of the steel producer. The cross section of most ingots approximates a square or rectangle with round corners but may be fluted to give additional hot strength to avoid corner cracks. The greatest dimension is always vertical, and all ingots are tapered. The exact cross section and vertical height are carefully selected for the job at hand. Steel of the rimmed and semikilled types (see Fig. 6) is usually cast in ingots with big end down, while killed steel is commonly cast in in-

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gots with a sink head or hot top, and with big end up. This procedure is employed in the latter case in order properly to feed the so-called "pipe" which forms during cooling and confine it to the hot top, which portion of the ingot is later sheared or cropped in an amount sufficient to give good sound metal. A bottom crop also is taken to ensure sound metal in that area of the ingot.

All steels, irrespective of type, are subject to certain internal variations from natural phenomena which occur as the steel solidifies in the mold. The character and magnitude of such variations and their effects depend upon the size of the ingots and may be generalized as follows:

Primary pipe in the upper portion of the ingot is caused by shrinkage of the metal in passing from the liquid to the solid state during solidification. Since the surface of the cavity becomes oxidized, rolling at a welding heat will not close it. Consequently, the pipe end of the ingot must be cut off and discarded. Under certain conditions, a shrinkage cavity known as secondary pipe is formed below the primary pipe but is not connected with it. Secondary pipe is usually associated with the evolution and entrapment of gases in the ingot during solidification. Because of its nature, however, it will normally weld in the rolling process and does not materially affect the quality of the finished product.

Another phenomenon associated with rate of solidification is segregation or nonuniformity in chemical composition within the ingot. Certain portions of the solidified metal contain either more or less of the elements originally contained in the liquid steel. Segregation in varying degrees is found in all types of steel ingots. The amount of segregation depends chiefly upon the type and composition of the steel, the teeming temperature, the ingot size, and the inherent segregating characteristics of the individual elements of the steel. These phenomena have been widely studied and controlled by special practices, proper discard, and other measures indicated by the intended end use of the product.

Steel Types

In most steelmaking processes, the primary reaction is the combination of carbon and oxygen to form a gas. If the oxygen available for this reaction is not removed prior to or during casting by the addition of silicon, aluminum, or some other deoxidizing agent, the gaseous products continue to evolve during solidification of the metal in the mold. Proper control of the amount of gas evolved during solidification determines the type of steel. If no gas is evolved and the liquid lies quietly in the mold, it is known as killed steel. With increasing degrees of gas evolution the products are known as semikilled steel and rimmed steel. The efficiency of several of the common deoxidizers in molten steel at 2900 F (1600 C) is shown in Fig. 8.

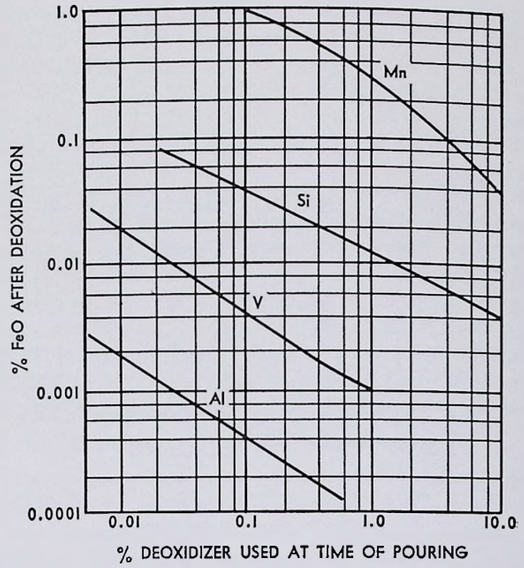


Fig. 8. *Effect of several common deoxidizers in molten steel at 2900 F*

Rimmed Steel

In rimmed steel there is a marked difference in composition across the section and from top to bottom of the ingot. The outer rim is ductile, and the amounts of carbon, phosphorous, sulfur, and nonmetallic inclusions in this rim are lower than the average composition of the whole ingot, while the amount of these constituents in the inner portion or core is higher than the average in the ingot as a whole. A marked evolution of gas during solidification of the outer rim is responsible for the typical structure of rimmed steel. When solidification begins, concentration of certain elements in the liquid core of the ingot increases, and at the core there is also some additional segregation toward the upper and central portions.

Rimmed steels normally contain less than 0.25% carbon, under 0.60% manganese, and 0.01% silicon maximum. Satisfactory rimmed steels do not retain any significant percentages of highly oxidizable elements such as aluminum, titanium, or silicon. Aside from the closing and welding of gas voids (blowholes), the structural pattern of the rimmed-steel ingot persists through the rolling process to the final product. Rimmed steel, which costs less to make than either killed or semikilled steel, is widely used as structural plate material. It is used also in strip form for production of electric-resistance-welded tubing. The rimmed portion is particularly suitable for welding by that process.

Capped Steel

The characteristics of capped steel and rimmed steel are similar, but capped steel represents an intermediate step between rimmed and semikilled steels. A deoxidizer may be added when the ingot is cast to effect a controlled rimming action. Capped steel is generally cast in big-end-down bottle-top molds using a heavy metal cap. Capped steel generally has a somewhat thinner rim, and the segregation in the upper central portion of the ingot is less than in rimmed steel. This method is not ordinarily used for steel containing more than 0.15% carbon.

Semikilled Steel

Semikilled steel has a variable degree of uniformity in composition. This type of steel is only partially deoxidized with silicon, aluminum, or both, to allow the evolution of sufficient gas to offset, entirely or in part, the shrinkage accompanying solidification. Since pipe cavities are minimized, semikilled steel is usually cast in big-end-down molds without hot tops. Most semikilled steels contain between 0.15% and 0.25% carbon, and approximately 0.05% silicon.

Killed Steel

Steel of the killed type is deoxidized almost completely, so there is essentially no evolution of gas during pouring and solidification. Consequently, a primary shrinkage cavity, known as pipe, forms during solidification, and this affected part of the ingot is discarded. Killed steel is usually cast in big-end-up ingot molds with hot top to supply the shrinkage cavity. There may be a secondary pipe in addition to the primary pipe, but this is reduced by the general use of big-end-up ingots.

Although killed steel has a more uniform composition than any of the other types, there are some variations in composition from surface to center and from bottom to top. All but a very few compositions of steel can be killed, if sufficient deoxidizing elements may be added to inhibit gas formation. Practically all alloy steels are of the killed type, as are most of the carbon steels containing more than 0.25% carbon. Such steels usually contain 0.10% silicon minimum. For certain uses, however, some steels are killed almost entirely with aluminum. B&W seamless-steel boiler tubes are made exclusively of killed steel.

Typical appearance and location of blowholes and pipe in rimmed, semikilled, and killed steels are illustrated in Fig. 6.

Atomic Structure of Iron

Pure iron at room temperature is highly ductile and is nearly comparable to copper in hardness. It is strongly magnetic (ferromagnetic) and in this respect differs from all other elements—although nickel and cobalt are magnetic to a milder degree.

Like all other metals, iron is crystalline in structure. All crystalline bodies may be considered as composed of atoms placed at the points of a space lattice (see Fig. 9). The smallest component unit of this lattice is known as a unit cell and is characterized by a regularity of structure. A crystal is composed of these unit cells, piled upon one another like bricks, and reproducing at their corners the points of the lattice. Different crystals have different types of space lattice. Analysis by X-ray diffraction shows that iron, on solidification from a molten state, crystallizes in the cubic space lattice.

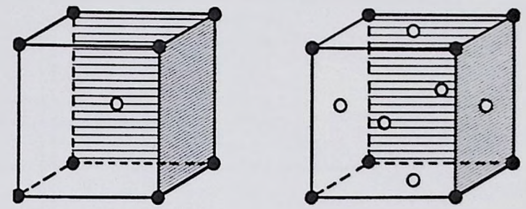


Fig. 9. *Space-lattice diagrams of crystalline structure (iron) showing arrangement of atoms in (a) body-centered cubic and in (b) face-centered cubic*

Iron may exist in one of two forms of cubic space lattice, body-centered or face-centered, so differentiated by the manner in which the atoms are arranged in each of them. Both are illustrated in Fig. 9. The body-centered cubic form has an atom at each corner and one at the center of the cell; the face-centered cubic form has an atom at each corner and one in the center of each of the six faces.

At room temperature, iron is composed of a body-centered cubic lattice (see Fig. 10). In this form it is known as "alpha iron" or "alpha ferrite," and it is soft, ductile, and magnetic. Upon heating above about 1415 F, alpha iron loses its magnetism but retains its body-centered crystalline structure. This structure changes to face-centered cubic at about 1670 F, at which temperature alpha iron is transformed to "gamma iron" and remains nonmagnetic. Continuing upward in temperature, another phase change occurs at 2570 F, when "delta iron" is formed. The latter is identical in crystal structure (body-centered) with that of the low-temperature alpha iron. It is magnetic and is stable to the melting point. There are no known phase changes in the liquid form (above about 2800 F). On cooling very slowly from the liquid state, the atomic rearrangements described occur again, but in reverse order.

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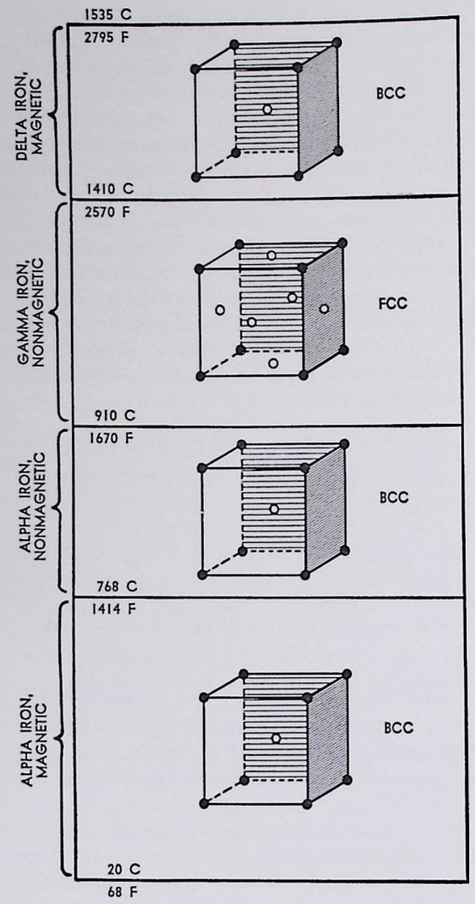


Fig. 10. *Allotropic transformations in pure iron*

The modifications of atomic structure which occur in iron during heating to and cooling from the melting point are called "allotropic" changes. The temperatures at which these changes take place are known as "transformation points" or "critical points." The range of temperature covered by the transformation points is called the "critical range."

Physical Metallurgy of Steel

The foregoing pertains to pure iron, which is difficult to obtain, is correspondingly expensive, and in any case does not have suitable characteristics for construction purposes. However, iron alloyed with carbon and such other elements as manganese, silicon, phosphorus, and sulfur represents steel, which is an economic material of construction. Since carbon in combination with iron is most effective in producing desirable characteristics, it has been brought under careful regulation in the manufacture of steel. Some of the effects of carbon in iron are illustrated graphically in the equilibrium diagram, Fig. 11.

Phase Changes in Steel

This diagram (Fig. 11) indicates the phase changes in iron-carbon mixtures at various temperatures. No reference is made to the time it takes, after the temperature is changed, for these phase transformations to reach the point of equilibrium shown. The addition of carbon to iron lowers the melting point (until 4.3% carbon has been added) and also, at the higher temperatures, changes the melting point to a range where solid and liquid coexist. The delta iron range is restricted, and delta iron as a single phase is eliminated in the presence of 0.1% carbon, although some delta iron will still persist up to about 0.5% carbon. Below the delta iron region, gamma iron will absorb carbon up to compositions bounded by S-E, the limiting solubility (solid-solution solubility). When gamma iron contains in solution measurable amounts of any second constituent, such as carbon, it is known as "austenite." The temperature at which austenite exists by itself decreases as the carbon increases, along G-S to the so-called "eutectoid point" (0.80% carbon, 1333 F), and then the temperature increases as the carbon increases, since the austenite is unable to absorb additional carbon except at higher temperatures.

Critical Points in Steel

Referring again to Fig. 11, the critical points in steel lie along A_1, corresponding to line P-S at 1333 F, and along A_3 corresponding to line G-S at temperatures varying with composition. As they occur in heating and cooling, these critical points are again subdivided as Ac_1 and Ac_3 (heating), and Ar_1 and Ar_3 (cooling), respectively. These designations have no place in an equilibrium diagram since they are related to time, another variable, the effect of which is shown later. Below the temperature of 1333 F, carbon is relatively insoluble in the "ferrite," as alpha iron is known when rendered impure by the addition of other elements. The relative solubility of carbon in ferrite as compared with its solubility in austenite may be judged by comparing lines Q-P and S-E. This difference in solubility of carbon in the one as compared with its solubility in the other is the crux of practically all steel heat-treating.

The data in Fig. 11 show the end points of the possible transformation which are approached but never attained short of an infinite time. Processes in fabricating steel, however, may involve periods of time from seconds (spot welding) to several days (heat-treating large vessels). To evaluate the impact of these processes on the internal transformation of the steel, it is necessary to know the effect of time. A typical isothermal transformation is illustrated in the diagram, Fig. 12.

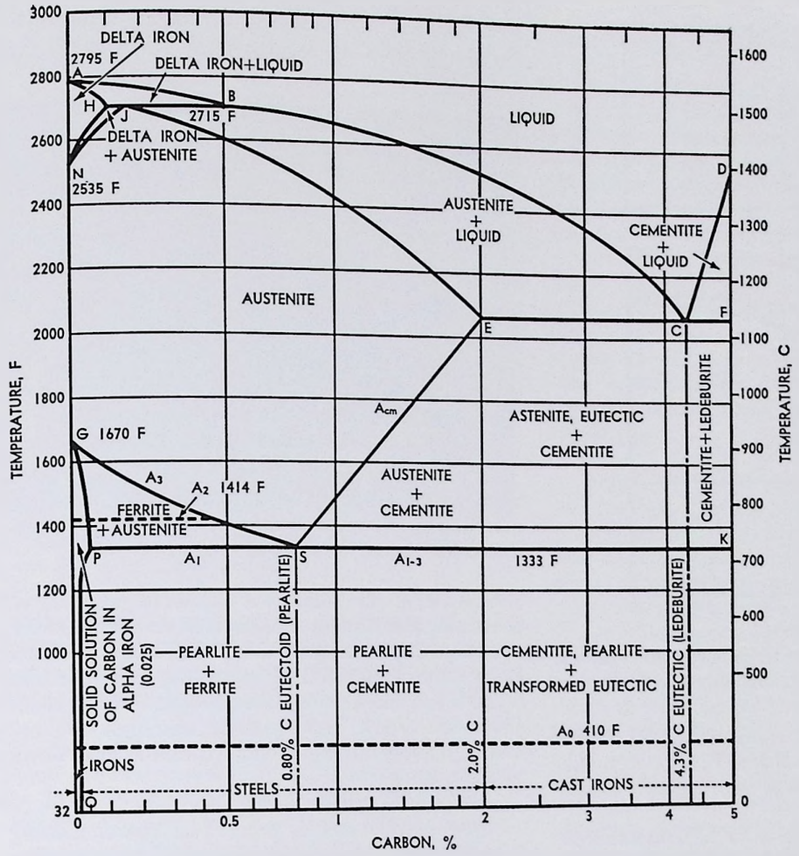


Fig. 11. *Equilibrium diagram showing phase solubility limits, carbon in iron*

Transformation in Steel

This diagram (Fig. 12) covers a specific composition of steel and indicates the time required for the progress of transformation from austenite to ferritic constituents at the various temperature levels. As an example, a hypoeutectoid* steel is heated to a temperature (about 1600 F) where it becomes completely austenitic, and it is then quickly transferred to and held in a furnace or bath at 700 F. From Fig. 11 it is evident that eventually ferrite and carbide should exist at this temperature, but Fig. 12 indicates how long this reaction takes. By projecting the time intervals during the transformation, as indicated in the lower portion of Fig. 12, to the top portion of the diagram, it will be noted that the austenite continues to exist for 3 seconds before anything happens; it then starts to transform, and at about 100 seconds the transformation is 50% complete; and at 700 seconds the austenite is entirely replaced by an agglomerate of fine carbides in ferrite.

*A steel containing less than 0.80% carbon. With 0.80% carbon it is known as eutectoid steel, and with more than 0.80% carbon it is known as hypereutectoid steel.

The symbols used in Fig. 12 are: A for austenite, F for ferrite, and C for carbide. At temperatures below about 600 F (the M_s line) this steel transforms to "martensite," a specialized form of ferrite and the hardest constituent of heat-treated steels. The "nose" of the curve in Fig. 12, at about 900 F, is of prime significance. It is clear that the transformation at this temperature is very rapid. Also, if this steel is to be quenched to form martensite (for maximum hardness), it must pass through the temperature range about 900 F very rapidly to prevent the transformation of some of the austenite to pearlite, which is much softer.

Alloying Elements

The effect of alloying elements on steel is indicated by the changes in the equilibrium diagram resulting from their addition to iron. Chromium first lowers the melting point of iron and then raises it as the alloy becomes leaner in iron. On the addition of chromium, the region of austenite (G-S-E-J-N in Fig. 11) is diminished, and the austenite phase is eliminated when the proportion of chromium reaches 13%. The addition of carbon (making a three-component sys-

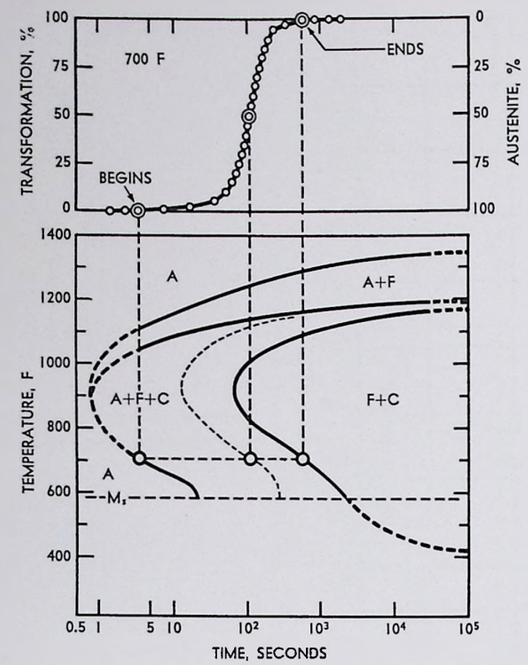


Fig. 12. *Typical isothermal transformation. Time required in a specific steel at 700 F taken as an example*

tem) changes these values considerably. At about 45% chromium and below about 1500 F, a nonmagnetic compound known as "sigma phase" appears, which renders the alloy brittle.

Nickel acts to widen the region of austenite, so iron alloys containing more than 25% nickel are practically austenitic at all temperatures. The action of all other alloying elements can be classed as (a) like chromium (ferrite formers), (b) like nickel (austenite formers), or (c) to form compounds.

Microscopic Examination

The microscope is traditionally the tool of the metallurgist for exploring changes in steel from alloy additions and thermal treatment. The surface of samples is ground flat and then polished to a mirrorlike finish. When the sample is placed under the microscope, the metal reflects light back to the eyepiece, and the discontinuities, which absorb some of the light, appear dark. Customarily, the samples are etched in mild acid solutions to bring out the characteristic appearance of: austenite, Fig. 13; ferrite, Fig. 14; pearlite, Fig. 15; or hard martensite in quenched steel, Fig. 16.

To the metallurgist, the structures shown in these photomicrographs are clear indices to the prior fabrication and to the mechanical properties. For good quality it is obvious, of course, that foreign matter

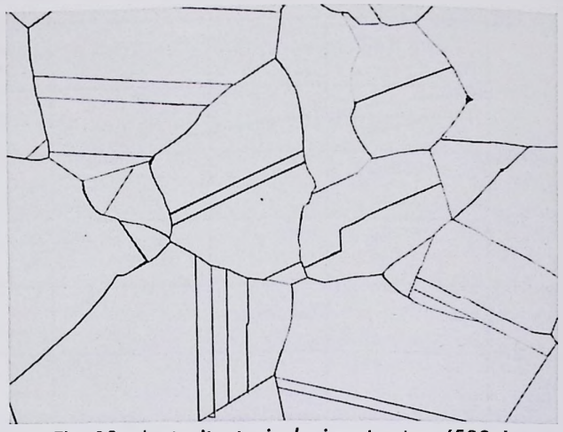


Fig. 13. *Austenite, typical microstructure (500x) reduced about ⅔*

in steel should be reduced to a minimum. The characteristic appearances of the photomicrographs mentioned above may be defined as follows:

Austenite. A solid solution of carbon in gamma iron, containing a maximum of about 2.0% carbon at 2090 F and about 0.80% carbon at 1333 F. It exists in ordinary steels only at elevated temperatures, but it is also found at ordinary temperatures in some stainless steels (18Cr-8Ni type). This structure has high ductility and high impact strength, which it retains even down to minus 320 F.

Ferrite. A commercially pure alpha iron containing a small amount of carbon (0.04-0.05%) in solid solution. This phase is soft, ductile, and relatively weak. Below about 40 F the impact strength is materially reduced. The tensile strength of commercially pure ferrite is of the order of 40,000 psi, the elongation is about 40%, and the hardness is from 90 to 95 Brinell.

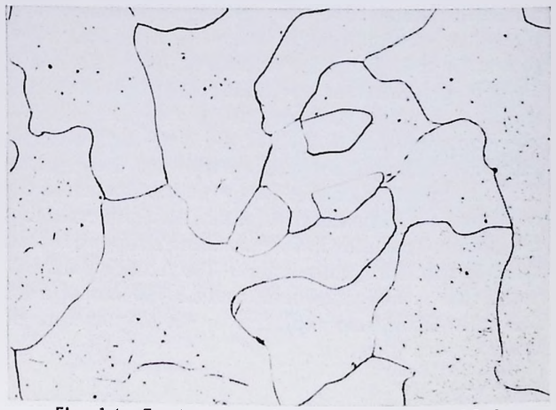


Fig. 14. *Ferrite, typical microstructure (1000x) reduced about ⅔*

Pearlite. A mixture, in lamellar form, of iron carbide (cementite) in ferrite, which occurs on slow



Fig. 15. *Pearlite, typical microstructure (1000x) reduced about ⅔*

cooling above the nose of the curve in Fig. 12. This condition is obtained by annealing or normalizing and generally represents an optimum high strength while retaining ductility and some machinability.

Cementite. Iron carbide, Fe_3C (cementite), a compound containing 6.67% carbon, is very hard (about 650 Brinell) and is extremely brittle. The ordinary occurrence of cementite is in slowly cooled hypereutectoid steels (carbon content greater than 0.80%).

Martensite. A very hard constituent obtained by the decomposition of austenite at low temperatures by rapid cooling.

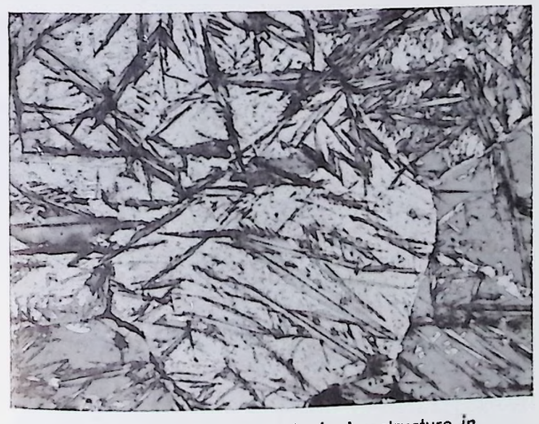


Fig. 16. *Martensite, typical microstructure in quenched steel (500x) reduced about ⅔*

Specific Effect of Alloying Elements

Steel alloys are the chief structural materials of modern engineering because of their wide range in properties to suit almost every specific application. These properties are affected directly not only by the characteristics and the amounts of the elements which, either alone or in combination, enter into the composition of the steel but also by their reaction as constituents under various conditions of temperature, fabrication, and use. For example, chromium increases resistance to corrosion and scaling; molybdenum increases creep strength at elevated temperatures; and nickel (in adequate amounts) renders the steel austenitic. The specific effects of the most important elements found in steel are outlined in the following:

Carbon. Carbon is not generally regarded as an "alloying" element because steel would not be steel without carbon. Nevertheless, it is appropriate in a discussion of alloying elements to first note the specific effects of carbon on the properties of steel.

In general, an increase in carbon content produces higher ultimate strength and hardness but lowers ductility and toughness of steel alloys. The curves in Fig. 17 indicate the general effect of carbon on the mechanical properties of hot-rolled carbon steel. Carbon also increases air-hardening tendencies and weld hardness, especially in the presence of chromium. In low-alloy steel for high-temperature applications, the carbon content is usually restricted to a maximum of about 0.15% in order to assure optimum ductility for welding, expanding, and bending operations. To minimize intergranular corrosion caused by carbide precipitation, the carbon content of aus-

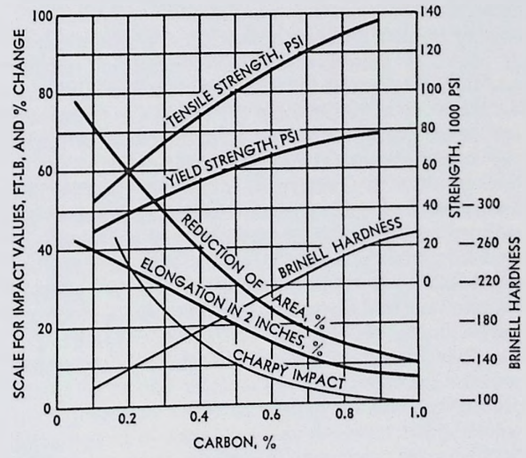


Fig. 17. *General effect of carbon on the mechanical properties of hot-rolled carbon steel*

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tenitic (18-8 type) alloys is limited in commercial specifications to a maximum of 0.08%, or even less, i.e., 0.03% in the newer extremely low-carbon grades used in corrosion-resisting applications.

No generalization is warranted regarding the effect of carbon on the long-time high-temperature (creep) properties of low- and high-alloy steels. In plain-carbon steels in the normalized condition, the resistance to creep at temperatures below 825 F appears to increase with carbon content up to 0.4% carbon; at higher temperatures there is but little variation of creep properties with carbon content.

An increase in carbon content lessens the thermal and electrical conductivities of steel and increases its hardness on quenching (see Fig. 18).

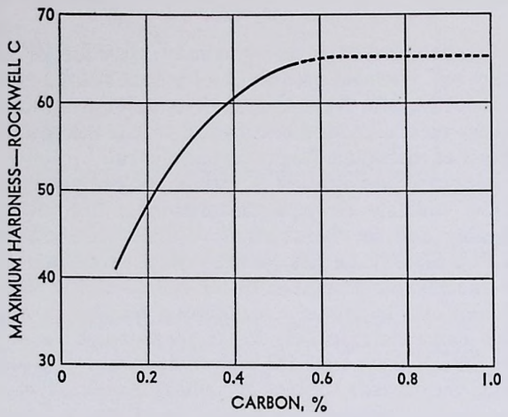


Fig. 18. *Order of maximum hardness attainable in steel with increase in carbon*

Phosphorus. High phosphorus content has an undesirable effect on the properties of carbon steel—notably on its resistance to shock and ductility when the steel is cold-worked. This embrittling effect, generally referred to as "cold-shortness," results from the tendency of phosphorus to enlarge the grain size and cause segregation. The harmful effect of phosphorus increases as the carbon content increases.

Phosphorus is effective in improving the machinability of free-cutting steels. This is related to its embrittling effect causing breakage of the chips on machining. In open-hearth (carbon) steels, phosphorus is limited to a maximum of 0.04%. In alloy steels intended for boiler applications, the permissible phosphorus content is still less. Its presence is objectionable for welding. Phosphorus has recently come into prominence as an alloying element (up to 0.15%) in proprietary low-alloy high-strength steels, where increased yield strength and resistance to atmospheric corrosion are primary requirements. In certain acids, however, high phosphorus may increase the corrosion rate.

Silicon. The usefulness of silicon is mainly because of its qualities as a deoxidizer and degasifier, hence it contributes to the production of sound steel. When added in amounts up to 2.5%, silicon increases the ultimate strength of the steel without decrease in ductility. Silicon in excess of this amount causes brittleness, and amounts higher than 5% make the steel nonmalleable.

Resistance to oxidation and surface stability of steel are increased by the addition of silicon. These desirable effects partially compensate for the tendency of silicon to lower the creep properties of steel. Silicon increases the electrical resistivity of steel and decreases hysteresis losses. Silicon steels are, therefore, widely used in electrical apparatus.

Manganese. In addition to being an excellent deoxidizer and sulfur neutralizer, manganese improves the mechanical properties of steel, notably the ratio of yield strength to tensile strength at normal temperatures. As an alloying element of steel, manganese serves as an inexpensive means of preventing "red-shortness" (brittleness), and it will improve rolling properties, hardenability, and resistance to wear. However, manganese increases the crack sensitivity of weldments, particularly so with steels of higher carbon content.

Unlike silicon, the effect of manganese is beneficial on the creep properties of steel. It does not appear to have any specific influence on the resistance to oxidation or corrosion of steel.

Chromium. This element is the essential constituent of stainless steel. Chromium has the effect of increasing yield and ultimate strength, hardness, and toughness of steel at room temperature. It also contributes somewhat to the strength of steel at high temperatures.

Resistance of the steel to wear or abrasion and increase in cutting ability are two of the outstanding effects of the addition of chromium. A steady improvement is noted in resistance to atmospheric corrosion and to attack by many reagents when the chromium content of steel is increased above 12%. The chemical properties of the steel, however, are affected by the carbon content; and, generally, the higher the chromium and the lower the carbon, the more resistant the alloy will be to certain types of corrosion. Chromium is also instrumental in increasing the resistance to oxidation at elevated temperature and, therefore, is of prime importance in steels intended for high-temperature service.

The addition of 1% or more of chromium may cause appreciable air-hardening in the steel. Air-hardening is a direct function of chromium and carbon content up to about 12% Cr. Low-carbon alloy steels containing over 12% Cr tend to become nonhardening, but the impact strength is reduced and the ductility is poor. Chromium lessens both thermal and electrical conductivities.

Nickel. Increased toughness, resistance to corrosion, and hardenability are the principal benefits when nickel (upwards of 5%) is added to steel. Nickel dissolves in the iron matrix in all proportions and, therefore, raises the ultimate strength without impairing the ductility of the steel. Nickel is particularly effective in increasing the impact properties—especially at low temperatures. It is a useful element in carburizing steels, and it is used to improve core properties in steels for roller bearings.

Probably the most important use of nickel as an alloying element is in combination with chromium in amounts of 8% Ni or more. By the addition of nickel, the high-chromium-iron-carbon alloys become austenitic in character. The various combinations of these two elements produce alloys (18-8 type) with properties which cannot be obtained with equivalent amounts of any of the elements alone. Such steels are resistant not only to corrosion by the atmosphere and other sources but also to oxidation at high temperatures, and they possess greatly enhanced creep strength.

Nickel has only a slight beneficial effect on creep properties of low-alloy ferritic steels. It reduces the coefficient of thermal expansion and diminishes the electrical conductivity of steel.

Molybdenum and Wolfram (Tungsten). Addition of molybdenum to steel increases its strength, elastic limit, resistance to wear, impact qualities, and hardenability. Molybdenum contributes to the "red-hardness" of steel, that is, the steel can be heated to a visible degree without materially lowering the hardness. It also adds to the resistance of steel to softening on tempering and restrains grain growth. Making chromium steels less susceptible to temper embrittlement is another beneficial influence of molybdenum.

The principal use of molybdenum in improving resistance to corrosion is in stainless steels of the 18-8 type. It improves the inherent resistance to corrosion of these steels in reducing chemical media and also increases their passivity under mildly oxidizing conditions. Experience indicates that under certain conditions molybdenum effectively reduces the susceptibility of stainless steels to pitting. Molybdenum is the most effective single alloying addition that increases the high-temperature creep strength of steel.

Wolfram (also known as tungsten) is similar in behavior to molybdenum. However, of the two, molybdenum appears to be the more effective in increasing strength for a given cost. Wolfram is an important element in high-speed cutting tools and in cemented-carbide tools.

Vanadium. While it is to some extent a degasifying and deoxidizing agent, vanadium is seldom used in that capacity because of its high cost. Vanadium, therefore, is applied chiefly as an alloying element in steel. Essentially a carbide-forming element which stabilizes the structure especially at high temperatures, it increases the strength, toughness, and hardness of steel. Vanadium minimizes grain-growth tendencies, thus permitting much higher heat-treating temperatures. It also intensifies the individual properties of other elements in alloy steels. In steels containing 0.5 to 1.0% molybdenum, small additions of vanadium (0.1 to 0.5%) produce a pronounced improvement in high-temperature creep properties if these steels are given certain high-temperature heat-treatments. Vanadium improves the cutting properties of tool steels.

Aluminum, Titanium, and Niobium (Columbium). Aluminum is an important minor constituent of low-alloy steels. It is an efficient deoxidizer and is almost universally used in the production of killed steel. When added to steel in appreciable quantities, aluminum forms tightly adhering refractory oxide scales and thus increases resistance to scaling. It is difficult, however, to add appreciable amounts of this element to steel without producing undesirable effects. It is generally agreed that an excessive quantity of aluminum has a detrimental effect on creep properties, particularly in plain-carbon steel. This is attributable to its grain-refining effect and, what is even more serious, to the acceleration which it exerts on spheroidization and graphitization of the carbide phase.

In the amounts that it is customary to add, aluminum does not increase resistance to ordinary forms of corrosion. Actually, because of its avidity for oxygen, steels of high aluminum content generally contain numerous alumina inclusions, which tend to produce pitting corrosion under conditions favorable for corrosive attack. Aluminum, however, increases resistance to oxidation when applied to steel as a surface coating, as in the calorizing process.

Titanium and niobium (also known as columbium) are potent carbide-forming elements. In addition, titanium is also a good deoxidizer and denitrider. These elements are most effective in the chromium-nickel austenitic alloys (18-8 type) where they react more readily with carbon than does chromium, allowing the chromium to remain in solid solution and in the concentrations necessary to maintain the "stainlessness" (corrosion resistance) of the steel. Titanium and niobium (or niobium plus tantalum) are sometimes used to reduce air-hardening tendencies and to increase resistance to oxidation in steel containing up to 14% Cr. These elements seem to have a beneficial effect on the long-time high-temperature properties of chromium-nickel stainless steels. Both niobium and titanium have been used in some of the new "superalloys" to improve high-temperature properties.

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Copper. Addition of copper in small amounts improves the resistance of steel to atmospheric corrosion and lowers the rate of attack in reducing acids. Copper is not resistant to sulfur compounds at elevated temperatures. Consequently, it is not ordinarily used in low-alloy steels intended for high-temperature service. Copper is added (up to 1%) in low-alloy constructional steels to improve yield strength and resistance to atmospheric corrosion, and also in some of the high-alloy steels to increase resistance to corrosion by sulfuric acid. Copper, in amounts between 2.0 and 4.0%, also has been added in certain complex high-alloy austenitic steels, in conjunction with molybdenum, titanium, and niobium, to cause precipitation-hardening, thereby improving the creep- and rupture-strength properties.

Some specific functions of alloying elements in steel are given in Table A in the Appendix, page 22-A1.

Heat-Treating Practices

The properties of steel may be altered by various thermal treatments that control its structure. These include annealing, normalizing, spheroidizing, hardening, and tempering, briefly described in the following:

Annealing. Depending upon the results desired, annealing may be performed in several ways. Steel may be annealed to meet specified requirements as to softness and ductility, to refine its grain structure, to remove internal stresses, to improve its machinability, or to alter its microstructure.

In full annealing, the steel is heated at a predetermined rate to about 100 F above the upper limit (locus of A_3, Fig. 11) of the critical range; it is held there for a suitable length of time, and then it is allowed to cool slowly (at a controlled rate) in the furnace. The microstructure of fully annealed hypoeutectoid steel (containing less than 0.80% carbon) consists of ferrite and pearlite.

When a full anneal is not necessary to relieve stresses set up in fabrication or to restore the ductility and improve the drawing properties of work-hardened steel, a "works" or "process" anneal is used. This treatment, sometimes referred to as stress-relieving, is carried out at temperatures below the critical range, usually between 1000 and 1300 F. By rule of thumb, the material is held at that temperature 1 hr per in. of maximum thickness and is then allowed to cool in still air.

Normalizing. This heat treatment is, in effect, a special annealing process—except that the steel is allowed to cool in still air from above the critical temperature. Normalizing is sometimes used as a preliminary step, before full annealing, to remove the

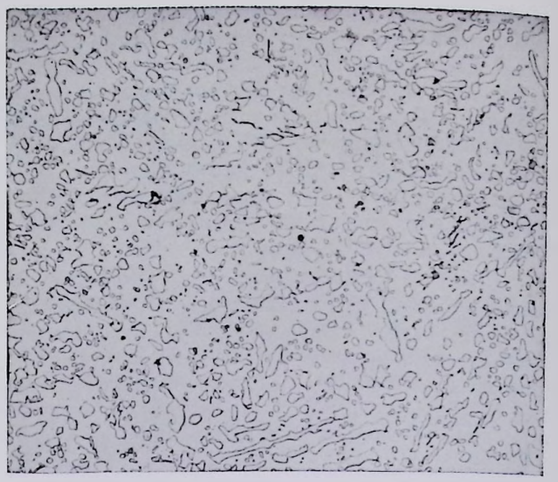


Fig. 19. *Spheroidization in steel. Typical microstructure (750x) reduced about ¾. The Fe_3C has been almost completely agglomerated*

effects of previous thermal treatments that may have rendered the structure of the steel unsatisfactory. Normalizing relieves the internal stresses caused by previous working and, while it produces sufficient softness and ductility for many purposes, yet it leaves the steel harder and with higher tensile strength than after full annealing. For the purpose of removing cooling stresses, normalizing is often followed by tempering (see below).

Spheroidizing. The usual object of spheroidizing (another type of annealing) is to soften the steel sufficiently to improve its machinability. Heating fine pearlite for a long time just below the lower critical temperature of the steel, followed by very slow cooling, will cause spheroidization—an agglomeration of the iron carbide (Fe_3C), which eventually becomes rounded and assumes the globular or spheroidal shape shown in Fig. 19. The properties of this product are similar to those of coarser pearlite. This treatment is most commonly applied to the higher carbon (above 0.45% C) and hypereutectoid steels.

Hardening (Quenching). When steels of the higher-carbon grades are heated to produce austenite and then cooled rapidly (quenched), the austenite decomposes into the constituent "martensite" (see Fig. 16). Martensite is formed at temperatures below about 400 F, depending upon the carbon content and the type and amount of alloying elements in the steel. It is the hardest form of heat-treated steels and has high strength and resistance to abrasion. Martensitic steels have poor impact strength and are difficult, if not impossible, to machine.

Tempering. Normalizing some air-hardening steels and liquid-quenching treatments impart the desired degree of hardness to the steel but also make it brittle. The object of tempering, a secondary treatment, is to remove some of that brittleness by allowing certain transformations to proceed in hardened steel. It involves heating to a predetermined temperature, always below the critical, followed by any desired rate of cooling. Some of the hardness of the steel is lost by tempering, but its toughness is increased, and stresses set up by quenching are eliminated. The usual tempering range for carbon steels is 400 to 750 F; the higher the tempering temperature the softer and tougher the steel will become. Some steels may become embrittled on slowly cooling from certain tempering temperatures. Steels so affected are said to be "temper-brittle." To overcome this difficulty, steels of that type are cooled rapidly (quenched) from the tempering temperature.

Mechanical Working of Steel

Metals may be shaped directly by casting or formed by mechanical working, which also imparts desirable properties. Metals may be worked either hot or cold. The initial working of the metal, as in forging, pressing, rolling, and piercing, is almost always a hot operation, while subsequent working, as in drawing, forming, and stamping, is usually a cold operation. Any mechanical work applied to the metal below its recrystallization temperature, which means at or about room temperature in most cases, is cold work. Mechanical work performed above the recrystallization temperature of steel is hot work, and the simultaneous annealing which occurs at that temperature prevents stress- or work-hardening. For satisfactory plasticity, hot work is usually performed above the transformation temperature range (A_3), which varies with the composition of the steel.

The temperature at which steel is mechanically worked has a profound effect upon its properties. Cold work increases the hardness, tensile strength, and yield point, but its indices of ductility—elongation and reduction of area—are decreased. The extent of the work-hardening, with progressive elongation of the grains in the direction of working, depends upon the amount of cold work. If the work-hardening caused by the necessary shaping operation becomes excessive, further work is impeded, and the metal must be given some sort of heat-treatment to restore its ductility.

Hot-working improves the quality of the steel by closing blowholes in the ingot and thus increases its homogeneity and soundness. The crystal structure is refined (reduced in size), which increases the ductility and toughness of the metal. Hot work does not harden the steel during the deformation process. It is a rapid and comparatively inexpensive method of production for tolerable dimensional accuracy and surface appearance.

Obviously, there is a great variety of steel-working equipment for shaping and forming steel products used in industry. The following deals briefly with the working of steel into plate, strip, tubing, headers, etc., for boiler construction.

Methods of Hot-Working

Forging. Forging is the working of metal by intermittent or continuous application of force. A forging hammer works the metal by successive blows, while in a forging press the pressure is applied slowly, and the steel is thoroughly kneaded in every direction.

Steam hammers used in boiler works are of the open-die type, but much forging of steel products is done in closed dies by drop forging. Hydraulic or steam-hydraulic presses are used to make large forgings. Nozzles, drum heads, saddles, and other boiler parts are usually formed on presses (see Chapter 24) using suitably shaped dies.

At the B&W Tubular Products Division, ingots are bloomed into square-section bars (blooms), which are then (after reheating) rolled into rounds for the manufacture of seamless tubing. When extra sectional thickness is required in a tubular element, the tube is upset in mechanical forging presses.

Rolling. The quickest method of reducing steel to shape is by rolling, and much the greater part of steel tonnage is so worked. The cast or forged metal in the form of ingot, slab, bar, billet, etc., is first brought to the proper temperature throughout and is then passed through rolls which revolve in opposite directions. The flow of metal is continuous and almost entirely in a longitudinal direction. The cross-sectional area is reduced, the internal structure is refined by reduction in grain size and change in shape (compared with cast or unworked metal), and the metal is shaped to the desired form.

Production of Plates. Plate steels are very important in boiler construction. The drum shell, drum ends (heads), certain drum internals, and certain structural supports are made from plate.

Plates are made from ingots of rectangular shape by using a slabbing mill, which performs the same function as a blooming mill but permits rolling the wider sections desired in the initial operation in the production of plates, sheet, and strip. After conditioning and reheating, the rough slabs are rolled to finished plates on either a sheared-plate mill or a Universal mill. Some extremely heavy plates may be rolled directly from the ingot. Sheared plate is rolled between straight horizontal rolls only and must be

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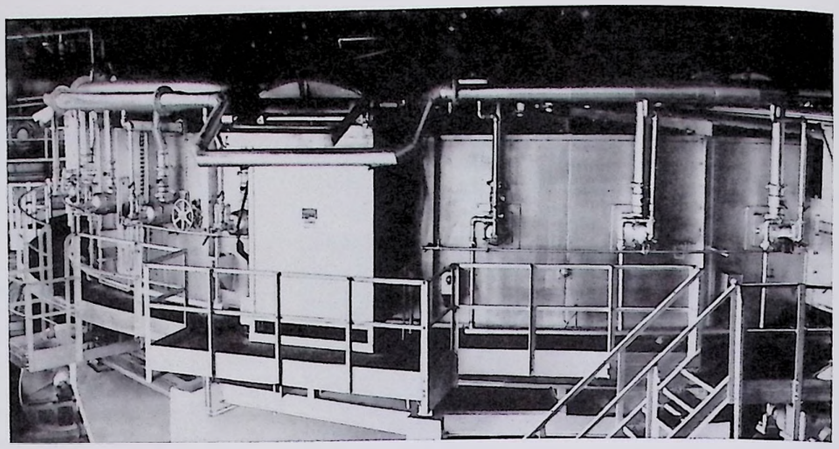


Fig. 20. *Rotary billet-heating furnace used to heat B&W tube-rounds*

trimmed to the finished size on all edges. Universal-mill plates are rolled between horizontal rolls and vertical rolls and are trimmed on the ends only. Mills of various designs are used for rolling plates.

For sheared-mill plates it is customary to roll the ingots or slabs in both directions (cross-rolling) to give uniform physical properties whether the test sample is taken parallel or transverse to the greatest dimension. On the Universal mill, plates are rolled in one direction only, the vertical rolls producing a straight, fairly parallel mill edge. The product of such a mill cannot be expected to have uniform properties in both directions. This metallurgical practice is important in the judicious selection of suitable quality plate material for various parts of the boiler. Drum plate, for instance, must be of higher quality and uniformity than plate used solely for structural purposes.

Tube Manufacturing

Many thousand feet of steel tubes of suitable size, shape, and material are required for a large boiler unit. These tubes are used in saturated surfaces including furnace walls and floors, and in superheaters, reheaters, economizers, and air heaters. Headers also might be classed as tubes, since the process of manufacture is similar, but the size range (9⅝ in. to 14 in.) is beyond that ordinarily associated with tubes.

Seamless steel tubing is the generally used standard for high-pressure applications. This is chiefly because of wall-thickness requirements, which, in many cases, are beyond the range suitable for manufacture by the resistance-welding method. Resistance-welded carbon-steel tubes may be used in boilers of moderate pressure. Although lap-welded steel tubing is now obsolete for boiler service, its use is permitted by the ASME code.

Seamless Tubes

Seamless tubes are made by piercing from solid rounds of carbon or alloy (Croloy) steels, without weld or seam. Tubes for boiler application are furnished either hot finished or cold drawn, depending upon size, tolerance, and finish desired. Metallurgically the two are quite similar: the difference is mainly in surface finish and the permissible tolerance in dimensions. To obtain a smooth even surface and close tolerances, stainless steel tubing, except in special instances, is finished by cold-drawing only. This process also permits control of grain size in final heat-treating, which is important in high-temperature application of austenitic stainless steels.

The production of seamless tubes requires special tools, careful control of heating, and exactness in procedure.

Centering and Heating. So that the piercer point may be accurately centered in relation to the piercing rolls, the hot-rolled conditioned round or machined-turned round (if of stainless Croloy), of a somewhat greater weight than the finished tube, is first sent to a centering machine, where a shallow hole is drilled in the center of one end. In some of the highly alloyed stainless steels, a small diameter hole may be drilled throughout the entire length of the billet to ease the load in the piercing mill and to preserve the piercer point.

The round, which may range in length from about 40 in. to 144 in. and from 3 in. to 8 in. in diameter, is then heated to the proper temperature in a modern rotary-hearth furnace (see Fig. 20).

Piercing. Piercing is done in machines especially designed for the purpose. The round is gripped between the faces of two revolving conical-shaped rolls that rotate in the same direction. Each roll is directly

driven by its own motor. By the rapid rotation and kneading action of the rolls, set at the proper angle, the round, somewhat reduced in diameter, moves forward spirally over the piercing point held firmly in position. The piercing operation is illustrated dia-

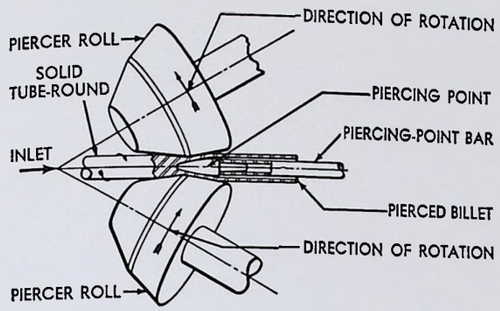


Fig. 21. *Piercing operation (diagrammatic) in B&W's cone-type piercing mills*

grammatically by Fig. 21. After piercing, the bar is withdrawn from the inside of the tube, and the rough pierced hollow may be reheated in a suitable furnace to the correct temperature for rolling.

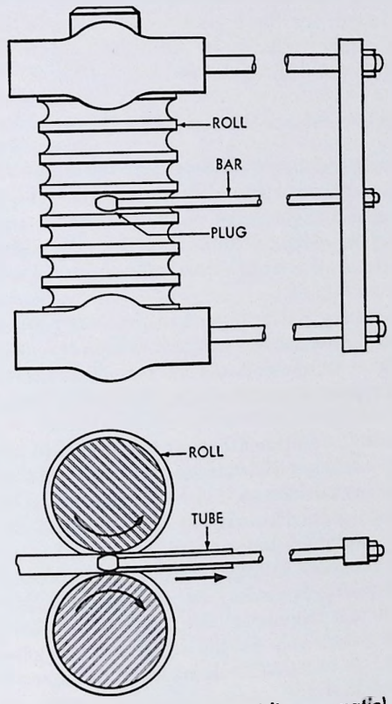


Fig. 22. *Rolling operation (diagrammatic) in tube-rolling mill*

Rolling. The function of the rolling mill is to lengthen the tube and reduce the wall thickness to the approximate dimensions required. A powerful ram forces the tube into the grooves of the rolls, the rotation of which carries it forward over a plug and bar, working the metal between the grooved surfaces of the rolls and the plug (see Fig. 22). Five, six, or even more passes with change of plugs may be required in rolling stainless Croloys. To make the wall uniform and to avoid ribs or overfills, the tube is turned 90 degrees after each pass through the rolling mill.

Reeling and Sizing. After rolling, the tube is in a semifinished condition, that is, of required wall thickness and approximate diameter but with a somewhat rough surface. Directly after rolling and while the tube is still hot, it is conveyed to a reeling machine (see Fig. 23), where it is forced spirally over a smooth mandrel, or "reeler plug," supported by a thrust bar on the outlet side of the mill. In rolling over this plug, light overfills and scratches are removed, and the tube is smoothed and burnished. The tube is also rounded and expanded a little in diameter, but the wall thickness remains approximately constant or, depending upon the pressure applied, is slightly reduced. After reeling, the tube is taken to a sizing mill.

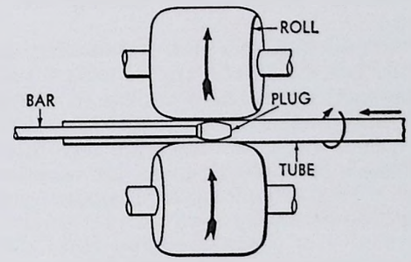


Fig. 23. *Reeling operation (diagrammatic) in reeling machine*

The function of the sizing mill is to hot-size the tube to its proper diameter. Its main elements are several sets of roll stands with single-grooved rolls set in tandem. Following the sizing operation, the tube is annealed or heat-treated, if required, and then straightened, cut to length, hydrostatically tested, and finally inspected. Appropriate samples are secured for tensile, hardness, and deformation tests, and the length, gage, and diameter of the tubes are checked.

To produce smaller tubes in sizes below about 2 in., the tube, after reeling and reheating to a working temperature, is reduced in diameter or increased in length in a reducing mill.

Cold-Drawing. Tubes requiring finer finish or closer dimensional tolerances and tubes of special or smaller sizes than can be obtained by the hot-finish process

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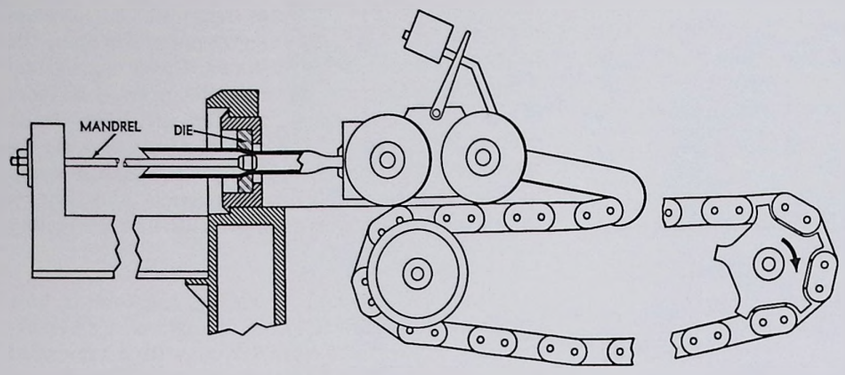


Fig. 24. *Drawbench for producing close-tolerance fine-surface-finish cold-drawn tubes*

are cold drawn. Tubing to be cold drawn is made in the same manner as hot-finished tubing. The cold-drawing is a subsequent additional operation.

The cold-drawing is done on a sturdily constructed drawbench fitted at its center with a heavy steel holder for the die through which the tube is drawn. The arrangement of the drawbench, tube die, and mandrel is illustrated diagrammatically by Fig. 24. The operation of annealing, pickling, lubricating, and drawing is repeated until the tube is of the desired size. Drawing reduces both diameter and wall thickness.

In the final operations after cold-drawing, an appropriate heat-treatment is applied to the tubes. Depending upon the particular analysis, this may be a full anneal, normalizing and tempering, stabilizing, or a quench treatment. Tubes are then descaled, straightened, cut to length, tested, and examined by inspectors familiar with the requirements for high-quality pressure tubes.

Extrusion Process for Seamless Tubes

Certain sizes and types of tubes, usually of the more "refractory" alloys for which rotary piercing is not well suited, are now made by the hot-extrusion process. This method of manufacture, based on the Ugine-Sejournet hot-extrusion process developed in France, was first used in the United States by B&W. In this process a hollow billet is confined in a container between the ram of the press and the die. A mandrel extending from the ram passes through the billet and the die. As the ram moves forward, the metal is squeezed out between the die and the mandrel, using a special glass lubricant.

Although the billet may be hollowed in the same press, prior to extrusion, by forcing the mandrel through the blank, this method is limited by the strength and stiffness of the mandrel. In an alternate method used by B&W, the billet is first pierced in a separate press to an oversize hole. This permits the use of a larger and stiffer mandrel, with less wander at the far end of the pierced billet. Use of a larger diameter mandrel is important because, in some cases, the ratio of extrusion mandrel length to diameter may be about 30 to 1. Use of a separate piercing press improves the concentricity of the final tube, since it depends upon the concentricity of the pierced billet prior to extrusion.

With present facilities it is possible to produce tube hollows ranging in size from 1½-in. to 3½-in. OD 0.160-in. to 0.600-in. wall thickness, and from 20-ft maximum length for the heavier wall to 60-ft maximum length for the lighter wall. These hollows are subsequently reduced by cold-drawing to finished tubing of the desired size.

Electric-Resistance-Welded (ERW) Tubes

Electric-resistance-welded steel tubing is made by forming flat strip into tubular shape (skelp) and welding the edges together in an electric-resistance-welding machine. A mill used for the continuous production of electric-resistance-welded tubing is shown in Fig. 25.

This tubing is widely used in stationary and marine boilers. Within the size range in which it is produced, it is equal to seamless carbon-steel tubing and is in some respects superior. It is, therefore, being used to an increasing extent by manufacturers of steam-generating equipment. Compared with seamless tubing, electric-resistance-welded boiler tubing has a smoother surface, a more uniform wall thickness, and less eccentricity. Because of softness and uniformity, welded tubes are easily installed.

The pressure tubing produced by electric-resistance welding is mainly of carbon steel, and both rimmed- and killed-steel strip are used. Rimmed steel is used extensively in the lower-carbon grades because of its excellent welding characteristics. In rimmed steels, B&W uses single-width strip to avoid welding rim to core, as would be the case when welding slit strip cut from wide strip. If the additional cost is warranted and killed steel is required by the specifications, slit strip may be used because of its more homogeneous structure.

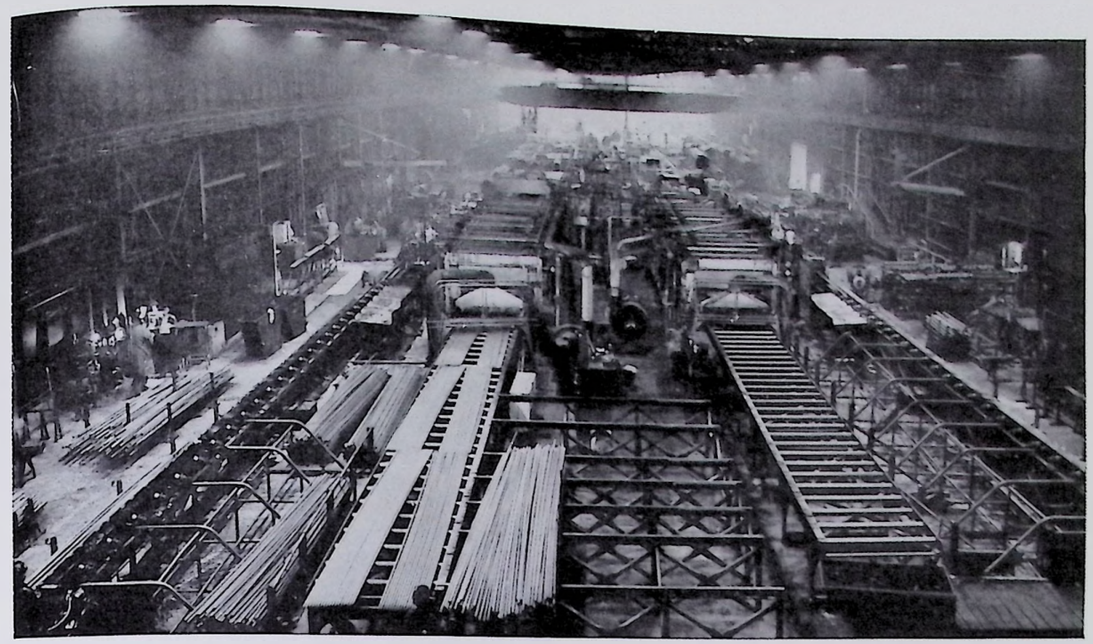


Fig. 25. *B&W electric-resistance-welded-tube mill for continuous tube-length production*

Electric-resistance-welded boiler tubing is commonly made in sizes from ¾-in. to 4-in. diam, in gages from 0.049 to 0.180 inch.

Hot-rolled strip, in coils, is used for most of the tubing produced by this method. The first operation is running the strip through a scale breaker under tension onto a recoiler. After recoiling, following the scale-breaking operation, the coil is backspun to open the individual wraps. The coil is then pickled, rinsed, and conveyed to the cold-rolling and sizing mill.

Cold-rolling is applied to the hot-rolled strip to smooth its surface and to obtain a uniform dimensional accuracy. Close control of the limits of thickness requires the use of a magnetic gage, running on the strip.

After cold-rolling, the strip is recoiled and the coil is fed into a leveler for flattening. The ends are then cut square and butt welded to the ends of the preceding and following coils to form an endless strip. Following the butt-welding operation, the moving strip is slit to the exact width required for the tube to be made. Width and thickness within close limits are essential for uniformly perfect welding. For strong and sound welds, slitting also provides fresh edges, eliminating rust and strip-edge damage.

The strip is folded into open-tube form (skelp) in a forming mill, preparatory to welding. In the resistance welder, beryllium-copper roll-type electrodes come in contact with the upper surfaces of the formed tube on either side of the seam, while pressure rolls located directly below the electrodes provide the pressure required to bring the edges of the tubing into contact. The outside and inside flashes (or upsets) formed in the welding operation are removed by special tools. The inside weld bead is trimmed within a specified height; the outside bead is cut off. Both of these operations are continuous as the tubing emerges from the welding mill.

On leaving the resistance welder, the tubing is accurately sized and straightened in a three-pass mill. As the welded tubing emerges from the sizing mill, it is cut to required length. The transverse welds at coil ends are not allowed as a part of the tube and are cut out. The tubes are then completely normalized in a controlled-atmosphere furnace to a uniform structure throughout, and the surface is given a rust-retardant oxide finish.

The tubes emerging from the normalizing furnace are conveyed through rotary straighteners discharging to the cutoff machines, where each tube is cut exactly to the required length. Coupons cut from the ends of tubes are tested mechanically, according to specifications. One coupon is flattened with the weld at the flattened edge, another coupon is expanded, and, if required, a third coupon is crushed longitudinally. In these tests, to pass inspection the coupons must not crack or split in the tube metal or open at the weld.

Other machines then perform operations as required in trimming to exact length, chamfering ends inside and outside, and reaming the inside bore at the ends.

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Inspection and Testing

Immersed ultrasonic testing is now being used to inspect the weld of electric-resistance-welded carbon-steel tubing. Here the sound is beamed through water into the weld of the moving tube a few feet beyond the welding head. Defects in the weld cause deviations in the normal pattern shown on a cathode-ray screen. Various alarm devices are connected into the cycle to alert the mill operators of possible defective material. In this case, ultrasonic testing is being used as an added inspection tool to the normal quality-control checks.

All tubes are subjected to hydrostatic pressure from 1000 to 6000 psi or as required by the specifications. Hydrostatic pressure is maintained for several seconds, and a shock stress is induced while under test pressure by striking the tube with mallets.

After hydrostatic testing, the tubing is ready for final inspection, including surface examination, micrometer check of dimensions, and a length check. The finished tubing, stamped with the Company's identification mark and the inspector's symbol, is then stenciled and bundled for shipping.

Quality and Metallurgical Control

To maintain a consistent standard of high quality in electric-resistance-welded tubing, a chemical analysis must be made of each heat of strip received. To preclude mechanical difficulty in forming and welding, camber and strip dimensions are also constantly checked. Etch tests are made to determine the amount of rim at strip edges of rimmed-type steel and to detect laminated strip. After cold-running, frequent hardness tests are also made to determine uniformity for forming. Tension tests are made at intervals and as required by specifications. Any defects or flaws in the weld are revealed by use of the magnetic-particle (Magnaflux) test.

Final inspection must include check of dimensions, visual observation for surface defects, burn-in, bead-trim height, scratches, and surface pits or slivers.

Air heater tubes are made essentially in the manner described in the foregoing, except that the hot-rolled strip is not sized by cold-rolling. Since the service is nonpressure, these tubes are not subjected to such rigorous or searching tests and are not generally normalized following welding.

Electric-resistance-welded boiler tubes are made to meet the requirements stipulated in specifications issued by the American Association of Railroads, American Bureau of Shipping, United States Navy, United States Coast Guard Marine Inspection, the American Society of Mechanical Engineers, and other governmental or private authorities.

Large-Diameter Thick-Wall Pipe

Large-diameter thick-wall pipe, from 8¾-in. to 27½-in. OD and wall thickness from ⅝ in. to 4 in., is fabricated at the B&W Barberton Works by piercing and drawing (see Chapter 24).

Welding in Boiler Construction

Prior to 1930, riveted drums were used for practically all water-tube boilers, although a few drums, necessary in early high-pressure applications, were forged and turned from the solid ingot because of the limitations of the riveted joint. Credit for the development of fusion welding, which eliminated the riveted drum joint, is noted in Chapter 24. In addition to this achievement of such far-reaching importance, B&W has contributed much to the improvement of welding. Among these contributions are:

First to apply radiography to production inspection of welded pressure-vessel seams.

Constructed the first furnace for the over-all stress-relief of welded boiler drums.

First to adopt alternating current for welding, in the fabrication of boiler drums and other pressure vessels, to avoid the troublesome magnetic interference of arc blow.

Compounded the first electrode flux made entirely of inorganic matter.

Manufactured the first covered electrodes for automatic deposition. This electrode permitted complete traverse of a longitudinal or circumferential seam without breaking the arc, and it was the first electrode that could be deposited with alternating or direct current of either polarity.

Welding Processes

A weld has been defined as a localized coalescence of metal, wherein coalescence is produced by heating to suitable temperatures with or without the application of pressure and with or without use of filler metal°. In this section the more important welding processes used in industry may be classified under two broad divisions—pressure welding and fusion welding.

Pressure Welding. The surfaces to be joined are heated to high temperatures and welded by applying pressure. No molten filler metal is added to the joint. Solid-phase welding of this type is exemplified by the old-time blacksmith weld. In the construction of boiler components, resistance butt welding of superheater and economizer tubing and the studding of various tubes fall into this category. Seam, spot, projection, and flash-welding methods also are in this category but are not used extensively in the manufacture of boiler components. If there is any molten metal in the weld zone, the amount is usually very small.

The source of heat energy for pressure-welding surfaces may be electrical, oxyacetylene gas, or a forge. The weld is produced by bringing the abutting heated surfaces together under pressure.

Attachment of studs, bolts, etc., to a tube or plate surface may be considered as a special case of pressure welding. In this method a small amount of molten metal is produced between the stud and the tube or plate surface by electric-resistance heating of the two parts at their point of contact. The stud is then projected against the work piece by suitable force. In this operation, control of timing, current values, and push-up force are of paramount importance. Most of the molten metal is ejected from the weld area, and the actual weld is really a plane of fusion between the stud and the tube or plate surface.

Fusion Welding. Of the many processes grouped under the general heading of fusion welding, the most generally used is arc welding. In arc welding, the two most popular types are known as (a) the open-arc method (using a coated rod), and (b) the submerged-arc method. Both methods are extensively used in boiler construction.

The original metal-arc-welding process was introduced near the end of the nineteenth century. In this process an arc is produced between a metallic electrode and the work piece. The heat energy of the arc then fuses not only the metallic electrode but also the surface of the work in contact with the arc. At first, using bare electrodes (unshielded arc), the weld metal lacked ductility and was excessively porous because of high oxygen and nitrogen content.

In the late twenties, flux coatings were developed for metallic electrodes. To produce weld metal essentially free from oxides and nitrides, the welding wire was covered with fluxing ingredients and combustible materials. In this way the molten metal was shielded from the surrounding air during the period of heat transfer through the arc, and undesirable oxides were cleansed or fluxed from the weld puddle.

Suitable Electrodes. Specific electrodes have been developed to enable the deposition of weld metal in a variety of positions and under various conditions. All of these coated electrodes have been classified by the American Welding Society and the American Society for Testing Materials, primarily according to usability and the type of deposit produced. Classification is based on the type of coating applied to the weld wire. For instance, for all-position welding of X-ray quality suitable for pressure work, the coatings of the E-6010 electrode, used with d-c reverse polarity, and the E-6011 electrode, used with a-c equipment, are of high-cellulose type, where the soundness of the weld metal depends primarily upon the shielding afforded by the combustion of some 30% of the cellulose. In welding structural assemblies where freedom from porosity is not an essential requirement, the E-6012 and E-6013 type of electrode used for d-c straight-polarity and a-c equipment, respectively, is usually preferred because of greater speed. Rutile, a natural titanium dioxide, is the primary constituent of these electrode coatings and serves as an excellent arc stabilizer, permitting deposition of weld metal in all positions at relatively high current rates.

In boiler construction most of the work is done in the downhand position, and for this purpose a group of high-speed electrodes, referred to as the E-6020 and E-6030 type, has been developed. Electrodes of this type carry a very heavy all-mineral coating, usually of the manganese-iron-silicate type, and can be used at very high current values to produce X-ray-sound welds suitable for pressure work.

A relatively recent addition to the group of coated electrodes is a lime rod, classified as E-6015 for d-c reverse-polarity welding or E-6016 for a-c welding. This type of electrode offers the highest degree of protection against cracking and produces the cleanest weld metal that can be attained by this process of welding. However, it is not used extensively in boiler construction because its general welding characteristics are inferior to the E-6020 and E-6030 type, particularly in the production of porosity-free weld metal in carbon steel.

Perhaps the most common electrode used in boiler work is a carbon-molybdenum type (E-7015) with minimum tensile strength of 70,000 psi in the stress-relieved condition.

An increasing proportion of boiler components is made of steels containing various amounts of alloying elements, from small amounts of chromium and molybdenum up to the highly alloyed stainless steels. For the welding of these alloy steels, electrodes are produced that will deposit weld metal essentially similar in composition to the parent metal. Electrodes corresponding in analyses to the entire range of Croloy steels (see addenda to Table D in the Appendix, page 22-A4) are available. Coatings on these alloy-steel electrodes are of the lime type. The use of lime-coated electrodes in chromium-alloy-steel welding will produce the cleanest and soundest deposits obtainable.

Submerged-Arc Method. Another method of fusion welding, by using a submerged arc, is of recent origin but is gaining rapidly in application in the production of pressure vessels. Both open-arc (with coated rods) and submerged-arc methods are essentially similar in that the quality, soundness, and mechanical properties of the weld metal depend to a large extent upon a flux. The primary difference is that, in using the open-arc method, the flux comes from the coating of the electrode and is deposited simultaneously

°"Welding Handbook," third edition, 1950, *American Welding Society.*

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with the weld metal of the rod, while in using the submerged-arc method, a bare rod fed from a coil through a welding head to the work strikes the arc, and the flux, supplied separately as a granular material, completely submerges the arc area.

The electric-current value permissible with a certain size electrode is primarily a function of the weight of flux used per pound of rod. The amount of flux that can be economically applied to a rod by an extrusion process is limited. A much greater amount of flux can be used with the submerged-arc method. The greatest advantage of submerged-arc welding over open-arc welding is that greater current values are possible. For instance, with ¼-in.-diameter wire, current values as high as 1500 amp may be used in the submerged-arc method, compared with a maximum of some 400 amp for a coated rod of the same size. Current values in excess of 2000 amp may be used successfully in submerged-arc welding with larger wire, $\frac{5}{16}$-in. and ⅜-in. diameter. The amount of flux molten in the submerged-arc method is roughly 1 to 1½ times the weight of the molten metal, while in the open-arc method the ratio of flux to core wire seldom exceeds 25%.

In production, plates up to 2 in. thick may be welded in two passes by the submerged-arc-welding method. These welds are clean, sound, free from defects, and have excellent physical properties. Moreover, the submerged arc may be used successfully for multipass welding of plate up to and even in excess of 6 in. thick.

Gas Welding. Oxyacetylene gas welding, another fusion-welding process, finds relatively little application in boiler construction in the United States. Compared with metal-arc welding, particularly for thicker materials, this process is expensive and slow. However, there are some applications, such as butt joints in light-gage tubing, where gas welding is used to advantage. In general, the use of gas welding in boiler construction is limited to the plain carbon steels.

Use of the atomic-hydrogen process of fusion welding is limited to light-gage materials, preferably not over ¼ inch. This fact and the inherent slowness of the process have restricted its use.

The inert-gas-arc process of fusion welding, in which helium, argon, or a combination of helium and argon supplies the shielding atmosphere, has somewhat the same limitations as the hydrogen arc and is rarely applied in boiler construction. However, it appears likely that a serious attempt will be made to develop this process, which may open up a new field in welding for boiler construction.

Brazing. The various methods of joining by the brazing process are used rarely in boiler work, since the strength of the joint, particularly at elevated temperatures, is usually too low.

Welding of a Power Boiler

In building a power boiler, thousands of pounds of electrodes of many different types and sizes may be used. The choice of electrode type and size depends primarily upon the composition, thickness, and joint design of the material to be welded. In all cases, welding stresses in a joint must be kept within a safe limit for good practice. Welds in structural members, brackets, supports, casing, and other components, where the finished part is subject to relatively low service stresses, present few complications.

Preheating. Perhaps the most effective way to counteract high stresses in deposited weld metal is to preheat the parts to be welded. The proper preheating temperature depends mainly upon the chemical composition and thickness of the material to be welded and to a lesser extent upon the welding method and type of electrode selected.

A joint in 1-in.-thick high-alloy (austenitic) steel, such as 25% Cr and 20% Ni, would not require preheating, while a joint in (ferritic) intermediate Croloy tubing of 2½-in. OD and ¼-in. wall thickness would require a minimum preheat of some 300 F. A 4-in. OD by ⅜-in. wall of SA-106 carbon-steel pipe would be welded without preheat, while a drum plate of SA-212 carbon steel 6 in. thick would be preheated to about 400 F prior to and during welding. The most severe preheat requirements are for high-carbon (1%) alloy castings containing some chromium, molybdenum, manganese, etc., where preheats as high as 1000 F are used.

Submerged-arc welding, compared with multiple-pass open-arc welding, may be done at some 25% lower preheat temperatures, since the cooling stresses are less severe because the heat input per inch of weld is substantially greater and consequently the rate of cooling of the molten metal is slower. The choice of proper preheat temperature also depends to a marked degree upon the wide variation in the behavior of weld metals deposited with different types of electrodes. Weld metal deposited with lime-coated mild-steel electrodes is practically unaffected by high cooling rates of the weld, while hot rods (E-6020—E-6030) are very sensitive to variations in cooling rates.

Method of Welding. Automatic welding, using either the submerged-arc or open-arc method, would be selected in preference to manual operation wherever the work can be done under an automatic welding head. However, manual welding must be used for certain types of joints not readily weldable automatically, and, of course, if the work can be positioned to permit its use, downhand welding with the downhand hot rod (E-6020—E-6030) would be preferable. All other welding has to be done in position, using all-position electrodes of the cellulose type.

The above choice of welding method and operation applies to all carbon steels and carbon-molybdenum steels. While submerged-arc welding has been used in some cases, the majority of welds in joining Croloy steels is made manually with the open-arc method. Stainless steel of the high-Cr-Ni type is welded almost exclusively by the open-arc method, using stainless coated electrodes of the proper composition. However, light-gage tubes formed from strip are welded in large quantities by inert-gas-atmosphere arc welding.

Postheat-Treatment. The postheat-treatment of weldments is again a function of the composition and thickness of the materials joined and, to a considerable extent, of the complexity of the structure. To avoid cracking, where heavy plates are involved, it is customary to stress-relieve the partially welded joint. Structural welding, on the other hand, is rarely heat-treated. Welded joints subject to the internal pressure of the boiler are postheat-treated, except plain carbon-steel or carbon-molybdenum-steel (C equal to or less than 0.20%) tubing or piping of ⅜-in. or less wall thickness and 4-in. or smaller diameter. All Croloy steels of the low-chromium air-hardening grades and most of the high-alloy steels are heat-treated after welding. The postheat-treatment temperatures may vary from 1150 to 1250 F for carbon and carbon-molybdenum steels to as high as 1950 F for some of the high-Cr-Ni austenitic steels.

See Table B in the Appendix, page 22-A2 ff, for recommended welding rod, preheating, and postweld heat-treatment for gas and arc welding of carbon and carbon-molybdenum steel and various ferritic and austenitic Croloy combinations.

Weld Quality. To assure consistently high-quality welds, all welding operators in the plant engaged in boiler work must first prove their ability by proper qualification tests in welding plates or pipes to meet the requirements of the *Welding Qualifications Section of the ASME Boiler and Pressure Vessel Code.* As a routine check of the quality of the work, samples of welds made during production should be taken from the job and tested in the laboratory.

To check mechanical properties of the deposited weld metal, all major welding on drums and other critical components is run over to attached test plates (see Fig. 5, Chapter 24), which are cropped and subjected to thorough laboratory tests.

Nondestructive Testing. By Code requirement, after welding and heat-treatment, all the main seams and some of the drum attachments must be examined by X ray or radiographed. By the use of powerful X-ray equipment, even very minute flaws or defects in the weld deposit can be detected and can then be cut out and rewelded as required. One-million-volt and 2-million-volt stations are included in the B&W X-ray equipment for shop examination. In the field or where inaccessibility precludes the use of X-ray equipment, examination is made by gamma rays from isotopes or radium. For the detection of flaws and defects open to the surface but too fine to be seen, magnetic-particle inspection (Magnaflux) and fluorescent-penetrant° inspection (Zyglo) are also used extensively to assure sound welds. The Magnaflux method is used only for magnetic (ferritic) steels, and the Zyglo method is used for nonmagnetic steels.

Mechanical Properties and Testing

Specifications standardizing all the conditions relating to the test piece and the method of testing have been formulated by the American Society for Testing Materials and other authorities. Holding to these specifications, it is possible to insure duplication of certain minimum mechanical properties that determine the acceptability of a steel for a particular use.

Tensile Test. The tensile test, a gradually applied unidirectional pull, determines the maximum nominal load that a material can withstand before actually breaking.

The relationship between stress (load per unit area) and the corresponding strain (deformation) in the test piece is illustrated in the stress-strain diagrams (a) and (b) of Fig. 26. The metal begins to stretch as soon as the load is applied, and during the period of the metal's elastic behavior the strain is directly proportional to the stress (Hooke's Law, elongation ∝ load ÷ area). Consequently, if the stress is released at any point along O-A (in either diagram, Fig. 26), the test coupon will return to its initial dimensions. However, if the stress is increased beyond the value A, the metal will no longer continue to be perfectly elastic (there will be a permanent elongation), and the linear relationship (proportionality) between stress and strain ceases. The value A is known as the proportional limit of the material, and in this elementary discussion it may be considered practically the same as the elastic limit, which may be defined as the maximum unit stress that can be developed just before permanent elongation occurs. The exact location of these limits involves laborious techniques and requires extremely sensitive measuring devices.

° In this method a small quantity of fluid, carrying a fluorescent dye, is applied to the surface of the weld and adjacent steel. The liquid is drawn by capillary attraction into any cracks or discontinuities appearing in the surface, and it is made visible by exposure to "black" (near ultraviolet) light.

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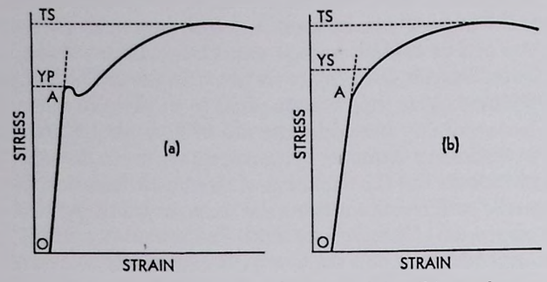


Fig. 26. *Stress-strain diagram; (a) mild steel (b) high-tensile steel and nonferrous metals*

The unit stress at which a ductile material suddenly continues to elongate without further increase in load is known as its yield point (YP). Since the true yield point for most metals is indefinite and, in common with the proportional and elastic limits, is difficult to determine with accuracy, these terms are being supplanted by "yield strength." Yield strength (YS), which can be readily determined, is the unit stress at which the permanent elongation of the metal reaches an arbitrarily specified value, usually 0.1, 0.2, or 0.5% of the gage length of the test piece.

If the loading is continued after the point of yield strength has been reached, a test piece of homogeneous composition and uniform cross section will be elongated uniformly over its length, with a corresponding reduction in area. Eventually a constriction or "necking down" will occur (in ductile material), usually in the region of the middle third of the test piece (Fig. 27). In other materials the necking down may not be localized but may occur as a more or less uniform reduction in area along the full gage length to instant of rupture. In all cases with ductile materials, however, an appreciable increase in elongation occurs over the portion of reduced area. The elongation before the specimen finally breaks will increase with the ductility of the steel. The maximum

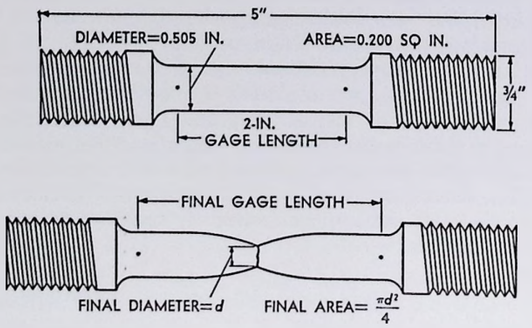


Fig. 27. *Tensile-test specimen before and after testing (ASTM standard)*

applied load in lb required to pull the specimen apart divided by the area in sq in. of the original cross section of the test piece is a value known as the ultimate or tensile strength (TS).

Ductility of the metal is determined by measuring the increase in length (total elongation) and the final area at plane of rupture after the specimen has broken. Ductility values are expressed as the per cent change of the original length and/or area. In general the strength of a metal decreases as the ductility increases. Conversely, very strong metals are likely to be hard and brittle, that is, lacking in ductility. To suit the requirements of an intended application, steels are mechanically worked or heat-treated to develop varying combinations of these properties, as noted earlier.

Hardness Test. Hardness may be defined as resistance to indentation under static or dynamic loads and also as resistance to scratch, abrasion, cutting, or drilling. To the metallurgist, the property of hardness is of considerable significance in its indication of other important characteristics of the metal: heat-

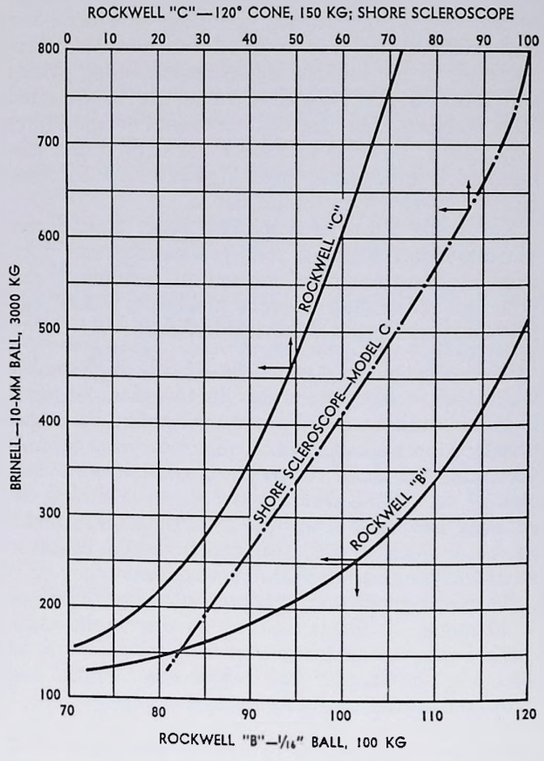


Fig. 28. *Hardness-conversion curves. Approximate relationship between Brinell, Rockwell C, Rockwell B, and Shore Scleroscope testing systems*

treatment, machinability, wear resistance, and fabrication.

Hardness is usually determined by using the Rockwell, Brinell, or Vickers (diamond pyramid) hardness testers, all of which measure resistance to indentation under static loads. A hardened steel ball, a spheroconical diamond, or a diamond pyramid is impressed into the material to be tested for hardness. The pressure is applied under a definite load and for a specified time. For the Rockwell machine, the hardness is read directly on a dial gage. Using the Brinell and Vickers machines, the diameter and diagonals, respectively, of the impressions, measured by means of a microscope, are converted to hardness numbers by reference to tables.

In special cases where it is desired to determine the hardness of microscopic constituents of the material, such as ferrite, pearlite, sigma, delta, etc., a microhardness tester (the Tukon machine) is used. In this test operation, which is completely automatic, the indentation is made by applying either a Knoop or diamond-pyramid indenter under a definite load and for a specified time. The diagonals of either type indentation are measured by means of a microscope, and the readings are converted to hardness numbers (Knoop or Vickers, depending upon the type of indenter used) by reference to tables.

The approximate relationship between the several hardness scales is given in Fig. 28.

Impact Test. Notched-bar impact tests are also important in the field of mechanical testing. These tests determine how well a material is actually able to equalize concentrated stresses, and they are sometimes taken as a measure of "toughness," a term difficult to define otherwise.

The two types of impact testing machines widely used are the Charpy and the Izod. In the Charpy test the specimen is a simple horizontal beam supported at both ends, while in the Izod test the specimen is treated as a cantilever beam. Standard impact bars for both tests are illustrated diagrammatically by Fig. 29. In both tests the bars are struck by a single blow of a swinging pendulum. The energy absorbed by the specimen is related to the height of rebound and can be read directly in ft-lb on a calibrated dial or scale.

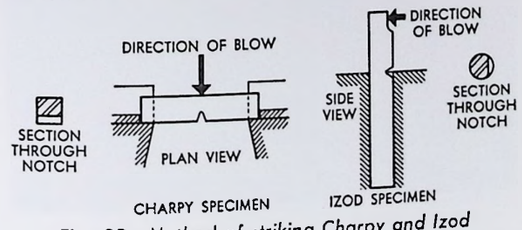


Fig. 29. *Method of striking Charpy and Izod notched-bar impact specimens*

Other Tests. Deformation tests are also made to determine the ductility of steel. These include cupping, bending, flattening, and flare tests. They furnish visual evidence of the good ductility of tube material.

Typical mechanical properties, at room temperature, of various B&W Croloy steels are listed in Table C in the Appendix, page 22-A4.

High-Temperature Properties and Testing

Tensile- or yield-strength data determined at normal temperatures cannot be used as a guide to the mechanical properties of metals at elevated temperatures. Even though such tests are made at the higher temperatures, the data would be inadequate for designing equipment for long-time service at elevated temperatures. This is so because continued application of load produces a very slow continuous deformation, which may be significant and measurable over a period of time and may eventually lead to fracture, depending upon the stress and temperatures involved. This slow deformation of continuously stressed metal, with time a very important factor, is called "creep."

The importance of this phenomenon is now widely recognized, and extensive studies of creep characteristics (including the stress-rupture time) at elevated temperatures have been made and are constantly under way, both by manufacturers and users of metal. It is the consensus among engineers that the creep and stress-rupture test data now available serve as the basis for a much more rational approach to the design of high-temperature equipment than was possible using data from short-time high-temperature tensile tests. Thus, for temperatures exceeding about 750 F, the S values (maximum allowable working stresses for ferrous materials in power boilers) set by the ASME are based on long-time creep and stress-rupture (creep to rupture) tests (see *Creep Test* below).

Tensile Test

Although largely superseded by creep and stress-rupture tests for design of high-temperature equipment, the short-time tensile test is useful in indicating the strength properties of metals up to approximately 750 F. This test also provides information on ductility characteristics helpful in fabrication.

The procedure used in conducting high-temperature tensile tests should conform to the requirements (Spec E 21) of the ASTM. In this test, a suitable electric-resistance furnace maintains the standard tensile specimen (see Fig. 27) at the selected constant temperature during application of the load. A constant pulling motion not exceeding 0.05 in. per minute is used, because rapid rates of loading may produce

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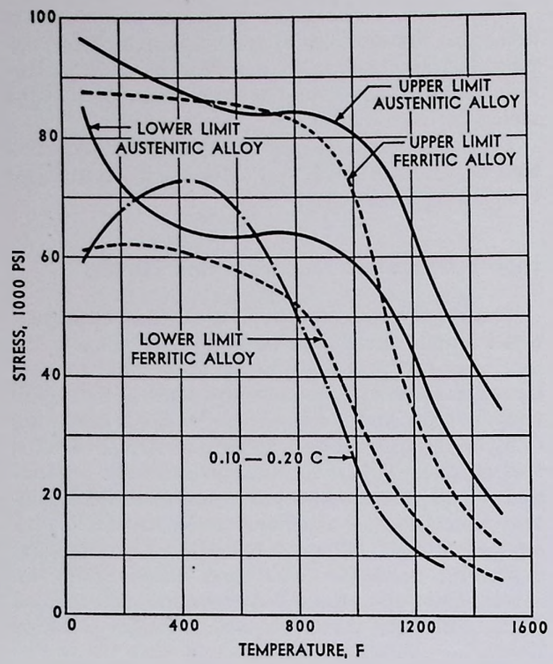


Fig. 30. *Tensile strength of various steels at temperatures to 1500 F*

inordinately high ultimate strength values. The tests are usually made on normalized or fully annealed material, corresponding to its actual condition as it enters service.

The ultimate strength of carbon steel and a variety of Croloys, determined by short-time tensile tests over a temperature range from 100 F to 1300-1500 F, is summarized in Fig. 30. In general, the results of these tests indicate that strength decreases with increase in temperature, although there is a region for the Croloys (austenitic and ferritic) between 400 and 900 F where strength is fairly constant. Some exceptions to the general rule are the increase in strength over that at room temperature of plain-carbon steel and certain low-alloy steels (with corresponding decrease in ductility) from 100 F to about 600-750 F, depending upon the material. The highest strength usually occurs in the zone of 400-600 F. What is termed the "blue-brittle" range, because of the bluish oxide film that forms on steel at these temperatures, occurs between 400 and 700 F. As the temperature is increased beyond 600-750 F, the strength of the carbon and some of the alloy steels falls off from that at room temperature with a corresponding increase in ductility.

Creep Test

It has long been known that certain nonmetallic materials, such as glass, undergo slow and continuous deformation with time when subjected to stress. The concept of creep in metallic materials, however, is comparatively new and began to attract serious attention in the early 1920's. Results of several investigations in the early twenties indicated conclusively that rupture of a metallic material could occur when it is subjected to a stress at elevated temperatures for a sufficiently long time, even though the load applied was considerably lower than that necessary to cause fracture in the short-time tensile test at the same temperature. Although there are still many gaps in the present theory of metallic creep, the creep characteristics established since 1930 for many metals and alloys have proved of great practical value in indicating the fitness of these materials for continuous service under stress at elevated temperatures. This information has also been useful in the development of still better alloys for such service.

In the United States the earliest attempt to investigate creep in a comprehensive manner was first made by B&W, beginning in 1926, when a fellowship was established at the Massachusetts Institute of Technology under the direction of Prof. F. H. Norton. Many steels now used successfully in power-generating units and in the petroleum-refining and chemical industries were tested and proved in the course of this investigation, using the best available equipment. Extensive studies of the creep characteristics of metals are now continuously under way in the Research and Development Laboratory of B&W at Alliance, O. This laboratory is completely equipped with modern creep- and rupture-testing facilities (see Fig. 31).

The general method of creep testing is specified by the ASTM (Spec E 22). A test piece is subjected to a fixed static tensile load while it is maintained at constant temperature. The deformation of the test sample is accurately measured at regular intervals during the test (see Fig. 32). The duration of the test may range from 1000 to 10,000 hr, or longer. A diagrammatic plot of the observed length of the specimen against elapsed time is generally of the typical form illustrated by Fig. 33.



Fig. 31. *Stand of B&W creep-test machines in laboratory at Alliance*

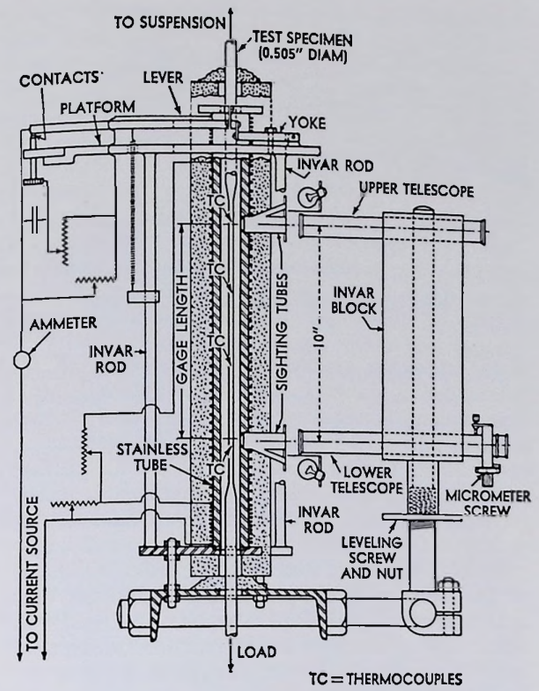


Fig. 32. *Creep-test station showing specimen, heating element, and telescopes for observing elongation*

The curve representing creep is divided into three stages. It begins after the initial extension (O-A), which is simply the measure of deformation of the specimen caused by the loading. The magnitude of this initial extension will, of course, depend upon the test conditions, varying with load and temperature, and will usually increase with temperature and load. The first stage of creep (A-B), referred to as "primary creep," is characterized by a decreasing rate of deformation during the period. The second stage of creep (B-C), referred to as "secondary creep," is usually characterized by extremely small variations in rate of deformation: hence this period is essentially one of constant rate of creep. The third stage of creep (C-D), referred to as "tertiary creep," is characterized by an accelerating rate of deformation leading to fracture. The important thing to the engineer is that these stages of creep always occur to greater or lesser degree. Just why they occur or what the actual mechanism is to produce them is not yet definitely understood.

In this country, to simplify the practical application of creep data it is the custom to establish two values of stress in lb per sq in. (for a material at a temperature) that will produce two corresponding rates of creep (elongation): one of 0.1% per 1000 hr and the other of 0.01% per 1000 hr, or, as usually expressed, 1.0% per 10,000 hr and 100,000 hr, respectively, as illustrated in the creep-test data for a number of metals of different composition given in Table D in the Appendix, page 22-A4.

For any specified temperature, several creep tests must be run under different loads (see typical curves, Fig. 34). The creep rate during the period of secondary creep is determined from these curves and is

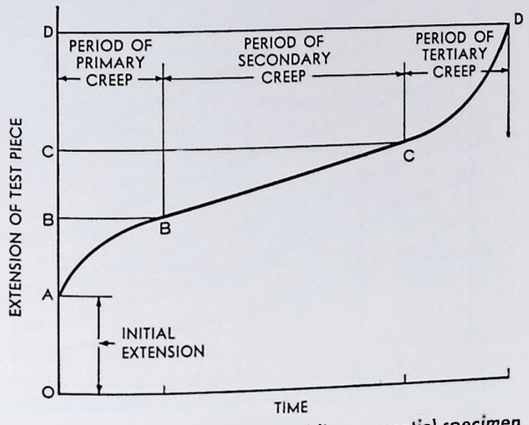


Fig. 33. *Typical creep-test (diagrammatic) specimen held at constant load and temperature*

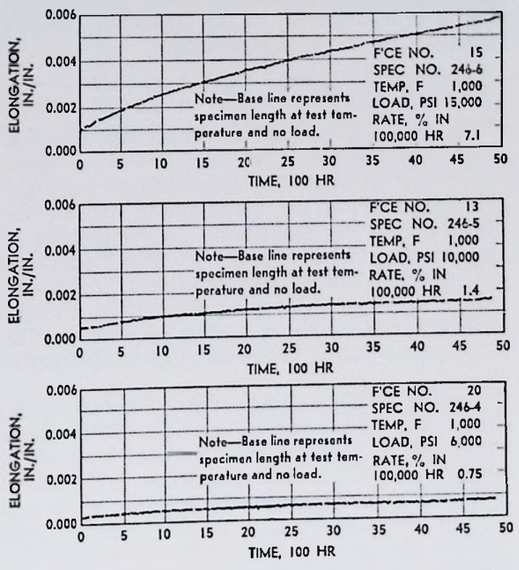


Fig. 34. *Typical time-elongation curves for Croloy 2¼ tested for three loads at 1000 F*

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plotted against the stress. When these data are plotted logarithmically on the same chart, the points for each specimen usually lie in a straight line. Thus, for tests of 2000 to 3000-hr duration it is possible by extrapolation to obtain the stress for a creep rate of 0.1% per 1000 hr and 0.01% per 1000 hr (see Fig. 35).

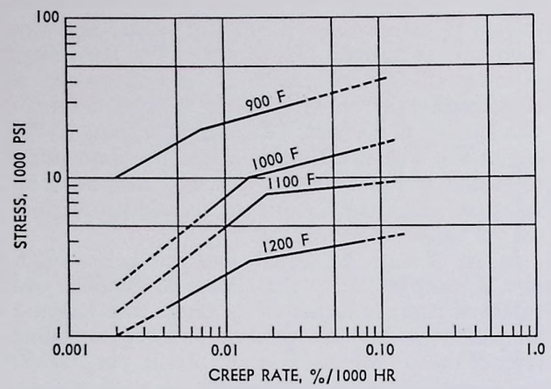


Fig. 35. *Design-stress values for 2¼ Croloy at different temperatures for creep rates in the range to 0.1% per 1000 hours*

The form of the creep curve depends upon the chemical composition and microstructure of the metal as well as upon the applied load and test temperature. The creep behavior of steel is influenced also by various metallurgical factors, as noted below. Tests of long duration provide the most reliable data, both as to the accuracy of the observation and the attainment of structural equilibrium.

Stress-Rupture Test

The stress-rupture test is essentially an overloaded creep test continued to actual rupture of the specimen. The more descriptive names "creep-to-rupture" and "creep-rupture" are now frequently used for this test. The time for rupture at any temperature is some function of the applied load. Since the modern concept of this test was first formulated, in 1937 and 1938, it has rapidly gained favor as a necessary supplement to the regular creep test in evaluating the load-carrying ability of metals at elevated temperatures.

Generally, the equipment employed in making the rupture test is the same as that used for the creep test. However, while the elongation of a specimen during a creep test usually varies from 5 to 250 microinches per day, the elongation during a rupture test may be 100 times greater, or even more. Consequently, different instruments for measuring the elongation of the specimen are required.

A logarithmic plot of stress versus time for fracture

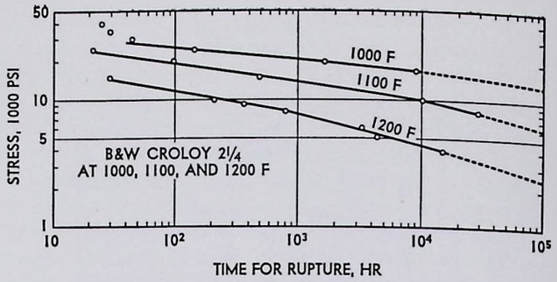


Fig. 36. *Typical stress-rupture curves*

of the specimen generally takes the form of the typical curves shown in Fig. 36. In general, rapid rates of elongation indicate a transgranular (ductile) fracture, and slow rates of elongation indicate an intergranular (brittle) fracture. As a rule, surface oxidation is present when the fracture is transgranular, while visible intercrystalline oxidation may or may not be present when the fracture is intergranular. Because of the discontinuities produced by the presence of intercrystalline oxides, the time to rupture at a given load may be appreciably reduced. The presence of intercrystalline oxidation is indicated in Fig. 36 by the change in slope of the curve for 1200 F.

The duration of the rupture test of a single specimen may be from a few hours to 10,000 hr, or longer.

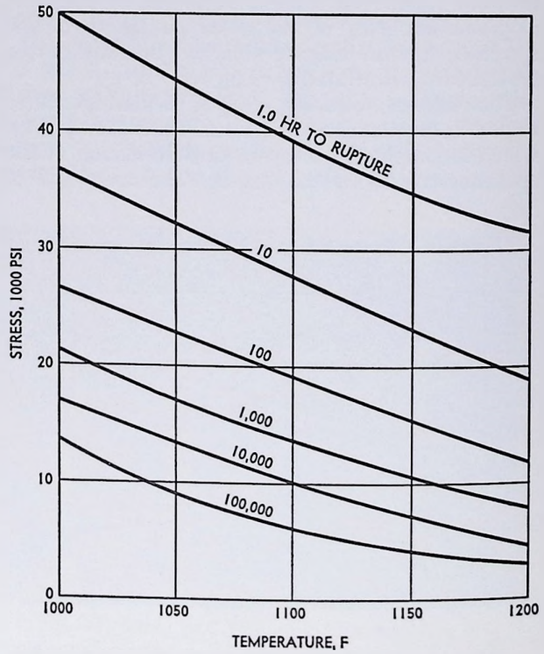


Fig. 37. *Stress at which fracture occurs, Croloy 2¼*

A complete creep-rupture test of a given material actually consists of a series of tests at constant temperature with each specimen loaded at a different level. The time for rupture of each specimen is noted. It is customary in the report of test data to give the stress for fracture in 100, 1000, 10,000, and 100,000 hr (see Fig. 37). Since very few tests are carried on for more than 10,000 hr, the values for fracture times longer than this are determined by extrapolation. At the Alliance Research and Development Laboratory, it is the practice to load the specimens so that one or more at each temperature will run for at least 5000 hr, and usually for 10,000 hr or more. The longer testing times give greater assurance that the material has reached structural equilibrium and also materially improve the accuracy of extrapolating the rupture strength for still longer times.

Some Factors Affecting Creep

Creep of metals is a sensitive property and may be profoundly affected by a variety of metallurgical factors. These factors are difficult to control commercially, and it is almost impossible to isolate any one factor in order to study its specific effects on creep. It is not surprising, therefore, that little progress has been made in establishing general rules regarding the creep behavior of metals.

Composition. It is generally recognized that the most direct way to improve the creep properties of a metal is by adding alloying elements. Carbide-forming elements, such as molybdenum, tungsten, and, to a lesser degree, chromium and vanadium, effectively enhance the creep resistance of steels. Increased carbon content is beneficial up to a temperature range of approximately 900-1000 F; above this range, however, variations in carbon content in most wrought steels appear to have little influence. Manganese and small additions of silicon have some effect in improving the resistance to creep at the lower temperatures. Aluminum, used as a deoxidizer in plain-carbon steel or as an intentional addition in alloy steels, greatly reduces creep strength at high temperatures. Nickel additions are beneficial when used in sufficient amounts to produce an austenitic structure, which is more creep resistant than ferrite. At the higher temperatures, austenitic stainless steels (18Cr-8Ni type) and certain special alloys have creep properties much superior to low- or high-alloy ferritic steels.

Grain Size. At the higher temperatures, experimental evidence strongly indicates that coarse-grained materials usually exhibit greater resistance to creep than fine-grained materials. Conversely, at the lower temperatures, fine-grain structures appear to be slightly superior. However, there are many exceptions to this rule, and no satisfactory explanation has been advanced to account for the specific effect of grain structure on the creep behavior of metals.

Heat-Treatment. The creep characteristics of metals are affected by heat-treatment, which controls grain size and the stability of structure. When a metal is exposed to elevated temperatures for a long period of time, there is a tendency, under the stress and heat to which it is subjected, for the initial structure to attain a more stable state. For instance, lamellar pearlite may revert during service to the more stable spheroidized structure, which, in turn, brings about rapid deterioration in the creep strength of steel. Upon prolonged heating in the 900 to 1200 F range, the carbide phase in plain-carbon and carbon-molybdenum steels may ultimately change to graphite, which, when highly localized is believed to account for certain failures noted in service. Obviously, therefore, the selection of the best type of heat-treatment for a particular high-temperature application should be based largely on the degree of stability that it would impart to the structure initially and also throughout the service life of the steel at elevated temperatures.

In general, experience has shown that, for service above approximately 950 F, annealing (or normalizing) is the most suitable treatment. Age-hardenable alloys, intended for service in which creep resistance is required, are usually subjected to a dual heat-treatment. In the initial heat-treatment, the precipitating phase is brought into solution; the final heat-treatment is to insure stability of the precipitated phase at the service temperature.

Surface Stability. Creep strength may be seriously impaired by surface reactions at elevated temperatures. By reduction in effective cross section from progressive oxidation or corrosion of the metal, the unit load is increased, and the rate of creep is accelerated. Under certain conditions, highly concentrated stresses may greatly accelerate and localize the mechanism of oxidation.

Surface stability may be enhanced by alloying additions of silicon, aluminum, chromium, or nickel. Surface oxidation may also be decreased by applying protective coatings to the metal.

Method of Manufacture. Minor variations in method of manufacture have markedly modified the creep behavior of steel. In general, the investigations made would seem to indicate that steel melted in the electric furnace is somewhat superior to open-hearth steel. This is probably because of less rigid controls and, therefore, wider variations in deoxidation practice usually encountered in the open-hearth process. It is also recognized that killed steel is more creep re-

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sistant than rimmed or similar nondeoxidized steel. However, deoxidation per se does not insure superior long-time high-temperature properties. For example, at 850 F, steels deoxidized only with silicon and having a coarse grain structure are much more resistant to creep than steels deoxidized with large amounts of aluminum and of fine grain size. The detrimental effect of aluminum results from its potency in causing grain refinement and also its acceleration of spheroidization and graphitization of the carbide phase. Both of these characteristics tend to lower creep resistance at elevated temperatures, while a coarse grain size and stability of structure tend to increase resistance to creep.

Steels Used in Boiler Construction

A considerable variety of both carbon and alloy steels is used in boiler construction.

Plate Steels

Since the temperatures do not exceed the saturated steam temperature, the significant mechanical properties of steel for steam drums are the tensile and yield strengths as determined at room temperature rather than the creep strength. Design is based on an allowable stress of ¼ of the minimum ultimate strength to 650 F. For special requirements and the higher temperatures, there are certain advantages in going to higher-tensile-strength carbon-steel plate or to alloy-steel plate. Above 650 F, reduction in working stresses is mandatory for carbon steels and some alloy grades of plate, while certain other alloy plate materials may be used for temperatures to 750 F without reduction in working stress. Use of flange-quality plate is not permitted at temperatures exceeding 850 F nor in thicknesses exceeding 2 inches. When made from plate, manway covers, handhole fittings, lugs, braces, and other parts subject to pressure must be of firebox or flange quality. Chemical composition and mechanical strength of several plate grades of carbon and low-alloy steels, approved by the ASME for construction of boilers and other pressure vessels, are given in Table 2. For stationary boilers, B&W mainly uses thoroughly deoxidized (killed) carbon steel to SA-212 specification. If required by certain service conditions for saving weight in marine work or because of design requirements, other grades, including alloy-steel plate, may be used. Typical microstructures of several plate steels are shown by Fig. 38.

Tube and Pipe Steels

The principal heat-absorption surfaces of the boiler are the steam-generating tubes, including water-wall furnace tubing and the economizer. Ordinarily, these sections of the boiler are made of hot-rolled low-carbon steel tubing, although small-diameter tubes or special sizes may be cold drawn and annealed. Seamless and electric-resistance-welded tubes are used interchangeably in generating tubes. However, seamless tubes are supplied for high-pressure units requiring a wall thickness difficult to form as skelp. Some of the specifications for ordinary boiler tubes, such as SA-83 and SA-178, do not stipulate whether rimmed or killed steel is to be used. As explained above under tube-manufacturing processes, the current general practice of B&W is to use killed steel for seamless tubes and rimmed steel for electric-resistance-welded tubes.

Medium-carbon steel (0.35% C max) finds considerable application for boiler and furnace-wall tubes,

TABLE 2

Composition and Properties of Plate Steels

Chemical Composition

ASME Spec	C	Mn	P	S	Si	Cr	Mo
SA-212	0.28-0.35	0.90	0.035-0.04	0.04-0.05	0.15-0.30	—	—
SA-285	0.15-0.30	0.80	0.035-0.05	0.04-0.05	—	—	—
SA-387°	0.21	0.80	0.035-0.04	0.04-0.05	0.15-0.30	0.50-0.80	0.45-0.65

NOTE: Percentages are maximum unless range is given.

Mechanical Properties

	Tensile Strength†	Yield Strength†	Elong in 2 in.‡	Elong in 8 in.‡
SA-212	65,000-85,000	35,000-38,000	24-21	21-18
SA-285	45,000-65,000	24,000-30,000	32-28	29-24
SA-387°	65,000-82,000	40,000	24-23	20

°Grade A. †Pounds per square inch. ‡In per cent.

NOTE: The chemical composition and mechanical properties given in this table represent a summary of the requirements for the various grades under the specifications noted.

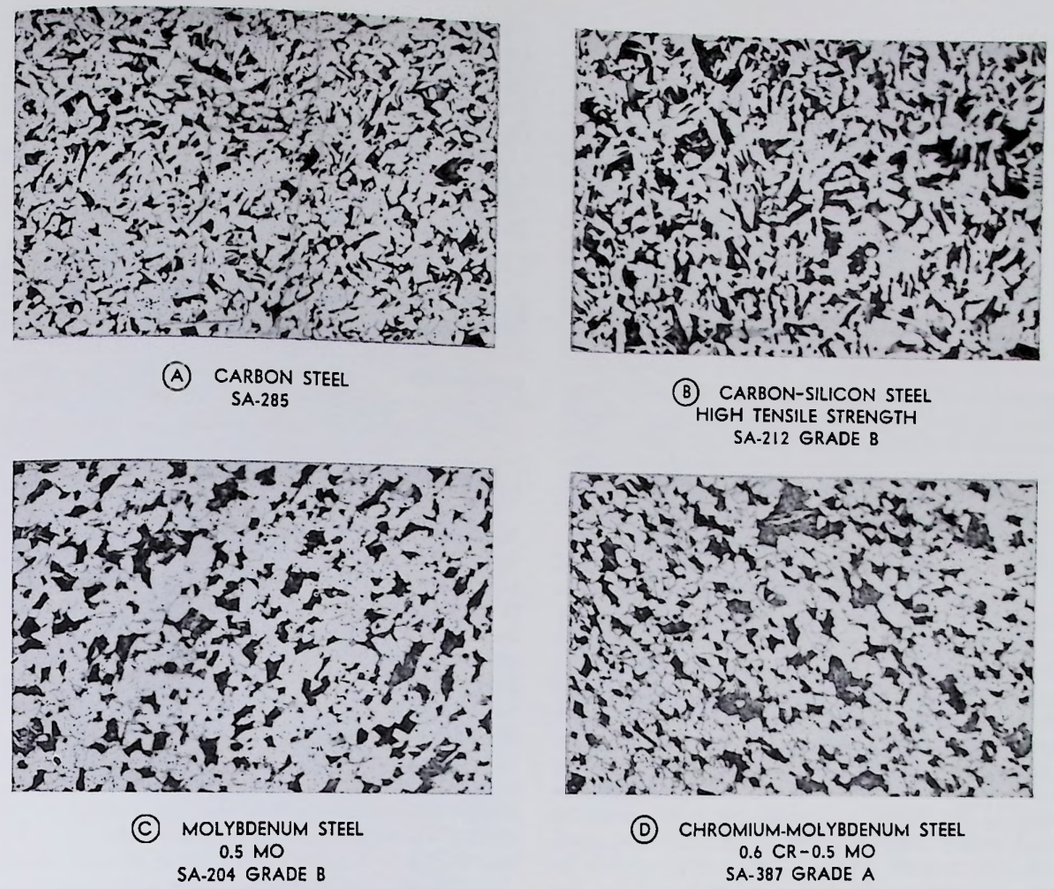


Fig. 38. *Typical structures of plate steels (100x). White matrix, ferrite; dark areas, pearlite; with carbide phase appearing in mottled areas (Widmanstätten pattern) in B, C, & D. Steel shown in A used principally in nonpressure parts. Steels shown in B, C, & D used in pressure parts*

since this grade will permit higher stress levels in the zone of saturation temperatures from 540 to 685 F encountered in service. The mechanical strength properties are superior to low-carbon steel, and, therefore, lighter-wall tubes may be used for equivalent pressures.

The highest metal temperature of pressure parts in the boiler is in the superheater. Consequently, it must be made of material having superior high-temperature properties and resistance to oxidation. Carbon steel is a suitable and economical material to about 850-950 F metal temperature depending upon pressure. Above this range, alloy steels are required because of the low oxidation resistance and the low allowable stresses of carbon steel. Sometimes two or more alloys may be used in the construction of the superheater. The lower alloy, such as carbon-molybdenum steel, is used toward the inlet section, while the better materials, the so-called low and intermediate chromium-alloy steels, are used toward the outlet, where the steam and metal temperatures are increasing. Stainless steel tubes are required in the hottest sections of the superheater to meet the operating conditions of modern high-pressure (1800-4500 psi, or more) high-temperature (1000-1050 F, and even 1150 F) central station boilers.

The B&W Croloy Steels. For many years B&W has been active in the development of tubing alloys for high-temperature applications. A considerable number of the presently accepted ASME standard steels for superheater practice were developed originally by B&W. These steels with the designation "Croloy," meaning chromium-containing alloy, are available in a graded series of intermediate Cr-Mo alloys and also in stainless Cr or Cr-Ni grades.

The intermediate Croloy steels (up to and including Croloy 9M) are complex chromium steels with

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varying contents of molybdenum. These alloy steels are magnetic and have normal thermal transformations. They may be considered "ferritic" steels in that their microstructure, in the annealed state, is composed mainly of ferrite (alpha iron) and carbides. In general, the behavior of these alloys is somewhat similar to that of high-carbon steel, although they are more ductile for a given strength because of the absence of the equivalent amount of carbide. Heating to or above their transformation temperature, required for forming and after welding, makes these alloys so susceptible to air-hardening that an annealing or stress-relieving (tempering) treatment is necessary to restore toughness and ductility following these operations. Addition of carbide-forming (stabilizing) elements, such as titanium and niobium, tends to restrict air-hardening of the Croloys on heat-treating at temperatures up to about 1700 F. Gradual hardening will take place above these temperatures.

Stainless Croloys are usually classified as of austenitic or ferritic type according to chemical composition and metallurgical characteristics.

Austenitic stainless Croloys are low-carbon chromium-nickel steels containing above 16% Cr with sufficient nickel to provide an austenitic structure at all temperatures. Moderate amounts of other elements, such as molybdenum, niobium, niobium plus tantalum, titanium, and silicon, may be added for special purposes. Austenitic Croloys are normally nonmagnetic but may become slightly magnetic when cold worked. These steels do not undergo transformations in the usual sense and therefore cannot be hardened by heat-treatment. They can be hardened and strengthened materially, however, by cold-working. Included in the austenitic group of stainless Croloys are the widely known 18-8 type steels (18% Cr and 8% Ni).

Ferritic stainless Croloys, usually referred to as "high-chromium irons" or "straight-chromium steels," are iron-chromium alloys containing chromium in excess of 11.5%. Compared with austenitic (chromium-nickel) Croloys, these steels are ferromagnetic, possess less ductility, are notch sensitive, and are subject to excessive grain growth on welding or other treatment at high temperatures. Despite these features which tend to curtail their utility to some degree, the high-chromuim irons occupy an important place in industry in both corrosion- and heat-resisting applications. The 12%, 18%, and 27% chromium steels are representative of the ferritic group of stainless Croloys.

Applications of Tube and Pipe Steels

The Croloy steels in widest use for superheater tubing in boiler service have been: (a) 2% Cr and 0.5% Mo, (b) 2¼% Cr and 1% Mo, (c) 3% Cr and 0.9% Mo, and (d) 9% Cr and 1% Mo. The most widely used stainless steel is the austenitic 18-8 stabilized with titanium, niobium, or niobium plus tantalum. The chemical composition of these and other grades of carbon and alloy steels for boiler and superheater tubing and steam piping is included in the addenda to Table D in the Appendix, page 22-A5.

Creep strength in the intermediate-alloy steels is not proportional to chromium content and is somewhat better in those of lowest chromium content. Therefore it becomes necessary, depending upon the anticipated metal temperatures, in choosing the optimum composition to have in mind both strength properties and resistance to oxidation.

The selection of an appropriate and economical material for construction of the superheater is dependent upon a number of factors. Because of the temperature drop through the steam film immediately adjacent to the inside surface, the metal of the superheater tube will usually be from 50 to 150 F hotter than the steam in the tube.

Dry steam is delivered to the superheater from the boiler at saturation temperatures ranging up to 675 F or more, depending upon the pressure at which the boiler is designed to operate. As the steam passes through the tubes, it may be superheated in some cases to a final temperature of 1050 or 1150 F. Carbon steels are reasonably resistant to scaling in steam and in combustion atmospheres up to about 1000 F but, under high stress, may suffer accelerated oxidation at somewhat lower temperatures. This attack generally takes the form of intergranular penetrations which, since the effective wall thickness of the tube is reduced, increase the unit stress and act as stress-concentration centers. Carbon-steel tubing in the superheater is therefore restricted to the moderate temperatures and pressures, while alloy steels are used for the more severe conditions.

To insure the long life required for satisfactory superheater design, the metal used must meet the following primary requirements at the anticipated operating temperature: (a) resistance to corrosion by steam, (b) resistance to corrosion by flue gas, (c) resistance to creep, and (d) resistance to stress rupture for a reasonable life period. To establish an adequate margin of safety and length of service life, these characteristics of the metal must be given due consideration in design. Economy dictates that the lowest alloy with properties suitable to the conditions should be used, stepping up from carbon steel to carbon-molybdenum steel and to the lower-chromium alloy steels as the temperatures increase. For metal temperatures approaching 1125 F, the lower-alloy steels up to and including 3% chromium are usually adequate. Chromium-molybdenum steel (9% Cr) and the stainless steels are used at higher temperatures, where conditions require either an increase in resistance to oxidation or higher load-carrying characteristics, or both.

As a general guide, where pressure stresses are not excessively high, the temperatures in Table 3 may be recommended as a safe maximum for various alloy steels. These temperature limits apply to the surface in contact with the flue gas and are based solely on the resistance-to-oxidation characteristics of the metal.

TABLE 3

Safe Maximum Tube-Metal Temperatures
(Oxidation-Resistance Basis)

Material	ASME Specification	Max Metal Temp°, F
Carbon steel	SA-210	950
Carbon molybdenum	SA-209 grade T1a	975
Croloy ½	SA-213 (SA-335 P2)	975
Croloy 1¼	SA-213 grade T11	1050
Croloy 2	SA-213 grade T3b	1080
Croloy 2¼	SA-213 grade T22	1100
Croloy 3M	SA-213 grade T21	1125
Croloy 5	SA-213 grade T5	1150
Croloy 9M	SA-213 grade T9	1200
Croloy 18-8 Ti	SA-213 grade TP321	1400†
Croloy 18-8 Cb	SA-213 grade TP347	1400†

°Of surface in contact with flue gases.

†Temperature limit may be modified depending upon character of fuel and other circumstances. It can be 100 to 200 F higher in low-pressure process-steam superheaters fired with low-sulfur fuel, but it should not exceed 1150 F when firing vanadium-bearing oil.

Stresses used for calculation of tube and/or pipe thickness should correspond to those given in Table P-7 of the ASME *Boiler and Pressure Vessel Code* for the respective materials. An abridgement of Table P-7 appears as Table E in the Appendix, page 22-A6 ff.

Steam Piping. The steam leads, or piping connecting the boiler and turbine, are highly important components of the power plant. Such piping should be properly designed and erected to accommodate thermal expansion and absorb vibratory stresses, and it should be so supported that it does not produce imposed stress on the boiler proper. While some mechanical joints are still used, it is now common practice to weld most of the joints between the boiler and the turbine. Carbon-steel piping is used for moderate pressures and for steam temperatures up to 800 F.

Carbon-steel pipe has shown a tendency to graphitization in the region of the welds at temperatures over 800 F, and it has been safe and prudent to restrict its use in this service to a temperature limit of this value°. Consequently, at these higher temperatures carbon-molybdenum steel has been commonly used for steam piping in many central stations. Long-time service, however, indicates that piping of this material also graphitizes, with a chainlike formation in the heat-affected area of welds and random formations in the body of the pipe proper. This phenomenon, the effect of which has made it necessary to reweld and replace a considerable amount of piping in the power industry, has been widely investigated. It has been found that the rate and extent of graphitization in piping materials are affected to some degree by steel-manufacturing practice. While it appears that graphitization is more liable to occur in materials which have received a considerable amount of aluminum during deoxidation, recent investigations indicate that, at somewhat longer time, it will also occur (at 900 to 950 F) in coarse-grain normal steel in which little or no aluminum has been used.

For greater resistance to graphitization under prolonged usage, the current trend in steam-piping materials is to use chromium-molybdenum steels. Steels of 0.5% Cr-0.5% Mo and 1% Cr-0.5% Mo are now commonly used in the range between 850 and about 1000 F, with a somewhat higher chromium content preferable in the range of 950 to 1050 F. Steel of 1¼% Cr-0.5% Mo has also been used to some extent. In several recently designed plants, steels of 2¼% Cr-1% Mo, 3% Cr-0.9% Mo, and, in some, even stainless steels of 18-8 Ti or 18-8 Cb alloy have been used for steam piping.

Steam piping of extreme wall thickness is made by boring solid forgings. Where moderate wall thickness is required, the piping, of seamless type, is made by the rotary piercing process. Precise practices in welding, hot-bending, and heat-treating are required in fabricating air-hardening alloys containing chromium, and metallurgical control is a predominant factor in securing the results desired. The steels presently used for steam piping and the corresponding nominal maximum steam temperatures are listed in Table 4. Allowable stress values are specified in Table P-7 of the ASME *Boiler and Pressure Vessel Code* (see Table E in the Appendix, referred to above).

°Graphitization has been detected in carbon-steel tubes in small amounts, but the experience has been that no failures have occurred, and consequently carbon steel is used for tubing at somewhat higher temperatures (see under *Tube and Pipe Steels*, above) than in the case of piping.

TABLE 4

Suggested Materials for Steam Piping

Material	ASME Specification	Nom Max Steam Temp, F
Carbon steel	SA-106	800
Croloy ½	SA-335 grade P2	950
Croloy 1	SA-335 grade P12	1000
Croloy 1¼	SA-335 grade P11	1050
Croloy 2¼	SA-335 grade P22	1100
Croloy 18-8 Cb	SA-376 grade TP347	1200
Croloy 16-13-3	SA-376 grade TP316	1200

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Forgings, Castings, Bolting, and Studs

A variety of parts for attachments, fittings, valves, and flanges may be made from carbon- or alloy-steel forgings and castings. While many of these parts are joined integrally to drums, water feed lines, steam lines, etc., by welding, there are also certain attachments which are joined with studs or by bolting. The material for bolts and studs must have high mechanical properties, and it is usually conditioned by normalizing and tempering or quenching and tempering. The tempering temperature is at least 100 F higher than the operating temperature in service. Furthermore, the material must be creep-resisting to minimize the relaxation in service, which would gradually loosen the fastening and necessitate retightening. Considerable study has been devoted to the subject of stresses and relaxation in bolts and studs, and a variety of bolting steels, suitable for various temperature levels and stresses, has been developed and is now in common use.

Heat-Resisting Alloys

High-alloy heat-resisting materials must be used for certain boiler parts that are exposed to high temperature and cannot be water cooled. These parts are made from alloys of the oxidation-resistant, relatively high-strength, chromium-nickel-iron type, many of them cast to shape as metallic baffles, supports, and hanger fittings. Oil burner impellers, soot blower clamps, and hangers are also made of heat-resisting alloy steels.

Deterioration of these parts is through conversion of the surface layers to oxides, sulfides, and sulfates, and in this condition they are referred to as burnt or oxidized. Experience indicates that steels containing 25% Cr-12% Ni or 25% Cr-20% Ni give reasonably good service life, depending upon location of the part in the flue gas stream and upon the characteristics of the fuel. Temperatures to which these metal parts are exposed may range from 1000 to 2200 F. Life may be shortened if exposed to flue gases from fuel oil containing vanadium compounds. Sulfur compounds formed from combustion of high-sulfur fuels are also detrimental and will reduce life. These may coreact in the presence of vanadium and cause greatly accelerated rates of attack, especially when the metal temperature of the part exceeds 1300 F. In these circumstances, there may be some advantage in using an alloy containing approximately 29% Cr-9% Ni.

Grades of Steel in Boiler Construction

The grades of steel commonly used by builders in boiler construction for tubes, pipe, plates, forgings, castings, and in bolting are covered by ASME specifications as listed in Table F in the Appendix, page 22-A10. Other steels in plate form, bars, and in structural shapes, used for nonpressure parts of boilers or for supports, are of usually accepted commercial standards.

Materials used in boiler construction should be obtained from selected vendors with long experience in the manufacture of the specific items required, such as steel plate, tubing, pipe, sheets, forgings, and structural steels.

Research and Development Facilities

For more than 50 years B&W has been actively engaged in the investigation of the behavior of steel under service conditions in the generation of power and in various industrial processes. Extensive laboratory facilities for metallurgical research, development, and control of quality in production are maintained at Alliance, O., Beaver Falls, Pa., and Barberton, O. A brief description of some of the equipment at these laboratories follows.

Metallographic Facilities. A variety of metallographs and microscopes (Fig. 39) are used by trained metallurgists to study the structural characteristics of metals. Microscopically observed characteristics, such as grain size, segregation, mode of occurrence and distribution of nonmetallic inclusions, and other heterogeneous conditions, have considerable bearing on the mechanical properties of the metal. By metallographic examination it is, therefore, possible to



Fig. 39. *Microstructure of metals determined by photography and direct observation*

predict with some assurance the probable behavior of a metal under specific conditions in service. This method is useful also in determining the reason for failure of metal parts in service, which may lead to improvements or better operating control.

Heat-Treating. Accurately controlled electric furnaces are used for grain-growth studies, carburizing tests, hardenability (Jominy) tests, experimental heat-treatments, and for scaling and aging tests of steels at various temperatures.

Melting. The laboratories are equipped with high-frequency induction furnaces for melting experimental heats of any combination of alloying elements desired. The melts produced in these units are processed into bars and billets for experimental use in determining the properties of various alloy combinations. To study the suitability of these metals for seamless tubing, sample billets are pierced in a special piercing mill. The final product is subjected to a series of carefully conducted tests designed to supply preliminary information on working characteristics and mechanical properties.

Mechanical Testing. The mechanical-testing laboratories are equipped with Brinell, Rockwell, Vickers, and Tukon hardness testers, and complete facilities for tensile, compression, impact, tensile-impact, and fatigue tests on metals at various temperatures.

A Hounsfield tensometer for tensile tests is in operation at the laboratory in Alliance. This unit can be operated by motor or by hand, and it can be used in either the horizontal or vertical position. The load capacity of the tensometer is 4480 lb. Specimens of 0.125-in. diameter and a minimum gage length of 0.447 in. can be tested in this machine (fitted with a suitable heating element) at temperatures up to 1650 F.

At Beaver Falls, an automatic hot-twist test machine is used to evaluate high-temperature working characteristics of steels. There are also dilatometers for determining the critical transformation temperatures and the thermal expansion of metals.

Creep and Stress-Rupture Testing. The Research and Development Laboratories in Alliance are equipped with 36 stands for creep tests, 32 stands for creep-rupture tests, and 60 for stress-rupture tests. If desired, the 36 creep-test machines can also be used for creep-rupture tests.

The temperature for 20 of the creep-test machines is regulated by multiple-circuit electronic controls. These controls are actuated by the expansion of the furnace tubes, and the temperature can be regulated to within plus or minus 1 F up to 1550 F. In these units a test bar of 10-in. gage length is used, and it is possible to measure an extension of less than 10 microinches (0.00001 in.). In the other 16 creep-test machines, a temperature up to a maximum of 1800 F can be maintained within plus or minus 2 F. The gage length of the test specimen is 3 in., and an extension of 5 microinches (0.000005 in.) can be measured.

The four stress-rupture furnaces are of annular design, and each contains 12 circular chambers for test specimens. The four multiple units are usually operated at different temperature levels up to 1500 F. Each furnace can be maintained at the desired temperature within plus or minus 3 F. There are 16 additional stress-rupture machines (single-station units) at the Barberton Works.

The creep laboratory is air conditioned, and a constant room temperature of 83 F, plus or minus 1 F, is maintained at all times. A gasoline-driven motor-generator set is available to carry the full load of the creep and stress-rupture equipment in the event of a power outage.

In addition to the creep and stress-rupture tests described above, equipment has been developed and is being used for testing tubular materials in tubular form, which more closely simulates the actual stress conditions encountered in operation. This presents a more realistic picture of the stress combinations prevailing on a tube in service.

Other Testing Facilities. The application of X-ray diffraction to metallurgical research problems and inspection by radiographic methods are standard procedures.

The Barberton Works Control laboratory utilizes the most modern and up-to-date X-ray equipment, including 1-million-volt and 2-million-volt stations, to inspect all the main seams in boiler drums after welding and heat-treatment.

X-ray equipment for diffraction, spectrometric, and fluorescence analyses (Fig. 40) is in constant use at Beaver Falls and Alliance. These units provide unique



Fig. 40. *X-ray equipment for diffraction, spectrometric, and fluorescence analyses of metals for quick results*

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facilities for rapid quantitative and qualitative identification of alloy components and compounds, determination of the mechanism of alloy formation, stress analysis, and related problems.

Two of the Company's laboratories are equipped with light spectrographs for analytical work, while in a third laboratory (Barberton Works) there is an electron microscope for special studies of metal structures at extremely high magnifications (see Figs. 41 and 42).

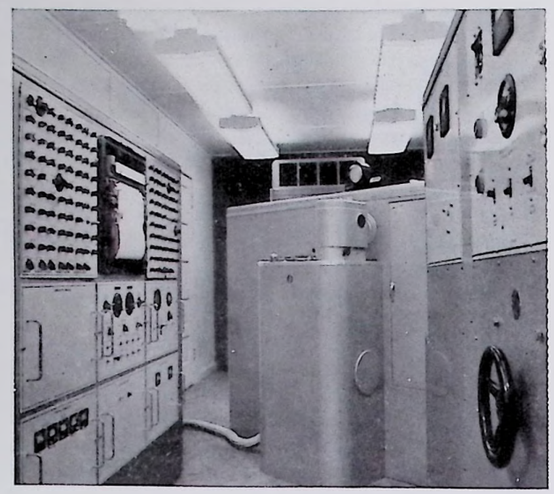


Fig. 41. *Light spectrograph for quick analysis of a wide variety of metals and other materials*

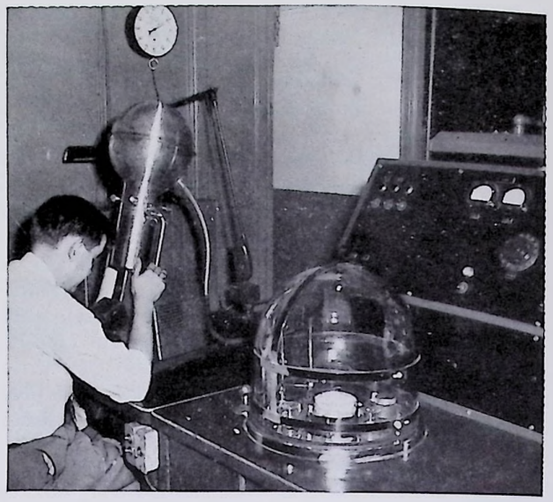


Fig. 42. *Electron microscope for extremely high magnification (beyond 2000x) in the study of metal structure*

23

SELECTION OF STEAM-PRODUCING EQUIPMENT

THE boiler designer must produce economically a steam-generating unit that will satisfy the needs of his customer. To produce the most economical steam-generating unit requires close cooperation between the designer and the user's engineering staff or consultants. This community of interest exists whether the unit is small or large.

By coordinating the various components—boilers, furnaces, fuel burners, fans, and controls—the boiler manufacturer has produced a limited series of standardized steam-generating units, particularly in the smaller capacities (up to about 50,000 lb of steam per hr), burning oil or natural gas. Within the proper scope of their application, these standardized units (some sizes shipped assembled) are simple and economical in performance. Beyond these limits, only partial standardization of some component elements is possible because of 1) the user's individual requirements and conditions for pressure, temperature, capacity, and fuel, and 2) the incidence of continual improvements as a result of the constant search for better over-all economy. Since the steam plant is to become the heart of the system it serves, with some 30 years of expected normal useful life, consideration should be given to every established improvement applicable to each new design.

The cost of fuel consumed throughout the life of a steam-generating unit is so great that the savings from even a slight increase in efficiency can easily amount to many times the investment required to obtain this increased efficiency (see *Evaluation* below). The time, effort, and cost to provide proved features of better operation in keeping with plant requirements and conditions are economically justified. As long as progress continues, this is why (generally in the larger-size categories) premature standardization is contrary to the ultimate best interests of the owner. Each project, therefore, warrants the attention and expense required to produce a coordinated steam-generating unit which will insure a dependable steam supply with maximum economy.

The requirements and conditions that form the basis for the plant designer's selection of equipment for a specific case can normally be outlined as follows:

1. Steam Requirements
   - a) Pressure and temperature
     - At points of use
     - At outlet of steam-generating unit
     - Allowable variations
     - Predictable future requirements
   - b) Rate of heat delivery (or steam flow)
     - To points of end use
     - To boilerhouse auxiliaries and feedwater heating
     - To blowdown
     - From outlet of steam-generating unit
     - Variations—minimum, average, and maximum
     - Predictable future requirements
2. Boiler Feedwater
   - a) Source and analysis
   - b) Temperature entering steam-generating unit
3. Geographical Considerations
   - a) Available fuels—analyses, costs, and future trends
   - b) Space limitations
   - c) Relation of new equipment to existing boilerhouse equipment
   - d) Restrictions of local laws and immediate neighborhood, present and predicted
   - e) Elevation above sea level
   - f) Earthquake possibility
   - g) Foundation conditions
   - h) Kind and unit cost of energy for driving auxiliaries
   - i) Climate
   - j) Accessibility for service
4. Operating Personnel
   - a) Experience level of workmen for operation and maintenance
   - b) Cost of labor

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5. Evaluation
   a) Of improved efficiency
   b) Of auxiliary power required
   c) Of various fixed charges
   d) Of many of the above items

Considerable time and effort are required to establish sufficiently accurate basic data with comprehensive consideration of engineering factors, judgment in planning for future expansion or changes, and evaluation of the tangibles and intangibles involved, so that the experience and craftsmanship of the boiler manufacturer can be applied to the full benefit of the plant designer and the owner. The user should, at the outset, decide who is to prepare these data. Lacking adequate and capable personnel, the services of consulting engineers are always available. Before the equipment can be selected, the basis of operation and arrangement of the entire steam plant must be planned, then construction drawings must be prepared for the foundations, building, piping, and walkways. Also the construction work must be coordinated and supervised.

In general, equipment selection cannot be reduced to the application of simple rules or formulas. Through cooperation with the boiler manufacturer, discussion in detail of the items referred to above will help in making intelligent decisions.

Pressure and Temperature

At points of use, the required steam pressure and temperature may be from 2 psi and 215 F (saturated) for heating, to 4500 psi and 1150 F for a large turbine generator. Some general suggestions for establishing the end-use requirements and their effect on the steam-producing equipment seem appropriate.

Steam for Process and Heating

The pressure of saturated steam (no superheat) used for process heating is such that the corresponding condensing steam temperature is somewhat above the required temperature of the materials to be heated. Generally superheat is of no value for this kind of service and is often undesirable because of its interference with temperature control. Reclaiming or devulcanizing rubber, where the rubber in a caustic solution is heated to 400 F by condensing saturated steam at 250 psi (407 F) in the jacket of the devulcanizer, is a typical example of process heating with steam.

Pressures of saturated steam for comfort heating of buildings range from 2 psi to as high as 80 psi in the case of "space heaters." It is seldom economical to distribute steam through long lines at pressures below 150 psi because of piping costs. Furthermore, the usual requirements for steam within the boiler-house for soot blowers, feed pumps, and other auxiliaries make it desirable to operate boilers at a minimum of 125 psi. Consequently few steam plants of any size are operated at pressures below 125 psi. If the pressure required at points of use is lower, it is common practice to use pressure-reducing stations at or near these locations. The central steam-heating systems in New York, Rochester, Pittsburgh, Cleveland, and Detroit are examples of this practice.

Where relatively small amounts of steam at normal plant-steam pressure but with superheat are required, it is usually better to use a separately fired superheater rather than to superheat the total plant-steam supply.

The pressure required at the outlet of steam-producing equipment for heating service usually ranges from 125 to 250 psi, and superheat is not required. For this service, boiler manufacturers have generally standardized on a pressure of 250 psi for small water-tube boilers.

It is customary American stationary-boiler practice to hold the main steam-line pressure practically constant for all loads, on the premise that this condition satisfies all pressure and quantity requirements of the steam-using equipment. Automatic combustion-control apparatus is accordingly designed to function on this basis. The main steam-line pressure is, of course, lower than the pressure within the boiler drum by the amount of the pressure loss through the nonreturn steam-outlet valve, the superheater (if used), and the boiler dry pipe. Since these losses vary substantially as the square of the steam flow, the internal boiler operating pressure will be at a maximum with maximum steam output. To avoid unnecessary losses and maintenance from frequent action of the safety valves, the first safety valve should be set to relieve at not less than the maximum internal boiler pressure plus about 5%. By law, the boiler design pressure cannot be less than the safety-valve relief pressure. The design pressure is thus established from a determination of the pressure required for the service.

Combination Heating and Power Service

Some manufacturing operations, as in paper and textile mills, in the production of chemicals, and in processing rubber, require mechanical or electrical power as well as steam for process heating. For such applications, studies are made of the relative merits and costs of 1) a plant where the power is purchased and steam is generated to supply the heating requirements only, and 2) a plant where steam is generated to supply both the power and the heating loads. A sound appraisal of the relative merits of the two alternatives requires a knowledge of the steam and power requirements, ability to correlate these requirements, economic studies, and good judgment (see Chapter 10). The following general summary may be of assistance:

1. The basic economic advantage in generating steam to supply both power and heating requirements is from the utilization of a much larger portion of the heat supplied in the fuel, by reduction or elimination of the loss to the circulating water in a condensing plant. This loss can be as much as 60% of the heat supplied in the fuel, even in a modern central station.

2. Despite this fundamental thermodynamic advantage, it is usually more economical to buy power when it is available at reasonable rates from a dependable source, except where:
   a) Waste fuels and waste heat, such as bagasse, blast-furnace gas, sawdust or hogged wood, and hot gases, are available at low cost from the plant process.
   b) The steam-heating and power demands are reasonably parallel and relatively large, i.e., 50,000 lb of steam per hr or more.

3. Variations in process-heat and power demands usually do not coincide. However, if the process-steam requirement is always the larger, the exhaust steam from a straight back-pressure machine can be supplemented with a pressure-reducing and desuperheating system. If this is not the case, there are two possible alternatives:
   a) Use of an extraction condensing machine.
   b) Generation of a part of the power in a straight back-pressure machine and purchase of the remainder of the requirement.

As an example of a by-product power installation under favorable conditions, the cost of power generated in a plant of 10,000-kw capacity with 12% fixed charges and fuel cost of 30¢ per million Btu will be in the order of 4 to 5 mills per kwhr.

The selection of steam pressure and temperature for by-product power plants requires study. Approximate steam conditions at the turbine or engine may range from 150 psi and 500 F in one type of factory application to 1800 psi and 900 F in other applications.

Power Generation

Steam engines and steam turbines are the alternative prime movers available for the generation of mechanical or electrical power. Except for the smallest isolated installations (below 500-hp capacity), the high-speed turbine predominates because of its compactness, efficiency, and low-cost features. However, since mechanical power in large amounts is seldom required locally, it is usually more economical to buy it as electrical energy from a central station. The present-day central stations can easily deliver a kwhr for each 9000-10,000 Btu supplied in fuel. The economy of the central station is evident when this heat rate is compared with a heat rate in the order of 50,000 Btu, or more, per kwhr required for a 100-kw noncondensing engine or turbine-driven generator set. Consequently, the bulk of the mechanical and electrical power used in this country today, for stationary service, is produced by relatively large steam-driven turbine generators—from 10,000 to 450,000 kw in size. For such installations the highest pressures, superheats, reheats, and vacuums are all used to obtain maximum economy.

To determine the pressure and temperature that give the lowest total power cost throughout the life of a proposed steam power plant, it is necessary to balance plant investment, fixed charges, fuel, and other operating costs for the various alternatives. The optimum may be steam at 400 psi and 700 F, as for a 500-kw unit, or 4500 psi and 1150 F with reheat for the largest units. For the latter, with current costs of money, equipment, fuel, and labor, the tendency is to select the highest pressures and temperatures that can be economically used with available materials for superheaters, reheaters, piping, valves, and turbine parts.

For optimum capacity and efficiency, it is good practice to maintain constant steam pressure at the turbine throttle. Consequently, the pressure at the superheater outlet varies with load and is equal to the pressure at the turbine plus the pressure drop from the superheater outlet to the turbine. This pressure drop is usually about 5% of the turbine pressure at maximum steam flow. It must include an allowance for the nonreturn valve, but in many modern single-boiler single-turbine installations a valve between superheater outlet and turbine throttle is not required.

The design pressure for the steam-producing equipment is determined by adding the following items to the turbine throttle pressure:

1. Pressure drop through piping and nonreturn valve at maximum flow.
2. Pressure drop through superheater and drum internals at maximum flow.
3. Ample allowance above drum operating pressure at maximum steam output for boiler safety-valve blowdown (about 5%).

Item 2 is wholly dependent upon the detail design of the steam-producing equipment as determined by the boiler manufacturer, and the plant designer should specify that the boiler design pressure be 5% above the maximum drum pressure.

The steam temperature specified at the superheater outlet is taken as equal to the temperature required at the turbine, since the loss in temperature from one point to the other is negligible. For optimum performance it is desirable that the steam temperature and pressure remain constant at all loads. Steam temperature variation and control for boilers are discussed in Chapters 12 and 13. Controlled superheat prevails for turbine sizes above 30,000 kw. In steel-mill installations, where the alternate firing with blast-furnace gas, oil, coal, or coke breeze tends to produce widely different steam tem-

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peratures for a given steam flow, control is particularly attractive.

For large turbine-generator units—80,000-kw capacity and up—using steam conditions above 1250 psi and 900 F, with a single boiler unit per turbine generator, the so-called "reheat" cycle is now popular because of the increased station thermal efficiency over the nonreheat cycle (see Chapters 10 and 12). It is standard practice to locate the reheater within the boiler enclosure. The reheater, receiving steam exhausted from an intermediate-pressure stage of the turbine, raises the temperature, usually to the initial turbine-inlet temperature, and delivers the steam back to the low-pressure section of the turbine. For installations of this type, the temperature and pressure of the steam leaving and re-entering the turbine and the load range over which constant temperature of reheat steam is to be maintained are established by the plant designer in cooperation with the turbine and boiler manufacturers.

Rate of Heat Delivery

To insure prompt fulfillment of all steam demands, i.e., the delivery of heat to all points of use at the required rates, it is necessary to select steam-producing equipment of sufficient capacity, range of output, and responsiveness. The demand may be steady, as in most space-heating systems, or it may fluctuate widely and rapidly, as in a heavy-forging plant. Many steam-heating processes, as in the initial heating of a liquid batch, require high peak flows of short duration. Rapidly changing rates of steam flow are usually required for a steel-rolling mill in generating electric power, which fluctuates with the load on the rolls. The steam-flow requirements should, therefore, be accurately established for:

Peak flow
Maximum continuous flow (usual steady maximum flow)
Minimum flow
Rate of change in flow

The peak load will, of course, establish the top capacity for the steam-producing equipment and all of its auxiliaries. For widely fluctuating loads it is advisable to establish the 15-minute peak. In most systems, peaks of shorter duration can be met by the "storage" of heat.

The range from minimum to maximum output is a most important factor in selecting pulverized-coal, oil, and gas-burning equipment. To maintain ignition and smokeless combustion for these types of firing, the range from minimum to maximum output must also be considered in designing the furnace. It is of somewhat lesser importance when underfeed stokers are used.

The required rate of change in steam flow (load) may materially affect the entire design of the steam-producing equipment. Within a specified load range, good response is easily obtained with well-designed firing equipment for Cyclone Furnace operation, pulverized coal, oil, gas, and with some stoker types. A multiplicity of pulverizers and burners will widen the available range, but, if there is a frequent change in load, the manual operation of cutting in and cutting out pulverizers and burners may be objectionable. It is therefore important that the designer, manufacturer, and operator all understand what can be done by full-automatic firing and what is required beyond that.

To select equipment to produce steam at a minimum total cost, it is also necessary to establish the probable "capacity factor"—the ratio of the average output to the rated output. The capacity factor serves as a basis for the exercise of judgment in establishing the value of incremental increases in the efficiency of steam generation. Obviously, the first cost necessary for high efficiency in a plant operating at continuously high output could not be justified for a plant operating only a few weeks per year, as in some branches of the food-processing industry.

Boiler Feedwater

Raw water from surface or undersurface sources invariably contains in solution some degree of troublesome scale-forming materials, free oxygen, and sometimes acids. The treatment of water for boiler use must be based on quantitative laboratory analyses of samples usually taken during the various seasons. The latter requirement is particularly true for flowing streams and small lakes, the maximum contamination occurring at the end of extended dry seasons.

Dissolved oxygen will attack steel, and the rapidity of attack increases sharply with rise in temperature. Hence there should be no oxygen in the water within a boiler. In modern deaerating feedwater heaters, the oxygen content can be reduced to 0.005 cc/liter (0.007 ppm) or less. In good operating practice, the remaining traces of oxygen are removed by chemical treatment in the boiler (see Chapter 21).

The temperature of the feedwater entering an economizer should be high enough to prevent condensation and acid attack on the gas side of the tubes. Dew point and tendency to corrosion vary with the sulfur content of the fuel and with the type of fuel-firing equipment (see Fig. 16, Chapter 11).

Availability of Fuel as a Factor

For the dependable and economical generation of steam, the equipment must be suitable for burning the fuel or fuels available at the particular site at a reasonable cost. The usual fuels are wood refuse, bagasse, digester liquor, peat, lignite, bituminous coal, anthracite, natural gas, oil, blast-furnace gas, coke, petroleum coke, and refinery gas (see Chapters 2, 3, and 20). Of these, bituminous coal, natural gas, and oil are the most important in steam production for power, and bituminous coal leads the other two.

The cost of delivered fuel is constantly changing because of fluctuations in price and freight rates, seasonal variations in availability of natural gas, and temporary shortages of coal or fuel oil. Reliance on a single fuel is possible for some users with dependable and abundant local sources. The majority of users, however, particularly utility companies, finds it advantageous to plan for the use of more than one fuel because of cost fluctuations or to assure continuity of operation should the normal supply be interrupted.

Natural Gas

The operator who has an assured supply of economical natural gas is fortunate. If natural gas only is burned, fuel storage facilities, ash hoppers, ashpits, and ash-handling equipment are unnecessary. Soot blowers can be omitted, dust collectors are not needed, and there is no stack-discharge problem. Control of heat input to the boiler furnace is simple. No consideration of ash deposits is necessary in arranging the heating surface for optimum heat transfer and draft loss. The total enclosure volume is at a minimum, and the adaptability for outdoor service is increased. The sole disadvantage of natural gas fuel is the potential hazard because explosive accumulations of gas from faulty valves or fittings or as a result of incorrect operation are not readily detectable. With well-trained operators, however, adequate safety precautions can be taken.

Blast-Furnace Gas

Blast-furnace gas, a by-product of blast-furnace operation, is usually the principal fuel for steam generation at steel mills. Its storage is impractical because of the large volume per unit of heat value (90 Btu per cu ft), and it is either burned immediately or wasted to the atmosphere. Since the supply varies with cycles of blast-furnace operation, either oil, pulverized coal, or coke is commonly used as a supplementary fuel. When the auxiliary fuels are used, it is desirable to water-cool the furnace and thus avoid excessive maintenance. Hot air should be used for combustion to maintain ignition of the blast-furnace gas in the water-cooled furnace.

In most modern plants using blast-furnace gas, precipitators are installed to eliminate iron-ore dust, lime, and coke particles, which, if carried over, tend to deposit on the heating surfaces, where they are difficult to remove during operation. Auxiliary pulverized-coal or coke-breeze firing is quite effective in removing these deposits. When the boiler is out of service, the deposits are readily removed by water-washing.

When a shift is made to the auxiliary fuel in a unit burning blast-furnace gas, there is a drop in superheated steam temperature because of the lower gas mass flow. Some type of steam temperature control is therefore necessary when constant steam temperature is desired, as with most turbines operating at 850 F or higher.

Fuel Oil

Oil fuel has many of the desirable features of natural gas—ease of handling, elimination of ash hoppers and ashpits, and relative freedom from stack-discharge problems. It does, however, require fuel storage, heating, and pumping facilities. In recent years, increasing difficulties have been experienced with residue oils because of changes in the sources of supply of crude petroleum and advances in oil-refining techniques. Fuel oil with high sulfur and vanadium content can cause very troublesome deposits in superheaters and air heaters (see Chapter 18). To minimize trouble from these deposits, care must be taken in the arrangement of the heating surfaces, and adequate soot-blowing equipment must be provided. In many instances provisions are made for the water-washing of air heaters. Design of superheater supports also requires special attention because of the attack of ash deposit from oils of high vanadium content.

Coal as Fuel

Coal is the most universally available and also the most complicated and troublesome of the major fuels. Its use involves storage and handling facilities, crushers or pulverizers, ash-disposal equipment, and frequently dust collectors. The boiler stack discharge may or may not be troublesome, depending upon the equipment installed and local conditions. Ash hoppers and soot-blowing equipment are essential components of the steam-generating unit.

The wide variations in the properties of coal and coal ash are well known. The over-all design of the steam-producing unit must be suitable for optimum performance in burning the particular coals available, with proper allowance for reasonable deviation from the coal analyses on which the design is based. Coals with high ash-fusion temperatures (see Chapters 2 and 18) are inherently suitable for burning on most types of stokers, and, when pulverized, in dry-ash-removal hopper-bottom furnaces. Low-ash-fusion coals are satisfactorily burned on chain-grate, spreader, and sometimes on underfeed stokers. Low-ash-fusion coals in pulverized form can require a furnace so large for satisfactory dry-ash operation that slag-tap designs may be more economical. These low-ash-fusion coals, in crushed form, can also be burned economically in the Cyclone Furnace with slag-tap ash removal. In any particular instance, the final selections of burning method and

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boiler type are based on a combination of technical factors, economics, engineering judgment, and sometimes personal preference. When a unit is to be added to an established plant, integration of the new with the existing equipment must be considered.

If coal is to be the fuel for a prospective installation, a tabulation should be prepared of the analyses of all the coals economically available. The equipment that will satisfactorily burn the coals within this range should then be selected, so that the owner is in a position to buy coal with the maximum dollar economy.

Coal-Burning Equipment

Detailed descriptions of various types of fuel-preparation and -burning equipment appear elsewhere in this book. The following survey serves as a summary of the principal points affecting the choice of method and equipment when burning coal.

Stokers

The single-retort underfeed stoker is versatile for a relatively wide range of bituminous coal classifications, but it operates best, particularly at low load, with "free-burning" coal that does not tend to coke. This stoker is flexible over a range of duty from banked fire to full load. With reasonably careful operation, there is no smoke or objectionable fly-ash discharge from the stack. A deep ashpit is not required. The main disadvantage is in the labor required to remove ash when burning coal with a low ash-fusion temperature (below 2200–2300 F ash-softening temperature on a reducing basis), which tends to clinker.

The principal advantage of the spreader stoker is that a wide variety of bituminous coal and also lignite can be successfully burned. It is sometimes preferred where rapid load changes are involved. There are two types of ash-removal equipment, the dumping grate and the moving grate. The former requires more manual labor, and the latter has the advantage of lower carbon loss.

With the spreader stoker there is a carry-over of coarse fly-ash and carbon particles through the boiler and to the stack. It is therefore necessary in most locations to install a fly-ash collector with the added operating cost of higher induced-draft-fan power. To reduce thermal loss, the collected ash is continuously reinjected into the furnace to burn out any remaining carbon. In selecting a boiler unit for spreader-stoker firing, care must be taken in the arrangement of tubes, supports, baffles, and other components to avoid coarse-particle erosion or sand-blast action. Boilers with straight-through gas flow are desirable for these conditions (see some B&W boiler types below).

The chain-grate and the traveling-grate stokers are well adapted for free-burning coals with high ash content. They are not sensitive to low ash-fusion temperatures, and maintenance is low. Coking coals should be avoided. A deep ashpit is desirable to reduce the frequency of cleaning. With suitable furnace arches, anthracite screenings and coke breeze can be burned successfully on chain-grate or traveling-grate stokers. Maximum burning rates approximating 34 lb per hr per sq ft of grate prevail. Fly-ash collectors are required except in isolated locations.

The once-popular multiple-retort underfeed stoker is seldom selected today for the higher steam outputs because of the generally more economical performance of pulverized coal for this duty. However, the underfeed stoker will burn many different varieties of coal, and it has a wide load range. At burning rates up to 40-50 lb per hr per sq ft of grate area, it can be operated without smoke and objectionable fly-ash or carbon discharge from the stack. A deep ashpit is required for this type of stoker.

At present, because of increased maintenance, the temperature of the preheated air for all types of stokers is limited to about 350 F. This condition tends to curtail the over-all thermal efficiency of the unit without an economizer.

For all types of stokers and their operation see also Chapter 16.

Pulverized-Coal Firing

While pulverized-coal firing can be adapted to small boilers, the disproportionately high incremental costs of pulverizers and other auxiliaries tend to establish a practical limit in steam output of not less than 50,000 to 60,000 lb per hr for this type of firing. There appears to be no presently recognizable upper limit of size for the pulverized-coal firing of boilers. The equipment can be designed to burn practically any bituminous coal and lignite. Anthracite can be successfully burned in pulverized form, but fuel preparation and furnace and burner arrangements require great care.

The over-all aspects of pulverized-coal firing, as applied to boiler units, may be summarized as follows:

1. Almost any coal mined in the United States is suitable.
2. Economically suitable for a very wide range in boiler size.
3. Proper coal preparation, including the removal of moisture, and the proper handling of the pulverized coal (direct firing and drying within the pulverizer preferred).
4. Hot air for combustion and for moisture removal (an air heater is required).
5. Handling of the ash refuse (slag from slag tap, dry ash, and fine flue dust).
6. Wide flexibility in operation and high thermal efficiency.

7. Dust collectors are required in most installations.

For a more complete discussion of pulverized-coal firing see Chapter 17.

Cyclone Furnace Firing

Cyclone Furnace coal firing has been established in recent years as a method parallel to pulverized-coal firing. At present, the range of coals suitable for this method is somewhat more restricted, and the method cannot be economically applied to quite as small a boiler. However, burning crushed coal eliminates the initial and operating costs of pulverizing equipment. The furnace and stack gases are substantially clean, so the additional costs of fly-ash collection systems and ash erosion of equipment are reduced, if not eliminated. These are attributes of a very high order. Cyclone Furnace firing with coal, oil, or gas is currently suitable for boiler sizes ranging from 100,000 to more than 5,000,000 lb of steam per hr.

See Chapter 28 for a more complete discussion of the Cyclone Furnace.

Space Limitations

New steam-producing equipment is frequently installed in an existing building. Boilers are made in a variety of shapes (see outline of boiler types below), and a careful study of space conditions and possible arrangement of the equipment is necessary for best over-all results. Space conditions sometimes have a direct bearing on the selection of the type of firing and number of steam-generating units for a given steam demand. Often the space occupied by an older inefficient boiler unit is entirely suitable for a modern unit of high efficiency, with double the steam output or more.

Relation of New to Existing Equipment

If the pertinent facts are established and studied, existing equipment can frequently be used for a new steam-producing unit. For instance, in a natural-draft plant, if there is an existing stack of sufficient area to accommodate the additional load, the unit selected should have a draft loss less than the available draft, to save the expense of a new stack. Where superheaters are used and the new boiler is to be connected to the old steam main, the steam pressure drop through the new superheater should be about the same as through the old superheaters, since all boiler drum safety valves (as required by most state laws) are set to relieve at the same pressure. Modern coal pulverizers and direct firing may be necessary to relieve the older overloaded equipment, so that needed steam-generating capacity can be installed without addition to buildings. If the new boiler unit is to be erected on existing piling and the conditions limit the allowable column loading, the unit can frequently be designed for column-loading distribution to match the existing footings and thus save expensive foundation work.

Restrictions, Local Laws, and Neighborhood

Restrictions by local laws and the nature of the surrounding area affect the selection of equipment mainly with reference to smoke or fly-ash discharge from stacks. Most towns and all cities will doubtless sooner or later enact strict ordinances to limit the allowable smoke and fly-ash discharge from boiler stacks. Consequently, except for extremely isolated locations, equipment should be selected that can be operated without smoke and without more than the recommended ASME fly-ash discharge of 0.85 lb of ash per 1000 lb of stack gas (at 32 F and 29.9 in. Hg, adjusted for 50% excess air). It is also well to discharge gases from boilers through relatively high stacks (150 to 200 ft above grade) to aid in the dispersal of any sulfurous gases and fly-ash still present. Sometimes stacks must be high enough to discharge the gases above surrounding buildings. The height of stacks in the vicinity of air fields may be limited for reasons of safety. Where earthquakes are prevalent, it is important that the boiler-unit supporting structure be suitably braced and in accord with local laws and regulations.

Elevation Above Sea Level

The site elevation materially affects draft loss, stack draft, fan capacities, and motor capacities (see Chapter 5). It also affects the suction head of fuel oil pumps. The site elevation may even make it impractical to employ stack draft.

Power for Driving Auxiliaries

Power is required for driving feed pumps, fans, stokers, and pulverizers. Feedwater heating is required for all steam plants. It is common practice in industrial process-heating plants to use sufficient steam-driven auxiliaries to provide enough exhaust steam for the feedwater heater and to use motor drives for the other auxiliaries. There are instances, however, where the demand for low-pressure exhaust steam is great enough so that all the auxiliaries can be economically steam driven and thus often avoid the expense of purchased power. Sometimes both motor and turbine drives are provided for each auxiliary—particularly appropriate for "cold" start-up.

In the large electric-generating plant, motor drives are generally used throughout for the auxiliaries. With the high efficiency of the main turbine generator, motor drives, even with allowances for transmission and switching, are more economical than small turbines. Output control and response of the driven auxiliary are also simpler and more effective with motor drive.

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Climate as a Factor

To save building costs, boilers are frequently installed outdoors in many parts of the United States, particularly where the climate is mild, as in the South. Farther north where the climate is moderate, as in New Jersey, New York, and Washington, savings in building costs have been made by semioutdoor installations with overhead protection, which have been quite satisfactory. Justification of outdoor operation is less important in areas of cold climate, as in Minnesota, Montana, and the Dakotas.

Operating Personnel

The American boiler operator is able to handle intelligently a wide diversity of power-plant equipment, as demonstrated by the thousands of successful plants operating with steam pressures and temperatures up to 2500 psi and 1100 F. This is partly the result of the usual close cooperation between the plant designer and the equipment manufacturer. However, there are cases where the operating personnel is relatively inexperienced and likely to remain so. In such cases it would be well to select simple and conservative equipment, even at the sacrifice of some efficiency.

Selection of Boilers and Related Equipment

Certain general guiding principles should be observed in the selection of boiler units and their auxiliary equipment. The most important of these principles may be summarized as follows:

1. The user must establish his present and, as far as possible, his future conditions and requirements (see above).
2. The user should have a working knowledge of the performance of the different components of the integrated unit under various conditions, particularly his own. Such knowledge should include an appreciation of the basic physical laws, the vagaries of fuels, the significance of the many different arrangements of heat-absorbing surface, methods of fuel burning, and handling of refuse. Guidance in acquiring this perspective appears throughout this book. See also section below on specific boiler types.
3. A sound evaluation should be made of the costs, to obtain benefits as outlined in the examples at the close of this chapter.

When hypothetical conditions are assigned for the selection of a steam-producing unit suitable for a particular service, there is always the risk of the omission of some dominant factor applicable to an actual case. Subject to such limitations, the following may serve as a simple example of the selection of a steam-producing unit for a particular service:

EXAMPLE

Steam-Producing Units for a Hospital

Pressure and Amount of Steam Required (at points of use)

Heating (radiators) pressure, psi	2
Load, winter peak, lb/hr	20,000
Season for heating, months	8
Laundry pressure, psi	100
Load, 8 hr per day, lb/hr	4,000
Sterilizer pressure, psi	65
Load, lb/hr	1,000
Kitchen pressure, psi	40
Load, lb/hr	1,000
Steam to feedwater heater, pressure, psi	1
Load, peak, lb/hr	1,400
Temperature of feedwater to boiler, F	212
Total steam from boiler	
Summer peak, lb/hr	6,320
Winter peak, lb/hr	27,400

Other Conditions

Condensate return, %	80
Temperature, F	180
Make-up water—Zeolite treated	
From softener, total solids, ppm	170
Available fuels	
Bituminous coal, Ohio and Pennsylvania	
In bunker, cost in $ per ton	8.50
Btu per lb as fired	12,500
Fuel oil #6	
In storage tank, cost in ¢ per gal	10
Btu per gal	145,000

General Conditions of Location

No critical space limitations
No existing equipment
Clean stack required
Elevation, 500 ft above sea level
Good foundation conditions
No earthquakes
Reliable source of electrical energy
Severe winter climate
Reasonably well-trained personnel

Predictable Future Conditions

Size may be increased 50% in 10 yr
Both coal and fuel oil will be available

Number and Capacity of Boilers

For the present, two, each rated at 20,000 lb/hr
In the future, one, rated at 20,000 lb/hr

Boiler Steam Pressure and Output

Boiler output pressure, 100 psi
With reducing valves
Estimated total steam output

At present requirements, lb/yr	77,360,000
Derived as follows:	
Heating—8 months (5840 hr) at 10,000 lb/hr avg, lb/yr	= 58,400,000
Laundry, Kitchen, Sterilizers: 3000 hr at 6320 lb/hr avg, lb/yr	= 18,960,000
Total steam, lb/yr	77,360,000

Fuel Cost Comparison per Year

Yearly efficiency burning coal, %	70
Yearly efficiency burning fuel oil, %	75

For coal, $\frac{77{,}360{,}000 \times 1000}{12{,}500 \times 2000 \times 0.70} \times 8.50 = \$37{,}580$

For fuel oil, $\frac{77{,}360{,}000 \times 1000}{145{,}000 \times 0.75} \times 0.10 = \$71{,}140$

The estimated saving is $33,560 per yr if coal is used.

It is evident from the above example that, for a complete installation of two boiler units, each producing 20,000 lb of steam per hr with all necessary auxiliaries included, at a cost of about $400,000 for coal-fired and about $360,000 for oil-fired units, the choice would be preponderantly in favor of the coal-burning units if economics is the dominant factor in selection.

For this service the type H boiler without superheater and with a single-retort stoker is well adapted (see boiler types below). At about 30 lb of coal per sq ft, hr, the grate area of the stoker would be 70 sq ft. Normally the stoker would be steam driven. The forced-draft fan could be motor driven or driven by a turbine exhausting into a low-pressure steam line. Two feed pumps, one motor driven and one steam driven, would be preferred. A stack suitable for the draft required for the three boilers (one future) at full load would have a diameter of 7 ft 6 in. and a height of 150 ft above the operating floor level.

The design of the above installation, it should be noted, is based on a heating load with long periods of extremely low-load operation. At this light duty, the all-refractory furnace of the type H boiler will be effective in abating smoke during the low-load summer operation, and with the conservative rating of the units the furnace maintenance cost should be low. Also, because of its simple design, without water-wall handholes and with few drum baffles, the boiler can be readily cleaned internally.

In a location where the price of oil is more favorable or where natural gas is an economical fuel, the type FM boiler (see below) should be considered. If the type FM boiler is selected, care should be taken to make sure that the burners and automatic controls suitably cover the load range without smoke.

Changes in or additions to the original conditions will naturally affect the selection of the main unit and components. For instance, in the above example had the solid fuel been a free-burning (Midwestern) coal, a spreader stoker probably would have been selected in preference to the single-retort stoker. If North Dakota (or Texas) lignite had been the fuel, a chain-grate stoker (with a considerably modified furnace with extensive ignition arches) would have been considered.

Again, if in the example additional steam-producing capacity was to be installed in a plant operating satisfactorily with straight-tube boilers burning oil with an assured future supply, the most likely selection would not be as above but would be similar to the existing units.

Some B&W Boiler Types and Attributes

Familiarity with various classes of steam-generating units and their components is essential for an intelligent choice of equipment. The following individually descriptive outlines are intended to serve as an introduction to a number of designs of boiler units covering a wide range of output, application, and performance characteristics. Many B&W boiler types have been omitted from the descriptive outlines because of space limitations and because, in a number of instances, they are in effect variations of the designs shown. Some are illustrated elsewhere.

In the selection of steam-producing equipment, a broad comprehension is necessary of matters relating to fuel preparation, fuel combustion, and the heat transfer and performance of superheaters, economizers, and air heaters, appearing elsewhere.

Boiler-Water Concentration

The maximum allowable boiler-water concentration (total solids in boiler water, ABMA recommended values) in relation to the pressure at the outlet of a steam-generating unit (applicable to the boiler types in the following descriptive outlines) is given below:

Maximum Allowable Boiler-Water Concentration

Pressure at Outlet of Steam-Generating Unit, psi	Total Solids, Parts per Million, ppm
0 to 300	3500
301 to 450	3000
451 to 600	2500
601 to 750	2000
751 to 900	1500
901 to 1000	1250
1001 & higher*	1000

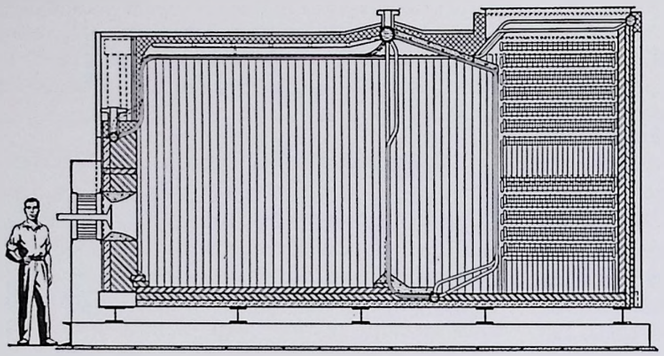
*Ed. Note: The industry, as yet, has not established final standards for supercritical operation.

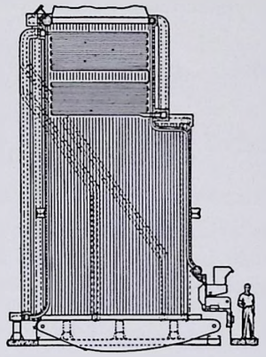
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Draft Loss in Various Boiler Types

Draft loss through a boiler unit depends upon many factors. Among these are: type of fuel, excess air, manner of gas flow in gas passes, and arrangement of convection surface of boiler, superheater, economizer, and air heater. In each case, the draft loss to be expected must be calculated for the final arrangement of the complete unit. Hence, specific draft loss data cannot be given in the following descriptive outlines of boiler types. The general observation, however, may be made that with the usual operating conditions the smaller boilers have a draft loss in the range from less than 1.0 to about 4.0 in. of water, depending upon the type and size. The larger higher-duty units operate with higher drafts. Approximate values are given in some instances.

High-Temperature-Water (HTW) Generator





General Description

Bent-tube unit, all-welded construction. Standard unit (left above) shop assembled, steel cased, pressure fired with oil or gas. Higher-capacity unit (right above) field assembled, steel cased, suction furnace for stoker firing.

Range in Size, Btu Absorption
Shop assembled, 10 million to 50 million Btu per hr; field assembled, 50 million to 200 million Btu per hr and higher

Design Pressure
Lb per sq in., 450

Operating Pressure
Lb per sq in., to 410

Water Discharge Temperature
300 to 450 F

Fuel
Oil and gas for pressure firing; coal with all types of stokers for suction firing

Operational Control
Manual, or automatic in varying degree; safety interlocks

Furnace
Water cooled; pressure type for oil and gas, suction for stoker firing

Dimensions Outside Setting (Std Unit), Approx
Smallest, 8 ft wide × 12 ft high × 14 ft front to rear
Largest, 11 ft wide × 12 ft high × 27 ft front to rear

Indicated Field of Application

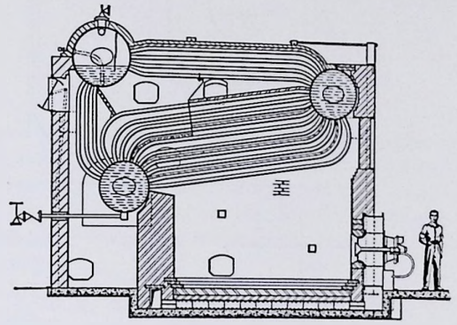
1. For high-temperature water heating or for process heating, where high economy or close temperature control are of prime importance.
2. Where low-pressure steam and domestic hot water are required for local use at any desired location in a system, generated in suitable heat exchangers.
3. Where high heat storage is desirable to provide continuity and uniformity of temperature control during peak loads, abnormal operation, and during long periods of banked loads, as for overnight or week-end shutdowns of plant operations.

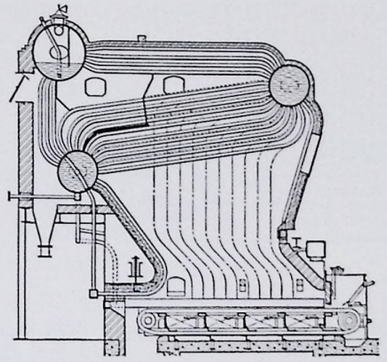
General Comments

The high-temperature-water generator is not a boiler, since no steam is formed other than the small amount initially required to pressurize the system, unless nitrogen is used for pressurization, in which event no steam whatsoever is formed. The HTW generator is actually a direct-fired forced-circulation water heater designed for low pressure drop, enabling operation with the system circulation pumps.

Units fired by gas or oil are readily converted to stoker firing, and vice versa, without major alterations. Foundation requirements are simple, a concrete slab sufficing for oil or gas firing and the normal provision for ash removal with stoker firing. Relocation can be accomplished without dismantling.

Type H Stirling Boiler





General Description

Bent-tube boiler unit, generally field assembled. Some small sizes can be shop assembled and shipped. Usually without superheater, though superheater may be fitted.

Range in Size, Steam Output
Lb per hr, to 50,000

Design Pressure
Lb per sq in., 160 to 300 (special above 300)

Operating Pressure
Lb per sq in., 15 to safety-valve setting

Steam Temperature
Saturation to approximately 175 F superheat

Fuel
Oil and gas; coal with all types of stokers; bagasse and wood

Operational Control
Usually manual

Furnace
Refractory; frequently water cooled for oil or coal firing

Dimensions Outside Setting, Approximate
Smallest, 6 ft wide × 15 ft high × 15 ft front to rear
Largest, 18 ft wide × 23 ft high × 25 ft front to rear

Indicated Field of Application

1. In the production of steam for heating, process, or power needs within the output and specifications noted above, using any of the fuels listed.
2. Where utmost simplicity and minimum first cost are prime factors.
3. Where use of low-cost fuels does not justify an economizer or air heater. However, either or both may be installed if warranted by the evaluation of the saving in purchased fuel.
4. Where simple field erection methods may be followed.

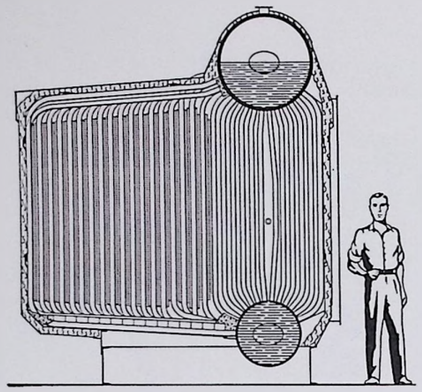
General Comments

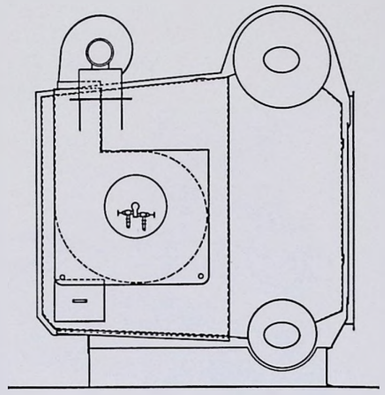
The type H Stirling boiler is essentially simple and inexpensive. It is particularly well suited for moderate-duty service and limited headroom. With the aid of the all-refractory furnace frequently used, good combustion efficiency can be maintained even at very low rates of output—often an important advantage when used for heating service.

The standard baffle for this unit is in checker form, which causes a tortuous flow of gas in the first pass, acts as a gas distributor, and materially increases the efficiency of the unit without appreciable effect on draft loss.

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Type FM Integral-Furnace Boiler





General Description

Standard steel-cased bent-tube unit, shop assembled and skid mounted. Forced-draft fan and burners integral with unit. Sometimes referred to as a "package boiler."

Range in Size, Steam Output
  Lb per hr, to 60,000 in increments of approximately 2000 lb per hr for smaller sizes and 5000 lb per hr for larger sizes
Design Pressure
  Lb per sq in., 250 for sat. service (standard) to 1050 with superheater (special)
Operating Pressure
  Lb per sq in., 15 to safety-valve setting
Steam Temperature
  Saturation to 700 F
Fuel
  Oil and gas, singly or in combination
Operational Control
  Steam pressure, feedwater flow, fuel, and air fully automatic; safety interlocks
Furnace
  Water cooled; pressure type; welded casing
Dimensions Outside Setting, Approximate
  Width and height, small sizes, 10 ft × 12½ ft, large sizes, 11 ft × 13½ ft
  Depth 7½ to 25 ft, depending upon size

Recommended Maximum Allowable Boiler-Water Concentration

(Usual drum arrangement, std baffles as required—otherwise ABMA rules apply)

Operating Pressure, psi	Total Solids, ppm
15 – 50	1250
50 – 250	3500

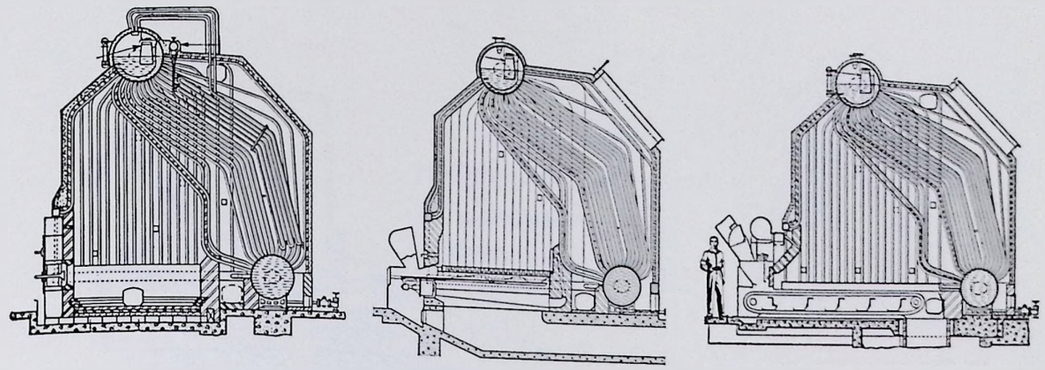
Indicated Field of Application

1. In the production of steam for heating, process, or power needs within the output and specifications noted above, using oil and gas fuel singly or in combination.
2. Where a boiler is needed quickly for either outdoor or housed location.
3. Where the boiler operators may be called upon for additional duties.
4. Where space is at a premium and a compact unit, including all auxiliaries, is needed.
5. Where removal of the boiler to another location may be contemplated.

General Comments

Since 1950 a very large number of type FM boilers, fired with gas or oil fuel, have been placed in service. This unit, completely assembled, can be shipped over any standard railroad in the United States except for the largest sizes, which are shipped with the fan unmounted and which may require special routing. It can be delivered promptly to meet sudden new or increased demands for steam. No special foundation is required, and for outdoor locations a stub stack is sufficient. For indoor locations, connection may be made to a regular stack.

Type FF Integral-Furnace Boiler



General Description

Standard steel-cased bent-tube unit, with water-cooled furnace and drum cyclones. Smallest size may be shop assembled. The unit is designed in a series of 5 different heights between drums, with provision for superheater. Fans are not integral.

Range in Size, Steam Output
  Lb per hr, to 50,000 in increments of 2000 lb per hr, or less
Design Pressure
  Lb per sq in., 160 to 600
Operating Pressure
  Lb per sq in., 15 to safety-valve setting
Steam Temperature
  Saturation to approximately 150 F superheat
Fuel
  Oil and gas; coal with all types of stokers
Operational Control
  Manual, or automatic in varying degree
Furnace
  Water cooled except floor; nonpressure type
Dimensions Outside Setting, Approximate
  Smallest, 7 ft wide × 13 ft high × 15 ft front to rear
  Largest, 17 ft wide × 23 ft high × 19 ft front to rear

Indicated Field of Application

1. In the production of steam for heating, process, or power needs within the output and specifications noted above, using any of the fuels listed.
2. Where a boiler is needed quickly for either outdoor or housed location.
3. Where space limitations require unusual proportions not met by type FM.
4. Where future conversion to an alternate fuel is contemplated—because of the simplicity of conversion with this unit.
5. Where considerable variation in fuel-ash characteristics is anticipated.

General Comments

The FF Integral-Furnace boiler is one of the well-standardized types widely used in industrial plants. With suitable firing methods, oil fuel varying in characteristics from bunker C to residual and bituminous coals of a wide range in per cent ash and fusion temperature may be burned successfully. Pulverized-coal firing is not used for this unit.

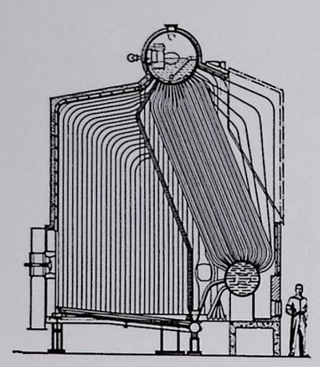
The range in vertical height of unit (a series of 5 standard dimensions between upper and lower drums), which in turn affects the other two dimensions, permits a size selection for comparable performance to suit many existing space conditions of headroom and floor area. The space conditions required for either spreader-stoker or chain-grate firing are comparable.

No major changes in the boiler unit are involved in converting from oil and gas to coal firing or the reverse. The "saturated" surface components are generally not affected. There are minor alterations to brickwork to suit the new firing equipment, and, if a superheater is present, it may be necessary to add or remove some surface to meet changed combustion requirements (chiefly excess air) of the new fuel.

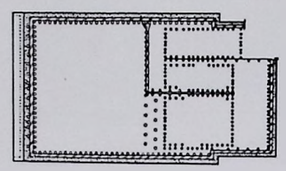
When fuel saving justifies the expenditure, economizers and air heaters may be added, as in most boiler types, to utilize low-level heat. However, of the more than 1500 type FF boilers placed in service in the period 1940-1960, less than 10 per cent were equipped with air heaters or economizers. The proportion of boilers equipped with these additional heat traps will tend to increase as fuel prices rise.

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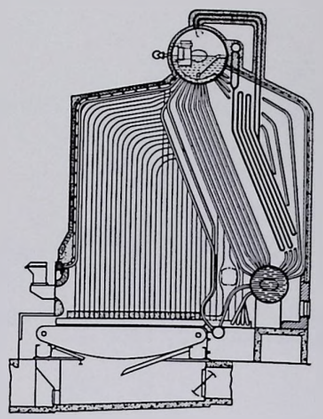
Type FP Integral-Furnace Boiler



SIDE VIEW THROUGH 3rd PASS, OIL & GAS FIRED



PLAN VIEW



SIDE VIEW THROUGH 1st PASS, SPREADER-STOKER FIRED

General Description

Standard welded-steel-cased bent-tube unit, with completely water-cooled furnace and drum cyclones. Assembled in the field. The unit is designed in one height (18-ft drum centers) and in 9 widths, with or without superheaters. Fans are not integral.

Range in Size, Steam Output
Lb per hr, to 80,000 in increments of approximately 5000 lb per hr
Design Pressure
Lb per sq in., 160 to 825
Operating Pressure
Lb per sq in., 15 to safety-valve setting
Steam Temperature
Saturation to approximately 850 F
Fuel
Coal with spreader, underfeed, or chain-grate stoker; oil and gas
Operational Control
Manual, or automatic in varying degree
Furnace
Completely water cooled; nonpressure type
Dimensions Outside Settings, Approximate
Smallest, 8 ft wide × 28 ft high × 22 ft front to rear
Largest, 17 ft wide × 28 ft high × 22 ft front to rear

Indicated Field of Application

1. In the production of steam for heating, process, or power needs within the output and specifications noted above, using any of the fuels listed.
2. Where a boiler is needed quickly for either outdoor or housed location.
3. Where coal ash is particularly abrasive, and stoker firing is preferable, in this size category, to pulverized-coal firing.
4. Where coal ash-fusion characteristics permit little or no uncooled furnace area.
5. Where stoker firing is to be supplemented by simultaneous oil and gas firing with auxiliary burners through the side walls.

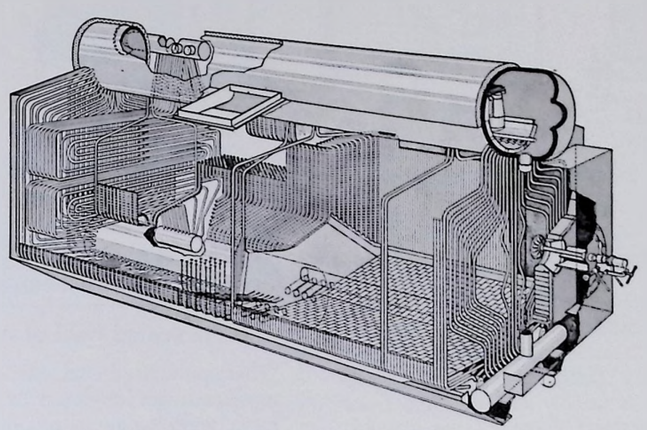
General Comments

The FP Integral-Furnace boiler is particularly well suited for stoker firing. With its completely water-cooled furnace, the rates of heat input and heat absorption may be somewhat higher than for the type FF boiler. Travel of the gas through the boiler has been arranged so that the abrasive particles are not concentrated in streams.

Oil and gas firing, either singly or in combination, is entirely suitable and may be used initially as an alternate to stoker firing. Permanent conversion from one fuel to another is as simple as for the type FF boiler. Installation of oil and gas burners in the side walls of a stoker-fired unit permits combination firing, either simultaneously with coal or separately. When firing oil and gas alone, the stoker must be protected with a suitable tile or ash coating. The unit is not designed for pulverized-coal firing. If the fuel saving justifies the expenditure, economizers or air heaters should be installed.

The type FP boiler is well suited for the production of steam for both industrial and power requirements. A highly trained operating staff is not required for its successful operation.

Type FO Integral-Furnace Boiler



General Description

Standard double-steel-cased bent-tube package boiler, with water-cooled furnace and drum cyclones. Shop assembled and skid mounted. Burners are integral, and forced-draft fan is floor mounted.

Range in Size, Steam Output
Lb per hr, 60,000 to 100,000 in increments of approximately 8,000 lb per hr
Design Pressure
Lb per sq in., 250 to 775
Operating Pressure
Lb per sq in., 15 to safety-valve setting
Steam Temperature
Saturation to approximately 750 F
Fuel
Oil and gas, singly or in combination
Operational Control
Steam pressure, feedwater flow, fuel, and air fully automatic; safety interlocks
Furnace
Water cooled; pressure type; inner and outer welded casing
Dimensions Outside Setting, Approximate
Width and height, all sizes, 12½ ft × 15½ ft
Depth 16 to 27 ft, depending upon size
Recommended Maximum Allowable Boiler-Water Concentration
In accordance with ABMA rules

Indicated Field of Application

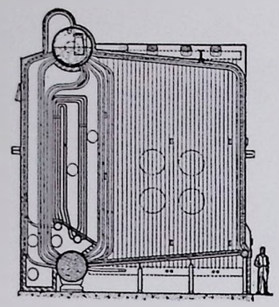
1. In the production of steam for heating, process, or power needs within the output and specifications noted above, using oil and gas fuel singly or in combination.
2. Where a boiler is needed quickly for either outdoor or housed location.
3. Where the boiler operator may be called upon for additional duties.
4. Where space is at a premium and a compact unit, including all auxiliaries, is needed.
5. Where removal of the boiler to another location may be contemplated.

General Comments

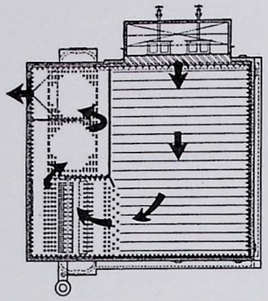
The FO Integral-Furnace boiler is designed specifically for pressure firing of fluid fuels, based on the many years of experience accumulated with the FM boiler. The larger capacities and the higher pressures and temperatures for which the FO boiler is designed dictated the development of an entirely new physical arrangement. No special foundation is required, and for outdoor locations a stub stack is sufficient. For indoor locations, connection may be made to a regular stack. The forced-draft fan can be located at any convenient place. When fuel saving justifies the expenditure, an air heater may be added, as in most boiler types, to utilize low-level heat.

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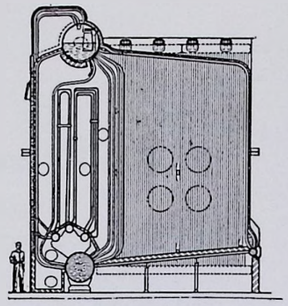
Type FH Integral-Furnace Boiler



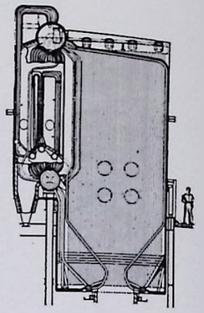
SECTIONAL SIDE VIEW, OIL AND GAS FIRED



PLAN CROSS SECTION SHOWING HORIZONTAL GAS FLOW PATH



SECTIONAL SIDE VIEW, PULVERIZED-COAL FIRED FLAT FLOOR FOR ASH RAKE-OUT



SECTIONAL SIDE VIEW, PULVERIZED-COAL FIRED WITH ASH HOPPER

General Description

Standard steel-cased bent-tube unit, with completely water-cooled furnace and drum cyclones. Assembled in the field. The unit is designed in 4 heights and in several depths (front to back) for each height. An air heater is usually specified, and there may also be an economizer. Fans are not integral. The gas flow is horizontal, similar to types FM and FP. See Fig. 6, Chapter 21, for cutaway view showing the principle of horizontal gas flow.

Range in Size, Steam Output
Lb per hr, 50,000 to 350,000 in small increments

Design Pressure
Lb per sq in., 160 to 1050

Operating Pressure
Lb per sq in., 15 to safety-valve setting

Steam Temperature
Saturation to approximately 910 F

Fuel
Pulverized coal; oil and gas, in combination or singly

Operational Control
Manual to complete automatic combustion and feedwater regulation

Furnace
Water cooled; pressure or nonpressure type

Dimensions Outside Setting, Approximate

Flat Floor

Smallest, 22 ft wide × 25 ft high × 14 ft front to rear

Largest, 39 ft wide × 39 ft high × 40 ft front to rear

Hopper Bottom

Smallest, 23 ft wide × 39 ft high × 16 ft front to rear

Largest, 39 ft wide × 66 ft high × 40 ft front to rear

Indicated Field of Application

1. Where pulverized coal, gas, or oil is to be used to produce steam for power or process requirements for a wide range in amount and conditions of steam.
2. For either outdoor or housed location.
3. Where conditions of space and economics require a compact unit to operate at high efficiency comparable to the larger specially built boilers.

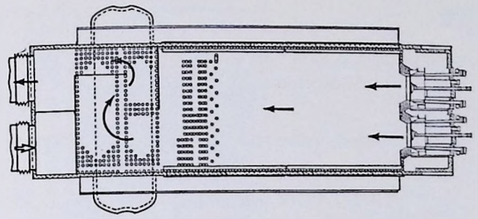
General Comments

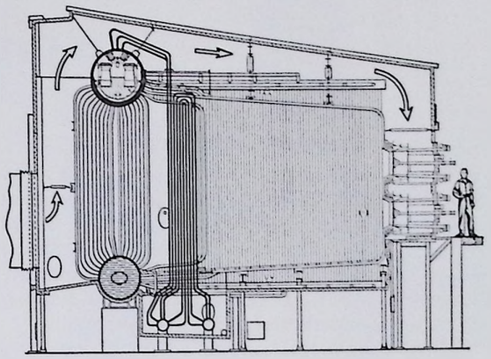
The FH Integral-Furnace boiler is particularly well suited for pulverized-coal firing and is widely used for maximum outputs from 50,000 to 350,000 lb of steam per hr. With its furnace construction of touching tubes, except for refractory areas around the burners, it is designed for moderately high-duty service. Heat liberation rates may be correspondingly high per sq ft of furnace area per hr for a coal ash-softening temperature of 2600 F, oxidizing basis (see Chapter 18 for a discussion of ash-fusion temperatures), with rates reduced for lower ash-softening temperatures. The allowable heat liberation rates are generally lower for the flat-floor than for the hopper-bottom arrangement.

Standard superheaters, both inverted and pendent types, are available to suit the specific application. Where constant steam temperature is required over a considerable load range, an attemperator is installed in the lower drum (see Chapter 13). Air heaters are invariably necessary to provide hot air for drying the coal and as an aid in combustion, and they are also usually justified by the fuel saving. With high fuel costs, economizers are also sometimes justified when the saving warrants the added investment.

In the flat-floor arrangement, normally suited only for outputs up to 100,000 lb of steam per hr, the ash is periodically hand-raked through doors in the wall opposite the burners and is removed by a transport system. For outputs above 100,000 lb of steam per hr or for low ash initial-deformation temperatures (2300 F, oxidizing basis), the hopper arrangement is better. With this design the ash is collected in a storage hopper below the furnace hopper and is periodically transported by pneumatic or hydraulic means to silos, pits, or final disposal areas.

Type PFI Integral-Furnace Boiler





General Description

Standard bent-tube unit arranged with gastight furnace and boiler for pressurized operation, with completely water-cooled furnace and drum cyclones. Maximum shop subassembly facilitates field erection. The unit is designed in 3 heights and in several widths for each height, with provision for a drainable-type superheater. Forced-draft fan is not integral. The gas flow is horizontal, similar to that of type FH.

Range in Size, Steam Output
Lb per hr, 80,000 to 450,000 in spaced increments

Design Pressure
Lb per sq in., 160 to 1150

Operating Pressure
Lb per sq in., 15 to safety-valve setting

Steam Temperature
Saturation to approximately 910 F

Fuel
Oil and gas, in combination or singly

Operational Control
Manual to complete automatic combustion and feedwater regulation

Furnace
Membrane-wall construction; pressure type

Dimensions Outside Setting, Approximate
Smallest, 11 ft wide × 30 ft high × 27 ft front to rear

Largest, 29 ft wide × 41 ft high × 35 ft front to rear

Indicated Field of Application

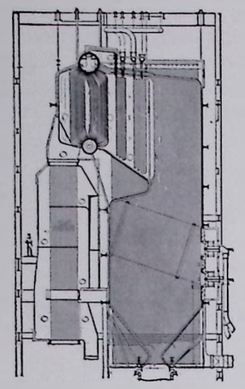
1. In the production of steam for heating, power, or process, within the output and specifications noted above with oil or gas firing, or both.
2. For either outdoor or housed locations; bottom supports for minimum space requirements; reinforced concrete or steel foundations.
3. Where economic conditions require high efficiency, sustained operation, and a minimum of attention and maintenance.

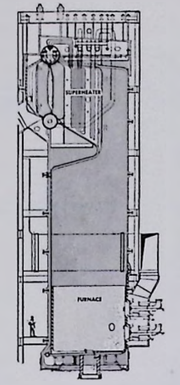
General Comments

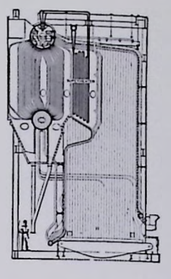
The PFI Integral-Furnace boiler is particularly indicated in areas where oil and gas fuels are commonly used. The completely water-cooled furnace provides a gastight unit suitable for pressure operation with a forced-draft fan only, simplifying control and operation and reducing maintenance. The unit is specifically designed to operate at high ratings for maximum steam output within a minimum of available space. Superheaters are completely drainable and arranged with wide tube spacing to provide a relatively constant steam temperature over a wide load range. Air heaters are normally provided and are justified by fuel saving.

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Stirling Two-Drum Boiler







General Description

Bent-tube boiler, partly standard, usually steel cased, with one 42-in. to 72-in. upper drum vertically above one 24-in. to 54-in. lower drum. The steam drum is fitted with cyclone separators. The unit is field assembled and is top supported, except in rare instances. It may be built with or without a superheater, but a superheater is usually required.

The boiler generating tubes may be 2 in., 2½ in., or 3 in. and are frequently swaged at ends to reduce the required thickness of drum plate. Tubes of 2½ in. swaged to 2 in. at ends and sidespaced at $3\frac{9}{16}$ in. on centers are a typical arrangement. The water-cooled furnace has 2-in., 2½-in., or 3-in. tubes spaced as required, usually touching.

When the ash is abrasive and for conditions requiring low draft loss, the gas flows from front to rear in a single pass. For oil or gas fuel, a multi-pass arrangement is common. Furnace width varies in 1-ft increments. There are several standard dimensions of furnace depth and between upper and lower drum centers.

Range in Size, Steam Output
: Lb per hr, to 1,000,000 (and higher, special) in small increments

Design Pressure
: Lb per sq in., 160 to 1050 (and higher, special)

Operating Pressure
: Lb per sq in., 15 to safety-valve setting

Steam Temperature
: Saturation to 1000 F

Draft Loss at Maximum Output
: In. of water, from 1 to 8 or more, depending upon size and conditions

Fuel
: Coal, Cyclone Furnace fired or pulverized, or with all types of stokers; oil and gas, singly or in combination, as regular or supplemental fuel; also waste-heat sources

Operational Control
: Manual, or automatic in varying degree

Furnace
: Water cooled; pressure or nonpressure type

Dimensions Outside Setting, Approximate
: Vary widely, from smallest, 6 ft wide × 8 ft high × 12 ft front to rear with oil or gas firing, to 40 ft wide for a 1,000,000 lb of steam per hr unit for pulverized coal, with other dimensions correspondingly large

Indicated Field of Application

1. In the production of steam for heating, process, or power needs within the output and specifications noted above, using any of the fuels listed.
2. Where coal ash is particularly abrasive with stoker and sometimes pulverized-coal firing.
3. Where fuel ash-fusion characteristics require a water-cooled furnace.
4. Where simplicity of change to accommodate another fuel and method of firing is essential because of a possible future fuel source.

General Comments

The range in steam output of the Stirling Two-Drum boiler is greater than for any of the other units considered in this series. Because of its range in output and its ready adaptability for burning most fuels, this unit is now widely used. It is particularly suitable for steam production with spreader-stoker firing up to the limit of output possible in using a stoker, in the order of 400,000 lb of steam per hr. In this application, the single-pass gas flow (no turns) helps to prevent concentration of abrasive particles and consequent tube damage.

All tubes can be entered from the two drums for inspection and cleaning.

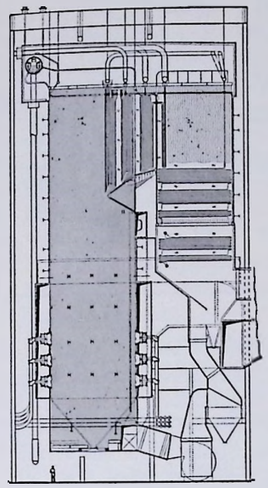
Where the additional cost is warranted to maintain a constant steam temperature at reduced loads, the superheater is designed for control of temperature by an attemperator of either the submerged-surface type installed in the lower drum or the spray type.

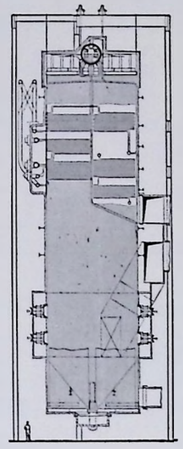
The water-cooled tubes in the side, front, and rear walls of the furnace are variously spaced, as required by the fuel characteristics, on from 6-in. centers to touching.

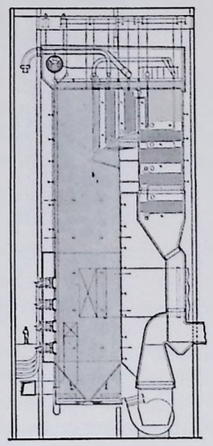
Hopper bottoms are usually fitted for pulverized-coal firing. A somewhat lesser height is required for Cyclone Furnace firing and slag-tap ash removal than for pulverized-coal firing and dry-ash removal. In the larger units, particularly with Cyclone Furnace or pulverized-coal firing, additional furnace surface is frequently provided by installing an intermediate or furnace division wall, as in the RB type of boiler (see below).

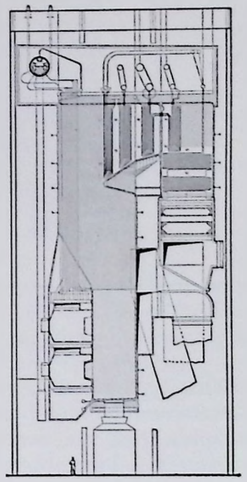
An economizer or an air heater, or both, may be installed if justified by an evaluation of the fuel saving and performance.

Radiant (RB) Boiler









General Description and Comments

The Radiant boiler derives its name from the fact that the heat absorption of the saturated surface is largely by radiant energy transfer. With its considerable range in output, it is used in the larger industrial plants and in utility stations. The various components are pre-engineered with enough flexibility to adapt the design to gradations of the various fuels and to the steam conditions selected by the user. The details of construction are not subject to change without considerable re-engineering.

This type of boiler generally finds application at subcritical superheater outlet pressures above 1200 psi, at steam temperatures above 950 F (both with and without reheat), and at a very wide range of capacities. The usual fuels are coal, oil, and gas, burned singly or in combination. The furnace is completely water cooled and may be designed for either dry- or wet-ash removal. One or more division walls may be used, depending upon the functional and economic requirements of the design. Superheater and reheater surfaces are pendent and/or horizontal.

Steam-temperature control is by gas recirculation. Gas tempering is used to control gas temperatures entering the superheater. Economizers and air heaters are required to obtain an efficiency commensurate with fuel costs. The physical size of the unit dictates field assembly, although many component parts are shop assembled.

Range in Size, Steam Output
: Lb per hr, about 300,000 to an undetermined maximum exceeding 10,000,000

Operating Pressure
: Subcritical, usually above 1200 psi

Steam Temperature
: As required, usually above 950 F

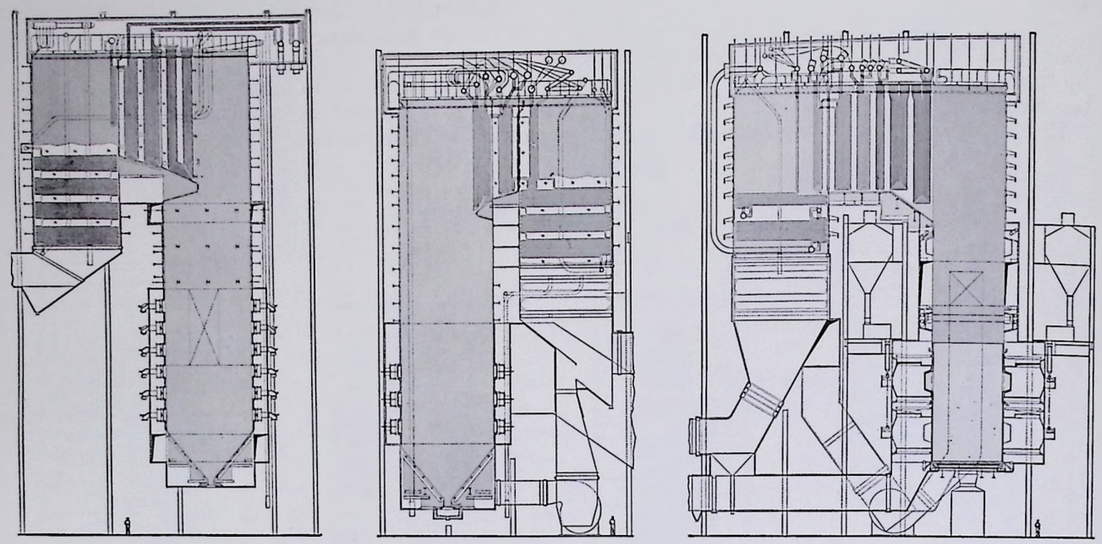
Fuel
: Coal, Cyclone Furnace fired or pulverized; oil; gas

Operational Control
: Usually completely automatic, including feedwater flow, combustion, and steam temperature

Furnace
: Water cooled; pressure or nonpressure type

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Universal-Pressure (UP) Boiler



General Description

This type of unit, variously known as the Universal-Pressure boiler and the Once-Through boiler, is applicable, functionally, to all commercial temperatures and pressures, subcritical and supercritical. Economically it is applicable to the larger sizes, to the higher subcritical pressure region, and to the supercritical pressure region.

The principle of operation is that of the "Once-Through" or "Benson" cycle. The working fluid is pumped into the unit as liquid, passes sequentially through all the pressure-part heating surface, where it is converted to steam as it absorbs heat, and leaves as steam at the desired temperature. There is no recirculation of water within the unit, and, for this reason, a drum is not required to separate water from steam.

Firing may be by oil, gas, or coal, the last either in pulverized form or crushed for Cyclone Furnace firing. The furnace is completely fluid cooled. It may be designed for suction or pressure operation. Ash removal may be either dry or wet. Furnace division walls are used in the larger units to effect a reduction in building volume, building steel, and platforms. Reheaters may be incorporated in the design for the reheat cycle.

Steam temperature is controlled by the fuel firing rate. Constant steam temperature can be maintained to one-third load with any fuel. Feed-pump speed and turbine throttle are used to control steam flow and steam pressure. The complete system of control is extremely simple and economical.

Indicated Field of Application

Same general field as Radiant boiler with qualifications as indicated in first paragraph of *General Description,* above.

Evaluation of Equipment Performance and Characteristics

Costs related to the performance and characteristics of equipment should be of continuing concern to the engineer and user. Equipment at an attractive initial cost, with high operating costs caused by low efficiency or other reasons, is frequently expensive over the period of its life. However, an increase in initial outlay can be justified in the long run only if the value of the improved performance is worth the extra expenditure. As a measure of justifiable expenditures, monetary values may be calculated for the performance and other characteristics of the equipment. These values are the bases for *evaluations* or cost comparisons.

Principal items of performance and other factors usually considered for evaluation in the selection of boiler units and related equipment are:

1. Efficiency.
2. Draft loss.
3. Air resistance.
4. Power required for fuel preparation (example, coal pulverizers).
5. Building space.
6. Pressure drop, superheater and economizer.
7. Maintenance (sometimes).

At the current cost of fuel, very approximate evaluations in capitalized costs for an improved performance in the order of 0.10 per cent change in efficiency, 1 in. of draft loss, 2 in. of air resistance, or 1 kwhr per ton of coal for pulverizing would justify a capital outlay, in each case, of 1.0 per cent of the cost of a complete boiler unit. On the basis of this rough approximation, an initial expenditure in capital cost of *up to* 1.0 per cent is justified, in each case, for improved performance of the order indicated in the first four items listed above. Building space may be similarly evaluated in the general range of from $1.00 to $2.00 per cu ft.

Superheater and economizer pressure drops should be considered on the basis of power required to operate feed pumps. The rough evaluation frequently made of this item (which includes a demand charge) is approximately $1.50 per 1000 lb steam per hr of unit output, per 1 psi pressure drop.

More precise calculations for the evaluation of efficiency and auxiliary power are demonstrated in the examples below.

Data Needed for Evaluation

Capitalization Rate

Capital evaluation of performance (i.e., what expenditure is justified to avoid a penalty or to obtain an advantage) is based on an acceptable capitalization rate set by the user's management. This rate, for instance, might vary from a low of 10% to a high of 30%.

To make these capital evaluations for various conditions of operation, it is necessary to know:

1. Fuel Cost per suitable unit, ¢/million Btu (MkB).
2. Heat Rate per kwhr, Btu/kwhr generated.
3. Purchased Power Cost, ¢/kwhr.
4. Demand Charge, usually $/installed kw.

Fuel Cost

Fuel cost is the cost of fuel delivered to point of use. This cost is used in evaluations of efficiency. It is also used in evaluations of power for the auxiliaries, when the power is produced within the plant.

Heat Rate

The heat rate of the plant is the fuel heat input required to generate a kwhr. It is used in evaluations of power for the auxiliaries, when the power is produced within the plant. When back-pressure turbines are used where the quantity of low-pressure steam for heating and processing is well above that required to generate the power for the auxiliaries, or when turbine-driven auxiliaries are used to provide exhaust steam, the thermal cycle is very efficient, the heat rate is low (4000-5000 Btu per kwhr, see Chapter 10), and the cost of auxiliary power is correspondingly low.

Purchased Power Cost

Purchased power cost is the cost of power required to operate the auxiliaries when the power is bought from outside sources.

Demand Charge

Demand charge represents the capitalized cost of any additional capacity in boilers, turbines, and generators necessary to generate the power required for the auxiliaries. The demand charge is not always used, but in many cases, especially in public utilities, its application is common practice. Where two or more different types of firing are used, the demand charge is based on the firing with the largest ratio of gas weight to heat input, since the maximum installed kilowatts to provide the greater fan power are required for this condition.

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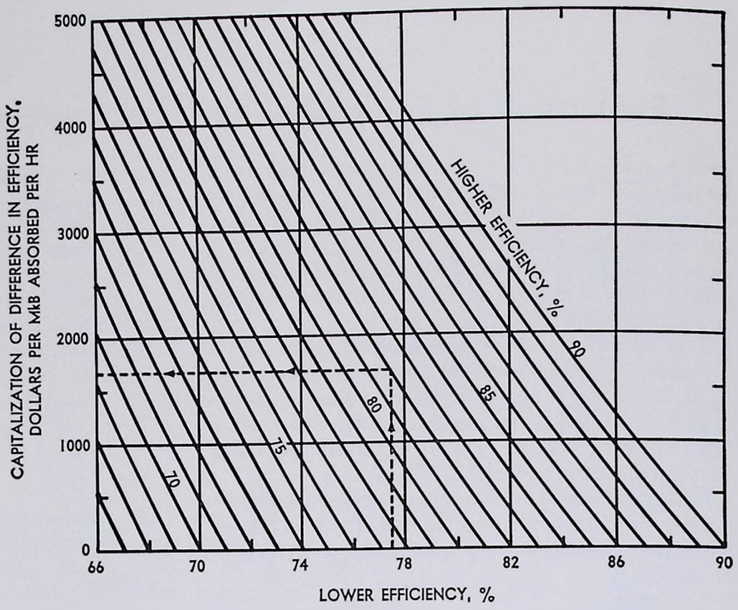


Fig. 1. *Capitalization in dollars per million Btu (MkB) absorbed per hr for changes in boiler efficiency. Base conditions: 8000 hr per year; fuel cost, 30¢ per MkB; capitalization rate, 10% per year. Use equation (1) in text for conditions other than base*

Efficiency and Fan-Power Evaluation

Capital evaluation curves for the unit and for fan power, over a range in efficiency at average present-day high-load-factor and three-shift operation with expensive coal or oil fuels, are presented in Figs. 1 and 2, respectively. The basic conditions applying to these curves are noted below in each case. For all other conditions, the evaluation of unit efficiency and fan power is established by use of the pertinent equations listed.

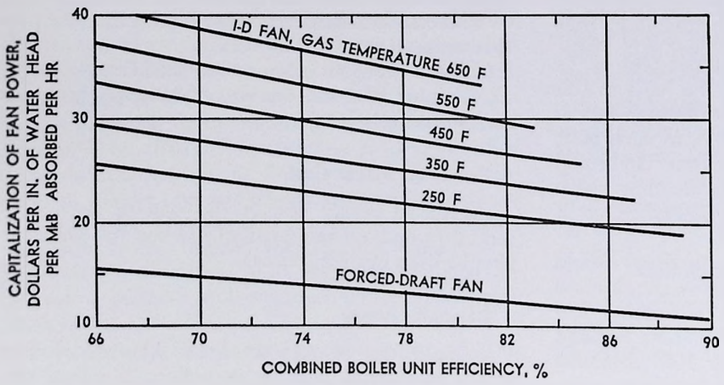


Fig. 2. *Fan-power capitalization in dollars per in. of water head per million Btu (MkB) absorbed per hr by boiler unit, for different gas temperatures and values of boiler unit efficiency. Base conditions are given in text preceding equation (2). Use equations (2) to (7) for conditions other than base*

Capitalization of Boiler Unit Efficiency

For Base Conditions (Fig. 1)
(1 MkB = 1,000,000 Btu)

Factors	Base
N = heat absorbed in MkB per hr (avg)	1
T = operating time, hr per yr	8,000
C = fuel cost, ¢ per MkB	30
Y = capitalization rate, % per yr	10

Capitalization of Boiler Unit Efficiency in Dollars

For Conditions Other than Base

$$\text{Capitalization} = \text{Value from Fig. 1} \times \frac{N}{1} \times \frac{T}{8{,}000} \times \frac{C}{30} \times \frac{10}{Y} \qquad (1)$$

Capitalization of Fan Power

For Base Conditions (Fig. 2)
(1 MkB = 1,000,000 Btu)

Factors	Base
X = I-D fan head, in. of water at 1,000,000 Btu absorbed per hr	1
V = F-D fan head, in. of water at 1,000,000 Btu absorbed per hr	1
G = gas weight, lb per 10,000 Btu input	10
A = air weight, lb per 10,000 Btu input	8
N = heat absorbed in MkB per hr (avg)	1
M = heat absorbed in MkB per hr (peak)	1
B = barometric pressure, in. Hg	30
T = hr per yr at N input	8,000
R = heat rate, Btu per kwhr	12,000
C = fuel cost, ¢ per MkB	30
Y = capitalization rate, % per yr	10
P = power cost, ¢ per kwhr	0.36
D = demand charge, $ per installed kw	150

Gas and Air Weights
(lb per 10,000 Btu input)

Firing	G	A
Natural gas	8.8	7.5
Fuel oil	9.1	7.8
Cyclone	9.4	7.9
Pulverized coal	10.1	8.0
Stokers	11.3	9.0
Blast-furnace gas	16.0	6.6

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Capitalization of Fan Power in Dollars

For Conditions Other than Base

(Base Values from Fig. 2)

When fuel cost applies:

$$= \text{I-D fan base value} \times X \times \frac{M}{1} \times \frac{G}{10} \times \frac{30}{B} \times \frac{T}{8{,}000} \times \frac{R}{12{,}000} \times \frac{C}{30} \times \frac{10}{Y} \quad (2)$$

$$= \text{F-D fan base value} \times V \times \frac{M}{1} \times \frac{A}{8} \times \frac{30}{B} \times \frac{T}{8{,}000} \times \frac{R}{12{,}000} \times \frac{C}{30} \times \frac{10}{Y} \quad (3)$$

When power cost applies:

$$= \text{I-D fan base value} \times X \times \frac{M}{1} \times \frac{G}{10} \times \frac{30}{B} \times \frac{T}{8{,}000} \times \frac{P}{0.36} \times \frac{10}{Y} \quad (4)$$

$$= \text{F-D fan base value} \times V \times \frac{M}{1} \times \frac{A}{8} \times \frac{30}{B} \times \frac{T}{8{,}000} \times \frac{P}{0.36} \times \frac{10}{Y} \quad (5)$$

When demand charge applies: (add to either of above alternates)

$$= \frac{\text{I-D fan base value}}{1.3} \times X \times \frac{M}{1} \times \frac{G}{10} \times \frac{30}{B} \times \frac{D}{150} \times \left\{\frac{M}{N}\right\}^a \quad (6)$$

$$= \frac{\text{F-D fan base value}}{1.3} \times V \times \frac{M}{1} \times \frac{A}{8} \times \frac{30}{B} \times \frac{D}{150} \times \left\{\frac{M}{N}\right\}^a \quad (7)$$

Feed-Pump Power Evaluation

Formulas for capital evaluation of feed-pump power required by superheater and economizer pressure drop are given below for the following conditions:

Capitalization of Feed-Pump Power

Factors	Base
FW_{avg} = lb fluid pumped per hr (avg)	1,000
FW_{peak} = lb fluid pumped per hr (peak)	1,000
$\triangle P$ = superheater or economizer pressure drop, psi	1
T = hr per yr at avg MkB per hr input	8,000
R = heat rate, Btu per kwhr	12,000
C = fuel cost, ¢ per MkB	30
Y = capitalization rate, % per yr	10
P = power cost, ¢ per kwhr	0.36
D = demand charge, $ per installed kw	150

Capitalization of Feed-Pump Power in Dollars

When fuel cost applies:

$$= 0.5 \times \triangle P \times \frac{FW_{peak}}{1{,}000} \times \frac{T}{8{,}000} \times \frac{R}{12{,}000} \times \frac{C}{30} \times \frac{10}{Y} \quad (8)$$

When power cost applies:

$$= 0.5 \times \triangle P \times \frac{FW_{peak}}{1{,}000} \times \frac{T}{8{,}000} \times \frac{P}{0.36} \times \frac{10}{Y} \quad (9)$$

When demand charge applies:

$$= 0.4 \times \triangle P \times \frac{FW_{peak}}{1{,}000} \times \frac{D}{150} \times \left\{\frac{FW_{peak}}{FW_{avg}}\right\}^a \quad (10)$$

Calculation of Evaluations for Capitalization

EXAMPLE No. 1

Evaluation for Two Boiler Units

Fuel cost, natural gas, oil stand-by, ¢ per MkB	12
Capitalization rate, % per yr	10
Demand charge, $ per installed kw	150
Heat rate, Btu per kwhr	12,500
Building space evaluation, $ per cu ft	2
Average steam flow, lb per hr	500,000
Peak steam flow, lb per hr	700,000
Operating time at avg steam flow, hr per yr	8,000
Barometric pressure, in. Hg	30
Heat absorbed by steam, Btu per lb	1,121
Total heat absorbed at avg rate, MkB per hr	560.5
Total heat absorbed at peak rate, MkB per hr	785.0

	Unit No. 1 (Base)	Unit No. 2 (Alternate)
Boiler efficiency at avg rate, gas fired, %	82.0	81.5
Draft loss at avg steam flow, gas fired, in. of water	4.5	5.5
Air resistance at avg steam flow, gas fired, in. of water	4.7	5.6
Draft loss, at peak steam flow, oil fired, in. of water	9.5	11.5
Air resistance at peak steam flow, oil fired, in. of water	10.0	11.9
Boiler exit gas temperature, F	320	340
Excess air leaving boiler, gas fired, %	17	17
Excess air leaving boiler, oil fired, %	17	17
Stated price of unit, $	900,000 (debit)	800,000 (debit)
Capitalization, in $, for:		
Difference in efficiency (0.5%)	Base	39,240
Fuel cost for I-D fan	Base	7,400 (debit)
Fuel cost for F-D fan	Base	3,360
Demand charge for I-D fan	Base	55,700
Demand charge for F-D fan	Base	27,300
Difference in building volume	7,000 (debit)	Base
Evaluated price (net)	$907,000 (debit)	$933,000 (debit)

In this example, under the conditions established for fuel cost, capitalization rate, demand charge, heat rate, and building space evaluation, Unit No. 2, of lower first cost, is actually the more expensive on an evaluated basis.

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Detailed Evaluation of Items Capitalized in Example No. 1

Difference in efficiency:

Value from Fig. 1 for gain of 1% in eff (from 81 to 82%) = \$350/N

Value for loss of 0.5% in eff (from 82 to 81.5%) = \$175/N

To correct for conditions of Example No. 1, use equation (1)

$$\text{Capitalization for difference in eff} = 175 \times 560.5 \times \frac{12}{30} = \$39{,}240$$

Fuel cost for operating fans:

Value from Fig. 2 for I-D fan at 320 F & 82% boiler eff = \$23.0

Value from Fig. 2 for I-D fan at 340 F & 81.5% boiler eff = \$23.5

To correct for conditions of Example No. 1, use equation (2)

$$\text{For I-D fan, Unit No. 1} = 23.0 \times 4.5 \times 785.0 \times \frac{8.8}{10} \times \frac{12{,}500}{12{,}000} \times \frac{12}{30} = \$29{,}800$$

$$\text{For I-D fan, Unit No. 2} = 23.5 \times 5.5 \times 785.0 \times \frac{8.8}{10} \times \frac{12{,}500}{12{,}000} \times \frac{12}{30} = \$37{,}200$$

$$\text{Additional capitalization required for Unit No. 2} = \$7{,}400$$

Value from Fig. 2 for F-D fan at 81.5 or 82% boiler eff = \$12.5

To correct for conditions of Example No. 1, use equation (3)

$$\text{For F-D fan, Unit No. 1} = 12.5 \times 4.7 \times 785.0 \times \frac{7.5}{8} \times \frac{12{,}500}{12{,}000} \times \frac{12}{30} = \$18{,}100$$

$$\text{For F-D fan, Unit No. 2} = 12.5 \times 5.6 \times 785.0 \times \frac{7.5}{8} \times \frac{12{,}500}{12{,}000} \times \frac{12}{30} = \$21{,}460$$

$$\text{Additional capitalization required for Unit No. 2} = \$3{,}360$$

Demand charges for fans: (Based on oil firing)

Use Fig. 2, as before, and equations (6) & (7) to establish values

$$\text{For I-D fan, Unit No. 1} = \frac{23.0}{1.3} \times 9.5 \times 785.0 \times \frac{9.1}{10} \times \frac{150}{150} \times \left\{\frac{785.0}{560.5}\right\}^2 = \$235{,}300$$

$$\text{For I-D fan, Unit No. 2} = \frac{23.5}{1.3} \times 11.5 \times 785.0 \times \frac{9.1}{10} \times \frac{150}{150} \times \left\{\frac{785.0}{560.5}\right\}^2 = \$291{,}000$$

$$\text{Additional capitalization required for Unit No. 2} = \$55{,}700$$

$$\text{For F-D fan, Unit No. 1} = \frac{12.5}{1.3} \times 10.0 \times 785.0 \times \frac{7.8}{8} \times \frac{150}{150} \times \left\{\frac{785.0}{560.5}\right\}^2 = \$144{,}300$$

$$\text{For F-D fan, Unit No. 2} = \frac{12.5}{1.3} \times 11.9 \times 785.0 \times \frac{7.8}{8} \times \frac{150}{150} \times \left\{\frac{785.0}{560.5}\right\}^2 = \$171{,}600$$

$$\text{Additional capitalization required for Unit No. 2} = \$27{,}300$$

Example No. 2

Evaluation of a Pulverized-Coal-Fired Unit—Arrangements A, B, C

(Base and Two Alternates)

Fuel cost, bituminous coal, ¢ per MkB	34.6
Capitalization rate, % per yr	18
Demand charge, \$ per installed kw	160
Heat rate, Btu per kwhr	13,000
Average steam flow, lb per hr	100,000
Peak steam flow, lb per hr	125,000
Operating time at avg steam flow, hr per yr	7,500
Barometric pressure, in. Hg	27.0
Heat absorbed by steam, Btu per lb	1,200
Total heat absorbed at avg rate, MkB per hr	120.0
Total heat absorbed at peak rate, MkB per hr	150.0

	A Base No Air Heater or Economizer	B Alternate Air Heater only	C Alternate Air Heater and Economizer
Boiler efficiency at avg rate, %	77.5	82.0	83.5
Draft loss at avg steam flow, in. of water	6.8	9.6	10.0
Air resistance at avg steam flow, in. of water	5.0	7.3	7.3
Boiler exit gas temperature, F	550	400	350
Economizer pressure drop, psi	—	—	10.0
Capitalization in \$ for:			
Increase in efficiency	Base	131,190 (credit)	173,360 (credit)
Fuel cost, I-D fan	Base	2,900	1,900
Fuel cost, F-D fan	Base	2,740 (debit)	2,500
Demand charge, I-D fan	Base	5,700	3,800 (debit)
Demand charge, F-D fan	Base	5,400	4,930
Fuel cost, feed pump	Base	Base	406
Demand charge, feed pump	Base	Base	834
Evaluated worth (net)	Base	\$114,450	\$158,990

A clear understanding of the net results in the above example is important: With A at base cost, alternate B at \$114,450 additional cost over A, and alternate C at \$158,990 additional cost over A, all three arrangements offer investments of equal financial return, and no definite choice is indicated on such grounds alone.

However, the selection of an alternate over the base arrangement is economically justified if the increase in bid price, attributable to the additional equipment of the alternate, is less than its evaluated net worth: The greater the difference, the greater will be the return on the investment.

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Detailed Evaluation of Items Capitalized in Example No. 2

Increase in efficiency: from A to B = 77.5% to 82%, and A to C = 77.5% to 83.5%

Values from Fig. 1 for A to B = \$1,680/N, and for A to C = \$2,220/N

To correct for conditions of Example No. 2, use equation (1)

For A to B $= 1{,}680 \times \frac{7{,}500}{8{,}000} \times \frac{130}{1} \times \frac{34.6}{30.0} \times \frac{10}{18}$ = \$131,190

For A to C $= 2{,}220 \times \frac{7{,}500}{8{,}000} \times \frac{130}{1} \times \frac{34.6}{30.0} \times \frac{10}{18}$ = \$173,360

Fuel cost for operating fans:

Values from Fig. 2 for I-D fans: A = \$31.7; B = \$25.2; C = \$23.3

To correct for conditions of Example No. 2, use equation (2)

I-D fan for A $= 31.7 \times 6.8 \times 150 \times \frac{10.1}{10} \times \frac{30}{27} \times \frac{7{,}500}{8{,}000} \times \frac{13{,}000}{12{,}000} \times \frac{34.6}{30} \times \frac{10}{18}$ = \$23,600

I-D fan for B $= 25.2 \times 9.6 \times 150 \times \frac{10.1}{10} \times \frac{30}{27} \times \frac{7{,}500}{8{,}000} \times \frac{13{,}000}{12{,}000} \times \frac{34.6}{30} \times \frac{10}{18}$ = \$26,500

I-D fan for C $= 23.3 \times 10.0 \times 150 \times \frac{10.1}{10} \times \frac{30}{27} \times \frac{7{,}500}{8{,}000} \times \frac{13{,}000}{12{,}000} \times \frac{34.6}{30} \times \frac{10}{18}$ = \$25,500

I-D fan for B, capital cost of fuel for additional power = \$26,500 — \$23,600 = \$2,900

I-D fan for C, capital cost of fuel for additional power = \$25,500 — \$23,600 = \$1,900

Values from Fig. 2 for F-D fans: A = \$13.2; B = \$12.5; C = \$12.2

To correct for conditions of Example No. 2, use equation (3)

F-D fan for A $= 13.2 \times 5 \times 150 \times \frac{8}{8} \times \frac{30}{27} \times \frac{7{,}500}{8{,}000} \times \frac{13{,}000}{12{,}000} \times \frac{34.6}{30} \times \frac{10}{18}$ = \$7,160

F-D fan for B $= 12.5 \times 7.3 \times 150 \times \frac{8}{8} \times \frac{30}{27} \times \frac{7{,}500}{8{,}000} \times \frac{13{,}000}{12{,}000} \times \frac{34.6}{30} \times \frac{10}{18}$ = \$9,900

F-D fan for C $= 12.2 \times 7.3 \times 150 \times \frac{8}{8} \times \frac{30}{27} \times \frac{7{,}500}{8{,}000} \times \frac{13{,}000}{12{,}000} \times \frac{34.6}{30} \times \frac{10}{18}$ = \$9,660

F-D fan for B, capital cost of fuel for additional power = \$9,900 — \$7,160 = \$2,740

F-D fan for C, capital cost of fuel for additional power = \$9,660 — \$7,160 = \$2,500

Demand charge for fan power:

Values from Fig. 2 for I-D fans same as above

To correct for conditions of Example No. 2, use equation (6)

I-D fan for A $= \frac{31.7}{1.3} \times 6.8 \times 150 \times \frac{10.1}{10} \times \frac{30}{27} \times \frac{160}{150} \times \left\{\frac{150}{120}\right\}^2$ = \$46,500

I-D fan for B $= \frac{25.2}{1.3} \times 9.6 \times 150 \times \frac{10.1}{10} \times \frac{30}{27} \times \frac{160}{150} \times \left\{\frac{150}{120}\right\}^2$ = \$52,200

I-D fan for C $= \frac{23.3}{1.3} \times 10.0 \times 150 \times \frac{10.1}{10} \times \frac{30}{27} \times \frac{160}{150} \times \left\{\frac{150}{120}\right\}^2$ = \$50,300

I-D fan for B, capitalized demand charge for added power = \$52,200 — \$46,500 = \$5,700

I-D fan for C, capitalized demand charge for added power = \$50,300 — \$46,500 = \$3,800

Values from Fig. 2 for F-D fans same as above

To correct for conditions of Example No. 2, use equation (7)

F-D fan for A $= \frac{13.2}{1.3} \times 5.0 \times 150 \times \frac{8}{8} \times \frac{30}{27} \times \frac{160}{150} \times \left\{\frac{150}{120}\right\}^2$ = \$14,100

F-D fan for B $= \frac{12.5}{1.3} \times 7.3 \times 150 \times \frac{8}{8} \times \frac{30}{27} \times \frac{160}{150} \times \left\{\frac{150}{120}\right\}^2$ = \$19,500

F-D fan for C $= \frac{12.2}{1.3} \times 7.3 \times 150 \times \frac{8}{8} \times \frac{30}{27} \times \frac{160}{150} \times \left\{\frac{150}{120}\right\}^2$ = \$19,030

F-D fan for B, capitalized demand charge for added power = \$19,500 — \$14,100 = \$5,400

F-D fan for C, capitalized demand charge for added power = \$19,030 — \$14,100 = \$4,930

Fuel cost for added feed-pump power: (for economizer pressure drop)

To calculate for conditions of Example No. 2, use equation (8)

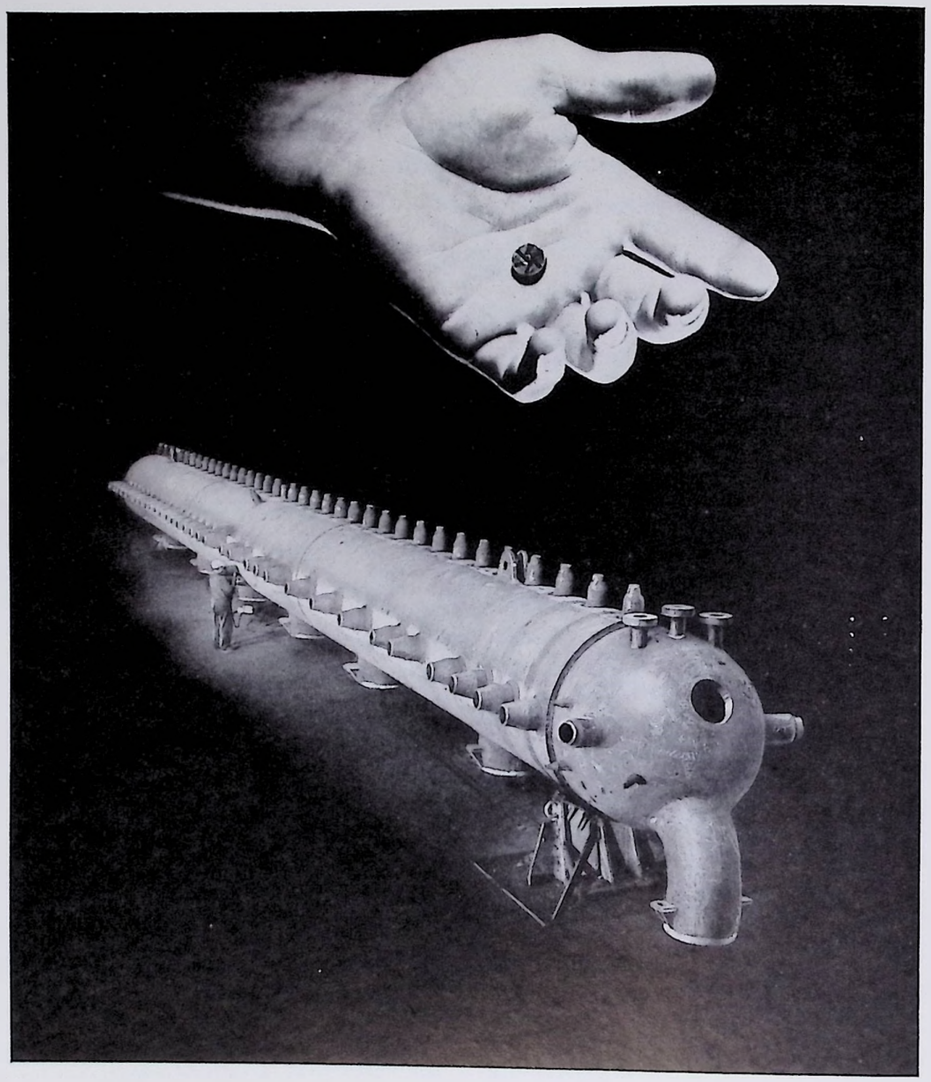
Feed pump for C $= 0.5 \times 10 \times \frac{125{,}000}{1{,}000} \times \frac{7{,}500}{8{,}000} \times \frac{13{,}000}{12{,}000} \times \frac{34.6}{30} \times \frac{10}{18}$ = \$406

Demand charge for added feed-pump power: (for economizer pressure drop)

To calculate for conditions of Example No. 2, use equation (10)

Feed pump for C $= 0.4 \times 10 \times \frac{125{,}000}{1{,}000} \times \frac{160}{150} \times \left\{\frac{125{,}000}{100{,}000}\right\}^2$ = \$834

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BOILER SHOPS ARE CALLED UPON TO MANUFACTURE PARTS AS SMALL AS AN OIL BURNER SPRAYER PLATE AND AS LARGE AS A 104-FT 230-TON DRUM

24

PRINCIPLES AND METHODS OF MANUFACTURE

FABRICATION of the pressure and nonpressure parts and of the auxiliary equipment for modern steam boilers involves a great variety of skills and unique manufacturing techniques and requires facilities for almost every metal-working and metal-forming process. Inasmuch as the majority of boilers, particularly in the larger industrial and utility applications, are specially designed to meet the customer's requirements, most of the pieces, subassemblies, and assemblies that make up the complete unit must be individually designed and shown on separate shop drawings. Thus, fabrication of steam boilers is almost pure "job shop" in character, with only a small segment that could be classified as a quantity-production operation.

The brief sections of this chapter cannot hope to give a complete picture of the many intricate manufacturing processes. Their purpose is to show some of the prominent equipment and procedures which are necessary for the proper manufacture of components of steam-generating equipment.

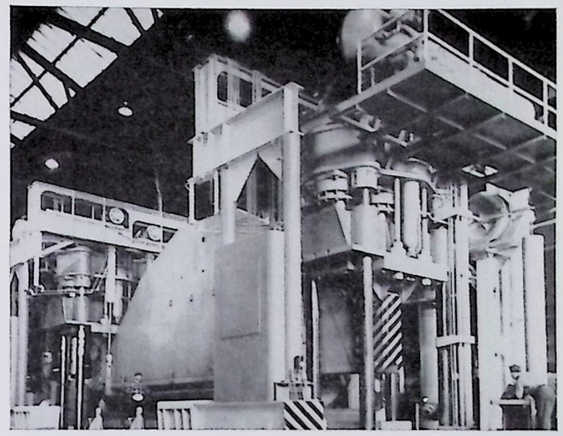


Fig. 1. *Large hydraulic press for forming heavy plate into half cylinders for pressure vessels*

Drum Fabrication

The frontispiece for this chapter shows a large steam drum for a utility boiler on the shop floor. In this case, the drum is made up of three cylindrical sections, or courses, and two hemispherically shaped heads. The many large stubs and nozzles along the length of the drum are connections to flow circuits for the steam-generating and steam-superheating surfaces of the boiler unit. The large nozzles along the bottom of the drum and the push-outs shown on the hemispherical head are connections from the drum to the downcomers, through which the water is carried to supply the various circuits in the generating system.

Fabrication of a drum of this type begins with the pressing of flat plate into half cylinders or the rolling of plate into full cylindrical shells. Fig. 1 shows one of the larger presses used for this purpose, and Fig. 2 a close-up of the pressing operation in the early stages. A plate-rolling operation in a set of

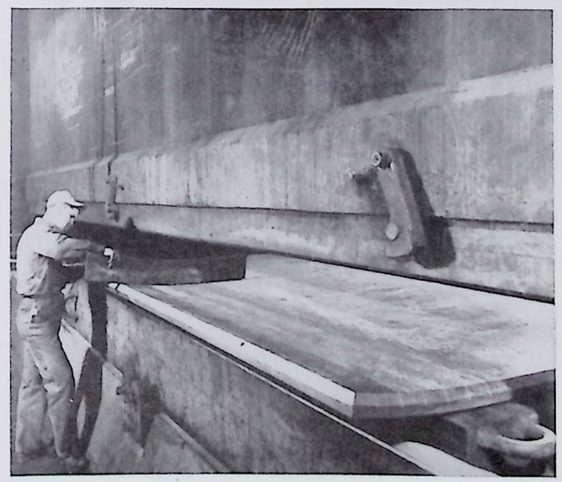


Fig. 2. *Template checking the contour of a drum plate being formed into a half cylinder*

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vertical rolls may be observed in Fig. 3. Rolling of plate into complete cylindrical sections is the process used for short vessels that can be made in a single course. For large drums, the normal procedure is to press plate into half cylinders or shells in lengths up to 42 ft and to form a cylinder by longitudinally welding two half shells together. The desired drum length is obtained by circumferentially joining as many such shells as are required.

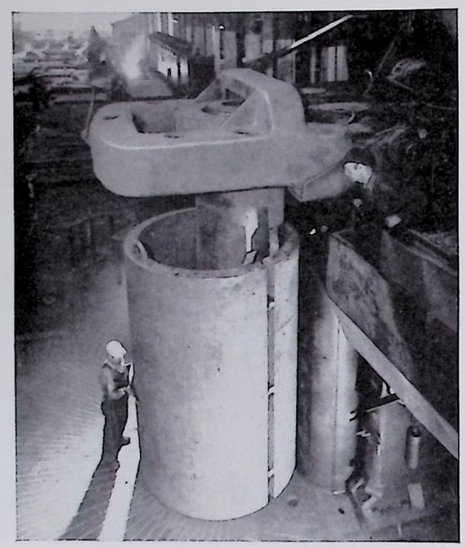


Fig. 3. *Checking the contour of a cylindrical section rolled from heavy plate*

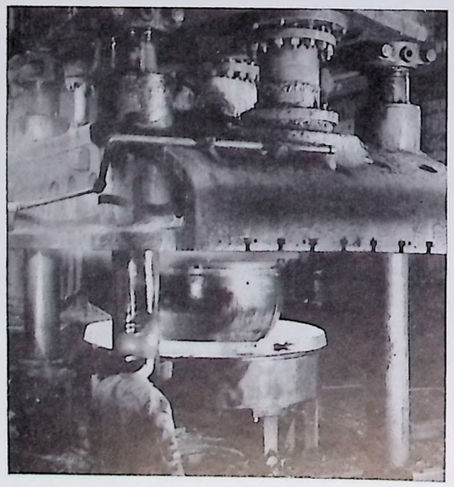


Fig. 4. *Hot-forming a drumhead from heavy plate in a special press*

Drumheads are formed from flat plate by hot-pressing with suitable forming dies. This operation is shown during its initial stages in Fig. 4. Drumheads are attached to the completed cylinder by circumferential welds.

No single feature in the construction of boiler drums and other pressure vessels has contributed more to the increase in size and functional capability of boilers than has the development of welding techniques. Based on extensive research during the years 1926-1928, The B&W Co. developed a welding process that consistently produced welds as strong as the parent plate and chemically comparable to it. A practical nondestructive method of testing to prove the weld was also developed. On September 17, 1929, a proposed specification for electric-arc welding of pressure vessels was submitted by the Company to the Boiler Code Committee of the ASME. In 1930 this committee issued complete rules and specifications for the fusion welding of drums for power boilers. In the same year the U.S. Navy also adopted specifications for construction of welded boiler drums for naval vessels.

Automatic techniques are used for the greater part of the welding on a boiler drum. Fig. 5 shows a double-submerged-arc welding machine in operation on a longitudinal seam. The filler metal, in wire form, is continuously fed to the arc area, which is completely covered by a granular flux to exclude air from the weld-metal area. During the operation, the electrode carriage moves along the seam from one end to the other. Many passes may be required, depending upon the plate thickness, the material specifications, and the form of the welding groove.

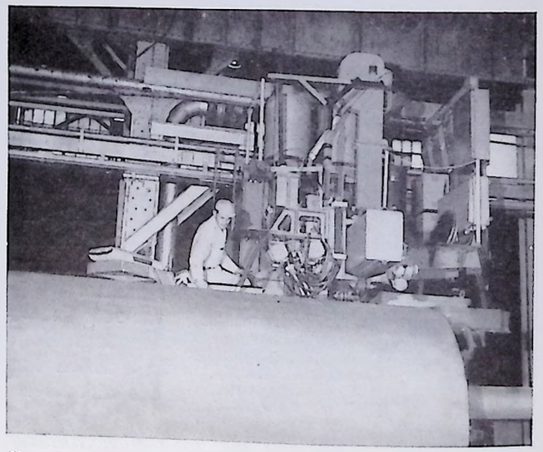


Fig. 5. *Welding longitudinal drum seam with double-submerged-arc machine. Note test plate at end of shell*

For submerged-arc circumferential welds, the drum is mounted on a turning device and rotated

while the welding head and the flux-applying equipment are stationary. Prior to and throughout all welding operations, a preheat is applied to the weld area to avoid detrimental stress conditions and metallurgical transformations. This preheat is maintained after welding until the vessel is stress-relieved.

In recent years advances in engineering designs and the development of high-strength high-temperature alloys have required new welding techniques. The atomic-energy programs uncovered certain inadequacies of existing methods. This new field is one in which the integrity of the system being fabricated must be beyond question, since even a minor weld failure could destroy the usefulness of an entire plant.

What was formerly considered sound welding is often unsatisfactory for nuclear application. Old techniques and procedures had to be discarded and new ones developed, especially for welding stainless steels. Thus today the use of an inert gas, such as helium or argon, to protect the welding zone from oxidation and at the same time eliminate slag entrapment, is a commonplace production method. Other developments that will contribute significantly to the quailty of the welded joint and appreciably increase the deposition rate of weld metal are currently nearing productive applications.

Nozzles and stubs are formed by hot-forging from a solid billet. Larger connections may be integral with the heads, as shown in the frontispiece, formed from rolled plate, or made of seamless tubing as are those shown along the bottom of the drum in the same illustration. Attachments to the drum are made by hand welding.

Every drum fabricated must be subjected to a stress-relief heat-treatment after all welding on the drum has been completed, and in most cases several heat-treatments are required during the course of construction of the drum. In instances where the application requires high impact strength, formed heavy plate is water-spray quenched to cool it rapidly from 1600 to 600 F through the transformation range. This heat-treatment is followed by normal stress-relief, which consists of heating to 1150 F, soaking for one hour per inch of thickness, and furnace cooling to 600 F. Fig. 6 shows the withdrawal of pressure-vessel parts from one of the large stress-relief furnaces.

Every inch of every seam weld in a drum and numerous other welds must be subjected to radiographic examination. For this purpose extensive X-ray equipment of a wide range of power, from portable machines rated at 400,000 volts to the 2,000,000-volt machine shown in Fig. 7, is provided. Supplementing the X-ray equipment are radioactive isotopes, such as cobalt-60, dye-penetrant methods of inspection for locating fine surface defects, and mag-

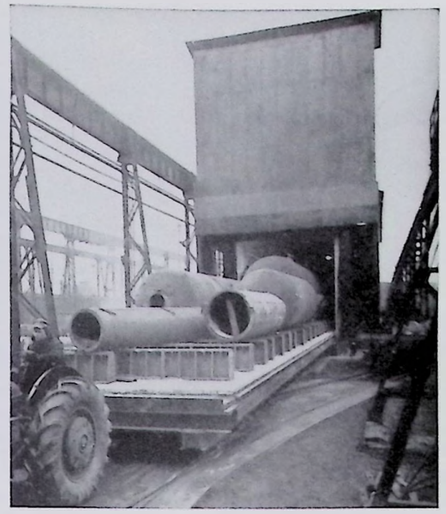


Fig. 6. *Stress-relieved pressure vessels emerging from heat-treating furnace for further fabrication*

netic-particle methods for defects just slightly below the surface of the material. Recently, reflectoscopic examination utilizing ultrasonic vibrations has been adopted as standard testing procedure on all heavy plate for pressure vessels.

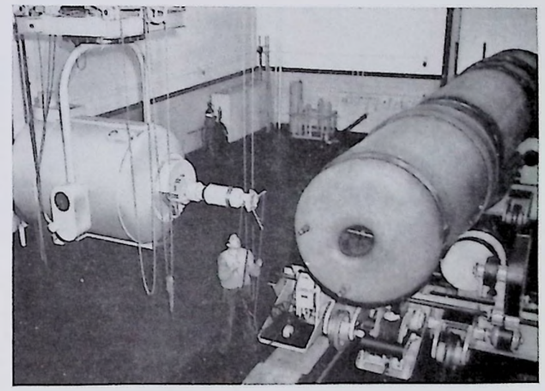


Fig. 7. *Registering 2-million-volt X-ray machine on circumferential seam of drum for radiographic examination of weld. Film cassette is positioned on inner surface*

Throughout the fabrication process of any pressure vessel, all work is conducted under the strictest surveillance of the Company's Quality Control Department to insure compliance with the ASME codes, Company standards, governmental standards, and customers' requirements. All materials used for pressure parts are certified as to specifications by wet chemical analysis, spectroscopy, physical tests for tensile, bending, and impact strengths, and metal-

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lurgical checks of the material. During the welding of longitudinal seams, a test plate, from which test coupons are later machined, is attached to one end of the drum and welded as an extension of the drum shell. Fig. 5 shows a test plate attached to a drum in position. A laboratory examination of the test-plate weld is made to check the quality of the drum weld. Other tests are for ductility, tensile strength, soundness, and chemical composition.

After all welding of pressure parts has been completed and found satisfactory, the vessel is hydrostatically tested at 1½ times the design pressure, in accordance with the ASME *Boiler and Pressure Vessel Code* requirements. The vessel is then ready for the finishing operations, which include installation of minor attachments and of the internals. Fig. 8 shows the internal-baffling arrangement for a large steam drum partially preassembled on the shop floor. This structure later will be knocked down and reassembled inside the steam drum for shipment with the drum.

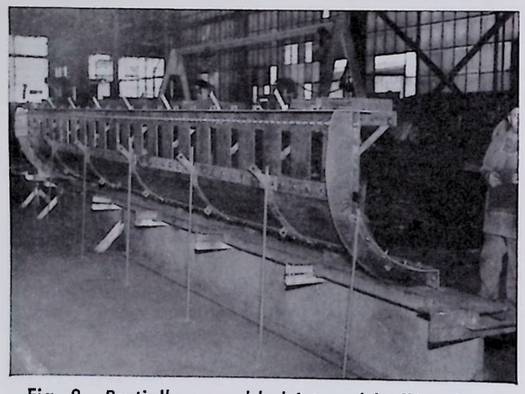


Fig. 8. *Partially assembled internal baffling for a steam drum*

Headers as Terminal Pressure Vessels

Headers are used extensively in modern steam boilers as a means for joining two or more boiler circuits. Headers are of smaller diameter (under about 24-in. ID) than drums and can be fabricated from seamless tubing or hollow forgings. Access to their interior is through handholes, whereas drums have manholes for ingress and egress of individuals. Fig. 9 shows a superheater header nearing completion of fabrication. Many bent tubes (stubs) and tee connections have been shop-welded to the 20-in. OD, 4½ in. thick, 50 ft long superheater header shown in Fig. 10. All welds to pressure vessels are completed in the shop, so that stress-relieving of welds can be done quickly and economically under controlled conditions in the Works.

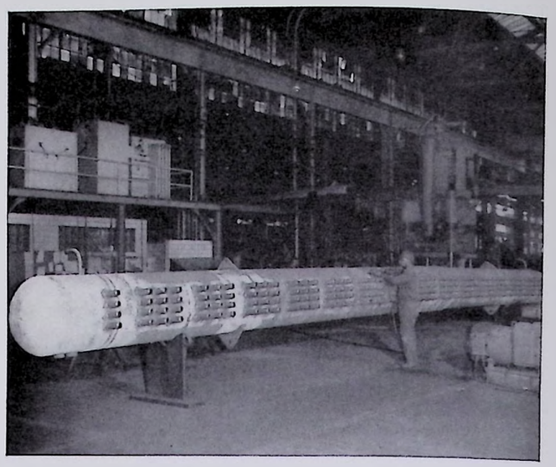


Fig. 9. *A large superheater header of low-alloy steel in the final stage of fabrication*

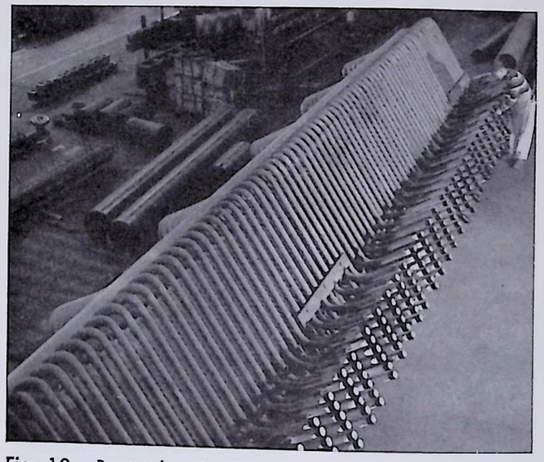


Fig. 10. *Bent tubes (stubs) and tee connections (opposite side) shop attached to a large superheater header*

Complete facilities are available for the manufacture of large-diameter heavy-wall tubular stock. This equipment consists of electric-arc furnaces for producing the required ingots, an 8500-ton press used for initial piercing of the ingots, a 1200-ton drawbench for reducing outside diameters and wall thicknesses and extending the lengths of the pieces, a multiplicity of engine lathes for machining the outside and inside diameters of the produced pieces, and numerous types of special test equipment for checking and controlling the quality of the finished products. Some of these various items of equipment are shown in Figs. 11, 12, and 13.

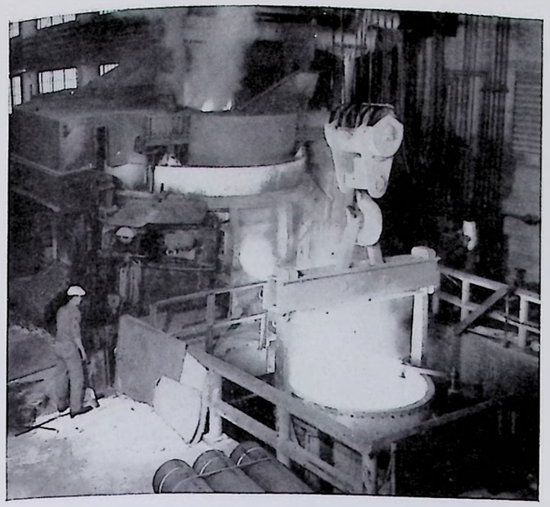


Fig. 11. *Tapping an electric-arc furnace preparatory to casting ingots for large hollow forgings*

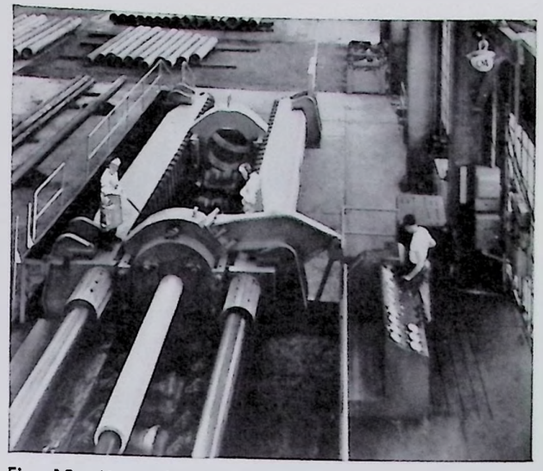


Fig. 13. *Huge drawbench converts 26,000-lb pierced ingot into long hollow forging*

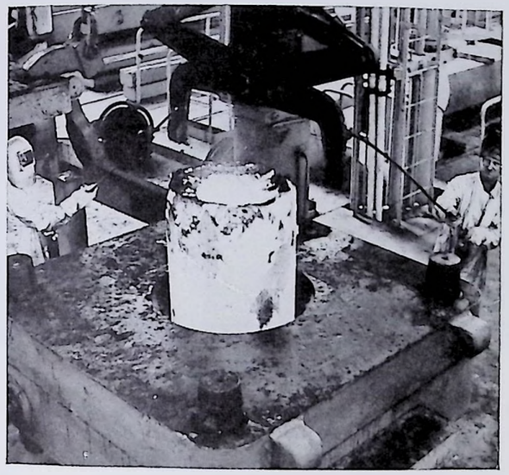


Fig. 12. *Cupped forging leaving piercer pot of vertical press, ready for drawing into tubular stock*

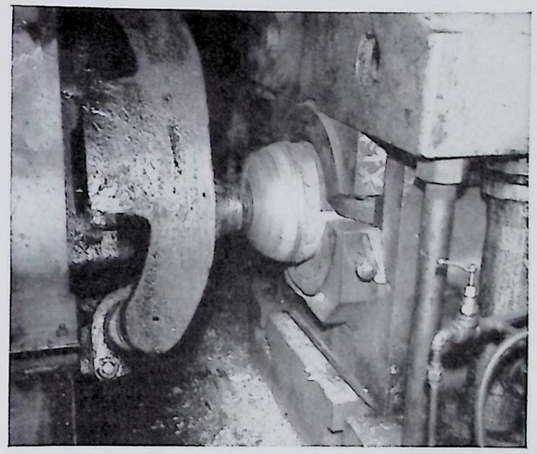


Fig. 14. *Header end closed by hot-spinning*

Fabrication of a large header, such as shown in Fig. 9, begins with circumferentially welding the required pieces of pierced and drawn hollow-forging stock to obtain the desired length. The ends of the header stock are then spun hot until closed, as shown in Fig. 14. To assure absolute tightness after spinning, the center portion of the closure is drilled out and replaced by a fusion-welded steel plug. For the larger headers, when the outside diameter exceeds the capacity of this spinning machine, the ends of the headers are closed with a forged hemispherical head in an operation very similar to that previously described for drumheads.

Completion of the header includes drilling and machining of tube and nozzle holes, fitting and welding the nozzles, stubs, and other attachments, heat-treating, cleaning, and finishing operations, all of which are conducted under strict quality control. Some of these operations are shown in Figs. 15, 16, and 17.

While the majority of headers today are circular in cross section, there are some applications where square headers are suitable. The fabrication process for these is similar to that described above, except that, prior to closing the ends, the header stock is changed to a square cross section by passing it through a set of rolls while at forging temperature. Also, the staggered, or sinuous, headers are still popular for certain types of marine boilers. Fig. 18

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Fig. 15. *Fusion welding nozzle connections to a large header*

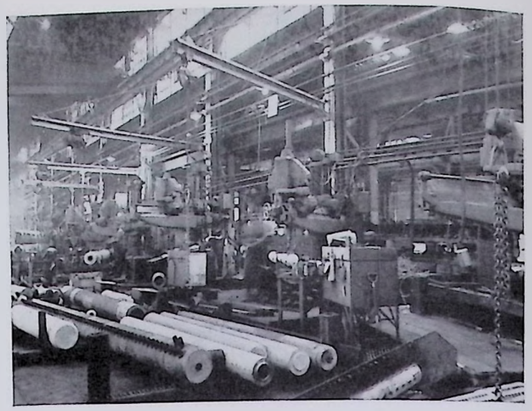


Fig. 17. *Bank of radial drills used for fabricating headers*

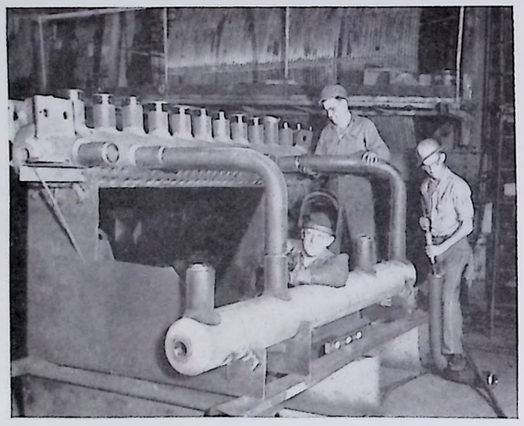


Fig. 16. *Shop assembly of two connecting headers*

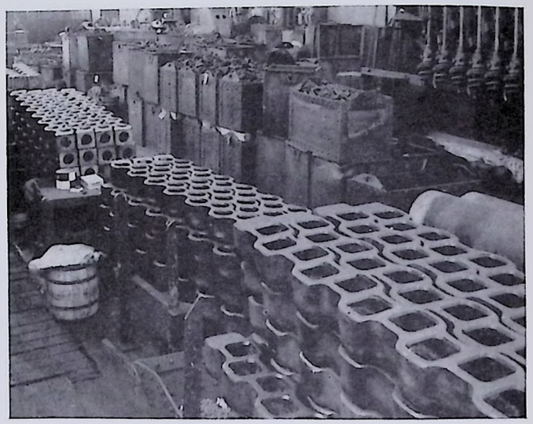


Fig. 18. *Partially fabricated sinuous headers for marine boilers*

shows a group of such headers. Fore and aft nests of sinuous headers are joined by straight tubes to form the convection sections of boilers. This arrangement gives convenient access for inspection and cleaning of the tubes. The particular shape of these headers is obtained by a forging operation in a stagger press equipped with exterior forming dies working against a collapsible sectional mandrel.

Tubes as Heat-Absorbing Surface

Tubes, in an almost endless variety of arrangements, serve as the heat-absorbing surface and provide flow circuits in today's steam boilers. Tubes range from 1 to 4½ in. in diameter and may be plain or fitted with studs for extended surface, as in the case of some furnace-wall tubes.

Functionally, tubes of a steam-generating unit may be divided into two general classifications: tubes for heating surface to heat water or convert water into steam, and tubes for heating surface to superheat the steam.

In smaller industrial boilers used principally for heating or process steam, the majority of the generating surface is of the convection type. For power boilers, on the other hand, the generating surface is

generally limited to the tubes forming the furnace and setting enclosures. Heat transfer is by radiation rather than by convection in such units.

For the larger units, past practice was to fabricate and ship each wall tube as an individual item. This required a piece-by-piece field assembly during erection. Today, preassembly of wall panels in the shop to facilitate field erection is rapidly growing in importance. The latest design of panel walls is a "membrane" construction of furnace and setting walls. This construction is shown for the major part of the preassembled wall section, Fig. 19, where the

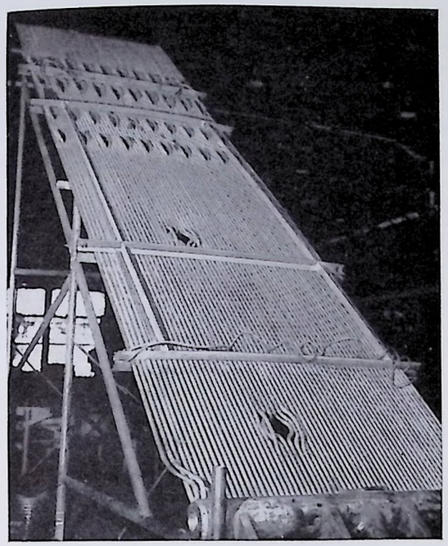


Fig. 19. *Membrane-wall assembly of steam-generating surface*

space between wall tubes, from the top header down to the lowest angles, is closed with a metal filler piece continuously welded on both sides to the adjacent tubes. The result is a gastight metal wall surrounding the enclosed volume. A close-up view of a membrane-wall panel is shown in Fig. 20.

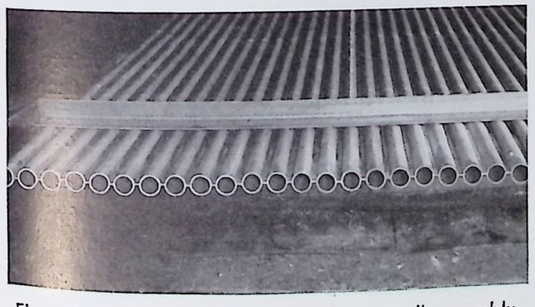


Fig. 20. *Close-up view of membrane-wall assembly*

This design eliminates the need for a pressure-tight casing, and reduces the cost of field erection. Walls of this type are assembled on a table in sections 10 ft wide and up to 90 ft long. The membrane is then welded to the tubes by the submerged-arc process, with a traveling head feeding a number of electrodes simultaneously. The machine for this operation is shown in Fig. 21.

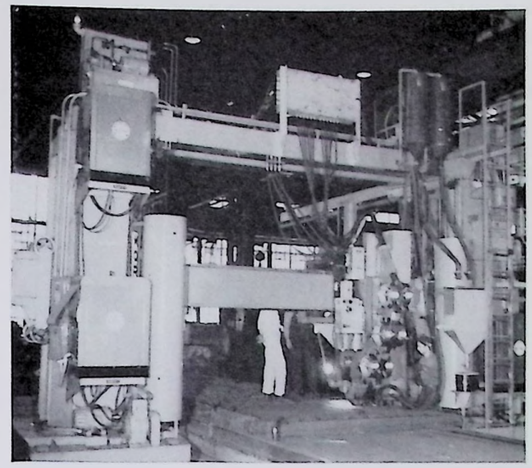


Fig. 21. *Machine equipped with trolley carrying six welding heads which operate simultaneously in fabricating a membrane wall*

Where bends in the panel are required, wall sections of this type are bent to required radius and degree after assembly of panels, using a multiple die

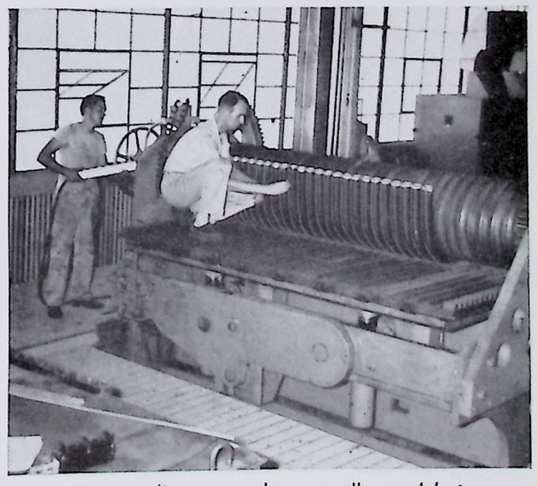


Fig. 22. *Bending a membrane-wall panel between special multiple dies*

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which may encompass as many as 60 tubes. Fig. 22 shows a membrane-wall section being bent in special multiple dies.

While wall-type surface, such as described above, is used to some extent for superheating purposes, most convection superheaters use continuous-tube sections. A typical section of the pendent type is illustrated in Fig. 23. Convection superheater platens are made from relatively short lengths of straight tubes which are first bent and then welded together to give the configuration shown. All of the bends are formed cold except the inside hairpin bend, which must be made hot.

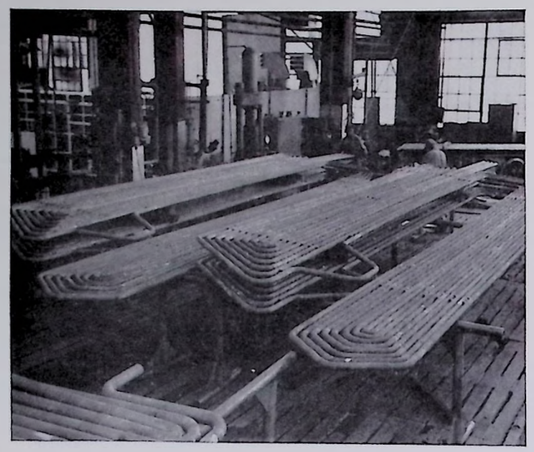


Fig. 23. *Pendent superheater platens ready for finishing operations*

Close return bends are made by first upsetting a section of straight tube to increase its wall thickness at the location for the bend and then hot-bending and sizing to the desired dimension. The prepared ends of the resulting canes are joined by an automatic butt welder to form the section. In addition to forming the weld, this machine also heat-treats the weld (see Fig. 24).



Fig. 24. *Automatic butt-welding machine for making continuous-tube platens. Note newly made weld and other welds in completed portion*

To complete the platen, supporting lugs are fitted and welded into position, a hydrostatic test at 1½ times the design pressure is applied, and the section ends are milled to exact lengths and beveled for field assembly. The sections are then painted and ready for shipment.

Structural Steel and Casings

As many as 4000 tons of structural steel may be required for the support of a modern central station boiler unit designed to produce 3 million lb steam per hr. The structural framing for the unit may reach a height of 290 ft from the column base to the top steel. This is the equivalent of a 28-story building. The area enclosed by the framing may exceed 13,000 sq ft. Most of the structural material required is readily available in standard mill shapes. However, in some cases the horizontal members that directly support the heavy loads must be fabricated from plate and standard-shape material.

Most boilers are enclosed with an inner casing of light-gage flanged sheet metal applied on the outer side of the wall tubes and welded to casing channels and buckstay supports in such manner as to provide for expansion and contraction in the boiler unit. Inner casing acts as a barrier to air infiltration on suction units and prevents outward leakage on pressurized units. It is not required with membrane-wall construction, as explained above. The boiler is insulated completely over the inner casing. On some units metal lagging is applied to the outside surface of the insulation to protect it from mechanical damage and to enhance the appearance of the unit and its housekeeping. It also provides protection against the weather on an outdoor installation.

Fuel-Burning Equipment

The B&W Co. manufactures a great variety of fuel-burning equipment for all types of fuel, including oil, gas, coal, combinations of these standard fuels, and many special fuels. This is one of the principal areas where standardization of product has been possible to a large extent and manufacturing techniques in some cases approach those of mass-production industries. One example is the manufacture of Cyclone Furnaces. Fig. 25 shows a number of these furnaces in the process of assembly on the shop floor.

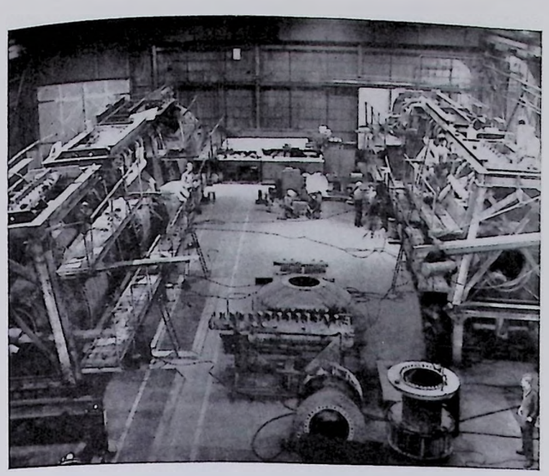


Fig. 25. *Cyclone Furnaces in the process of assembly*

Cyclone barrels range from 5 to 10 ft in diameter and are composed of semicircular-shaped tubes joined to form a cylinder (see Fig. 26). Note the detail of the throat area in this and the preceding figure. Since large quantities of duplicate tubes are required to form a cyclone, jigs, fixtures, and power-operated equipment are used to minimize the human effort for these repetitive operations.

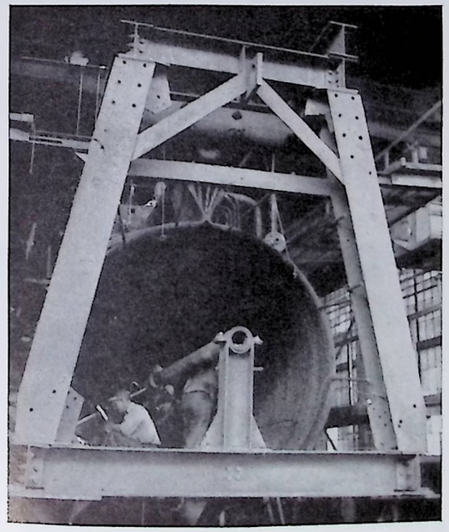


Fig. 26. *Interior view of a Cyclone Furnace during shop assembly*

Cyclone Furnaces are completely shop assembled in erection frames that permit stress-relief of the finished product. This enables close quality control to insure that ASME Code and Company standards are met. Before shipment to the job site, plastic chrome ore is applied to the stud tubes on the inside of the furnace barrel and the outside is covered with a protective skin casing.

The B&W coal pulverizer is another well-standardized product of the Company. Fig. 27 shows a partially assembled coal pulverizer on the shop floor.

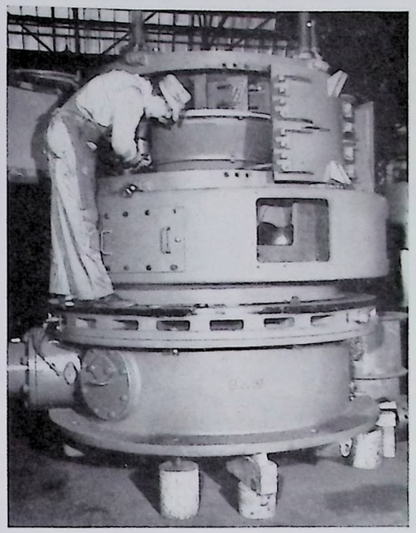


Fig. 27. *B&W pulverizer nearing completion of shop assembly*

The outer casing or housing of this pulverizer is a steel weldment formed from rolled plate and other standard shapes. The base of this machine is a large gray-iron casting manufactured in the Company's foundry. Within this base, the main spiral bevel gear and its driving pinion are mounted. During operation, these parts are completely flooded with lubricating oil, and the need to maintain cleanliness of this oil requires close manufacturing tolerances to insure that the seal separating the driving and grinding parts of the pulverizer shall successfully prevent coal dust from entering the lubricating system. The grinding rings are cast of wear-resistant alloy. Chill blocks, inserted in the mold at the grinding surface of the ring, cause the rapid cooling necessary to develop the required high wear-resistant quality. Pulverizer balls are produced by hot-forging from billets of selected alloy steel (see Fig. 28). After forging, the balls require heat-treatment to insure proper hardness and wear characteristics.

There are many operations in the production of a pulverizer. Some parts require machining, such as the seals and gearing described above. Others too

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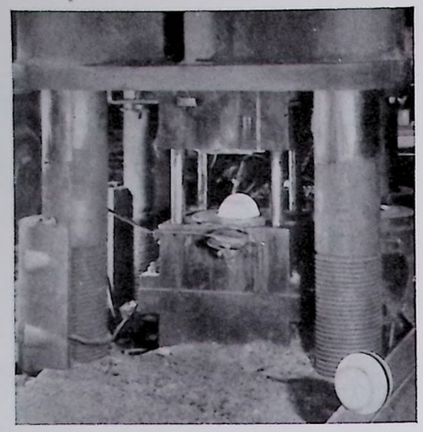


Fig. 28. *Press used to forge pulverizer balls from alloy-steel billet*

hard for machining, such as the grinding rings of a pulverizer, must be ground to the finished dimensions. Pressing, rolling, welding, bending, shearing, and drilling are examples of additional operations required. Each pulverizer fabricated is shop assembled and run-in prior to shipment.

Tooling

There are more than 2000 machine tools in the shops of the Boiler Division of the Company. These include practically every type of standard machine tool, such as vertical and horizontal boring mills, milling machines, planers, drilling equipment, turret and engine lathes, grinders, and slotters. In addition to these standard tools, there are a number of tools designed, and in many cases built, by the Company to perform unusual operations. Some of these machines are shown in Figs. 29 through 32. As is evident from examination of these illustrations, the outstanding characteristic of the B&W shops is the ability to fabricate heavy equipment.

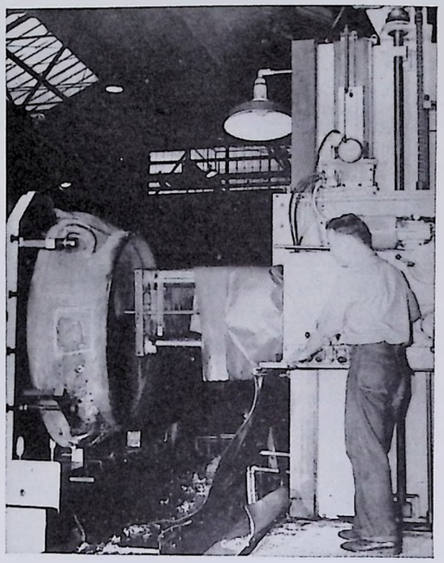


Fig. 30. *Gun drill makes extremely accurate holes through 12-in. plate in less than 6 minutes*

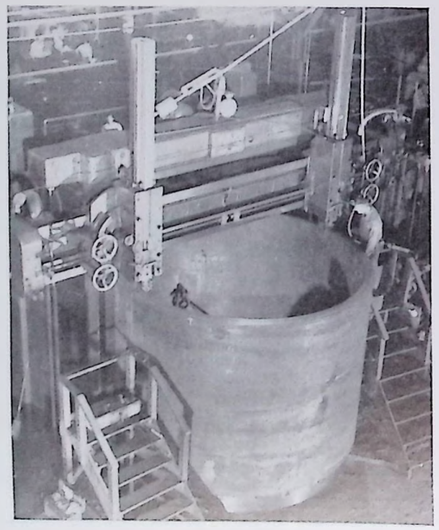


Fig. 29. *Machining the edge of a pressure vessel in a vertical boring mill*

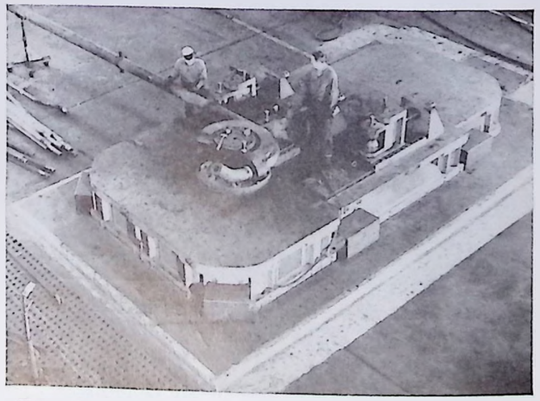


Fig. 31. *Cold-bending a large-diameter heavy-wall tube. Note out-of-plane bend at end of tube*

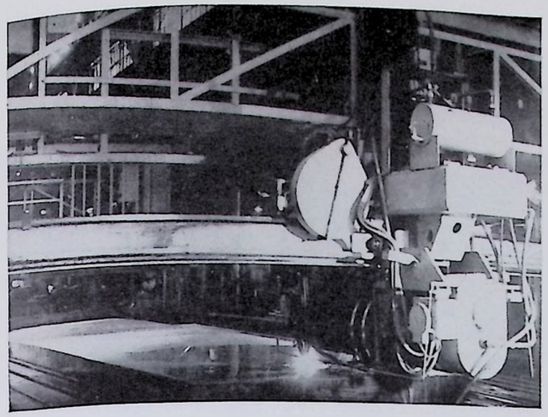


Fig. 32. *Machine for spot welding a thin sheet of stainless cladding to heavy steel plate for subsequent fabrication into a lined pressure vessel*

Nuclear Components

In recent years the increasing participation of the Company in programs for the development of nuclear-power applications has served to demonstrate the need for additional tooling, new manufacturing methods, and an adequate site for the assembly of components. Fig. 33 is an interior view of the newly completed building that was specially constructed for assembly operations. It is supplied with filtered air and maintained at a positive pressure to minimize the entry of dust.

Heat exchangers are an important part of a nuclear power plant. They are built to exacting specifications. Figs. 34 and 35 show a U-shaped heat exchanger in the process of assembly. The multitude of tubes must be properly positioned and assembled for maximum fluid tightness. After the tube-expanding operations shown in Fig. 35, the protruding tube ends are trimmed and seal welded to the tube sheet.

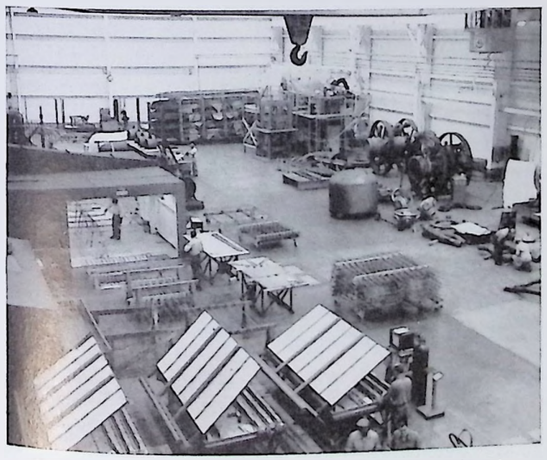


Fig. 33. *Facility for assembly of nuclear-application components*

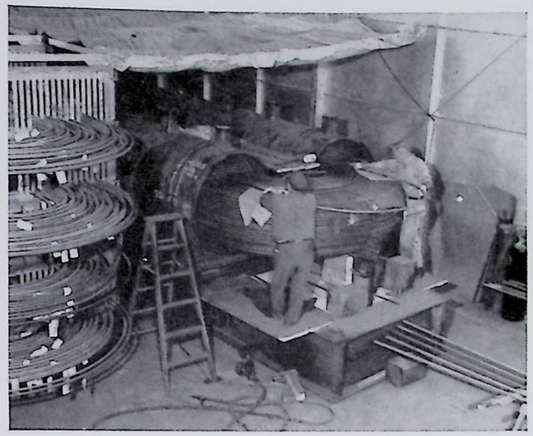


Fig. 34. *Installing tubing in a U-shaped heat exchanger for nuclear application*

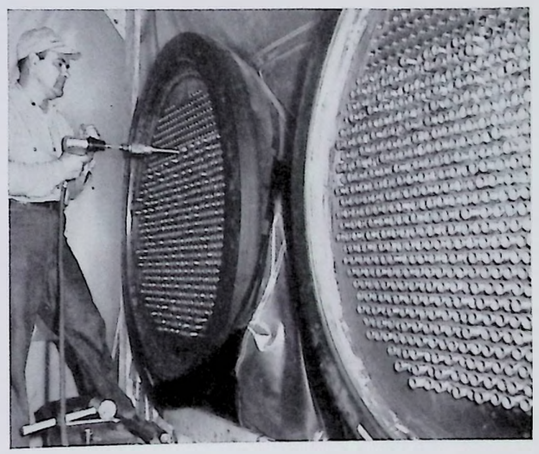


Fig. 35. *Expanding tubes into tube sheets of heat exchanger shown in Fig. 34*

The heaviest equipment and the vital part of a nuclear plant is the reactor itself. Fig. 36 shows the head of a nuclear reactor vessel in its final stages of machining. Prior to forming, the interior of the head was clad with stainless steel by the machine shown in Fig. 32.

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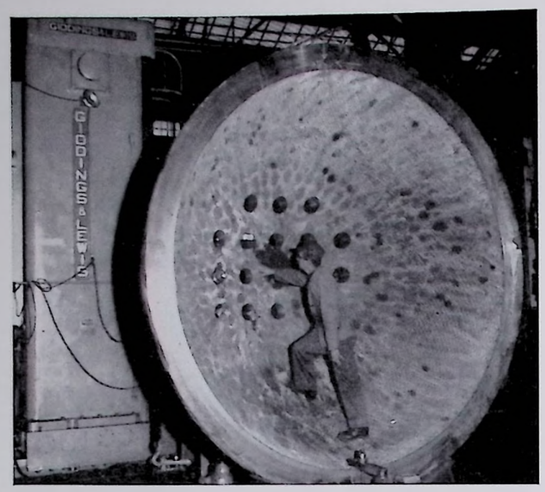


Fig. 36. *Head for a nuclear reactor vessel formed from B&W clad plate*

Certain components must be clad after they have been formed. Fig. 37 shows a large flange for a nuclear reactor being clad internally with stainless steel by the submerged-arc automatic-welding process.

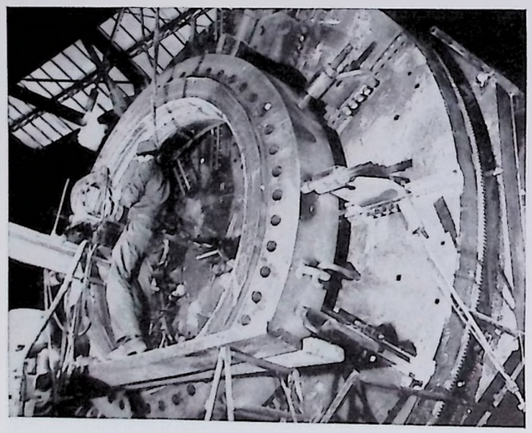


Fig. 37. *Applying stainless steel cladding to the inside surface of a flange*

Another heavy component of a nuclear reactor pressure vessel is the closure head through which the control rods must operate. Finishing operations on such a head are illustrated by Fig. 38.

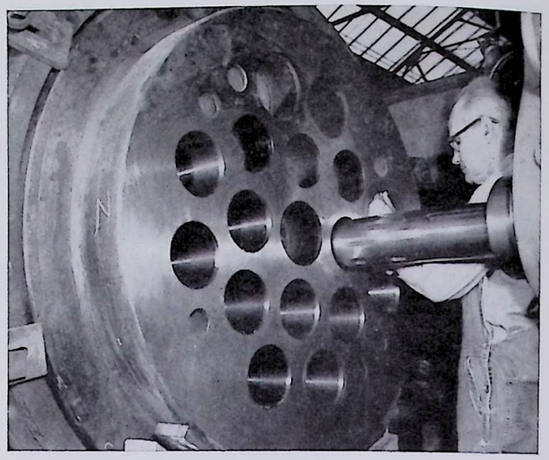


Fig. 38. *Horizontal boring mill finishing the inside surfaces of control-rod holes in a closure head for a nuclear reactor pressure vessel*

General

Practically all of the various components manufactured by the Company must be built to close tolerances. This is particularly true of equipment and parts that are shop assembled, such as pulverizers, Cyclone Furnaces, drums, and headers. It is also important for the thousands of parts which are shipped to an erection site remotely located from the manufacturing base. Such parts as these must match other parts in the field, as errors or inaccuracies require extensive and costly correction and cause delay in the erection process. These considerations, as well as the requirements of the ASME Code, various governmental agencies, the customer, and insurance companies, lead to emphasis on inspection and quality control of the extremely heavy equipment which characterizes the operations of The B&W Company.

25

PRINCIPLES AND METHODS OF INSTALLATION

A few boiler types are small enough to permit shipment completely assembled. Some marine and industrial boilers of moderate capacity are in this class. They usually weigh less than 50 tons and can be loaded on a single freight car. Larger industrial and central station units are shipped as components in various stages of fabrication and subassembly, as in Fig. 1. A large central station boiler may weigh 5000 tons and require several months and three hundred cars to ship the material to the site. The field assembly of a high-capacity steam-generating unit requires efficient, well-engineered, and well-organized erection methods to permit installation of the equipment within a reasonable time and at a minimum cost without sacrifice of quality.

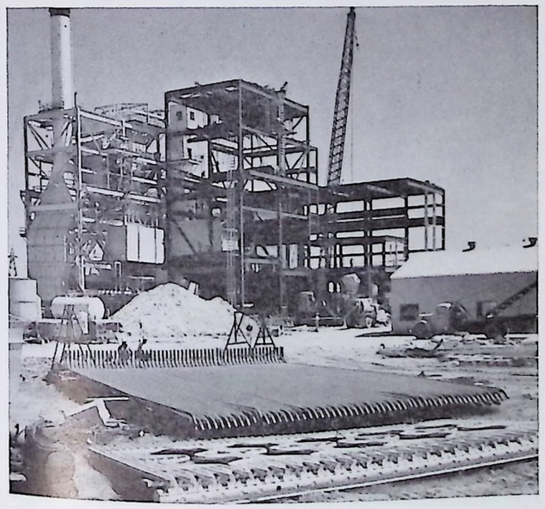


Fig. 1. *Erection work starts on a central station boiler unit*

The erection of a steam-generating unit represents a sizable part of the investment and has a tremendous influence on its functional performance. The manufacturer must have an erection organization to develop techniques that will complement the designer's skill and the fabricator's craftsmanship. Boiler components must be on the job site at the right time with the necessary tools, equipment, manpower, and supervision for assembly.

Good staff work is essential in planning and coordinating the installation of a steam-generating unit. Accurate estimates are needed for: 1) determination of the duration and cost of erection; 2) intelligent planning of all phases of the job; and 3) sales pricing purposes. Schedules coordinate shop fabrication with field erection in such manner that time is utilized to best advantage.

Design details are reviewed as they are developed to avoid difficult assembly operations. In order to have adequate tools, equipment, and facilities available on the job when needed, preparations are made well in advance of material shipments. Periodic visits to the site are made by erection personnel to reserve and arrange space for storage of material and parts and to insure unrestricted access to the building for drums and other large pieces. The Company maintains a corps of trained erection superintendents to take charge of the many different types of projects. As the job proceeds, liaison is maintained between the field superintendent and the several other departments of the Company whose efforts must be coordinated for maximum effectiveness. A cost-control system, with other staff functions, keeps management continuously advised on progress and costs.

Time and Cost Estimates

There is no fixed relationship between boiler capacity in lb of steam per hr and cost of field assembly. The factors that influence the design of boilers are too numerous. The fuels to be burned determine the furnace shape and volume, with consequent variations in size and weight of component parts. Higher pressures result in greater tonnage of materials. Higher temperatures require more field welds in alloy tubes and pipes. Design improvements and

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additions for increased efficiency affect installation costs. Coal-fired boilers with soot blowers and ash- and slag-removal devices cost more to build than oil- or gas-fired units of similar capacity.

A breakdown of weights and quantities of components must be available to estimate the field-erection cost of a steam-generating unit. The basic quantities are multiplied by man-hour units that have been compiled from reports on similar boilers erected during the previous five-year period. Man-hour units on a tonnage basis are used for structural supports, drums, headers, tubes, casing, superheaters, economizers, and air heaters. Man-hour units on other bases are applied to operations like welding, stress-relieving, and tube expanding. The sum of the various unit totals equals the man-hours of productive labor required. A further sum, usually a percentage, must be added for general operations, supervision, field equipment, maintenance, and cleanup, as these items cannot be charged to individual components. This total can reasonably be tempered by factors pertaining to site conditions, geographical location, and available time for completion. Total man-hours are multiplied by field-labor rates, with additions for insurance, taxes, and other overheads, to obtain dollar costs. Field-labor rates vary so widely that it is preferable to use man-hour requirements rather than dollars in comparative studies of erection costs.

While there is no definite relationship between steam capacity in lb per hr and the cost of field assembly of boilers, there is an approximate relation between the kw capacity of the electric-generating equipment and the erection cost of the boilers serving it. This approximate relationship holds even though the boilers may be of different types, reheat or nonreheat, and designed for different temperatures and pressures. Field-erection requirements for coal-fired boilers vary from 1 to 1½ man-hours per kw capacity; for gas-fired boilers, from ¾ to 1¼. These requirements do not include allowances for structural supports, dust collectors, ash-removal equipment, and power piping.

The elapsed time needed to erect a complete steam-generating unit is largely a function of man-hours. There is a limit, however, for any given size of unit, to the number of men that can be used during any given stage of erection. If more than the optimum number are employed, the lack of adequate places for them to work results in excessive costs. Fig. 2 shows the number of weeks required for various man-hour expenditures on single boiler units and the maximum number of men that can be employed efficiently at peak operations. Fig. 3 shows a manpower graph for a job of 100,000 man-hours employing 75 men at the peak of construction. These schedules are based on single-shift operation with a

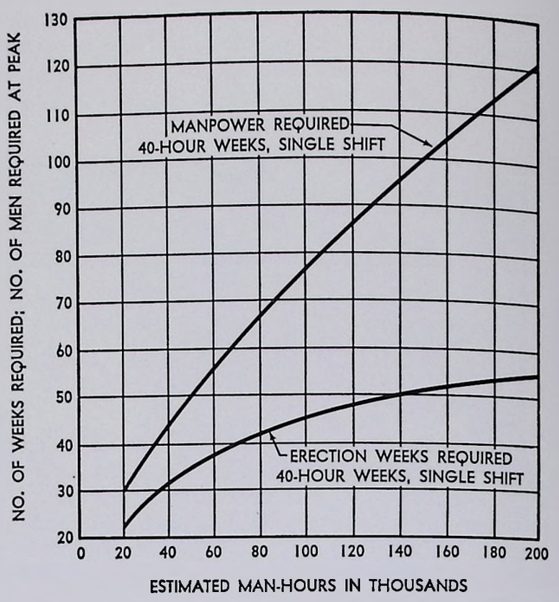


Fig. 2. *Graph showing number of weeks and number of men required for boiler erection related to total man-hours*

40-hour week. The erection period can be reduced by multiple shifts or overtime. These procedures increase total erection costs because of inefficiency and higher wage scales.

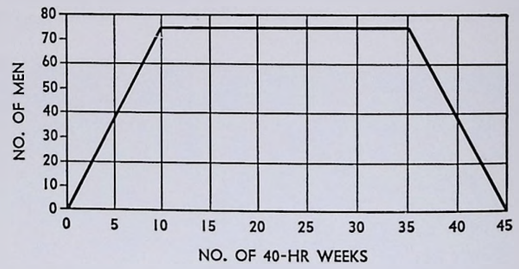


Fig. 3. *Manpower graph for boiler requiring 100,000 man-hours for erection*

Erection Schedule

After the sale of the boiler and following a study of contract requirements, a field-erection schedule is prepared and given to the Works to serve as the basis for shop fabrication and shipment. This schedule shows, in considerable detail, when each component will be needed at the site. The projected sequence of erection steps must make allowance for the expected manufacturing time, the shipping characteristics, the accessibility of the site, and the time available to complete the assembly. Certain general sequences are mandatory because of size, location within the structure, and method of support of the parts. The erection schedule, with additional information from

the engineering, purchasing, and fabricating departments, is incorporated in a production schedule. This schedule must be distributed within the Company and the customer's organization in order to coordinate boiler erection with other plant construction work.

Review of Design Details

The early design drawings are reviewed by erection personnel with representatives of engineering, manufacturing, and shipping departments for maximum effective coordination. The dimensions of shop-assembled components, such as furnace-wall panels, burner casings, and duct work, are limited by clearances on railroad and shipping rights of way. Shop assembly reduces field work and, when supplemented by ground assembly at the job site, leads to lowest construction costs. Design details are reviewed to assure the best shop- and field-assembly requirements. Preliminary plans for hoisting heavy members determine the lifting-lug and hanger requirements. Within the limits of practical economics, these attachments are generally made in the shop. Otherwise, advance information is obtained to provide them as part of the tool equipment for the job. During the design review, the locations of field welds in tubes, pipes, and other pressure parts are determined. Such joints must be few in number, consistent with size limitations, and so located as to leave clearance for welding and stress-relieving operations.

Selection of Tool Equipment

Many types of power equipment are needed for unloading, hauling, and positioning the heavy assembled sections and components of modern boilers. The amount of material that can be handled manually is almost negligible. Between 25 and 50% of the manpower is employed on welding and stress-relieving operations during various stages of construction. Equipment for this work must be available when needed. Construction buildings, trucks or flat cars for moving material, and hundreds of hand tools and hardware are necessary for general construction work. The exact pieces of equipment furnished to a particular job are determined by economic considerations. Where schedules are tight or job-site conditions unusually severe, a greater outlay for equipment may be justified than under more favorable circumstances.

The B&W Co. has a large assortment of derricks, mobile cranes, winch trucks, hoisting engines, welding machines, air compressors, and other construction tools upon which to draw. With the exception of cranes, this equipment is shipped to and from the warehouse and maintenance center at Barberton, O.

Preparations at the Job Site

Details of storage, handling, and erection must be thoroughly reviewed on the job site with the customer and the contractor before starting shipment. Provisions must be made for:

1. Adequate storage space near the site, varying from a few hundred sq ft to several acres, depending upon the size of the boiler. (The storage area should be served by railroad tracks with a spur leading to the erection area. Roadways must be available for movement of material by trucks and for access by movable cranes.)
2. An erection field office, tool and material room, and a change room for the workmen (see Fig. 4).
3. Compressed air and electric power suitably located at the boiler site.
4. Drinking water and sanitary facilities.
5. Stairways and walkways to permit safe access and working conditions around the boiler.
6. A derrick (if required) on the roof for raising material (see Fig. 5).
7. Omission or removal of structural members to give necessary access for erection of large components.
8. Opening in buildings, such as hatchways in the roof and large doors in the walls at ground level.



Fig. 4. *Temporary field office used by erection force during assembly of boiler unit*

Some boiler installations are outdoors with no permanent roof; others are in enclosed buildings. In either case a decision must be reached concerning the weather protection to be furnished by each contractor on the job. Foundations for boilers and auxiliary equipment are usually designed and constructed by the customer or by an engineering firm specializing in this type of work. Drawings indicating size and location of anchor bolts, conduits, openings, and the loads to be carried are supplied by the equipment manufacturers. All foundations must be completed and checked for compliance with specifications and instructions before erection work is started. This avoids subsequent costly delays, interferences, and alterations.

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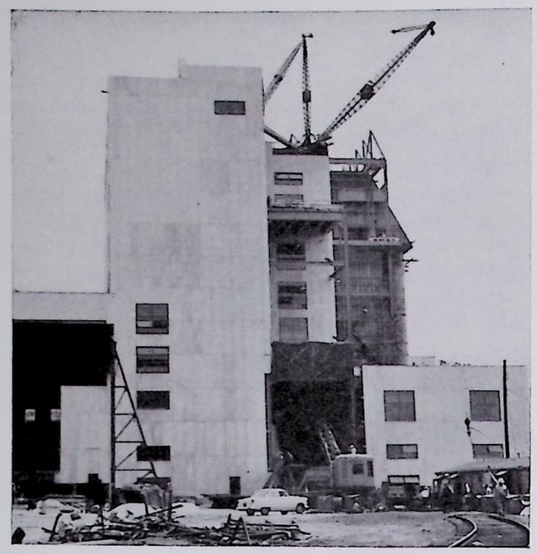


Fig. 5. *Stiff-leg derrick located on roof of building for handling boiler material*

Material Handling and Storage

When shipments from manufacturing plants arrive at the job, a crew must be on hand to unload them. The proper placing of parts and equipment in storage is essential. Most of the material is stored outdoors, with sufficient blocking to keep it out of dirt and water. Large machine components, fans, pumps, valves, and similar items may be stored outdoors if waterproof coatings are applied to protect finished surfaces. Welding electrodes, instruments, small fittings, and similar items are moved into buildings or waterproof sheds. The storage yard is divided into numbered plots, and records indicate the location of the various parts stored. A numbering system is also used for inside storage bins. Fig. 6 shows a typical material-storage yard.



Fig. 6. *Boiler unit components in field storage convenient for assembly at boiler site*

Contents of freight cars and trucks are checked against material lists. A material inventory must be made before work is started to disclose any shortages of components. This check should include minor items. Shortages must be eliminated to prevent interruption of the work. Planning is greatly simplified if all material is at the job site before erection is started. This is seldom achieved, so material arrival is usually scheduled to satisfy assembly sequences and to provide working points for the largest possible working force. There must be a great many places where men can work, with alternate places to move when difficulties arise.

Derricks, cranes, hoists, welding machines, and other tools are delivered with the early shipments of material. These are set up for use as the organization expands and as assembly operations develop.

Erection Procedure

A central station steam-generating unit usually is composed of a drum, water-cooled furnace, two-stage superheater, reheater, economizer, air heater, fuel-burning equipment, fans, flues and ducts, soot blowers, control equipment, and many other auxiliary items (see Fig. 7). All of these must be properly

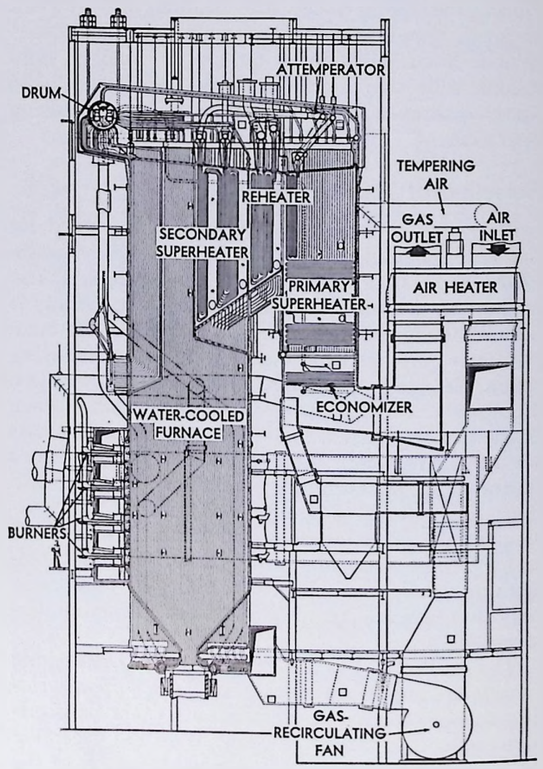


Fig. 7. *Side view of boiler unit indicating nomenclature of components*

positioned, assembled, and supported to form a complete unit. The procedure followed is of utmost importance in conserving time, manpower, and money.

Structural Supports. Erection of the steel supports is the first step in building the unit. This is done by conventional methods using cranes and derricks. Some members may be omitted temporarily to allow access for drums, tube assemblies, and other large pieces. The omission of members must not jeopardize the stability of the structure. Stairways and walkways are completed to the fullest possible extent to provide safe access to all parts of the unit.

Since many of the structural members govern the location of boiler and auxiliary components, utmost care must be taken in the alignment of supporting steel with established center lines. After the structure has been aligned and plumbed with guy wires and bracing, the members are rigidly connected with high-tensile bolts or rivets. High-tensile bolts, tightened by a procedure that insures full strength at each connection, are increasingly preferred to rivets.

Drum. The drum is the first major boiler component to be put in place after the structural steel has been erected. The drums for the largest boilers built today exceed 200 tons in weight and 100 feet in length. To clear building steel, the drum is usually raised out of level by as much as 60 degrees. Fig. 8 gives load and hoisting-angle relationship. The tackle block attached to the higher end carries considerably more than half the weight of the drum. Drum rigging must be absolutely safe, and details are worked out when tool equipment is selected for the job. Fig. 9 shows a typical drum-lifting arrangement using lugs which

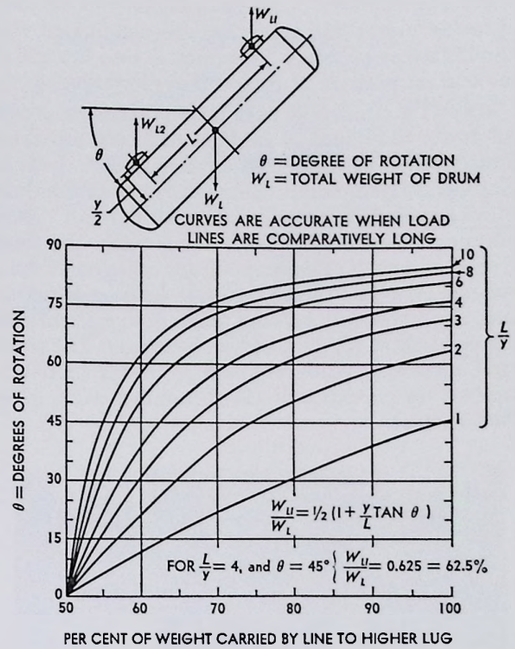


Fig. 8. *Load and hoisting-angle relationship as applied to lifting symmetrically supported drum*

were shop-welded to the drum. Attachment pieces for the top of tackle blocks are located on temporary steel members above the boiler structure. The tackle blocks are of special design to permit simple linkage connections at both top support and drum attach-

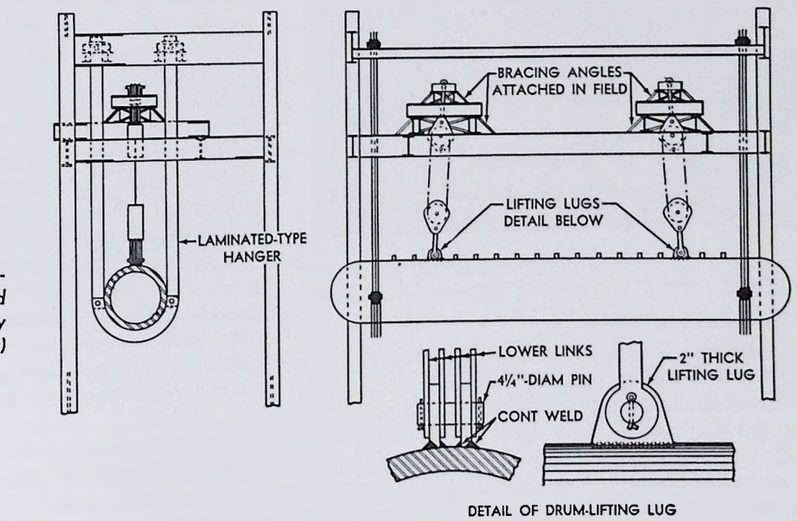


Fig. 9. *Schematic arrangement of lifting lugs and tackle for hoisting heavy drums (30 tons and over)*

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ment. The capacity of hoisting engine and the sizes of tackle blocks and wire rope are determined with ample factors of safety. The drum is thus brought to the correct position at its specified elevation.

When the drum is in final position—sometimes over 200 ft above ground—it is suspended from top structural steel. It is connected by means of U bolts or plate straps that permit linear movement of the drum with temperature variations. Fig. 10 shows a typical drum and structural-support arrangement. The drum is used as a major anchor point for other boiler components. For this reason accurate location is important, and temporary bracing prevents movement during erection of the remainder of the unit. The structural members, platforms, and stairways that were omitted for clearance are then installed to complete the structure.

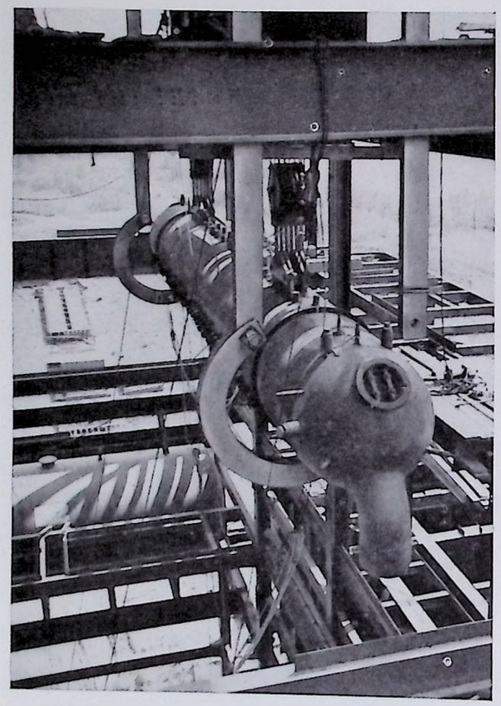


Fig. 10. *Typical arrangement of hangers and structural supports for steam drum. Drum shown is 66-in. diam, 52 ft long*

Downcomers, Headers, and Connecting Tubes. Large pipe downcomers connecting the drum with lower furnace headers are erected as access permits. These pipes are shipped in the longest possible lengths to avoid costly field welds. Large downcomers are often placed in the boiler well while steel is being erected and are moved to final location when convenient. Otherwise, downcomers must be shipped in shorter

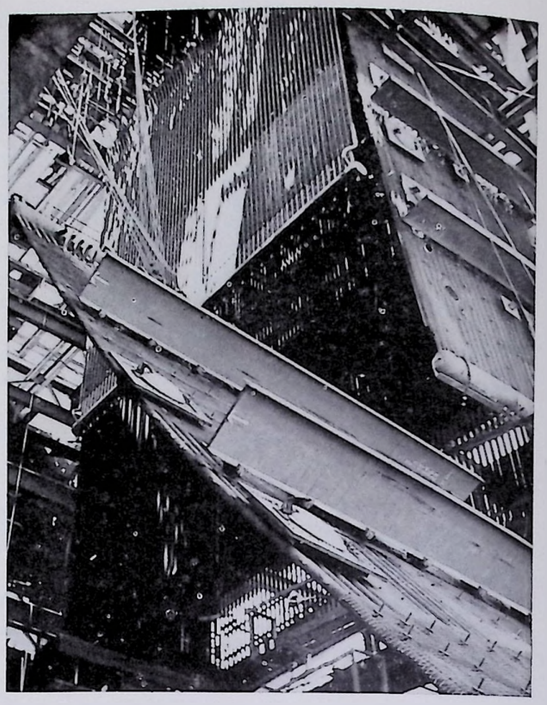


Fig. 11. *Raising furnace-wall section into position. Buckstays and other parts were attached during ground assembly*

lengths and sufficient steel omitted to give them clearance during erection.

Boiler furnace-wall, superheater, and reheater tubes are usually supported from headers to which they are connected. The headers are in turn supported from overhead steel through hanger rods attached in a manner that allows lateral expansion. Headers are braced in position by temporary struts welded to building steel. Placing of hanger rods and upper headers usually begins when the drum is raised. The upper ends of these rods rest on rocker washers, which allow them to swing without bending as the unit expands. The curved surface of the washer is oriented for that purpose.

Tubes and pipes connecting the top headers to the drum or to each other are now placed in position, while unrestricted space is available for hoisting them. During the same period, work is started on the erection of furnace-wall tubes. When all tubes for a wall are installed and aligned, exterior connections for buckstays, tie bars, doors, and soot blowers are welded to them. Superheater, reheater, and economizer surfaces are installed in the order dictated by design features of the particular unit.

To reduce cost and elapsed time of field erection, a large percentage of furnace tubes are shop-assembled into panels. These panels have sufficient tie bars

attached to hold tubes in line. Where field conditions permit, these panels may be assembled on the ground into complete wall sections with headers, casing, and buckstays attached (see Fig. 11).

Tube Connections. Tubes are connected to drums or headers either by expanding them into holes or butt welding them to stubs installed in the shops (see Fig. 12).

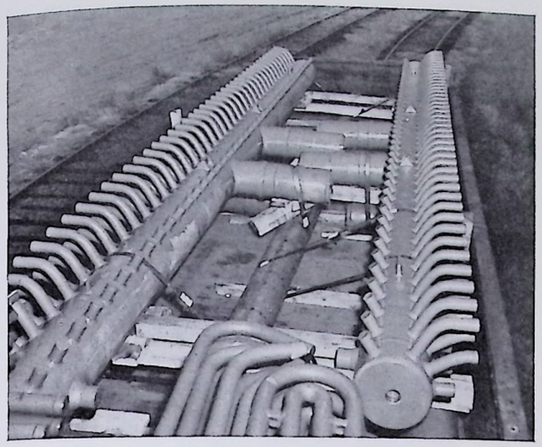


Fig. 12. *Superheater headers showing shop-welded stubs*

Design standards of The B&W Co. specify expanded joints for all low-pressure boilers. Certain special designs in this low-pressure range may call for an expanded joint reinforced by seal welding. Strength-welded stub-tube connections are specified for all boilers designed for 1200 psi and above. Similar stubs are used for all economizer headers and for all superheater headers designed to operate at 850 F or higher regardless of pressure.

If the end of a tube is inserted into a hole in a plate and is then plastically expanded by internal pressure, removal of the pressure leaves the tube bearing tightly against the plate. The result is called an expanded joint. A tool for making such a joint is shown in Fig. 13. The rolls being set at a slight angle to the body cause the tapered mandrel to feed inward when it is turned. The rolls then develop the internal pressure which expands the tube. The expanded joint presents a simple and economical way of fastening tubes into low-pressure boilers. Under direct axial loading, it is almost as strong as the tube. It does not remain fluidtight, however, when subjected to bending loads or wide temperature fluctuations.

The expanded joint can be made leakproof by a seal weld between the tube and the tube sheet. This is done on low-pressure boilers where tubes are designed for unusual loading or temperature conditions. Seal welding is done only in accordance with

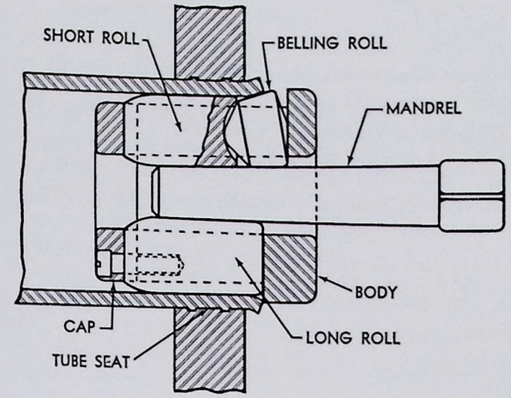


Fig. 13. *Tube expander assembly at completion of operation*

design drawings following an approved welding procedure.

Boilers designed for higher pressures and temperatures have tube connections made by strength welding. The ASME Code requires stress-relieving of strength welds. Therefore, these connections are made by welding tube stubs to the header or drum, which is then stress-relieved in a furnace. The tube stubs are prepared for field butt welding. The number of handhole openings is reduced to those required for inspection.

Field Welding. The welds that are made in erection of a steam-generating unit can be divided into two classes—those for assembly of plate and structural materials, and those for assembly of tubes, pipes, and other pressure parts.

Welding on plate and structural materials must be neat in appearance and of sufficient cross section for structural requirements. Operators performing this work must have the ability to do the job at hand, but as a rule no formal qualification tests are required. Inspection is usually limited to visual examination for defects and deficiencies in size of welds.

Welding that involves boiler pressure parts must conform with requirements of Section I of the ASME *Boiler and Pressure Vessel Code*. Paragraph P-112 of the Code limits the size of welds that may be made without stress-relief or radiographic inspection. Paragraph P-102 outlines requirements of welds exceeding the limits of paragraph P-112. The Code prescribes qualification tests for procedures and operators, as well as inspection requirements. These subjects are discussed below (see page 25-9 ff).

Welding Schedule. Welds that must be made by qualified operators and that require inspection under the ASME Code are listed on a welding schedule, which gives the following information:

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1. Description of parts to be welded.
2. Material specification.
3. Diameter and thickness of material.
4. Type (socket or butt weld) and position (horizontal or vertical).
5. Number of welds.
6. Welding process (metal-arc, oxyacetylene, tungsten inert-gas).
7. Type of electrode or filler metal.
8. Diameter of electrode or filler wire.
9. Preheat temperature.
10. Radiographic examination.
11. Stress-relief.
12. Welding-procedure specifications and process-qualification references.

Types of Welds

Butt Welds in Tubes and Pipes

Tubes and pipes are usually shipped with ends machined for butt welding and capped. Field preparation is limited to cleaning. Backing rings (see Fig. 14), made to fit accurately into a machined section of the tube end, assist in producing a welded joint with smooth internal surface and good strength characteristics. When alterations are necessary, special tools are available for remachining the welding edge and backing-ring seat. Butt welds made without backing rings are acceptable under the Code and are more desirable in the high-heat-input zones of furnace tubes. Omission of the backing ring makes alignment of the tube ends more difficult and results in a welding process which is more time-consuming. With or without a backing ring, the two ends of a joint must be thoroughly cleaned, accurately aligned, and welded in accordance with procedural specification.

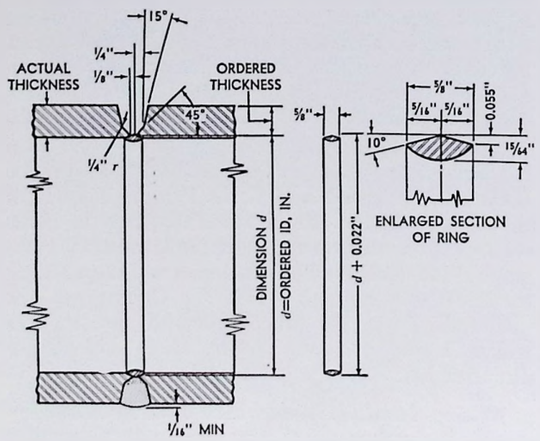


Fig. 14. *One form of backing ring frequently used for butt welds, pipes, and round-header sections*

Socket and Seal Welds

Socket welds of the type shown by Fig. 15(a) are used for joining small valves and fittings. They provide a simple connection that is easily aligned and welded. They are not suitable for use in heat-absorbing surfaces, and larger sizes of socket-weld fittings are uneconomical.

Seal welding is used to make mechanical joints fluidtight. The strength of the connection may be developed by pipe threads or by the rolled joint. It is desirable to use a counterbored fitting, Fig. 15(b), and avoid welding directly onto pipe threads. The throat of a seal weld is limited by the Code to $\frac{3}{16}$ in. minimum and $\frac{3}{8}$ in. maximum, and no thermal stress-relief is required.

Fillet Welds

A fillet weld is simply a weld between two surfaces at right angles to each other. Fillet welds used to join attachments for insulation, refractory, and casing to pressure parts must meet requirements stated in paragraph P-186 of the Code. They should

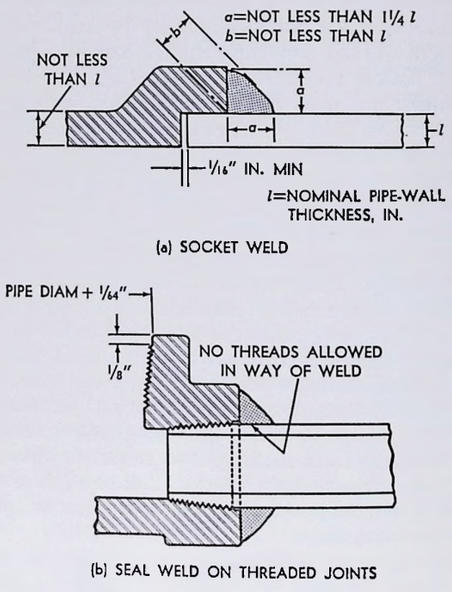


Fig. 15. *Sketch (a), typical socket weld for pipe 3 in. and under in size. Sketch (b), typical seal weld for threaded connection*

be made strictly in accordance with job drawings, which indicate size and location for adequate strength and heat transfer. Random welding to boiler pressure parts is to be avoided, particularly on headers or drums.

Welding Processes

The direct-current metal-arc process is used for most field welding, and coated electrodes are used for filler metal. Equipment and electrodes are available for an almost unlimited variety of applications. Except for the small amount of work done by other processes described below, all pressure-part welds are made by the metal-arc process.

Oxyacetylene is used only for joining certain thin-wall tubes of small diameter or in locations not accessible for metal-arc welding. The quality of weld metal deposited is slightly inferior to that of the metal-arc process, and the training of operators is more difficult. It is not recommended for use on alloy materials because of undesirable physical properties. The nature of the equipment and of the process makes oxyacetylene welding valuable for extremely close-quarters work encountered on some repair jobs.

When tubes or pipes must be butt welded without internal backing rings, the beveled ends are fused together by a tungsten arc shielded with inert gas. This is referred to as tungsten inert-gas welding. The intense localized heat fuses the thin edges and leaves a smooth internal surface almost free of oxide film. After the ends are fused, the remainder of the weld is completed with conventional arc welding.

Portable welding guns are available for attachment of pins, studs, and a variety of shapes to steel surfaces. Welding energy is supplied from a d-c welding machine. The design of the gun and a timing device make the process almost automatic, requiring a minimum of operator training. For hand application, maximum size is usually limited to the equivalent area of a ½-in.-diam stud.

Qualifications: Welding Procedures, Operators

All welding of pressure parts for power boilers must be done by qualified operators following qualified procedures. An operator qualification is proof that the operator can deposit sound weld metal in given positions, following an established procedure. A procedure qualification certifies that designated materials can be welded by following a specific technique. Section IX of the ASME Code states the variables that require additional qualifications for the operator and procedure. It also outlines tests that must be made in each case.

Although the number of process qualifications for every conceivable type of weld is almost unlimited, the number required in actual field practice is not large. Procedure qualifications cover certain groups of base or parent metals and a limited number of weld positions. Other variables, such as filler metals, welding currents, and edge preparations, are usually limited within each procedure specification. The Code requires that each manufacturer or contractor set up his own procedure specification and make tests to qualify these procedures.

In a matter of three or four working days, a competent welder can make the test samples that will qualify him for all of the welding necessary on a boiler. The sample welds for operator qualification are usually welded at the job site and sent to a laboratory for testing. If the samples pass tests outlined in Section IX of the Code, a certificate of operator qualification is issued by the manufacturer or contractor.

Thermal Stress-Relieving

Paragraph P-112(c) of the Code states the limitations of diameter, thickness, and material of welds that must be stress-relieved by heating. Paragraph P-108 specifies that the welded part shall be heated uniformly to 1100 F or higher if required by material characteristics, held at that temperature for one hour per inch of thickness, and allowed to cool slowly in still air to 600 F. In stress-relieving tubes or pipes, the width of the heated band shall be at least three times the width of the widest part of the welding groove, but in no case less than twice the width of the weld reinforcement.

The exact temperature to which a material must be heated for adequate stress-relief and the length of time at which this temperature must be held are usually stated in welding specifications, and these may be in excess of minimum Code requirements. The Code states that the parts shall be heated uniformly and cooled slowly. In general practice, it is considered safe to increase the temperature of pipe at the rate of 4000/OD deg F per hr and decrease it at the rate of 400/thickness or 3000/OD deg F per hr, whichever is lower (thickness and OD are in inches). Stress-relieving temperatures are allowed to deviate ± 25 F from specifications.

To reduce the cooling rate of deposited metal and avoid cracks in the weld, the base metal is preheated before welding. The preheat temperature will vary with the type of material and thickness, as called for on the procedure specification.

The method of heating for preheat and stress-relief is largely a matter of economics. There are several ways in which pipe can be heated, and it is only important that it be heated uniformly and that the temperature be properly controlled. The following paragraphs describe the methods usually employed

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in field erection. The references to stress-relieving equipment also apply to preheating.

Flame Heating

The simplest way to heat a weld to stress-relieving temperature is with an open torch. This method does not give the exact control of temperature that is sometimes required. When properly used, torches fired with butane or natural gas will produce uniform heating of small tubes and pipes. The extremely high-temperature flame of an oxyacetylene torch may result in severe overheating in spots and, therefore, is not generally used for stress-relieving.

A muffle furnace (see Fig. 16) fired with gas or oil will produce uniform heating with simple equipment. Such furnaces are easy to make and are widely used where natural gas is readily available. The disadvantages of muffle-furnace heating are: a) space required around the pipe, b) lack of flexibility for clearing external connections on the pipe, and c) the careful operation required to obtain uniform heating.

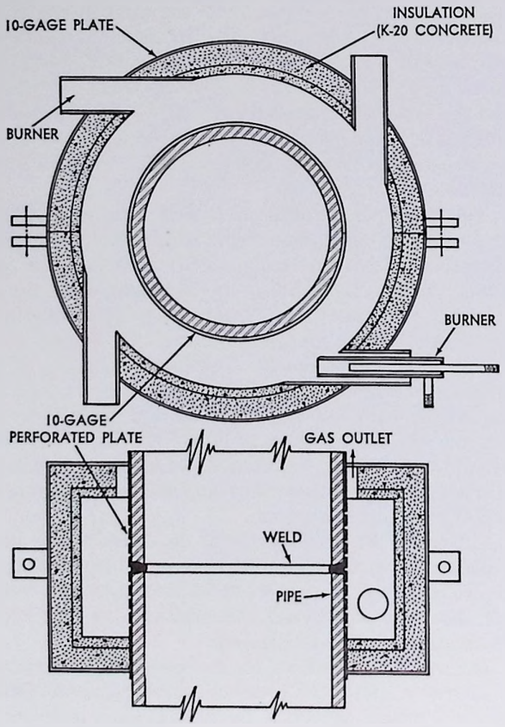


Fig. 16. *Application of a gas-fired muffle furnace to stress-relieve the zone of and adjacent to a butt weld*

Electrical-Resistance Heating

Several types of electrical-resistance heaters can be adapted to stress-relief work. Where a large number of small tubes is involved, commercial heating elements can be bent to shape and fitted into canisters (see Fig. 17). This arrangement is convenient for heating tubes 2½ in. and less in diameter to temperatures under 1375 F.

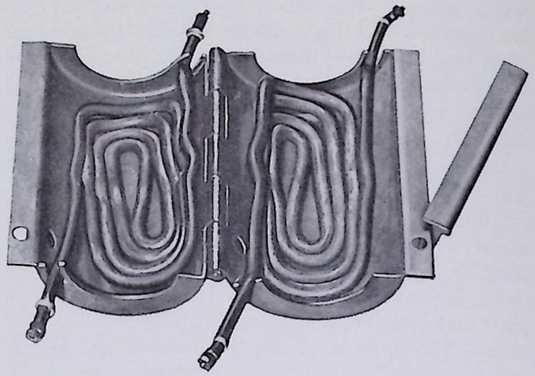


Fig. 17. *Electrical-resistance heating elements in hinged canister for quick application to stress-relief and preheat*

One of the resistance heaters widely used on larger tubes or pipe is the wrap-around type of spiral-wound alloy wire covered with short porcelain insulators. Heaters of this type usually operate on 220 volts and are made in capacities from 3 to 24 kw. One or more such heaters may be used for stress-relieving a weld. To reduce heat loss, insulation is applied over the outside of coils after they have been wrapped around the pipe. These heaters can also be mounted in hinged canisters for application to a given size of pipe.

Wrap-around resistance heaters have a low initial cost, and electrical connections are fairly simple. They can be adapted to a variety of work and will produce uniform temperatures that can be controlled with exactness. On the other hand, deterioration is fairly rapid when used on large thick pipe or at temperatures above 1200 F. Other types of heaters are more rugged and economical on large volumes of work.

Electrical-Induction Heating

If an insulated conductor is wrapped around a pipe and attached to a source of alternating current, the pipe becomes the secondary circuit of a transformer system. The currents induced in the pipe manifest themselves as heat. The analysis of this phenomenon is quite complex, but the practical application to heating of pipe is simple. For safety and convenience, 60-cycle current at 20 to 90 volts is obtained through a variable-voltage transformer. The conductor is a

copper cable covered with asbestos sleeving. The size of conductor must be selected to induce the required current at available voltages. A layer of heat insulation is applied to the joint to be stress-relieved, and the conductor is wrapped over the insulation. No insulation is applied over the induction coil, which remains fairly cool. The amount of heat applied is controlled by varying the transformer voltage.

Equipment required for induction heating (see Fig. 18) includes the transformer, copper cable for the induction coils, and rubber-covered cables for connecting the coils to the transformer. Transformers and cables can be connected in multiple circuits so that standardized equipment can be used for a variety of work sizes. The initial cost of equipment is high, but it is very rugged. Since heat is generated within the body of metal, the induction process is particularly valuable for heating large heavy-wall pipe to a uniform temperature.

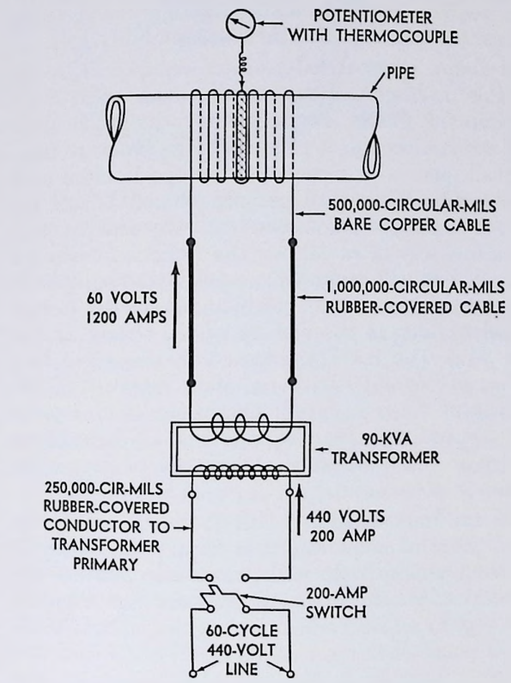


Fig. 18. *Application of high-capacity electrical-induction heating equipment for stress-relief and preheat of large thick-walled pipe*

Determination of Temperature

In preheating and stress-relieving, the temperature is usually measured with thermocouples and a potentiometer. Recording potentiometers can be used if permanent records of preheat and stress-relief temperatures are required. For both resistance and induction heating, commercial equipment is available for controlling the rate of increase or decrease in temperature.

Crayons are sometimes used for the rapid determination of surface temperatures. Each crayon is marked for the temperature at or above which it will leave a liquid smear on the heated surface. They are available for the range of 125 to 2000 F in increments of 50 F.

Codes and Inspections

ASME Code

Boilers built by The B&W Co. and most other boilers built in the U.S. comply with the ASME *Boiler and Pressure Vessel Code*. The rules of this Code for the safe construction of boilers have no legal status until adopted by a state, municipality, or other legally responsible body. To a considerable extent the Code is administered by the National Board of Boiler and Pressure Vessel Inspectors, which is made up of the chief inspectors of the states and cities that have legally adopted the Code.

Although the Code covers new boiler construction, there are many types of repairs which it does not cover. The National Board has formulated a *National Board Inspection Code*, which makes suggestions for uniform laws pertaining to inspection and includes *Rules for Repairs by Fusion Welding*. The National Board Rules, like the ASME Code, have legal status only when properly authenticated by governmental authority. The adoption of both the Rules and the Code has been general throughout the country, and, with few exceptions, they are the bases for all legal requirements.

The National Bureau of Casualty and Surety Underwriters issues a booklet, *Synopsis of Boiler Rules and Regulations*, listing the cities and states that have laws covering installation and operation of steam boilers. Other pertinent information includes names of administrative officials, fees, licenses, and inspections required.

To comply with the Code, boilers must be erected by or under the supervision of an organization authorized to use the ASME power boiler stamp or the ASME assembly stamp. The requirements for authority to use these stamps are given in the Code.

Inspection Requirement

A boiler manufactured and erected in accordance with the Code must be inspected at certain stages by an authorized inspector. The term "authorized inspector" refers to a person licensed by the state or city in which the boiler is erected and who has a National Board commission. Much of the inspection work is done by insurance companies, whose inspectors comply with these requirements.

When the inspector is satisfied that construction complies with the Code, he will sign data sheets per-

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taining to the unit and witness the stamping of the boiler pressure parts. All stamping required by the Code is usually done in the shops. Data reports pertaining to shop fabrication are signed by the inspector who followed that part of the work. The inspector who covers field erection signs data reports pertaining to assembly and witnesses any stamping required by governmental authority. The field inspector must be able to certify that all welding, as well as construction and materials, conforms to the Code. He is able to do this largely on certification of the manufacturer or contractor, whom the Code makes responsible for quality of welding.

The inspector must be given an opportunity to assure himself that welding operators and welding procedures have been properly qualified. He must also have an opportunity to inspect the welding in progress and satisfy himself as to its quality. The number, time, and duration of these inspections are usually determined by mutual agreement between the inspection agency and the manufacturer before the job is started. The authorized inspector must witness the hydrostatic test on the unit, and field data sheets are usually signed at that time.

Further inspections are required by the manufacturer or contractor to assure him that all workmanship meets his standard of quality. Inspections for the erecting organization are made by the job superintendent, welding technicians, foremen, and the welding operators, sometimes in company with the authorized inspector. Frequent and sometimes continuous inspections are made to insure that:

1. Each operator is qualified under the Code for the welds he makes.
2. Procedure specifications are rigorously followed.
3. Each joint is accurately aligned and fitted before welding is started.
4. Each weld bead is cleaned of slag and the weld is free of slag porosity and undercutting.
5. The finished weld is of correct contour without undercutting or overlapping.

Nondestructive Tests

Radiographic Inspection

Radiography makes use of short-wave radiations, such as X rays or gamma rays, to penetrate opaque substances. As applied to a weld in pipe, part of the rays will penetrate the pipe and some will be absorbed, depending upon the thickness and density of metal. If defects or discontinuities exist, there will be a variation in the absorption of the rays. When these variations are recorded on film, they produce an image or X-ray shadow picture of the interior defect. Such a shadow picture is called a radiograph.

Paragraph P-112 of the Code requires fusion-welded circumferential joints in pipes, tubes, and headers to be radiographed when their dimensions exceed those given in 1, 2, and 3 below:

1. 16-in. nominal pipe size or 1⅝-in. wall thickness for parts to contain steam, and 10-in. pipe size or 1⅛-in. wall thickness for parts to contain water. In each case the welds shall not be in contact with furnace gases.
2. 6-in. nominal pipe size or ¾-in. wall thickness when welds are to be in contact with furnace gases but not subject to radiation from the furnace.
3. 4-in. OD or ½-in. wall thickness when welds are to be in contact with furnace gases and subjected to radiation from the furnace.
4. The diameter and thickness limitations apply independently.

Paragraph P-102(h) of the Code outlines procedure requirements for making radiographs of welds. It also sets standards for their acceptability.

In shops where fixed-position equipment is applicable, radiographs are usually made with X-ray apparatus. A gamma-ray source is more practical for field work, although it requires more exposure time. A small piece of a radioactive isotope is often used as a source. These isotopes are obtained from the Atomic Energy Commission and are ordered to a given intensity of radiation. The radiation desired is such that it will make the required radiographs in a reasonable length of time without danger to personnel working in the vicinity of the operation. For field work The B&W Co. uses ⅛-in.-diam ⅛-in. long pellets of cobalt-60 with radiation intensity of 500 millicuries when received. One piece of this material, suspended in the center of a 24-in.-diam 1⅛-in.-wall pipe will produce a radiograph of proper exposure in 45 minutes.

To facilitate handling and to avoid losing the small piece of cobalt-60, it is kept permanently in an aluminum capsule with cords attached for manipulation. When not being used for film exposure it is kept in a lead container weighing about 200 lb. At all times when the capsule is out of its lead container it must be in the care of a technician who has been thoroughly trained in radiography, including procedures for protecting himself and others from the harmful effects of the radioactive source. Adequate barriers, shields, and warning signals must be set up when the capsule is in use, to protect those in the vicinity.

In making exposures with gamma rays, two films are generally used in each film holder, or cassette, which is placed on the side of the plate or pipe opposite to the radiation source (see Fig. 19). Both films are studied to distinguish details in the picture

of the weld from minor film-process markings. A defect, being less dense than the remainder of the weld, will permit more radiation to pass through, thus causing a darker image to appear on the film. The Code requires that welds shall be radiographed with a technique that will determine size of defects with thickness equal to or greater than 2% of the base metal. In order to prove the effectiveness of the technique, a metal gage equal in thickness to 2% of that of the base metal is included in the picture. Use of this gage, known as a penetrameter, is also a Code requirement.

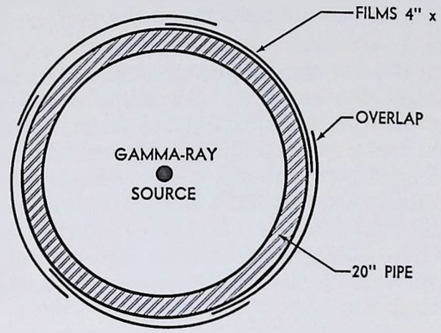


Fig. 19. *Position of gamma-ray source relative to film and material to be examined*

The more obvious weld defects can be seen on a radiograph without difficulty. The detection of small cracks and slag inclusions requires the services of a technician with considerable training and experience. The man who makes the radiographs on the job is usually qualified to interpret them. Defects revealed by the film are chipped out and repaired. Subsequent pictures are made until the welds are acceptable. The films are then reviewed by a technician at the manufacturing headquarters of The B&W Co. They are finally accepted at the job site by the authorized inspector representing the state. Films are held for a period of ten years after the boiler is completed.

Magnetic-Particle Inspection

If an electric current is passed between two points on a plate, it will set up a magnetic field at right angles to the path of the current. A crack or other defect located on or near the surface will cause a leakage field, forming local poles along the edges of the defect. Magnetic powder, sprinkled on the surface while the current is flowing, will be attracted to such locations and clearly outline the defect. This process is known as magnetic-particle inspection.

In welding heavy-wall pipe there is always the possibility that small surface cracks may develop, which cannot be detected by visual or radiographic examination. Although it is not a Code requirement, it is desirable that welds of sufficient importance for radiographic examination also be examined by magnetic-particle inspection.

In its simplest form, the equipment required for this process is a source of low-voltage current, such as a 300-amp welding machine, two conductors with copper electrodes, and a supply of magnetic powder. To prevent arc burns, a circuit breaker is installed in series with one electrode.

The weld surface must be dry, and it should be ground and polished before inspection is begun. The copper electrodes are placed in good contact with the metal, one on each side of the weld. The circuit is closed preferably by a push button mounted in a fixture that holds the two electrodes at proper spacing. A light layer of powder is dusted on, and excess powder is blown off. Any defects noted are marked for repairs. The current is interrupted before the electrodes are moved. By moving them in an oblique pattern, as shown in Fig. 20, the entire weld can be examined with a minimum number of moves.

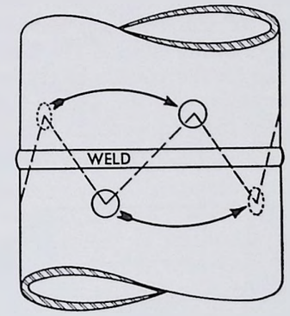


Fig. 20. *Positioning of electrodes oblique to weld for optimum detection of weld defects on or near the surface with magnetic-particle inspection*

For optimum results with this type of inspection, the operator must be trained by conducting tests on sample welds with known defects. A procedure must be established that can be duplicated on production welds.

Handhole and Manhole Fittings

Handholes and manholes are provided in boiler components for access to tubes that must be expanded during erection and for subsequent inspection, maintenance, cleaning, and repairs. Gasketed handhole fittings are used in headers designed for pressures and temperatures up to and including 1200 psi and 850 F, respectively (see Fig. 21). For higher pressures and temperatures a welded cup-type closure is used (see Fig. 22).

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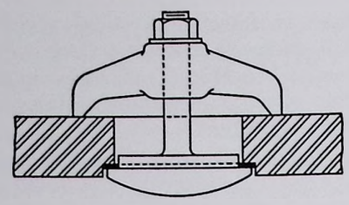


Fig. 21. *Gasketed handhole fitting used to close header openings under conditions of moderate pressure and temperature*

When gasketed fittings are installed, it is important to have all surfaces clean and free of burrs. The studs, nuts, and washers should be lubricated with a mixture of graphite and kerosene. Handhole fittings should be tightened initially by applying a torque of approximately 300 ft-lb. It is desirable to retighten handhole fittings during hydrostatic test of the boiler, applying the same torque as initially used.

Welded-type handhole fittings are designed to enter the header through a master opening. The master cap is of similar design but shaped to enter the header through the hole that it closes. In each case the shoulder of the cap, which seats on a machined surface of the opening, provides the mechanical strength of the closure. The ⅜-in.-throat seal weld of the cap on the outside of the header maintains fluid tightness of the joint. Headers with welded stubs for tube connections usually have one master handhole at each end for inspection.

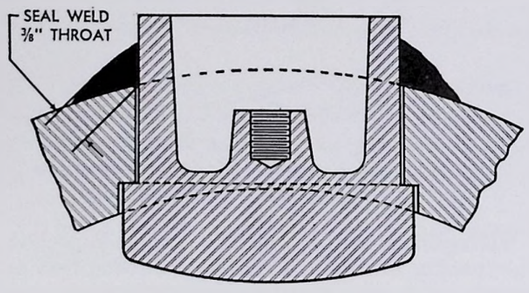


Fig. 22. *Welded handhole fitting used to close header openings in higher-pressure boilers*

Manhole openings in drums are either circular or elliptical, depending upon design conditions. Woven gaskets are used on manhole covers for units of design pressure up to 500 psi, metallic gaskets for higher pressures. Woven gaskets are coated with a mixture of graphite and kerosene; metal gaskets are installed dry. Manhole covers are usually attached to the drum with a hinge bar. Although the covers are interchangeable, the outside yokes are fitted to the contour of the drum. The cover, yokes, and drumhead are match-marked to insure that all parts will be replaced as originally fitted. Manhole studs should be tightened when installed and again under hydrostatic test pressure, with a torque of about 400 ft-lb.

Assembly of Flanged Joints

The general precautions taken in the installation of handhole and manhole fittings are followed in the assembly of flanged joints. However, nothing is gained by tightening a flanged joint under pressure. Sufficient and uniform tightening of flange studs is of primary importance. A feeler gage can be used to check the alignment of mating surfaces and to obtain uniform pressure on the gasket.

Stress in the studs of a flanged joint must be sufficient to resist internal pressure and exert a bearing load on the gasket as determined by design. Flanged joints on B&W boilers are designed for an optimum stud stress of 30,000 psi. The stress in studs may be measured as they are tightened, by checking the elongation. An increase of 0.001 in. per in. of stud length measured between centers of nuts will produce 30,000 psi stress. This increase may be measured with a dial indicator mounted in a frame, as shown in Fig. 23.

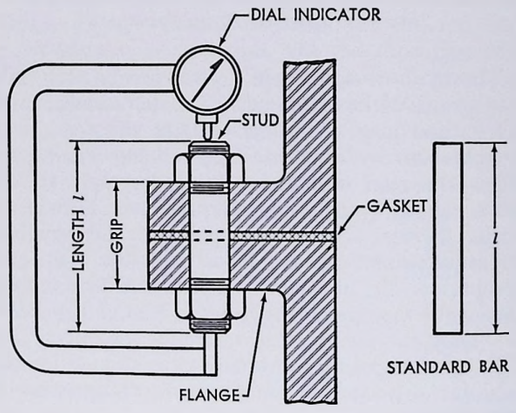


Fig. 23. *Dial-indicator type of calipers used for measuring elongation in flange studs*

If the surfaces of threads and nuts are smooth and uniformly lubricated, a relationship between torque and stress can be determined experimentally. This will help to save time in checking stress, particularly when a great many studs are involved.

Internal Cleanliness of Pressure Parts

Many problems can develop during operation because of dirt or other foreign matter in the boiler circulating system. During the fabrication of drums, tubes, headers, pipes, or other pressure parts, the

shops take steps to insure that they will be clean internally. For boilers of welded-stub construction, the ends of all tubes and pipe as well as the stubs in drums and headers are closed with metal caps and sealed with plastic tape. This makes it unnecessary to clean these components in the field. Inevitably there will be a few closure caps displaced during unloading and handling. In such instances the erection crew must examine the components involved and replace the closures at the earliest possible convenience. During assembly, care must be taken that no foreign material enters the pressure-part system.

For boilers with expanded joints, caps are installed only on tubes that might become filled with water during outdoor storage. The pressure parts of such boilers are searched before handhole and manhole closure fittings are installed. Reasonably straight tubes between headers or drums are probed by passing a ball of correct size through them. Bent tubes which would not clear themselves of loose foreign material by gravity are probed by blowing a tight-fitting sponge through them. Headers are checked with a light and mirror immediately before the extreme end handhole fittings are installed.

The use of oil on water surfaces of boilers must be avoided. A water-soluble lubricant is used on tube expanders and tools for machining tube ends.

Hydrostatic Testing

When all pressure-part connections have been completed, the boiler is tested at 1½ times the design pressure. Before the test is applied, a final inspection is made of all welding. External connections are completed, including all fittings within Code requirements. Connections for flanged safety valves are usually blanked. Welded safety valves are closed with an internal plug furnished by the valve manufacturer.

Water for hydrostatic testing must be clean and at a temperature of not less than 70 F nor greater than 100 F. Condensation on tube surfaces makes inspection difficult when the water is colder than the ambient temperature. Nondrainable superheaters and reheaters are filled with demineralized water or condensate, if available. As the boiler is being filled it is vented at every available connection, to allow water to fill every circuit.

After the boiler is filled, hydrostatic pressure is applied, usually by a pump made for testing purposes. If a boiler feed or other high-capacity pump is used, precautions must be taken to keep the pressure under control at all times. While water is being pumped into the boiler, continuous inspection is made to observe any leakage conditions that might need repair or become dangerous. During the hydrostatic testing of any vessel, safety precautions must be observed and nonessential personnel kept from the test area.

Test pressure must be held long enough to allow minute leaks to be observed. Six to eight hours are usually considered sufficient for a high-capacity boiler. Following this holding period, pressure is retained for an inspection period. During final inspection every area of the pressure system is viewed for leakage and repaired if necessary, to make the unit 100% tight.

Following inspection of the unit it is drained, and all nondrainable parts are protected from freezing if that possibility exists. The refractory, insulation, and casing work are completed in areas left open for test inspection.

Assembly of Nonpressure Parts and Auxiliaries

While pressure parts are being assembled and tested, work is in progress on nonpressure parts and auxiliary equipment. These include air heaters, fuel equipment, fans, duct work, refractory, insulation, and casing. These components require a large part of the manpower used on the unit. They must be scheduled to allow for access of large pieces and to take care of items that must be assembled in a definite sequence. Thus, air heaters are finished in time to allow for erection of flues and ducts. Pulverizers are located early enough for piping, drives, and other auxiliaries to be completed.

Refractory Materials

The furnace and other wall areas of high-duty boilers are made almost entirely of water-cooled tubes. In order to control burner performance and slagging characteristics, refractory materials processed from chrome ore are applied to wall areas in furnaces. This material is applied to areas having studs welded to the tubes, and the contour of the refractory follows that of the studded surface (see Fig. 24). It is available as plastic chrome ore for hand application with wooden mallets or as Kro-might-Gun for application with gunniting equipment.

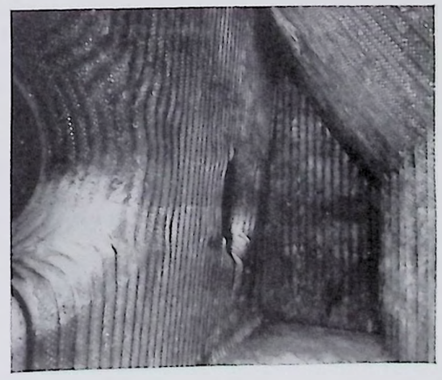


Fig. 24. *Application of plastic chrome ore to furnace wall, showing contour of studs welded to tubes*

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One of several types of plastic refractory materials is applied to the outside of tubes for insulation or sealing purposes. When inner casing is to be applied directly against the tubes, the refractory serves principally as an inert filler material for the lanes between tangent tubes. It has a binder which cements it to the tubes, and it is troweled flush with the surface of the wall (see Fig. 25). When casing is to be applied as a jacket or enclosure over an area of tubes, the refractory serves as a layer of sealing material against air leakage. For this arrangement it is generally applied in thicknesses of 1 to 1½ in. over expanded metal lath fastened to the tubes by studs welded on about 18-in. centers.

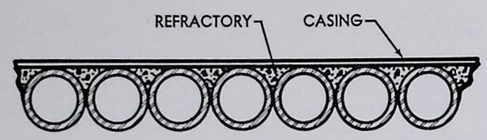


Fig. 25. *Sketch showing application of refractory to close-spaced tubes with inner casing*

Smaller boilers have furnaces constructed of tubes on wide-spaced centers backed with a layer of brick or tile. Brickwork of this type is supported by the pressure parts and held in place by studs welded to the tubes. The brick is insulated and made airtight by various combinations of plastic refractory, plastic or block insulation, and casing.

The principal point of concern in the application of refractory materials is to obtain high-quality workmanship. Details of construction are clearly outlined on drawings and instructions, which also designate the exact materials to be used. These materials must be applied to correct contour and thickness, without voids or excessive cracking. Skilled mechanics and close supervision are essential.

Insulation

The selection of type and thickness of insulation applied to surfaces of a boiler unit is an essential part of the design work. Specifications are written and drawings made that indicate the materials to be applied in each location and the method of attachment.

Insulation standards for B&W boilers specify either insulation blocks or insulation blankets reinforced with metal lath or wire fabric. Some plastic insulation is used for irregular surfaces. The prime requisites for all boiler insulation are that the application be tight, free of voids, well anchored, and reinforced where necessary. The finished job must present a neat workmanlike appearance.

Expansion of insulation with temperature is negligible in comparison with expansion of boiler components. In order to prevent large gaps from developing in the applied insulation, anchors must be provided at relatively close intervals. The blocks are held to these anchors by means of lacing wires (see Fig. 26). As the boiler expands, the space that develops between blocks is minute and does not tend to accumulate in one location.

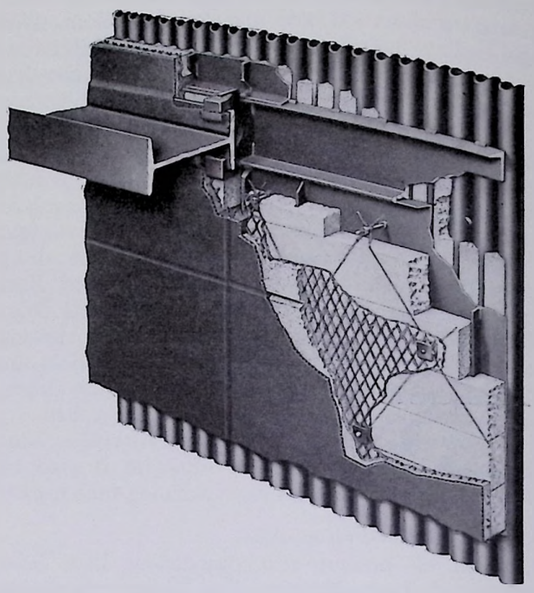


Fig. 26. *Section of furnace wall showing inner casing, insulation, and hard finish*

Ductwork with heavy stiffeners at frequent intervals presents a difficult insulating job. This problem is usually solved by welding steel reinforcing material over the outstanding ribs. Road mesh made of 8-gage wire welded into 4-in. squares is frequently used. When insulation is applied at an appreciable distance from a hot surface with a long vertical run, there is a tendency for circulation of air to occur in the void. This increases heat transfer into the insulation, which is undesirable. Where vertical runs exceed 10 ft, horizontal barriers are installed between the ductwork and insulation.

When insulation is applied on the outside of flues and ducts, some type of finish is needed for protection from erosion as well as for appearance. In some instances a plastic coating is applied, but the trend is toward sheet-metal lagging.

Casing

The primary purpose of casing on a steam-generating unit is to prevent air leakage. It is referred to as inner casing (Figs. 25 & 26) when it is installed in close contact with the pressure parts. So-called outer casing is installed outside of refractory and insulating materials.

The flanges of inner-casing panels are welded to horizontal channel stringers welded to the wall tubes on about 5-ft vertical centers. Various design details are used for irregular surfaces, but the objective is to keep the casing in as close contact with saturated pressure parts as possible. When the boiler goes into operation, the casing will expand with the tubes without the need of slip joints. The flexibility of casing flanges and corrugated panels takes care of temperature lag between casing and tubes.

When inner casing is applied to furnaces designed to operate under positive pressure, it is known as pressure casing. For this type of operation, air testing of the entire unit is required during construction to assure complete welding, with leakage brought to an irreducible minimum. The air test is performed by blanking the air and gas ducts and pumping the boiler setting to a predetermined pressure. A check of pressure drop within a given time interval will determine acceptability of the casing.

If the unit is to operate under furnace suction, the design of door and soot blower openings and other penetrations can be greatly simplified, at the expense of an extremely small air infiltration to the unit.

Insulation is applied over the outside of inner casing and is usually protected by sheet-metal lagging. All B&W boilers designed for capacities above 100,000 lb of steam per hr have inner casing.

For smaller boilers that are equipped with neither inner casing nor membrane walls, external casing is used over refractory and insulation that has been applied directly against tubes and other pressure parts. The casing will operate at much lower temperatures than the tubes and must have a number of expansion and slip joints to compensate for temperature differentials. This type of casing can be applied to boiler and furnace walls with tubes on wide spacing backed with tile or brickwork and insulation. Since external casing need not be closely fitted to contours of tubes, it can be installed more easily than pressure casing. The differential expansion requirements, however, present design difficulties on high-duty boilers.

Almost without exception, boiler casings today are of welded construction. It is essential that the strength of the weld equal the strength of the plate, that the welds be free of cracks and pinholes, and that they present a good appearance. With pressure casing, all seams are checked for air leakage under pressure by brushing them with soap solution. Nonpressure and external casings are given a thorough visual inspection.

Mechanical Equipment

The field installation of fans, stokers, pulverizers, feeders, and their driving motors follows standard methods for setting and alignment of machinery. In most cases equipment manufacturers furnish special instructions that apply to the alignment of bearings, couplings, and other parts, on each specific piece of apparatus.

Principal points of concern in erecting mechanical equipment are: 1) accurate location to center lines and elevation, 2) cleanliness and alignment of bearings, 3) alignment of couplings, and 4) homogeneous grouting between base plates and concrete foundations.

There are many handbooks which cover these general subjects, and lack of information on procedure is seldom a matter of concern. Of more importance is the assignment of mechanics who will follow instructions in complete detail, with close supervision to assure compliance with instructions.

Preoperation Inspection

After erection work is completed and the unit is ready for operation, a final over-all inspection is made. Preferably this is a joint inspection by representatives of the erection and service, or operating, organizations. Internal cavities of the boiler are checked to see that all debris has been removed. Of special importance is the removal of combustible materials that might present an explosion hazard during initial firing.

Tubes are given a final inspection for alignment, particularly superheater tubes which might interfere with soot blower operations. Movable connections between tubes are examined for expansion clearances and to see that all pin-and-clevis attachments are properly anchored.

Externally all components of the unit are checked for expansion clearances. Obscure corners are examined for any construction blocking or bracing that might have been left in place. Points of particular concern for expansion clearances are platforms and walkways constructed adjacent to external members that move with the boiler.

On most units construction schedules do not permit completion of all insulation before the unit goes into operation. It is important that work be completed on areas that will be inaccessible with the unit in operation.

Tools and Equipment

The extensive erection and maintenance programs carried on by The B&W Co. require a considerable inventory of tools and equipment. This includes everything usually associated with construction work, from small hand tools to air compressors, large hoist-

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ing engines, and heavy mobile cranes. A warehouse is located in Barberton, in which is housed a maintenance and repair facility, adequate storage space for equipment held in readiness for prompt shipment to jobs, and a machine shop for the development of tools unique to boiler erection or maintenance. In many instances these special tools are not available from any other commercial source of supply. Some specific examples of this category include tools for refacing handhole and manhole gasket surfaces, removing seal welds from tubes and handhole closures, and preparing tubes for welding.

Trained personnel select and schedule the shipment of tools and equipment to jobs, and records are kept that permit the identification of pressure parts for boilers of any age or type. This, together with the fact that all special tools are retained in stock even when they may be considered obsolete by modern standards, makes possible the repair of any B&W boiler ever built.

26

RESEARCH AND DEVELOPMENT

RESEARCH per se is instinctive in man to increase his knowledge. It is an essential factor in progress, which accounts for its broad use in practically all phases of human endeavor. Knowledge is the ultimate goal of all research. It is the foundation for the development of processes and products beneficial to mankind. Research represents the concentration of intellectual and practical creative ability to the solution of specific problems.

In the early days man utilized his inventive genius primarily to stay alive. Examples of fundamental importance among the early inventions are the wheel and the plow. With the advancement of civilization as a result of invention, the needs of man increased, and consequently inventions became more numerous and more complex.

Development is the natural corollary of invention. It results from keen observation and insight into natural phenomena. Diligent effort and much experimentation are necessary before most inventions are brought to practical usefulness. Often this development is achieved through considerable cost to those with faith in the ultimate outcome and is successfully completed only because of their determination to overcome the many obstacles present.

Engineering Research

Engineering research had its origin in the application of inventions to the improvement of military status. It was not until the middle of the eighteenth century that attention was diverted to peaceful applications. Since then, the technical advances of industry have exerted an increasing influence on social development. Today, technological progress affects the welfare and destiny of mankind throughout the world.

Research is defined by Webster as "critical and exhaustive investigation or experimentation having for its aims the discovery of new facts and their correct interpretation, the revision of accepted conclusions, theories, or laws, in the light of newly discovered facts, or the practical applications of such new or revised conclusions."

Research may be divided into three broad categories: 1) pure research, 2) background research, and 3) applied research and development. The boundaries between these classifications are neither simply nor sharply defined. Study, however, reveals that each category has a distinct function and requires different treatment for maximum results.

Pure research, largely performed in universities and government facilities, advances general knowledge and understanding of natural laws. The results, with subsequent applied research and development, improve existing industries or create new processes.

In background research, accurate data on scientific development are prepared and compiled for general reference. This information is used extensively in both pure and applied research to correlate previously collected data that are pertinent to the particular field under investigation.

Industry is primarily interested in applied research and development. This differs from pure research in several important respects. The aims of applied research are normally immediate and explicit. Generally the work can be planned and organized, characteristics not present with pure research. The expense involved in applied research and development is heavy, but not as high as in pure research. Since industry in general must measure the expense in terms of results, the economic justification for continuing the effort in any particular field must be evaluated carefully at all times.

Purposes of Industrial Research and Development

The recognized purposes of industrial research and development are:

1. To improve the quality of products.
2. To develop new materials, processes, or devices for existing or new markets.

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3. To develop new uses for existing materials, processes, or devices.
4. To abate dangers or nuisances.
5. To conserve natural resources.
6. To effect savings in cost of production or use.
7. To prevent or cure troubles of production or use.
8. To assist in standardization.
9. To improve customer or public relations.
10. To render special or unique service to the country in time of national crisis.

In the case of steam generation, the field for applied research and development covers all phases of the utilization of any and all fuels for the production of heat, the containment and flow of fuels, the transfer of heat to working media at high pressures and temperatures, the discovery of new materials, and the improvement of existing materials of construction and methods of manufacture.

The frontiers of this field of endeavor are constantly advancing, and each step brings on new and more difficult problems. The solution of these problems requires the skillful use of scientific and technical knowledge in coordinated efforts by qualified personnel of widely varied training and interest. Skilled personnel, adequate facilities and instrumentation, planned and orderly procedures, and suitable working space are all essential for optimum results in research and development work (Fig. 1).

Projects Proposed for Research

There is no dearth of ideas for research projects. Proposals emanate from all departments of the Company, from customer organizations, from related industries, and from fields of basic science. The economic potentialities of all such projects and the available facilities and personnel for handling them satisfactorily must be evaluated. For each research project selected, organization and control are necessary to assure the procedure most beneficial to its success. However, serious restrictions on individual enterprise and imagination must not be imposed.

Many of the proposals can be appraised by analytical study, calculation, and previous experience. Sometimes an idea can be tested by a small-scale laboratory demonstration to prove or disprove the validity of its essential features. Where promising results are obtained, a further trial in pilot-scale equipment may be justified to coordinate the additional factors of size, timing, and practical implementation.

It is important in any research investigation to make systematic observations and to keep careful records. Through the correlation of the data obtained, the potential value of the project or constructive ideas for the solution of some of the difficulties may be revealed. Furthermore, new avenues of approach to other methods or apparatus may be suggested, which may prove to be equal or superior to those initially followed. Even negative results can be of value in avoiding an expensive large-scale application of an intriguing, but false, premise.

Unit Cost of Power Reduced

During the past half century, developments in the electrical industry have resulted in a consistently lower cost of power to the consumer. This reduction in cost can be attributed almost entirely to improvements in the technology associated with power generation, to which the boiler industry has made many important contributions, such as: the mechanical firing of fuel to eliminate hand labor, heat traps to improve boiler efficiency, water-cooled furnace walls to reduce maintenance and to overcome unit-size limitations, and the suspension burning of pulverized coal to increase the steaming capacity of the unit. Supplementing these developments, meters and automatic-control equipment have had a marked influence on performance because of their use in obtaining improved efficiency in the combustion of fuels.

During this same period there was great progress in the use of superheated steam. Steam temperatures and pressures steadily increased as a result of the knowledge from previous experience and from the development of new materials of construction, welding methods, steam purification, and water chemistry. At present, alloy materials are being systematically explored to learn more about their use limits, so that their properties can be utilized more efficiently and more economically in the construction of boilers and other pressure vessels. A variety of alloy compositions is being studied to develop new compositions that will further extend the limits of operating temperatures and pressures. By such means, improvements in steam-cycle efficiency are realized, and the unit costs of power production are further reduced.

All of these advances required the solution of difficult and, at times, frustrating problems during their development. Prejudices and misconceptions from lack of knowledge and experience frequently resulted in obstructive or abortive modifications. However, the great gains based on fundamentally sound ideas have survived the rigorous test of time and point the way to further advancement.

Continuous Research Necessary

There is still much to be learned concerning the constitution and strength of all materials of construction, particularly metals; the effect of trace amounts of certain elements in alloy compositions; the influence of steel-mill processing and shop-fabricating procedures on the microstructure of materials; and the changes brought about over long periods of time by temperature, stress, and contacting fluids. Direct measurement of creep and stress-rupture properties (Fig. 2) of existing materials and more rigorous anal-

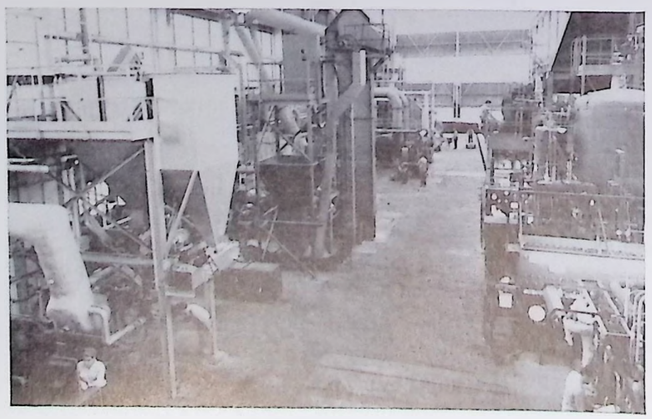


Fig. 1. *Test area for pilot-plant experiments of massive equipment. Various test arrangements for the study of the feeding and combustion of fuels appear at the left. A 2-ft Cyclone Furnace (see Fig. 3 for more detail) is shown in the left foreground. Equipment for oil-burner testing is located in the background. Facilities for the study of heat transfer and steam generation are shown at the right. Supercritical-pressure heat transfer and flow are under study in right foreground*

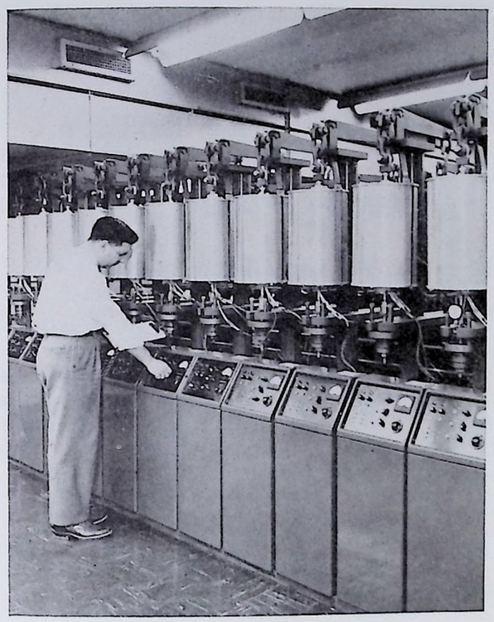


Fig. 2. *Creep and stress-rupture laboratory. A row of stress-rupture machines for determining, under uniaxial stress and controlled temperature conditions, the long-time load-carrying ability of metals at high temperature. A weighted lever arm maintains constant load on the specimen for the duration of the test.*

The specimen is placed in a cylindrical, insulated, high-temperature furnace, and a constant temperature within ±2 F is maintained by a magnetic temperature control at the base of the machine. This very reliable and sturdy control can operate for tens of thousands of hours without repair or loss of accuracy. There are no electronic tubes or moving parts. To insure high accuracy of temperature control and to keep all of the sensitive equipment clean and in good operating condition, the laboratory space is air-conditioned and the temperature is held at 83±1 F throughout the year

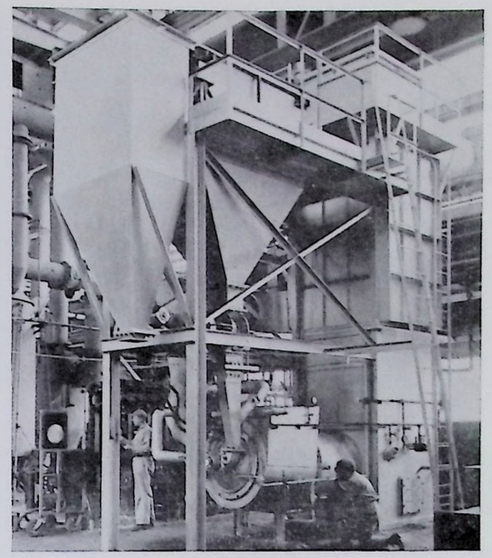


Fig. 3. *Pilot plant of Cyclone Furnace used to study fuel suitability and the effect of design modifications. Further studies are made on larger-diameter units at the Research Center and in the field*

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ysis and measurement of stress distribution in pressure vessels and other structures help to reduce the "ignorance factor" of design and to establish more exact use limits.

It is anticipated that dependence upon coal and its derivatives as the principal source of energy will steadily increase, with the probability that the quality of the coal available, as defined by the ash content and the ash-fusion properties, will deteriorate. When this happens, difficulties will be experienced in all methods of firing and in the transfer of heat to the absorbing media. The comparatively recent development of the cyclone principle of burning crushed coal (Fig. 3) represents a striking advance in the utilization of low-grade coal and is a notable example of the fruits of engineering research.

Other products of recent engineering research may be cited, such as improvements in devices for obtaining and measuring steam purity; new processes for coal gasification; recovery of pulp-mill chemicals and by-product heat values; pebble heaters for elevating the temperature of gases or vapors beyond the limits of metallic heaters; utilization of waste materials for their fuel and other values; and methods of alleviating the difficulties imposed upon heat transfer equipment by fuel ash. All of these recent developments are now being improved by current active research effort.

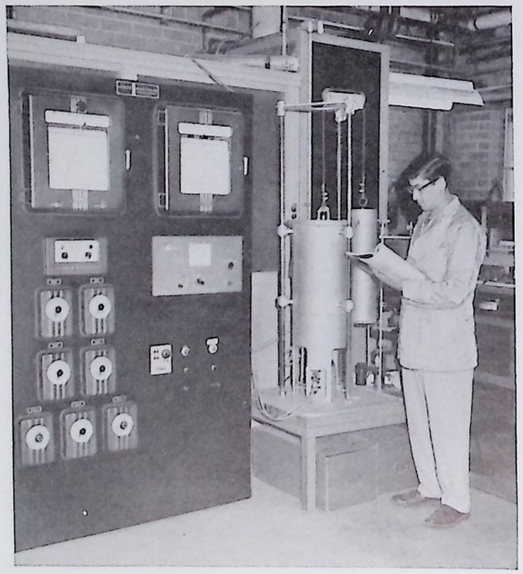


Fig. 4. *Differential Thermal Analyzer for studying the effect of high-temperature reactions on boiler operation, performance, and life. It enables a clearer understanding of what takes place at various temperature levels within the boiler and is helpful in determining the effectiveness of adjustments*

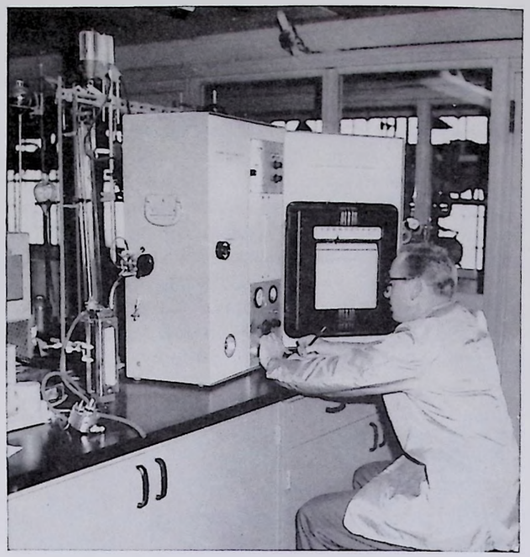


Fig. 5. *Gas Chromatograph for qualitative and quantitative analysis of gas samples and vapor mixtures. Concentrations as low as a few parts per million are measured*

Application of Nuclear Energy

No statement regarding research, particularly as it pertains to power generation, would be complete without reference to atomic energy. For an insight into the accomplishments as well as the research and development work by B&W in the field of nuclear energy application to the generation of power, see Chapter 27.

Summary

Research in the power-generating field requires the coordination of facilities and diverse technical talents for the expeditious and economical solution of every problem under investigation. At the Research Center of The B&W Co. there are mechanical, electrical, chemical, welding, and fuel engineers, stress analysts, chemists, spectrographists, metallurgists, physicists, and ceramists, all of whom are available for any research problem under consideration. Coordination of the efforts of such scientific and technical manpower with the aid of modern diversified facilities is the backbone of present-day industrial research.

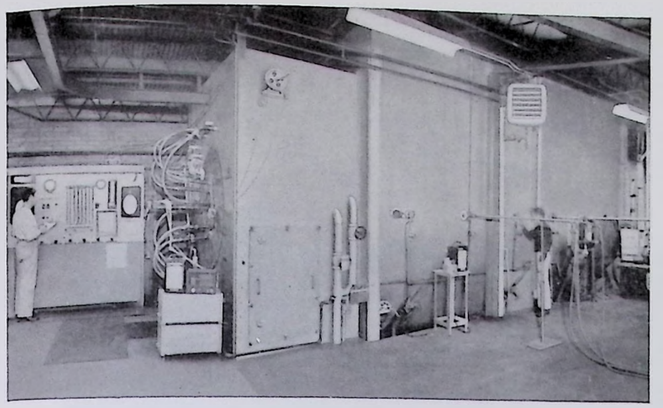
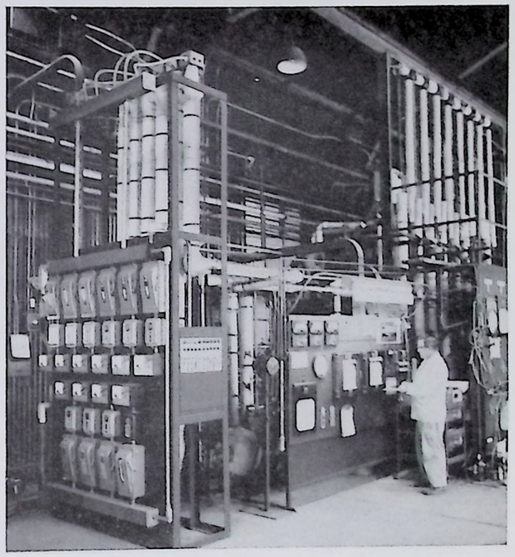


Fig. 6. *Oil- and Gas-Burning Test Tunnel for studying the performance of burners, lighters, and flame-failure detection equipment. The tunnel is instrumented for the study of heat flux, heat absorption, flame and metal temperatures, and combustion-gas composition. Flame characteristics are observed visually and can be recorded permanently by colored motion pictures*

Fig. 7. *The Testing of Nuclear Steam Boilers. Test facilities at the Research Center for the study of materials and processes used in the design, construction, and operation of equipment for nuclear energy applications*



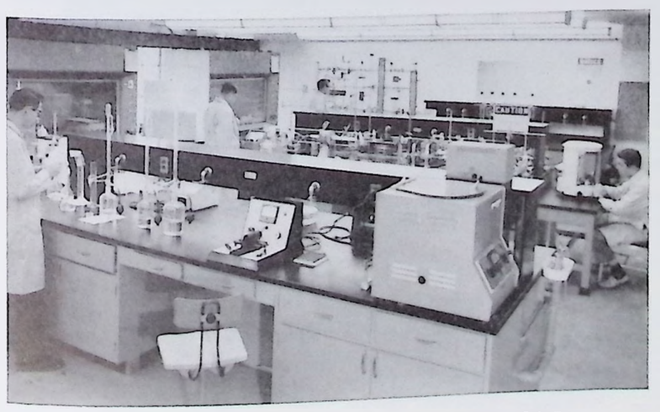
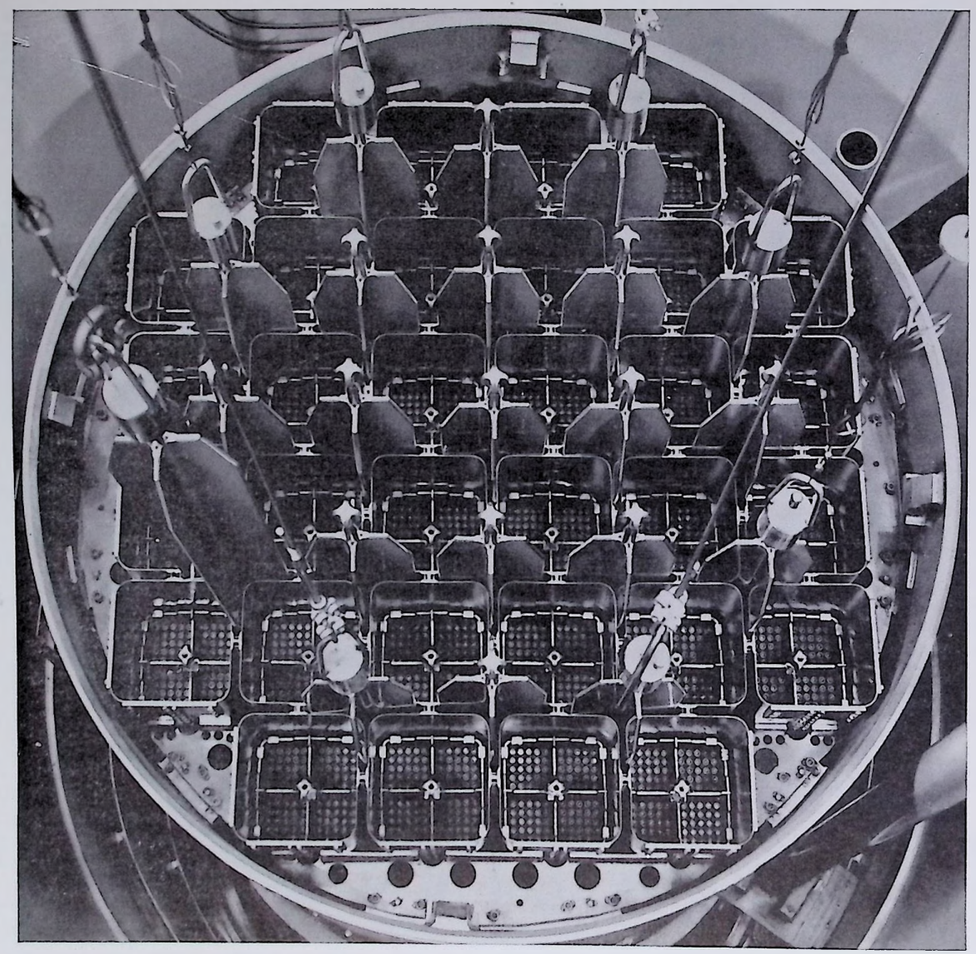


Fig. 8. *Section of the Radiochemical and Radiometallurgical Laboratory at the Research Center, where studies of radioactive substances are made. The laboratory is equipped with specially designed AEC-approved hoods, for the protection of personnel, and exhausts suitably filtered to prevent contamination of the surrounding atmosphere. Wastes are accumulated in special disposal tanks, and remote-handling equipment minimizes the danger of contamination*

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THE 62-INCH-DIAMETER 66-INCH-HIGH REACTOR CORE FOR THE N.S. SAVANNAH PREPARATORY TO TESTS AT B&W'S CRITICAL EXPERIMENT LABORATORY

27

NUCLEAR POWER SYSTEMS

THE discovery of a practical method for the release of nuclear energy disclosed a new power source of tremendous importance. The device in which this energy is released is called a nuclear reactor, which is, in effect, a furnace that generates heat by the interaction of nuclear particles rather than by chemical reaction. At present the only nuclear reaction capable of sustained and large-scale production of heat is fission, or the splitting apart of a heavy nucleus upon absorption of a neutron.

A nuclear reactor is currently defined, therefore, as an apparatus in which fission can be initiated and maintained at a controlled rate in a self-supporting chain reaction. It includes, among other things: fissionable material (fuel), such as uranium-235, uranium-233, or plutonium-239; a coolant to remove heat; a control system; and usually a moderating material to slow down neutrons. It often includes a reflector to conserve escaping neutrons.

The heat generated by a nuclear reactor may be transferred to a working fluid for use in conventional power-plant equipment. Steam is of such dominating significance in these operations that the boiler manufacturer is uniquely qualified to design and build reactors and their accompanying steam-generating apparatus.

Characteristics of Atomic Nuclei

The operation of nuclear reactors depends upon various types of interaction of neutrons with atomic nuclei. An atom consists of a positively charged nucleus surrounded by a number of negatively charged electrons, so that the atom as a whole is electrically neutral. Chemical energy, such as that obtained from the combustion of coal or oil, results from a rearrangement of the atoms caused by a redistribution of the electrons. In the processes taking place in a reactor, leading to the release of atomic energy, only the atomic nuclei are involved; the electrons may be neglected. For this reason, the term *nuclear energy* is frequently used as a more precise alternative to the earlier name, *atomic energy*.

Nuclear Fission

At the heart of the reactor is a mass of fissionable material, some of the nuclei of which split apart upon absorption of neutrons. The fission products, or fragments, emerge from this reaction at very high speeds and carry most of the energy in kinetic form, which, in turn, is rapidly converted to heat. After the self-sustaining reaction has begun in an assembly containing sufficient fuel, more than a "critical mass" (see *Glossary of Nuclear Terms*), it continues because an average of 2.5 neutrons are emitted by each fissioning nucleus and at least one of these is made to fission one neighboring nucleus. The result is a chain reaction, feeding on its own neutron yield, which continues as long as the critical mass of fuel is maintained.

Principal Reactor Parts

Reactors are usually composed of fuel, moderator, coolant, control system, structural material, and shielding.

Fuel. The fuel for a nuclear reactor is limited to one of the three fissionable isotopes: U-235 (from ore), U-233, and Pu-239 (both bred in reactors). It is contained in the core of the reactor, where the major portion of the heat is generated. Fuel elements must be protected with a suitable jacket or cladding because uranium and plutonium are very active chemically and are readily attacked by air, water, and liquid metals, which are the coolants commonly used. Cladding also prevents the escape of fission products and other toxic materials from the fuel element to the coolant.

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Moderator. The energy of most of the neutrons emitted by a fissioning nucleus is extremely high, approximately 1 to 10 million electron volts (1 Mev = 1.52×10^{-10} Btu) compared with its energy of about ⅛ ev when at equilibrium with its surroundings at room temperature. Since neutrons are electrically neutral they are very penetrating, especially at high velocities, and the fast neutrons can easily escape from the reactor. Those that do escape are lost and useless. To improve utilization of neutrons, their speed is often reduced by a "moderator," which is a material containing nuclei of low mass number, such as hydrogen or carbon. When neutrons strike the nuclei of the moderator there is an elastic collision; they give part of their energy to the moderator and bounce off at a reduced speed. At this lower velocity the probability of neutron capture by uranium atoms (resulting in fission) increases, and the chain reaction is more readily sustained.

Coolant. The heat generated in the reactor core at any appreciable power level (Btu/hr) must be removed. The function of the reactor coolant is to transport the heat to the outside, where it may be used for purposes such as the production of power. In a nuclear power plant, therefore, the coolant is the essential link between the reactor itself and the conventional power-generating equipment, such as a steam turbine. Ordinary (light) water, heavy water, carbon dioxide, and liquid sodium are among the more frequently used reactor coolants.

Control System. A fission chain reaction is not difficult to control. Some core configurations are self-regulating, once temperature and power levels are set. However, a separate control system is incorporated in the reactor structure. It consists of rods or sheets of neutron-absorbing material, mounted in the core so that they can be moved in and out quickly and precisely. Boron, hafnium, and cadmium, which have high neutron-absorption cross sections, are frequently used as control materials.

Structural Material. In a heterogeneous reactor (solid fuel elements) the fuel is contained and held in place within the flowing coolant by means of structural materials. These materials must have good mechanical properties at high temperature and substantial corrosion resistance. In addition, the structural materials in the core and reflector of a thermal reactor should have low neutron-capture cross sections. Magnesium, aluminum, and zirconium have low cross sections, but stainless steel (with a slightly higher cross section) is a frequently used reactor structural material because of its outstanding mechanical and corrosion-resistant properties.

Shielding. One additional feature is essential to every nuclear reactor, although not an actual working part. This is the massive shielding that must completely surround the reactor to absorb penetrating neutrons and gamma rays and thus protect the operating personnel. Neutrons are attenuated best by water or other hydrogenous materials, such as paraffin or polyethylene. Gamma rays are attenuated by very dense materials. Ordinary concrete is an inexpensive material that meets these requirements and is widely used in conjunction with water or sheets of steel, lead, or other metals. It is the shield that makes even a small reactor look impressively large.

Reactor Types

At present the building or planning of reactors is directed toward one of the following objectives: 1) the production of heat and power, 2) the production of plutonium, uranium-233, or radioisotopes, 3) the production of neutrons for research, or 4) the experimental development of reactors.

Reactors can also be classified (a) according to type of fuel used: 1) natural uranium, which contains 0.7% U-235, 2) enriched uranium, which contains a higher percentage of U-235, 3) Pu-239, or 4) U-233; or (b) according to the energy of the neutrons at the time they are captured by the fuel to induce fission: 1) fast, 2) thermal, or 3) intermediate.

In fast reactors, which have no moderator, neutrons are captured by the fuel immediately after they are generated. This type of reactor has certain advantages in neutron economy but requires a large fuel inventory. Losses are reduced by capturing some of the fast neutrons in a breeding blanket (see below).

Thermal reactors contain sufficient moderator to slow down the neutrons, so that they are in thermal equilibrium with their surroundings when captured. Since the probability of capture by fissionable material is high in the thermal range, thermal reactors generally require less fuel for a critical mass than do other reactors of the same output.

Intermediate reactors contain a reduced amount of moderator and, therefore, an increased amount of fuel.

A further classification of reactors can be made according to the physical condition of the fuel: 1) heterogeneous, or 2) homogeneous. Heterogeneous reactors have solid fuel elements separated from moderator and coolant in definite geometric patterns. The fuel in homogeneous reactors may be: 1) aqueous solutions of uranium salts, 2) solutions of uranium in molten metals, 3) fused salt mixtures containing uranium, or 4) molten plutonium with molten metal diluents.

Another reactor classification distinguishes between the *breeder* and the *converter*, depending upon the amount of new fissionable fuel produced by the absorption of neutrons in the fertile material, uranium-238 or thorium-232. A breeder reactor produces more fissionable material than it consumes, whereas a converter produces less than it consumes.

Many other classifications of reactors follow from the numerous design parameters shown in Table 1.

TABLE 1

Reactor Design Parameters

Neutron Energy
Fast, thermal, or intermediate

Geometry
Heterogeneous or homogeneous

Fuel
Uranium-235, uranium-233, or plutonium-239

Fertile Material
Uranium, thorium, or none

Fuel Regeneration
Less than consumption (converter), more than consumption (breeder), or no regeneration (burner)

Coolant
Pressurized water, pressurized heavy water, boiling water, boiling heavy water, air, carbon dioxide, steam, helium, nitrogen, sodium, sodium potassium eutectic, diphenyl, terphenyl, lead, bismuth, or fluid fuel

Moderator
Water, heavy water, graphite, diphenyl, terphenyl, beryllium, or beryllium oxide

Structural Material
Aluminum, zirconium, stainless steel, or carbon steel

Activities of The Babcock & Wilcox Company

B&W is engaged in most aspects of nuclear power development. Activities extend from the design and fabrication of complete power, test, and research reactors to the development of experimental units and the fabrication of component parts and fuel elements.

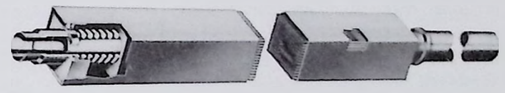
As a leading manufacturer of boiler equipment, B&W has specialized since 1867 in harnessing thermal energy for useful purposes. Boilers of its manufacture provided steam for the first central power station in the U.S. in 1881. B&W's long record of specialized achievement in heat transfer, fluid flow, pressure containment, and metallurgy led to its selection, in 1943, as a supplier of heavy components for the Manhattan Project.

When controlled fission became a reality, B&W rapidly broadened its interest to encompass the many aspects of nuclear reactor development and manufacture. Diversified projects for the U.S. Navy, the U.S. Atomic Energy Commission, the AEC's National Laboratories, the U.S. Maritime Administration, utility companies, and industrial groups both here and abroad have provided B&W an additional wealth of information and experience.

Recognizing the expanding needs of the nuclear field, B&W established a separate Atomic Energy Division in 1953 to develop new and improved reactors and associated components. It has executed many contracts, private studies, and laboratory developments related to reactor systems and components, including reactor plant-construction projects, conceptual design and evaluation, and related reactor development.

In 1956, the first unit of the Company's Nuclear Facilities Plant (NFP) for the production of fuel elements and cores was completed, and production capacity has since been increased manyfold. It was this country's first privately owned and operated fuel-element fabrication facility. Fig. 1 shows typical power, research, and test-reactor fuel elements manufactured at this plant, which also produces complete reactor cores, special fuel elements, control rods, and other component equipment.

Also in 1956 the first unit of B&W's Critical Ex-



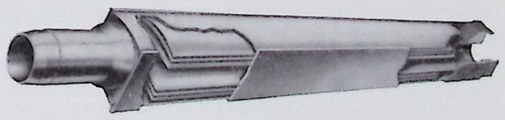
a) B&W FUEL ELEMENT FOR PRESSURIZED-WATER POWER REACTORS



b) B&W FUEL ELEMENT FOR N.S. SAVANNAH POWER REACTOR



c) B&W FUEL ELEMENT FOR RESEARCH REACTOR



d) B&W CONTROL ELEMENT FOR RESEARCH REACTOR



e) B&W FUEL ELEMENT FOR MATERIALS-TESTING REACTOR

Fig. 1. *Power, research, and test-reactor fuel elements*

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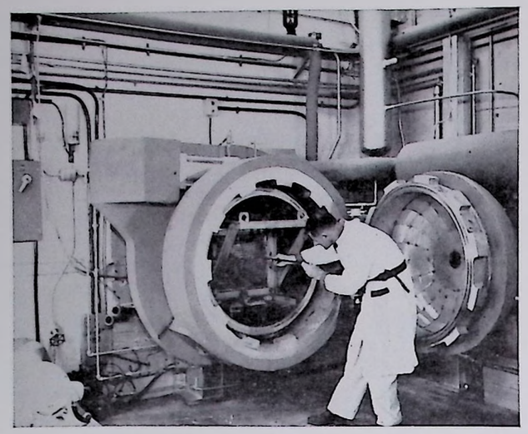


Fig. 2. *Hot-exponential facility for testing core materials under conditions simulating operating temperatures and pressures*

periment Laboratory was put into operation. In this facility, the operating characteristics and performance of full-scale power-reactor cores are determined, as a necessary verification of design calculations. This is done through operational study, at virtually zero power, of a mock-up of the proposed reactor core. Just as wind tunnels furnish reliable and inexpensive previews of the aerodynamic characteristics of the aircraft, this facility enables B&W engineers and scientists to analyze proposed reactor cores in action and to make practical improvements before the design of the reactor is finalized.

The Lynchburg Pool Reactor is used for a variety of experiments and design checks. A hot-exponential tank (Fig. 2), or autoclave, is coupled with this reactor to determine nuclear parameters of water-cooled cores at operating temperatures.

Consolidated Edison Indian Point Station

In meeting the challenges and responsibilities of providing electricity to nearly nine million people in a 600-square mile area, which includes New York City and most of Westchester County, Consolidated Edison Company of New York, Inc. constructed one of the nation's first large-scale atomic power plants at Indian Point, 24 miles north of New York City, on the Hudson River (Fig. 3).

After considering several reactor types, Con Edison in 1955 selected for development a pressurized-water thorium-uranium converter reactor. This concept was proposed by The B&W Co., the contractor for basic nuclear engineering and the designer and manufacturer of the major items of nuclear plant equipment.

Thorium in Power Reactors

The reasons for the interest in the use of thorium and the ultimate attainment of the thorium-uranium-233 fuel cycle are many:

1. Thorium occurs quite abundantly in easily mined deposits of monazite in a number of countries, notably India, Brazil, and Australia.
2. Conversion of thorium makes a better nuclear fuel for thermal reactors than does conversion of U-238.
3. Thorium metal has fair tensile strength, good mechanical-working properties, and machining characteristics comparable to those of mild steel. The oxide (ThO_2) is very stable even at elevated temperatures.
4. Thorium can be used in many forms as a fertile or blanket material in power breeder reactors.
5. The operating temperature of thorium in reactors is not limited by allotropic transformation as is the case for uranium, where the alpha-beta phase change still limits the performance of certain reactors.

Station Features

The general layout of the station follows that of conventional steam plants. The station will develop an electrical output of 275,000 kw from 2,200,000 lb steam/hr generated in the boilers at 405 psi and heated to 1000 F in separately fired superheaters. The heat rate of the steam cycle, exclusive of reactor efficiency, is 10,700 Btu/kwhr.

Nuclear Boiler Plant

The nuclear boiler plant is comprised chiefly of 1) the reactor and its heat transfer system containing valves, piping, pumps, boilers, and pressurizer; 2)

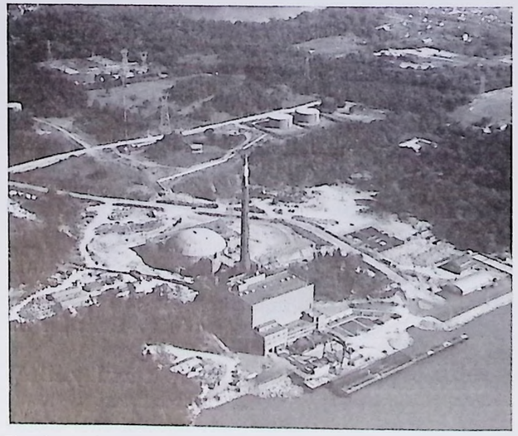


Fig. 3. *Indian Point Station during construction, July 1961*

reactor safety and control system; 3) safety containment vessel and auxiliary equipment; 4) primary and biological shielding; 5) reactor service facilities containing laboratories and auxiliary equipment; 6) fuel-element charging, discharging, and storage facilities; 7) coolant treatment and make-up systems; and 8) waste disposal facilities for collection, storage, and disposal of solid, liquid, and gaseous wastes.

The reactor operates at 1485 psi and is cooled by the circulation of water in four coolant loops operating in parallel. Coolant in the amount of 52.8×10^6 lb/hr enters the reactor at 486.5 F and leaves at 519 F to transfer the heat of the nominal reactor-plant rating of 585,000 kw thermal (2×10^9 Btu/hr). The first reactor core, composed of 120 fuel elements, will have a life of approximately 500 full-power days.

Plant Layout

An aerial view of the Indian Point plant is shown in Fig. 2. The concrete-shielded 160-ft-diam 1-in.-thick carbon-steel containment vessel eliminates the possibility of dispersing radioactive material in the unlikely event of a system rupture. It houses the nuclear reactor proper, all of the high-pressure high-temperature coolant and auxiliary systems, and a 70-ton gantry crane. Personnel are not permitted inside the containment vessel during normal plant operation.

The concrete wall surrounding the sphere is designed to attenuate the core gamma dose during full-power operation and the water activation gamma from the primary loops to less than ¾ milliroentgen equivalent man/hr (10% of laboratory tolerance for humans) at the external shield surface. This affords adequate protection to personnel in nearby locations.

A fuel-handling building, for storing and handling both new and used fuel and control-rod elements, is adjacent to the waste-disposal building which houses the equipment for the chemical-process system. The principal functions of this system are to 1) collect, contain, and treat the various solid, liquid, and gaseous wastes from the nuclear reactor plant and its auxiliary system; 2) provide decontaminated water for re-use in the primary-coolant system; 3) keep impurities to less than one ppm total solids in the primary-coolant water so as to prevent radioactivity build-up; and 4) supply primary-coolant water of sufficient quality for mechanical seals on such apparatus as the control-rod drives.

Reactor Core

The active portion of the reactor core approximates a right circular cylinder 8.3 ft in height and 6.5 ft in mean diameter. Its composition is determined by the

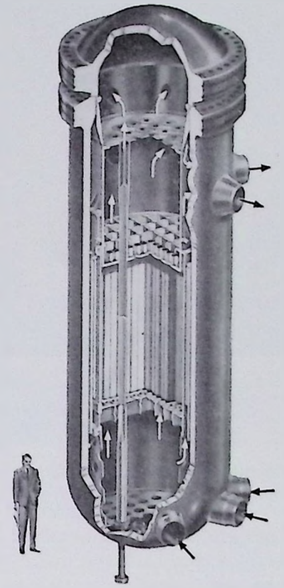


Fig. 4. *Cutaway view showing core and internals of pressurized-water internal-thorium-converter reactor at Indian Point*

requirements of heat transfer, structural integrity, excess reactivity for adequate life, and the need for an ample negative temperature coefficient, which tends to slow down the fission process on rise in temperature and thereby improve the stability of the reactor from the standpoint of control.

Each of the 120 fuel elements (Fig. 1a) contains 195 fuel rods arranged in a square lattice. The rods are stainless steel tubes loaded with fuel pellets. The fuel-element can is made of Zircaloy-2 sheet.

Thorium-232, the fertile material, does not itself undergo fission but absorbs neutrons to form Th-233. This isotope of thorium decays to protactinium-233, which in turn decays to the fissionable isotope U-233, which then fissions to produce heat energy. During the expected operating life of the Con Edison core—500 full-power days—approximately 239 kg of the fissionable isotope U-233 are produced.

Reactivity control of the reactor is accomplished by a combination of (a) hafnium cruciform control rods driven by electromechanical drives through the bottom of the reactor vessel, (b) a soluble neutron poison (boric acid) in the primary cooling system when the coolant is below operating temperature and (c) incorporation of a small amount (250 ppm) of burnable poison (boron) in the stainless-steel fuel-rod cladding.

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Fig. 5. N.S. Savannah, world's first nuclear-powered merchant ship, launched July 21, 1959

Primary-Coolant System Components

The primary-coolant system consists of four parallel loops which circulate demineralized light water through the reactor vessel and the steam boilers. Each loop consists of two canned-motor pumps, two swing check valves, a steam boiler, and connecting piping. The design conditions for all components are 1800 psi and 650 F. The components of the primary-coolant system are designed in accordance with applicable sections of the ASME *Boiler and Pressure Vessel Code* and the ASA *Code for Pressure Piping*. The principal material of construction for all components coming in contact with coolant water is type 304 stainless steel.

The general arrangement of the reactor pressure vessel is shown by Fig. 4. Four inlet connections are located in the bottom hemispherical head and four outlet connections in the upper portion of the shell. There are twenty-one penetrations for control-rod nozzles in the bottom hemispherical head. The top closure head is secured by studs. To assure tightness, the closure head and vessel are seal welded to a semi-toroidal-shaped stainless-steel seal. Concentric cylindrical stainless-steel thermal shields are mounted inside the pressure vessel to reduce thermal stresses in the vessel wall during the reactor operation by absorbing gamma radiation emanating from the core. Insulation around the pressure vessel reduces the heat transfer to the neutron-shield tank.

The steam boiler in each coolant loop consists of a U-tube lower heat exchanger and an upper separator drum with connecting risers and downcomers. The tubes are rolled and welded into tube sheets clad with stainless steel. The upper drum is located above and on the center line of the heat exchanger and is connected to it by risers and downcomers for natural circulation of the secondary steam-water mixture. The normal steady-state output of each steam boiler is 145,000 kwt (498×10^6 Btu/hr). Steam flow (saturated) at the upper-drum outlet is 550,000 lb/hr at 405 psi and 449 F.

A pressurizing system establishes and maintains primary-coolant system pressure within prescribed limits during warm-up, steady-state operation, load changes, and shutdown. The pressurizing system consists primarily of a pressurizer tank, heaters, surge line, spray line, and spray head.

Nuclear Merchant-Ship Reactor

The first American nuclear merchant ship, the *N.S. Savannah*, was built through the combined efforts of the U.S. Maritime Administration and the U.S. Atomic Energy Commission. Major requirements included a conservative design, proven reactor concept with a long-life core, use of commercially available materials and equipment wherever practical, minimum capital and operating expenditures, and dependable and safe operation of the ship in normal commercial trade. The power plant was also designed to permit continued nuclear research, because it is basically a floating laboratory and test ship built to prove the feasibility of nuclear power for merchant ships. The B&W Co. contracted to supply the nuclear propulsion plant, the nuclear part of which it designed and built.

The *N.S. Savannah* (Fig. 5) is a single-screw geared-turbine combination passenger-cargo vessel 595 ft long, with a beam of 78 ft, a draft of 29 ft-6 in., and a displacement of 21,800 tons. The ship, constructed by the New York Shipbuilding Corp, has accommodations for 60 passengers and will carry about 10,000 tons of cargo. The designed sea speed is 20 knots.

The reactor system is the pressurized-water type, consisting of a single reactor with two main coolant loops and two steam boilers. Primary water at a pressure of 1735 psi generates saturated steam at 455 psi at full power. This steam expands through the main turbines to the condenser, where pumps return the condensate through the feed system and to the boilers. There are two 1500-kw turbine generators which provide auxiliary power for all propulsive and ship services. A low-pressure heat exchanger furnishes a maximum of 7500 lb steam/hr for hotel services and cargo heating.

For emergency take-home power, a 750-shp electric motor has been installed that can be coupled to the main gearing. Two 750-kw diesel generators designed to start automatically upon failure of the turbine generators, provide sufficient power for the take-home motor and other essential electrical loads. One oil-fired package boiler, capable of supplying 7500 lb steam/hr at 150 psi, furnishes emergency steam for heating and fire protection.

Pertinent design data for the power plant are given in Table 2.

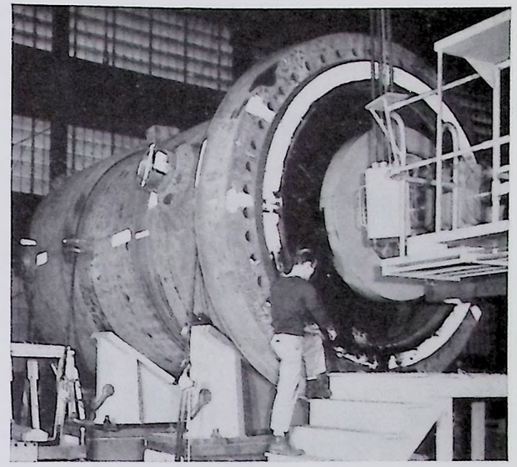


Fig. 6. Machining N.S. Savannah reactor vessel in Boiler Division plant at Barberton, Ohio

TABLE 2

Design Data for N.S. Savannah Power Plant

Load	Normal	Maximum
Shaft power, shp	20,000	22,000
Shaft speed, rpm	106.7	110
Turbine inlet press., psi	455	435
Condenser vacuum, in. Hg	28.5	28.5
Feedwater temp, F	347	343
Total steam, lb/hr	242,200	261,460
Total electric load, kw	1,900	1,900
Reactor heat gen, kw	63,500	69,000
Reactor press., psi	1,735	1,735
Reactor water flow, 10^6 lb/hr	8	8
Primary water mean temp, F	508	508
Reactor water Δt, F	22.8	24.8

The reactor is fueled by 7100 kg of uranium with an average enrichment of 4.4%. The uranium oxide in the form of pressed sintered pellets is contained in stainless steel tubes. Each of the 32 fuel elements (Fig. 1b) contains 164 of these tubes. Individual stainless steel boxes, which are semipermanently fixed to the lower support plate, surround each element and form channels in which the control rods move. These boxes direct the coolant flow through the fuel elements and regulate the amount of coolant passing through the channels.

The reactor is controlled by rods of cruciform section (see Frontispiece) made of stainless steel plates containing boron enriched to 90% in the B-10 isotope for increased neutron absorption and clad in plain stainless steel sheets. The rods are actuated by top-mounted electromechanical-hydraulic drives. Zircaloy-2 followers, fastened to the bottom of the absorbing rods, occupy the rod channels in the core as the rods themselves are lifted out. Thus the control rods are replaced by the Zircaloy followers rather than by water that would produce high power-peaking in the adjacent fuel rods.

The principal nuclear requirements for the core were (a) lifetime should be at least 52,200 megawatt-days (mwd) and (b) the control rods should hold down the core, when it is new and at room temperature, with approximately 3% in reactivity to spare.

A critical experiment with the actual elements was carried out at the Lynchburg Critical Experiment Laboratory before the elements were delivered to the shipyard. Experiments to confirm the reactivity values of the control rods were completed at the same time.

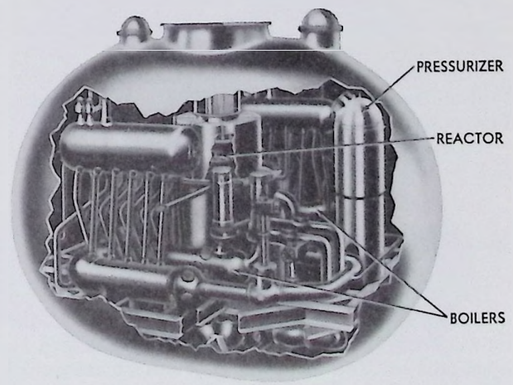


Fig. 7. Model of nuclear boiler plant of N.S. Savannah

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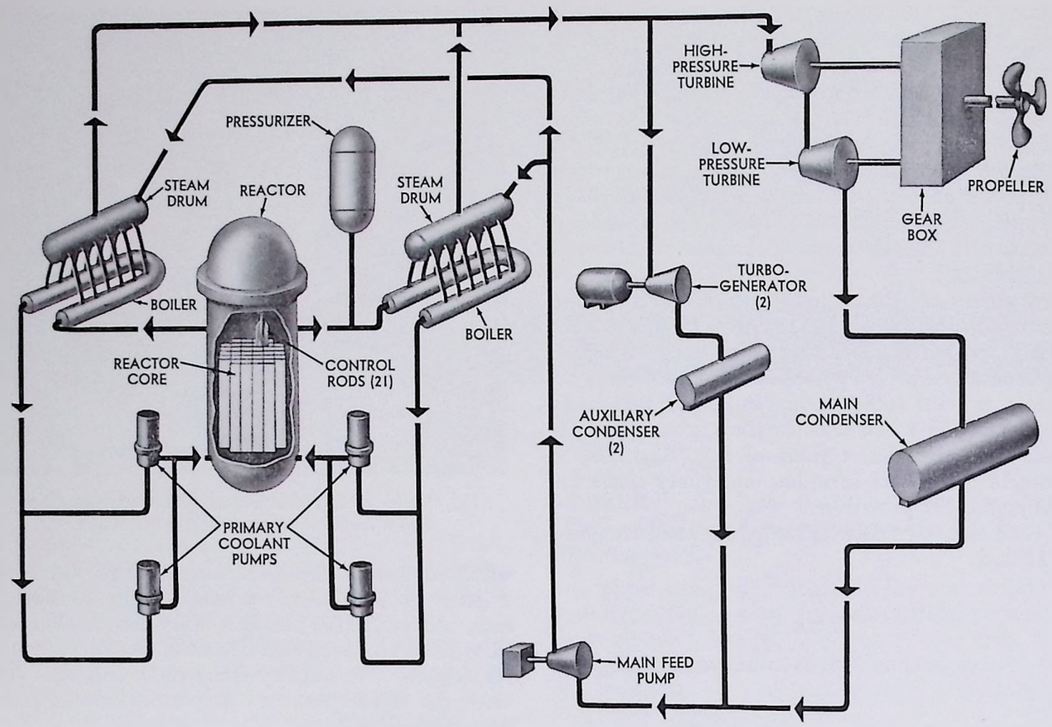


Fig. 8. *Flow diagram of N.S. Savannah propulsion system*

The instrumentation and control system for the reactor follows conventional design practices. The safety system provides for scramming the reactor under specified conditions. In addition, a manual scram switch is placed on the console.

The entire primary system is enclosed in a steel containment shell (Fig. 7) which furnishes support for both the primary and the secondary shielding. The secondary or biological shield reduces the radiation in crew and passenger spaces to a level less than the maximum value permitted for occupational exposure by the International Commission on Radiological Protection and the National Committee on Radiation Protection.

A flow diagram for the nuclear power plant of the *N.S. Savannah* is shown schematically in Fig. 8.

Research Reactors

Development of nuclear reactors for industrial and university research has proceeded in parallel with the development of power reactors. B&W developed an open-pool reactor design that provides a maximum of experimental flexibility for scientific and research workers.

A typical B&W pool reactor is shown in Fig. 9. A cluster of fuel elements is suspended from a movable bridge that spans an open concrete pool filled with highly demineralized water. The concrete sides of the pool are formed from high-density concrete to provide radiation shielding for reactor personnel. The

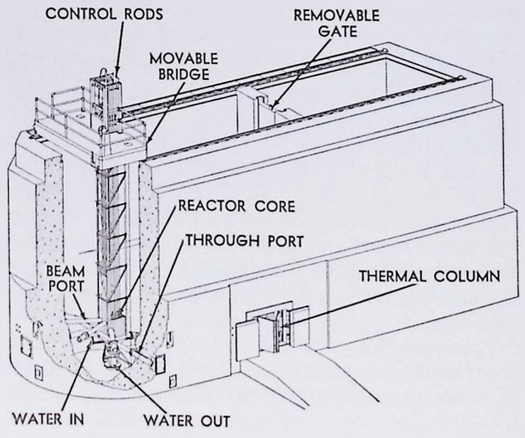


Fig. 9. *B&W open-pool research reactor*

water is the coolant for the fuel elements and the moderator for neutrons. The core is sufficiently submerged to provide radiation shielding for those working on or near the bridge.

The bridge is moved to one of several operating positions, and the reactor is then started up and operated from a separate control room overlooking the pool. Control rods, containing neutron-absorbing material, are slowly withdrawn until the reactor becomes critical and reaches the desired operating power level. The power is maintained at a constant level by continuous manual or automatic control of these rods. Thermal power up to 5000 kw (17.1×10^6 Btu/hr) can be produced in a pool reactor, and at this power approximately 4×10^{13} fast and slow neutrons/cm^2, sec are available for neutron research at the core.

At all power levels above 100 kw (0.34×10^6 Btu/hr), forced-circulation cooling is used. The pool water is drawn down through the fuel elements and through a header to a heat exchanger and then returned to the pool. Below 100 kw the header is disconnected and the fuel elements are cooled by natural convection of the pool water.

Three primary methods are provided for utilizing the neutrons for research work. Dry watertight tubes, 6-in. and 8-in. in diam, extend radially through the concrete pool walls to the core. Beams of both fast and slow neutrons passing through the tubes are available for many types of experiments in the working area around the outside of the concrete pool. At another operating position, neutrons from the core are passed through a $4' \times 4' \times 7'$ array of graphite blocks, called a thermal column. Graphite has the property of reducing the velocity of the fast neutrons, and thus a large-volume source of slow (thermal) neutrons is available within and in front of the thermal column. Experiments can also be carried on within the pool itself. Small samples to be irradiated may be placed in containers and positioned close to the fuel elements or sent through a pneumatic tube system to a location at one side face of the core. Large experiments may also be assembled on the floor of the pool and irradiated after moving the reactor core to a position adjacent to the experiment.

Spectral-Shift Control Reactor

Water reactors in 1963 enjoy the distinction of having reached the most advanced state of development of all reactor types. Statistically reliable experience will soon be available. Unfortunately, nuclear power is economically feasible today only in limited locations—primarily those of high fuel cost. In an effort to break this high-cost barrier, B&W has developed a variation of the pressurized-water reactor (PWR) known as the Spectral-Shift Control Reactor (SSCR). The thermal characteristics of the SSCR are identical in nature with those of a conventional PWR. It has the same limitations with respect to burnout and fuel melting and the same potential for thermal efficiency.

The SSCR is a concept wherein reactivity control is accomplished by varying the concentration of a heavy-water light-water mixture in the primary coolant loop. The addition of heavy water, D_2O, changes the relative number of neutrons at particular energies (i.e., shifts the neutron spectrum), resulting in more neutrons at slightly higher energies and more absorption in fertile material. This increased absorption in the fertile material allows a larger amount of fissionable material to be controlled (i.e., the reactivity of the fuel lattice decreases with an increasing deuterium-hydrogen ratio because the "slowing-down power" for D_2O is only 13% of that of light water).

The SSCR is similar to pressurized-light-water reactors already constructed and operating. Moreover the SSCR incorporates not only many of the demonstrated technological advances achieved to date in conventional water reactor systems but has the added advantages of improved neutron economy, longer core life, and increased power density. Fig. 10 shows a model of the nuclear portion of an SSCR plant designed for 390,000-kw electrical capacity.

The heat source, or reactor core, for an SSCR is shown in Fig. 11 with the closure head in place and the drive mechanisms mounted above.

The Spectral-Shift Control Reactor should offer the utility user the advantages of lower capital and fuel costs with additional safety and flexibility.

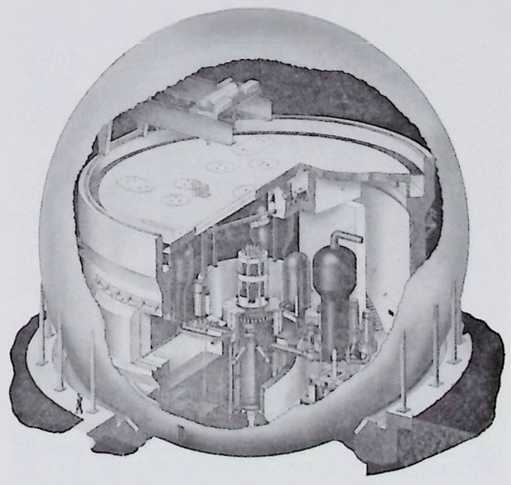


Fig. 10. *Design drawing of containment vessel housing an SSCR nuclear boiler plant of 390-mw electrical capacity*

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Development Activities

The achievement of economically competitive nuclear power requires the successful completion of development programs directed toward the reduction of both fuel and capital costs.

Reactor Cores

The Company is undertaking extensive development and testing work to reduce fabrication costs for oxide fuel elements and to confirm their high burn-up and heat-release capabilities. Particular emphasis is placed on core designs of Zircaloy-clad UO_2 and ThO_2-UO_2 for the Spectral-Shift Control Reactor and on achievement of its potential for low fuel costs through full utilization of the fissionable isotope, U-233, formed in thorium by neutron capture.

High-Temperature Fuel Elements

As support for advanced reactor designs involving high-temperature coolants, investigations of ceramic fuels clad with graphite and high-temperature alloys are being undertaken. Because of the elevated temperature involved, factors of creep corrosion and rapid chemical reactions between materials are of prime importance and must be investigated along with the effect of radiation. The low neutron-capture cross section of graphite has stimulated special efforts to prove its suitability as a core cladding and structural material.

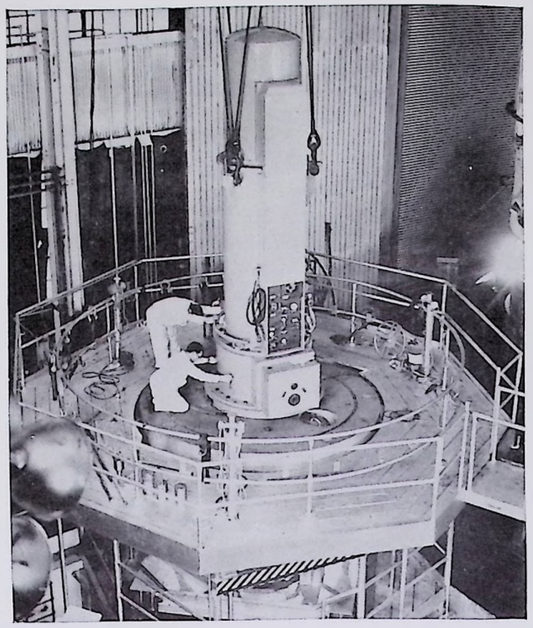


Fig. 12. *Shielded fuel-transfer cask on rotary manipulator*

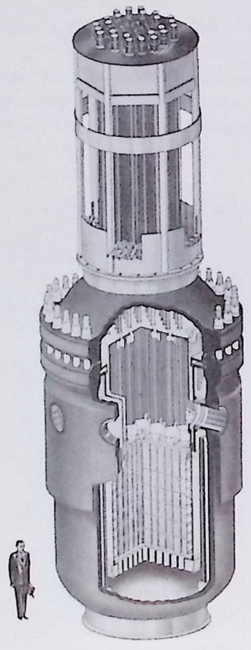


Fig. 11. *Cutaway view showing core internals of Spectral-Shift Control Reactor for plant design shown in Fig. 9*

Control Elements

The safety of any reactor depends primarily upon the speed and reliability of the control system which acts to introduce poison into the core, thereby reducing the reactivity. Boron has been the most popular choice for a poison material. However, since both helium and lithium are produced when a boron atom absorbs a neutron, undesirable dimensional swelling and reduction in ductility occur. An alloy of stainless steel and boron (about 2% of boron) was successfully fabricated by the Company and in-pile tested under expected operating conditions. Because of its high cost and present unavailability, hafnium is rarely used in commercial reactors even though it is an excellent control-rod material. The search is continuing for a material which is high in neutron-absorption cross section, readily available, and which does not exhibit the unfavorable irradiation behavior of boron. The most promising recent developments have been with the rare earth oxides, such as those of europium, gadolinium, dysprosium, and samarium.

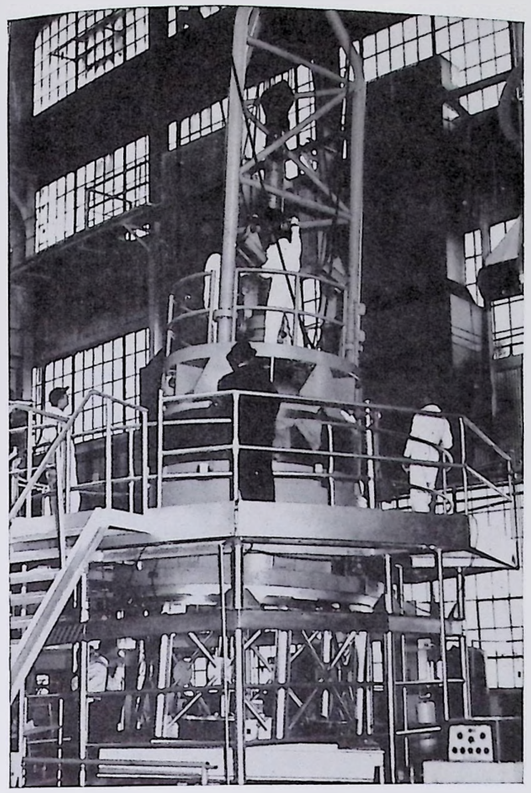


Fig. 13. *Transfer cask removing reactor internals*

System Materials

The structural materials used for construction of reactor primary systems must be developed and selected on the basis of design and Code requirements as influenced by the coolant being circulated and the radiation and temperature environment.

Components

B&W supports a continual program of development activities directed toward improvement of its nuclear products. A major share of this work is performed to solve present and anticipated problems associated with nuclear components. Some of the major development activities in this area include reactor-closure design, tube to tube-sheet welding, clad-nozzle manufacture, ultrasonic inspection, methods of analysis, and steam separation.

Handling Equipment

During the course of the *N.S. Savannah* contract, the Company developed, fabricated, and tested unique and practical refueling equipment. Fig. 12 shows the shielded fuel-transfer cask on the rotary manipulator. Fuel-handling equipment is operable when the ship has a list of 4°, a condition that simulates its worst attitude at dockside during refueling operations.

The cask contains an integral fail-safe interlocking hoisting mechanism which operates within a primary and secondary cooling system. Panel controls permit the operator to spray-cool the fuel element during transit from the reactor to the cask and then to fill the cask with water for transport. With an extension piece added to increase its length, the same cask is used for removal of control rods or fuel elements.

Fig. 12 shows the reactor-vessel-internals transfer cask in place on the reactor-vessel-flange mock-up during the removal of a 17,000-lb portion of the reactor internals, with the transfer cask operating at a 4° angle to the vertical. This 27-ft-high cask assembly is so designed that the shield portion will telescope, thus doubling the shield thickness to permit removal of a smaller, but hotter, upper-grid-plate assembly. An internal-hoist and tool-column assembly permits handling of vessel internals with specified combined ship motions of heave, roll, pitch, and drift.

Recognizing the disadvantages of the traditional method of tightening the nuts on large studs or bolts by first heating the latter to a predetermined tem-

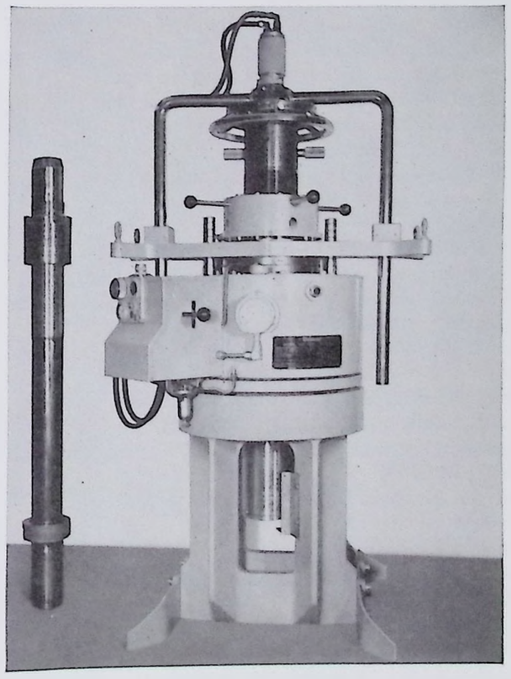


Fig. 14. *A 600-ton stud tensioner for the N.S. Savannah*

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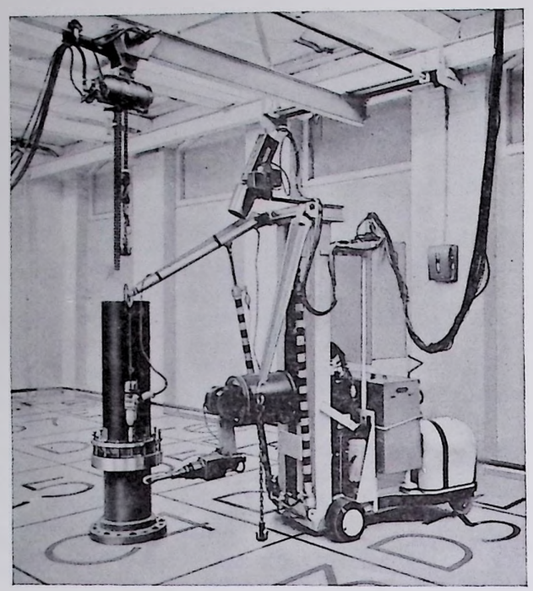


Fig. 15. *A prototype manipulator unit*

perature, the Company designed the stud tensioner (Fig. 14), which performs the equivalent operation in ⅕ the time. The basic principle involved is one of using force to stretch the stud elastically a predetermined amount prior to tightening the nut.

The 600-ton-capacity unit shown tensions studs by an hydraulic-powered drawbar to within 0.001-0.002 in. of the desired elongation. Completely powered in all operations, a single control will (a) power-drive the drawbar on and off a stud, (b) retract a stud within the body of the tensioner to permit a welder or seal-weld cutter to pass, (c) reinstall and tension the stud, and (d) raise the tensioner for relocation to the next stud. Air-powered trollies move this two-ton tensioner around the reactor vessel.

In addition to the specific equipment described above, B&W has developed special nuclear equipment, such as control-rod drive-shaft seals, seal plugs, and mobile manipulator equipment. The prototype manipulator unit shown in Fig. 15 was designed and built by B&W to provide information for the design of an advanced robot mechanic to do various and sundry divergent jobs, such as lift a ton, turn a key in a lock, or accomplish a variety of jobs under conditions of environment that prohibit direct access to humans. Performance is controlled from a closed room which precludes visual observation of the equipment. A TV camera with autozoom lens is mounted on top of the mobile manipulator truck to allow close viewing of the manipulator arm and hand. The installation of another TV camera provides an over-all view of the mock-up area.

The arrangement of the control center is shown in Fig. 16. The section on the left supplies power for the manipulator. The two handles on the extreme right control the manipulator and the manipulator truck. The two central handles control the crane.

B&W has also developed an improved command system for its offerings in the robot field. This system utilizes four electronic units, one for each of the following; transmitter timer, transmitter coder, receiver timer, and receiver decoder. The two receiver units are of miniaturized design so that local shielding can be readily provided for high-radiation-field use. These basic units are assembled from standard electronic component groups that are readily replaceable because of the unitized plug-in construction. A 60-function receiver (60 completely independent commands which can be individual and/or simultaneous) occupies approximately half a cubic foot. Two types of transmission systems are employed, depending upon the application. An induction-loop transmission system is preferred for operation in a defined area, but a radio link can be used for greater range and flexibility. The induction-loop system requires no FCC licensing, and the unit is virtually interference-free. This system has been prototyped and is now being thoroughly tested.

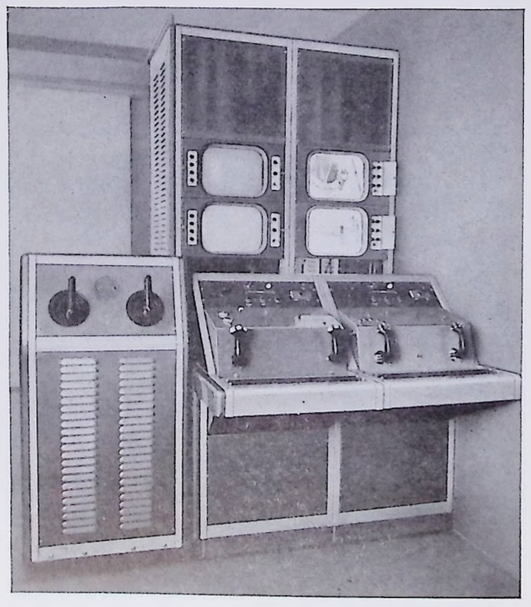


Fig. 16. *Operator's console and TV monitoring equipment*

GLOSSARY

of Nuclear Terms

Atom. Chemically, the smallest electrically neutral constituent part of an element that can engage in a chemical reaction. Physically, it consists of a positively charged nucleus surrounded by a compensating number of negative electrons in various orbits.

Atomic Nucleus. A composition of two kinds of primary particles, protons and neutrons.

Breeder Reactor. One that produces more fissionable material than it consumes.

Breeding. The process whereby a fissionable species is utilized as a source of neutrons to produce more nuclei of its own kind than are used up. This is the function of a breeder reactor.

Chain Reaction. A fission chain reaction where the neutrons produced by the fission of an atomic nucleus (two or more neutrons on the average) cause the fission of other atomic nuclei, which in turn propagate the fission reaction in the same manner. Not all the neutrons released by the capturing nuclei produce new fissions, because of leakage, absorption of impurities, and other losses. A self-sustaining chain reaction requires that at least one of the neutrons emitted in each average fission be captured to produce another fission.

Converter Reactor. One that is similar to a production reactor except that it utilizes the fissionable material produced to augment its lifetime.

Converting. The process whereby neutrons are used to transmute thorium-232 into uranium-233, or uranium-238 into plutonium-239. Less specific than breeding.

Critical. Term used to describe the condition of a nuclear reactor in which a fission chain reaction is being maintained at a constant rate. A reactor in which the rate of fissioning is more than that required for criticality is said to be "supercritical," and one in which the rate is not sufficient to sustain a chain reaction is termed "subcritical."

Critical Mass or Size. The mass of nuclear fuel for which a chain reaction proceeds at a constant power level. This is related, among other things, to the volume it occupies, or size.

Cross Section, Nuclear Cross Section. The effective area of the target nucleus exposed to bombardment by nuclear particles, used to express the probability of a particular nuclear process or reaction.

Electron. An elementary particle orbiting about the nucleus and balancing a proton positive charge.

Enriched Uranium. Uranium containing a higher than the normal percentage of U-235.

Fission, Atomic. The splitting of the nucleus of a heavy atom, as uranium or plutonium, into nuclei of lighter atoms—the fission products.

Fission Products. Elements that result from atomic fission. In addition to uranium and plutonium, these may consist of more than 40 different radioactive elements—barium, iodine, cerium, arsenic, silver, tin, cadmium, and others.

Fissionable Material. Any element or isotope, the nuclei of which can be caused by nuclear bombardment to undergo nuclear fission and to produce a fission chain reaction.

Fuel Burn-Up. As a reactor continues to operate, the fissionable material is gradually used up and the reactivity will decrease accordingly. This is known as fuel depletion, or burn-up, and amounts to a loss of about 1 gram of U-235 per megawatt-day of operation. Burn-up is normally expressed in terms of megawatt-days per metric ton of uranium metal or uranium oxide.

Half-Life. The length of time it takes a radioactive element to decay to the point where one half of the original material remains.

Heavy Water. Water in which the hydrogen of the water molecule consists entirely of the heavy hydrogen isotope of mass two. It is used as a moderator in certain types of nuclear reactors.

Isotope. One of a group of two or more nuclides having the same number of protons (the same atomic number) but different numbers of neutrons.

Moderator. A substance, such as graphite or heavy water, used in a reactor to slow down neutrons from the high energies at which they are released in fission to lower energies at which they cause fission more readily.

Multiplication Factor, k. The ratio of the number of neutrons produced by nuclear fission to the number of neutrons absorbed in the reactor.

Neutron. An electrically neutral nuclear particle of mass number 1.

Neutron Cross Section. A measure of the ability of a material to interact with neutrons by scattering, capturing, or being fissioned by them.

Neutron Flux. A term used to express the intensity of neutron radiation, usually used in connection with the operation of a reactor.

Nucleon. A general term for a proton or a neutron.

Nuclide. An atomic species characterized by the constitution of its nucleus, i.e., the numbers of protons and neutrons and energy content.

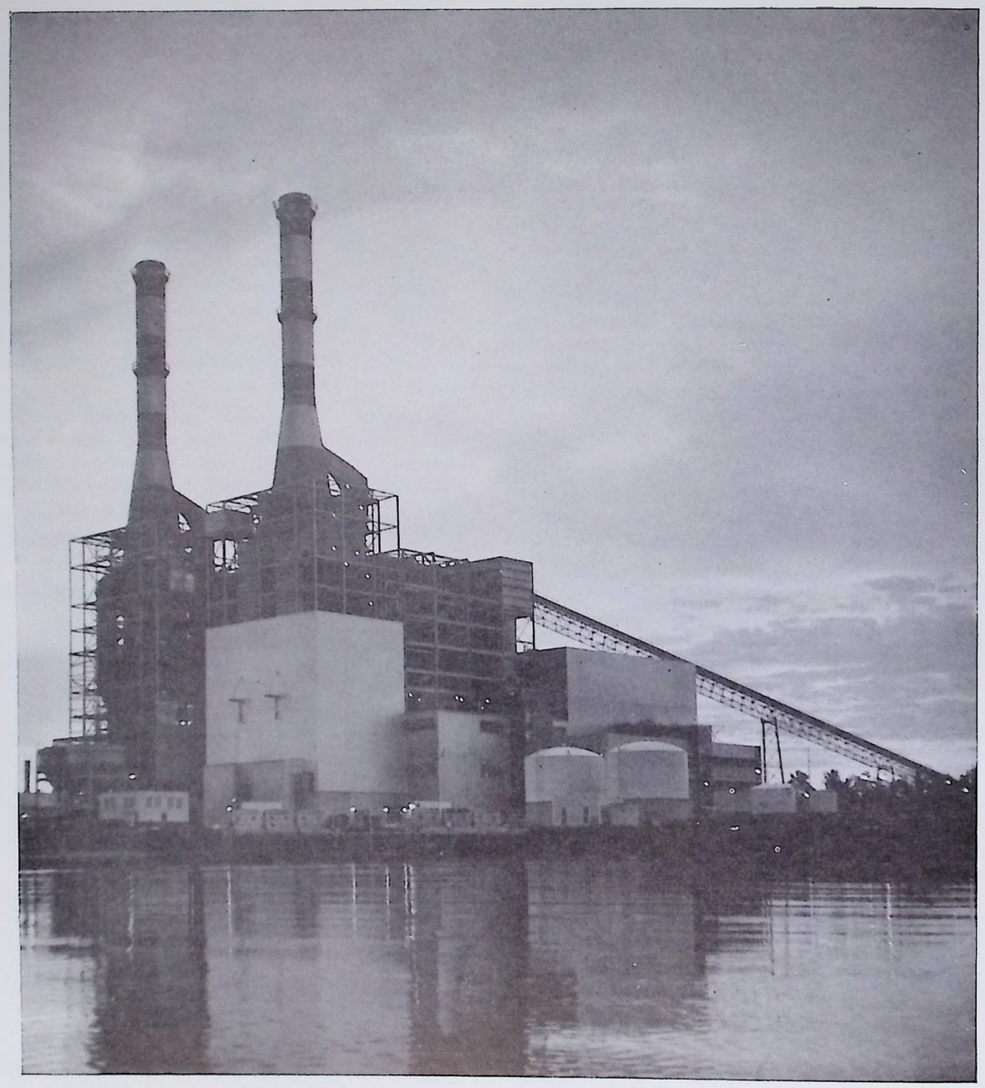
Proton. A positively charged elementary particle of mass number 1.

Reaction Cross Section. The effective area presented by a target nucleus to a bombarding neutron under defined conditions and for a given result.

Reactivity. The amount by which the multiplication factor exceeds one is called "excess multiplication factor," reactivity, or excess k and is usually expressed in terms of per cent of the multiplication factor.

Thermal Reactor. One in which most of the fissions result from the capture of thermal neutrons. In intermediate and fast reactors, most of the fissions result from the absorption of intermediate and fast neutrons, respectively.

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CYCLONE FURNACES FIRE THIS EASTERN CENTRAL STATION

28

THE CYCLONE FURNACE

OVER the years continued advances have been made in the methods of burning coal. The motivating factors were mainly economics, growth of industry, and higher standards of living. Economics demanded the efficient use of the cheapest coals available. Growth of industry and higher living standards required more steam-generating units of higher capacities than those attainable with the older types of equipment. In the early 1920's intensive work on pulverized-coal firing was started to overcome the limitations of the stokers available at that time. The advantages that pulverized-coal firing offered were:

1. The use of the most economical size of coal available.
2. The removal of capacity limitations imposed by stokers.
3. Improvement in response to load.
4. The ability to burn all ranks of coal from lignite through anthracite.
5. Ease of combination firing of oil and gas with coal.
6. An increase in thermal efficiency because of lower excess air for combustion and lower carbon loss than that with stoker firing.
7. A reduction in operating labor requirements.

Today, pulverized-coal firing is highly developed and is considered an excellent way to burn many types of coal, particularly the higher grades and ranks. However, in its development and application, experience has shown this method of firing to have certain limitations and undesirable features, such as:

1. Large power consumption required for driving pulverizers.
2. High maintenance costs for pulverizers.
3. Excessive fly-ash discharge from the stack (approximately 80% for dry-ash-removal furnaces without dust collectors).
4. Erosion of boiler pressure parts by fly-ash entrained in the flue gases (unless low gas velocities are maintained).
5. Erosion of induced-draft fan blades and scrolls, even when the fans are located after dust collectors.
6. Lower availability and increased maintenance because of eroded components.
7. Relatively large furnace volumes required for good combustion.

For a clear understanding of the last item, a brief review of combustion and the functions of a furnace may be helpful. Very simply, combustion of coal is a chemical reaction in which carbon combines with oxygen to form CO_2. This gas tends to blanket the coal particles and retard further combustion. To maintain rapid combustion, the CO_2 must be scrubbed away and every particle of coal brought rapidly into intimate contact with additional oxygen.

Pulverized-coal firing requires that the coal be reduced from 1- or 2-in. diam lumps to a powder, so fine that approximately 70% will pass a 200-mesh screen, which requires a large amount of power. The finely pulverized coal is then very intimately mixed with combustion air in the burner; however, after this initial mixing the tiny coal particles are merely carried along in the air stream, and very little additional scrubbing by the air occurs. Thus, further contact of oxygen with the coal must be largely by diffusion. The furnace consequently has to be relatively large to give the necessary retention time for oxygen to diffuse through the blanketing CO_2 layer to reach the coal particles, and, at the same time, temperatures must be sufficiently high to complete combustion. After combustion, since the residual ash particles are much smaller than even the original tiny pulverized-coal particles, the former are easily carried along with the flue gases from the furnace and through the boiler setting.

At the same time the pulverized-coal-fired boiler furnace also has the function of cooling the com-

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bustion gases, so that when they enter the convection surfaces they are below the temperature at which slagging occurs. This function conflicts with that of maintaining the high temperatures necessary to complete combustion. It would therefore be preferable to separate these functions and use the boiler furnace for cooling only. This would require a separate small combustion chamber where high turbulence and temperature may be maintained.

For many years engineers recognized this need and actively explored basic changes in the design of furnaces and fuel-burning equipment to improve combustion and furnace performance. In addition, significant changes in the availability and use of coal further increased the need for new designs. For example, demands for the higher grades of coal have depleted many seams, and many others have been reserved for metallurgical and other uses. Mechanization in coal mining has increased the ash content of mined coal. Washing is widely used to lower ash and sulfur contents; however, this is an added expense (see Chapter 15). The industrial growth of the Western portion of the country, rich in reserves of subbituminous and lignitic coal, is rapidly increasing the consumption of these lower ranks of coal. This has furthered the need for equipment fully suitable for the lower grades and ranks of high-ash low-fusion-temperature coal.

The Cyclone Furnace is an outgrowth of efforts by many engineers to meet these needs and to overcome some of the difficulties encountered with other firing methods. It is a significant step in improving the methods of firing fuels and bettering the functional, design, and performance characteristics of boilers and related equipment.

Principle of Operation

The Cyclone Furnace (Fig. 1) is a water-cooled horizontal cylinder in which fuel is fired, heat is released at extremely high rates, and combustion is completed. Its water-cooled surfaces are studded and covered with refractory chrome ore. Coal crushed in a simple crusher, so that approximately 95% will pass a 4-mesh screen, is introduced into the burner end of the cyclone. About 20% of the combustion air also tangentially enters the burner and imparts a whirling motion to the incoming coal. Secondary air with a velocity of approximately 300 fps is admitted in the same direction tangentially at the roof of the main barrel of the cyclone and imparts a further whirling or centrifugal action to the coal particles. The combustible is burned from the fuel at heat release rates of 500,000 to 900,000 Btu/cu ft, hr, and gas temperatures exceeding 3000 F are developed. These temperatures are sufficiently high to melt the ash into a liquid slag, which forms a layer on the walls of the cyclone. The incoming coal

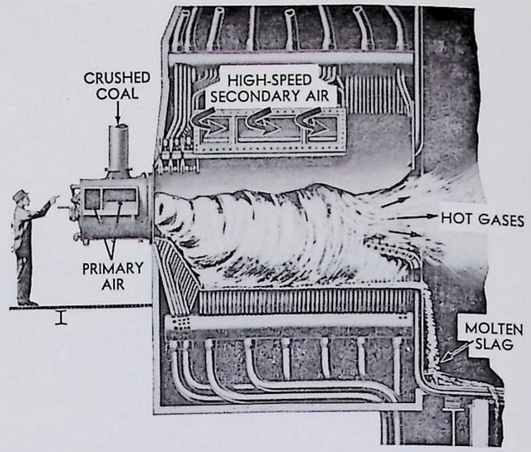


Fig. 1. *The Cyclone Furnace, in the form of a horizontal cylinder, is completely water cooled by connection to the main boiler circulation. All combustion gases leave through the re-entrant throat at the rear. Molten slag drains from the bottom at the rear through a small opening into the adjacent boiler furnace*

particles (except for a few fines which are burned in suspension) are thrown to the walls by centrifugal force, held in the slag, and scrubbed by the high-velocity tangential secondary air. Thus the air required to burn the coal is quickly supplied, and the products of combustion are rapidly removed. Even though the release of heat per cu ft in the Cyclone Furnace is very high, the total amount of heat actually absorbed is relatively low. This is because of the small amount of surface in the cyclone and the insulating properties of the covering slag layer. Heat absorption rates range from 40,000 to 100,000 Btu/sq ft, hr, as shown in Fig. 2. This combination of high heat release and low heat absorption insures the high temperatures necessary to complete combustion and to provide the desired liquid slag covering of the surface.

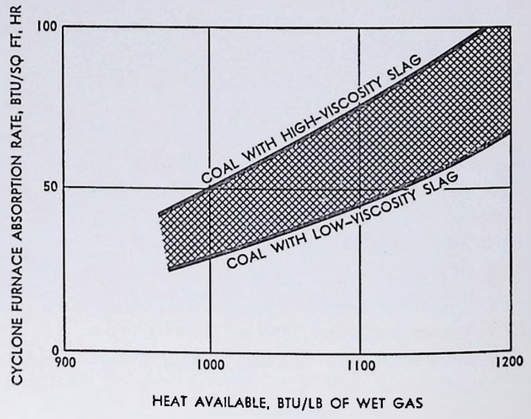


Fig. 2. *Heat absorption rates in the Cyclone Furnace*

The gaseous products of combustion are discharged through the water-cooled re-entrant throat into the gas-cooling boiler furnace. Molten slag in excess of the thin layer retained on the walls continually drains toward the rear and discharges through the tap hole (shown in Fig. 1) to the boiler furnace, from which it is tapped into a slag tank and disposed of in conventional manner.

The results of this method of combustion are that the fuel is burned quickly and completely in the small cyclone chamber and the boiler furnace is used only for cooling the flue gases. Most of the ash is retained as a liquid slag and tapped into the slag tank under the boiler furnace. Thus, the quantity of fly-ash is low and its particle size so fine that erosion of boiler heating surfaces has never been experienced even at high gas velocities.

Suitability of Fuels for the Cyclone Furnace

In the production of power, no one method of burning fuel has proved suitable for all classifications of fuels under all conditions, and the Cyclone Furnace is no exception; however, its range in use of fuels is relatively broad. For example, of the 500,000 tons of bituminous and lignitic coal mined in this country in 1950, approximately 70% is considered suitable for use. Further, the poorer and cheaper grades of coal generally have fuel characteristics which are well suited. These factors are of considerable economic importance in the production of power. In buying fuel for Cyclone Furnace boilers, there are fewer restrictions in the selection of coal, and full advantage can be taken of the competitive fuel market and the fuel availability in the area of the plant. Fuel oils and gases, along with many other by-products from coal, petroleum, and wood, are suitable for firing (see *Operating Results*).

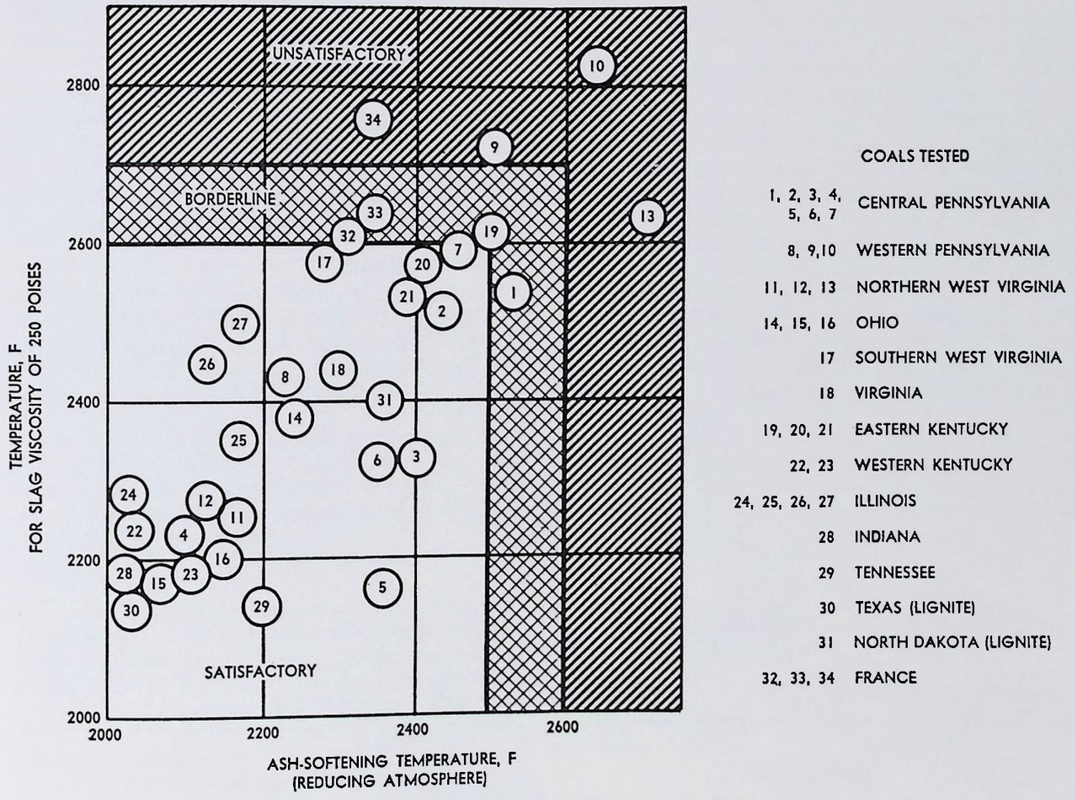


Fig. 3. *Tests of coal suitability for the Cyclone Furnace (600 F combustion-air temperature)*

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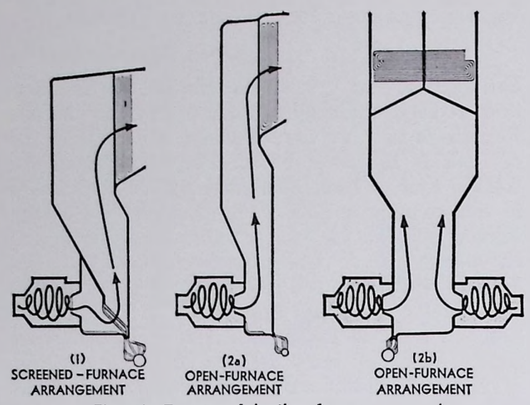


Fig. 4. *Types of boiler furnaces used with Cyclone Furnaces*

The suitability of most solid fuels is dependent upon the amount of moisture, the volatile matter, and the quantity and type of ash. Based on experience with both commercial and experimental units, it has been found that, for satisfactory combustion of solid fuel, the ash must form a liquid slag layer in the cyclone and the rate of deposit of fuel particles in the slag layer must not exceed the rate of combustion. With a normal combustion-air temperature of 600 F, it has been found that the slag characteristics will be suitable (see Fig. 3) if the slag formed from the coal ash has a viscosity of 250 poises or less at 2600 F. Also, the volatile matter of the coal should exceed 15% on a dry basis, in order to obtain the required high combustion rate. For coals of very high moisture content it may be necessary further to limit the coal ash-softening temperature and slag viscosity. There is considerable give-and-take in these characteristics, and an increase in the temperature of the combustion air will generally permit extending these limits. Of the various criteria for establishing suitability of solid fuel for firing, the most significant is the viscosity of the slag formed from the ash. For most coals the slag viscosity may be calculated from the analysis of the ash and the ash-softening temperature in a reducing atmosphere, as discussed in Chapter 18. Ash content and ash quality are not criteria in the case of liquid or gaseous fuels, as a liquid slag coating of the furnace is not required to burn them.

Design Features

Boiler Furnace

There are two general types of gas-cooling furnaces developed for the Cyclone Furnace, namely, the screened furnace and the open furnace. These are shown in Fig. 4. In unit (1) a water-cooled slag screen divides the boiler furnace into lower and upper sections and serves to remove an additional amount of dust from the flue gases. In units (2a) and (2b) the products of combustion from the Cyclone Furnaces discharge directly into the boiler furnaces. For smaller units, sufficient firing capacity is usually attained with Cyclone Furnaces located in only one wall (see 2a). For large units, furnace width can often be reduced by installing them in both the front and rear boiler-furnace walls (see 2b).

Cyclone Furnace Capacities

The size and number of Cyclone Furnaces depend primarily upon the boiler size and the load flexibility required. They are built commercially in sizes ranging from 5-ft diam through 10-ft diam, with an allowable range of maximum heat input per furnace of approximately 100 million to 500 million Btu/hr.

Coal Preparation

There are two general types of coal preparation and feeding arrangements (see Fig. 5), the direct and the bin or storage systems. The latter is preferred for most bituminous coals when the plant layout permits. With this system coal is crushed in a central

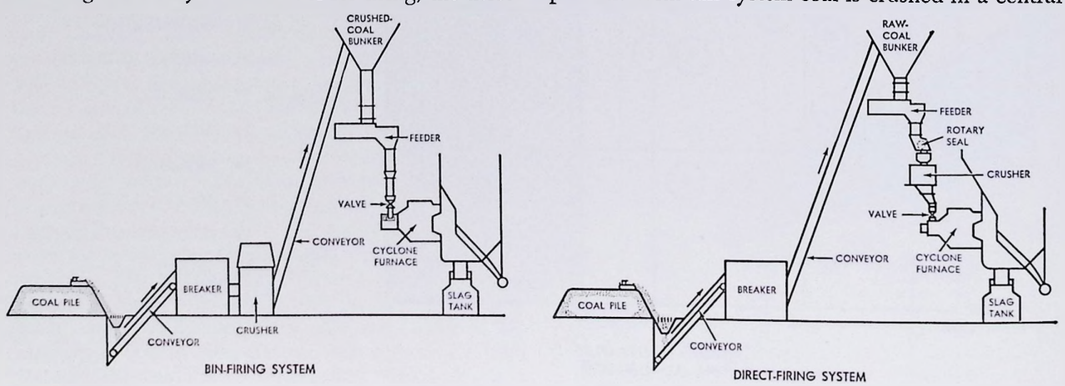


Fig. 5. *Bin-firing and direct-firing systems for coal preparation and feeding to the Cyclone Furnace (schematic)*

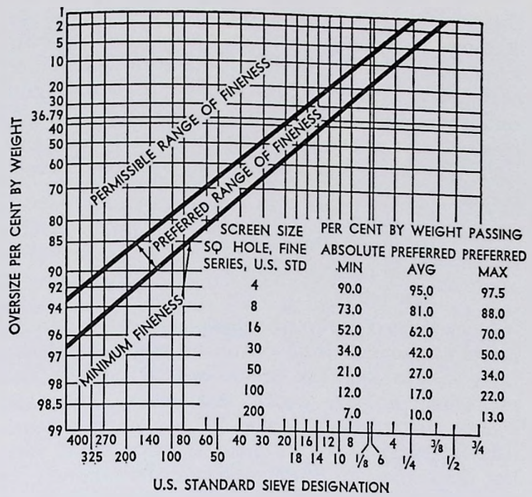


Fig. 6. *Sizing of crushed coal fired in the Cyclone Furnace*

preparation plant to a sizing suitable for firing, and the crushed coal is delivered to the bunker. The range of sizing of crushed coal required with either system is given in Fig. 6. Because the crushed coal is relatively large in particle size, the hazards associated with pulverized-coal storage systems do not exist. The only precaution necessary is to provide adequate venting of the bunkers to insure removal of the small amounts of combustible gases released from freshly crushed coal of certain types. With the bin system the amount of equipment in the boiler room is reduced, and short crusher outages can be accommodated without interrupting boiler operation.

The second method of coal preparation is the direct system, which has a separate hot-air-swept crusher located between the feeder and the burner of each Cyclone Furnace. The advantage of this system is that the intimate mixing of coal and hot air in the crusher helps to dry the coal. This improves crusher performance and ignition with high-moisture coals, such as lignite. It is also often easier to accommodate the direct system in existing plants, where the coal-handling equipment cannot readily be adapted to the bin system.

Coal Feeders

The coal feeder normally used is a drag type, shown in Fig. 7. An angled cutoff plate is provided at the coal discharge from the feeder to insure a uniform rate of feed. This is necessary because the coal is burned almost instantaneously when it reaches the Cyclone Furnace, and fluctuations in feed are reflected in combustion. The rapidity of combustion makes the Cyclone Furnace very responsive to load demands, and it has been demonstrated that the rating will change just as quickly as the coal-feeder

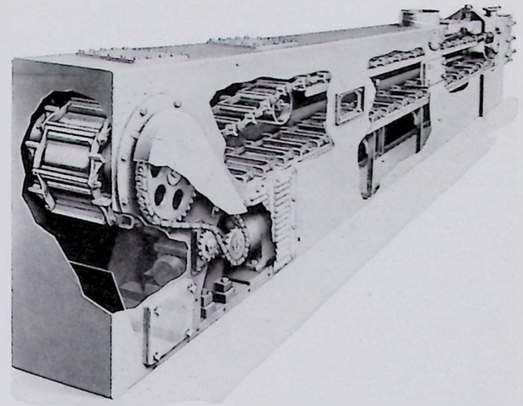


Fig. 7. *Drag feeder for the Cyclone Furnace*

speed. Belt feeders with continuous weighing devices have been used to serve the dual function of coal scale and feeder.

Oil and Gas Burners

As mentioned above, the Cyclone Furnace is excellent for the burning of oil or gas. These fuels can be burned at ratings and with performance equal to those with coal firing. Streams of oil may be injected either into the secondary-air stream or through the front coal burner (see Fig. 8), where the oil is picked up and atomized by the high-velocity air. Gas is fired through flat ports located in the secondary-air entrance to the cyclone. The gas burners and the oil burners located in the roof may be left in place when coal is fired, which permits switching from one fuel to another by remote control without having to remove the Cyclone Furnace from service. This feature makes the Cyclone Furnace well suited for firing multiple fuels. For installations where a unit may be operated for some time on oil or gas but will later be converted to coal, it is possible to arrange for firing oil or gas with air pressures well below those required for coal, thus affording a considerable saving in power during the period when the alternate fuels are burned. For such installations, 40 to 50% of the total air is admitted as primary air

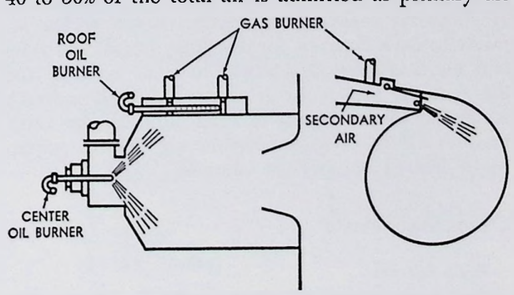


Fig. 8. *Oil and gas burners for the Cyclone Furnace*

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compared with 15% for coal firing. With the lower air pressure, atomizing oil burners are used.

Slag-Handling Equipment

Slag-handling equipment for a Cyclone Furnace boiler is similar to that for a pulverized-coal slag-tap boiler, except that since the per cent of ash recovered in the Cyclone Furnace is higher the capacity of the slag-handling equipment must be correspondingly greater. There are two principal types of handling in use, as shown in Fig. 9. The first consists of one or more large storage tanks located under the furnace floor, into which slag is continuously tapped, quenched, and allowed to accumulate. It is withdrawn at intervals and discharged to a storage area, where it is recovered for sale or other use. The second type of slag handling is similar with the exception that a smaller storage volume is provided under the furnace floor, and the slag is removed continuously as it discharges into the water-quench tank. The advantages of the continuous handling system are that less space is required under the boiler furnace for the slag tanks and that no routine operating labor is required.

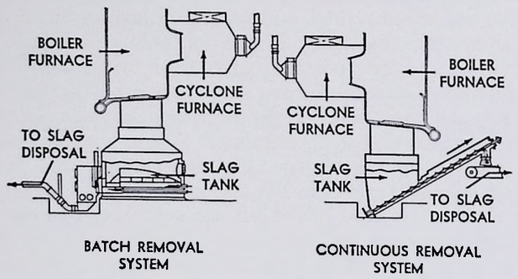


Fig. 9. *Slag-handling systems for Cyclone Furnace boiler*

Combustion Controls

Automatic combustion controls for Cyclone Furnace boilers are generally based on maintaining equal feeder speeds (feeders are calibrated to give equal volumetric flows for equal speed) and equal total air flows to individual Cyclone Furnaces. The air flows are measured by venturi meters in the air ducts to each furnace. In this way, equal coal rates and air flows are maintained to each cyclone, and the over-all excess air is controlled with a boiler meter based on steam flow-air flow in the usual manner. Often oxygen recorders are used as operating guides to monitor the controls.

Operating Results

Fuels Burned

The first commercial Cyclone Furnace boiler was designed to burn Central Illinois coal and was installed at the Calumet Station of the Commonwealth Edison Co., Chicago, in 1944. Since then this method of burning has been incorporated in many units throughout the United States and Europe. In this country coals of the following range of constituents have been burned in commercial Cyclone Furnace boilers:

Moisture, %	2 to 40
Volatile matter (dry), %	18 to 45
Fixed carbon (dry), %	35 to 75
Ash (dry), %	35 to 75

As mentioned previously under *Suitability*, a number of by-products from petroleum and many waste fuels, such as bark, can be successfully burned. The petroleum products (coke and pitch) range in volatile matter from as low as 5% for some petroleum coke to as high as 60% for pitch. Because of the very low ash content of these petroleum by-products, it is usually necessary to add a slag-forming material or to also burn coal with suitable ash characteristics to provide the necessary slag coating of the cyclone barrel. For the 5% volatile coke it is also necessary to burn 5 to 10% auxiliary oil or gas to stabilize ignition.

Coal chars resulting from the low-temperature carbonization of bituminous or lignitic coals have

Fig. 10. *Cyclone Furnace unit with screened boiler furnace. Maximum continuous steam output 1,523,000 lb per hr, 1050 F, 2400 psi design pressure. Five 9-ft-diameter Cyclone Furnaces per boiler*

sufficient ash and volatile matter and are suitable for firing, provided the ash viscosity characteristics are suitable. Fig. 10 shows one of three units designed for burning coal char of 15% volatile matter content and bituminous coal.

Two units designed to burn bark and bituminous coal in amounts up to approximately 50% by weight of each are in operation in a large Southern paper mill. To burn the bark successfully, it is necessary to fire sufficient coal to supply the slag coating in the cyclone and to maintain sufficiently high temperatures to provide the necessary fluidity of the slag. Preparation and feeding of bark are discussed in Chapter 19.

Power Requirements

Since the only coal preparation is crushing, the power required is low compared with that for pulverizing coal. To offset this, the forced-draft-fan power required is relatively high, the Cyclone Furnace air pressure drop being in the range of 20 to 40 in. of water compared with 2 to 10 in. of water for pulverized-coal burners. The net result is that power requirements for a Cyclone Furnace boiler are comparable to those for pulverized-coal firing. Fig. 11 shows the comparative power requirements for a wide range of coals. It will be seen that, for the low-grindability low-heating-value bituminous and lignitic coals, the power required is substantially less than that for a pulverized-coal-fired unit. However, for high-heating-value high-grindability bituminous coal, the power is greater.

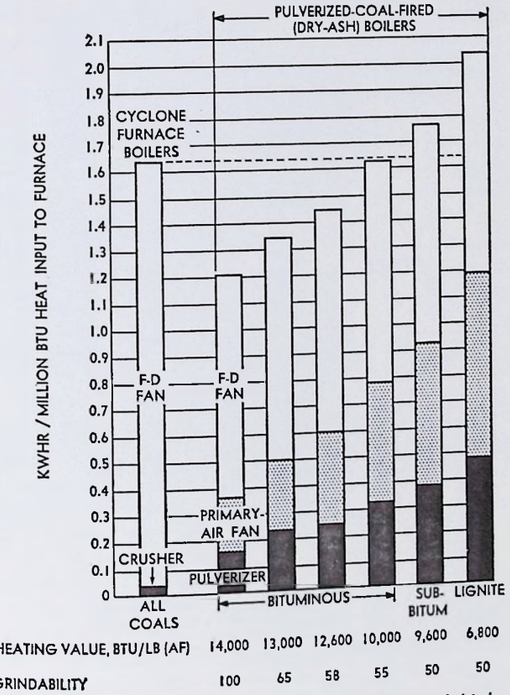


Fig. 11. *Auxiliary power requirements of typical high-capacity pressure-fired Cyclone Furnace and pulverized-coal units*

Combustion Efficiency

The excess air required for satisfactory combustion is less than 10%. However, where automatic controls are used and particularly where there are several Cyclone Furnaces for one boiler, excess air is usually maintained between 10 and 15% to insure that no individual cyclone is operating with insufficient air. When operating with coal of suitable sizing and with 10% excess air, the loss in efficiency from unburned combustible has been found to be less than 0.1% with most coals.

Ash Recovery and Dust Collectors

The results of dust-loading tests from a number of operating Cyclone Furnace units are given in Fig. 12. The dust loading of the flue gas is in the range of 5 to 15% of the ash in the coal, compared with about 80% for a dry-ash pulverized-coal-fired unit. For many installations the dust loading for the

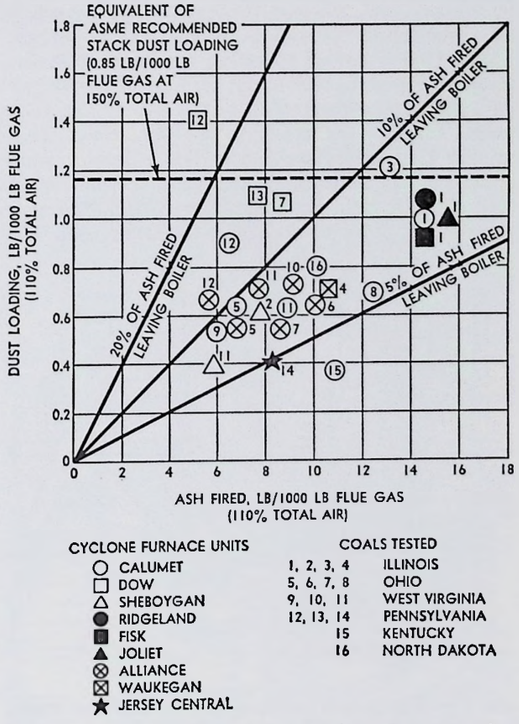


Fig. 12. *Fly-ash emission from Cyclone Furnace units (test results)*

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Cyclone Furnace without a dust collector is within the ASME recommendation of 0.85 lb per 1000 lb of flue gas at 150% total combustion air. This lower dust loading is important not only from the standpoint of reduced dust nuisance but also because of the reduction in boiler cleaning requirements and the elimination of erosion of convection heating surface and induced-draft fans. Low dust loading also permits the boiler to be designed for higher gas velocities through the convection banks, resulting in better heat transfer.

It is common practice to equip pulverized-coal-fired boilers with high-efficiency dust collectors to eliminate the dust nuisance. This, however, does not change the amount of fly-ash passing through the boiler. The dust loading of a pulverized-coal-fired unit with a dust collector is but little lower than that of a Cyclone Furnace boiler operating without benefit of dust collector. This comparison is illustrated in Fig. 13 for a large central station Cyclone Furnace unit and a pulverized-coal-fired unit arranged for dry-ash removal. Both units are of 200-megawatt capacity, each consuming approximately 1800 tons of coal per day and producing 360 tons of ash. With pulverized-coal firing approximately 290 tons per day of fly-ash passes through the boiler to the dust collector. The latter traps approximately 260 tons of fly-ash per day, allowing 30 tons, or approximately 8% of the ash in the coal, to discharge to the atmosphere. Under similar operating conditions with the Cyclone Furnace, only about 36 tons of fly-ash per day, or approximately 10% of the ash in the coal, passes through the boiler to the atmosphere. If a smaller dust collector of only 80% efficiency were added to the Cyclone Furnace unit, the amount of dust discharged to the atmosphere would be 7 tons per day, or 2% of the original ash in the coal. The dust from the collector may be returned to the Cyclone Furnace and recovered as slag, thus effecting a complete elimination of fly-ash handling and the near elimination of dust emission from the stack.

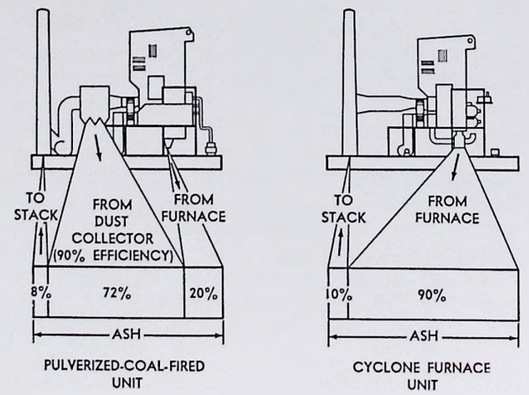


Fig. 13. *Comparison of fly-ash emission from typical large dry-ash-removal pulverized-coal-fired unit and Cyclone Furnace unit*

Typical Installations

Although Cyclone Furnace units were first used in central stations, their use has been extended to a wide variety of industries. Fig. 14 shows a unit installed in a large Northern industrial plant. The unit incorporates the direct-firing system, and for maximum ash recovery, the boiler furnace is of the screen type. Slag is collected in a large quench tank with batch discharge to storage.

Fig. 15 shows a smaller unit installed in a Southern chemical plant. This unit incorporates the bin

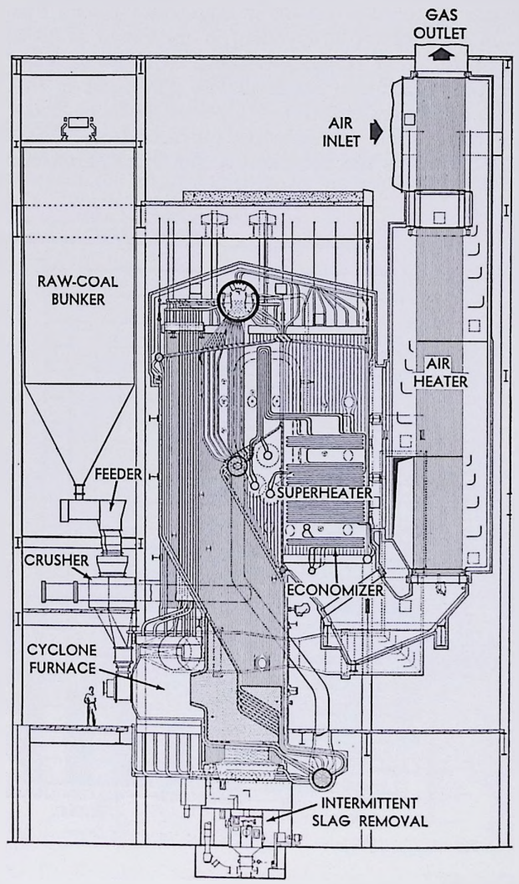


Fig. 14. *Cyclone Furnace unit with screened boiler furnace, direct-firing system, and intermittent slag removal*

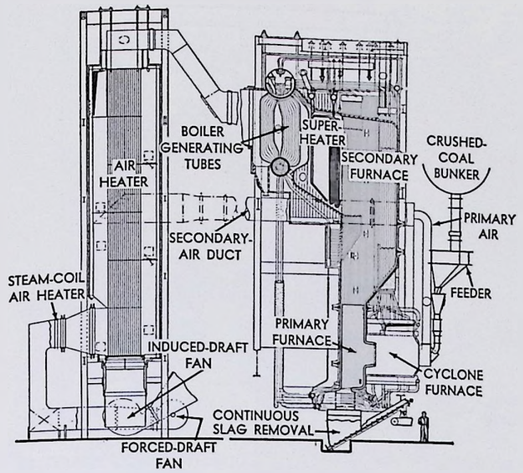


Fig. 15. *Cyclone Furnace unit with open boiler furnace, bin-firing system, and continuous slag removal*

system for firing and the open boiler-furnace arrangement. Slag is handled by a continuous removal system. Both this and the preceding unit are designed for pressure operation (without induced-draft fan).

The designs of Cyclone Furnace units have followed the general trend of the power industry to higher steam temperatures and pressures and larger capacities. Fig. 16 shows a 175,000-kw-capacity Universal-Pressure once-through unit with opposed Cyclone Furnace firing for subcritical pressure operation at 2475 psi, 1050 F superheat, and 1000 F reheat. Fig. 19 of Chapter 1 shows a Universal-Pressure once-through unit with opposed Cyclone Furnace firing for subcritical pressure operation at 2450 psi, 1053 F superheat, and reheat to 1003 F. This unit, one of the largest in the world, serves a 700,000-kw turbine.

Operation

Start-Up

The Cyclone Furnace can be started by the continuous firing of coal, oil, or gas. Coal is ignited by a permanently installed gas-lighting torch or a retractable oil-lighting torch inserted into the front secondary-air port. With coal firing, the usual load range for good operation of an individual furnace is from 100 to 50% of rated capacity, depending upon the ash characteristics; however, for short periods, such as start-up, loads of 20 to 25% can be carried with some increase in unburned combustible. The load range for continuous operation of the entire Cyclone Furnace unit is from full to approximately half load, again depending upon the ash characteristics and the furnace arrangement. For short periods, such as overnight load drops, lower ratings can be carried by allowing the slag tapped from the Cyclone Furnace to accumulate and solidify on the boiler-furnace floor. When the load is increased, this slag will melt and tap in the usual manner.

Operating Labor

Typical operating-labor requirements for a large central-station Cyclone Furnace unit are tabulated below. These are approximately the same as for an equal-size pulverized-coal-fired unit, as the operating-labor requirements are primarily dependent upon the extent of automation, and either type of firing is well adapted to automatic control.

Position	Men per Shift
Supervisory shift engineer	One for entire plant
Fireman	One per boiler
Auxiliary equipment operator	One per two boilers
Auxiliary equipment helper	One per two boilers

Maintenance

The principal items requiring maintenance are the coal crusher and the Cyclone Furnace burner. Crusher maintenance includes replacement of hammers and grid bars at yearly or less frequent intervals, depending upon the arrangement and the type of coal fired. In the burner, the coal is accelerated to

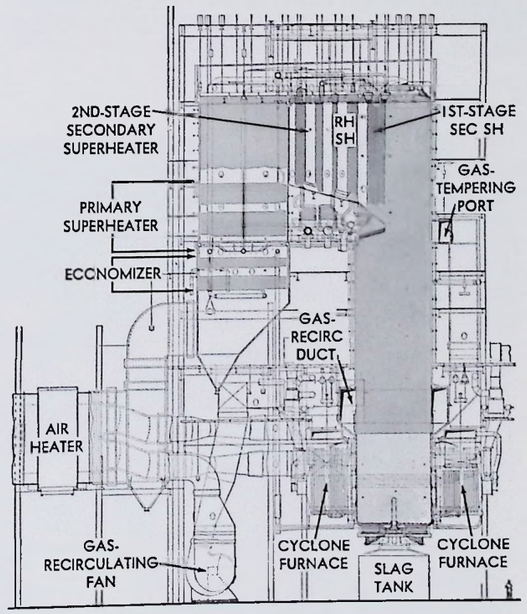


Fig. 16. *Universal-Pressure once-through boiler with open boiler furnace and opposed Cyclone Furnaces*

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the high velocity necessary to throw it against the slagged surfaces of the cyclone barrel. This high velocity causes erosion of the burner, which is minimized by the use of tungsten carbide or other erosion-resistant replaceable wear liners (Fig. 17). Since these liners normally last a year or more, they can be replaced or built up during regularly scheduled annual outages.

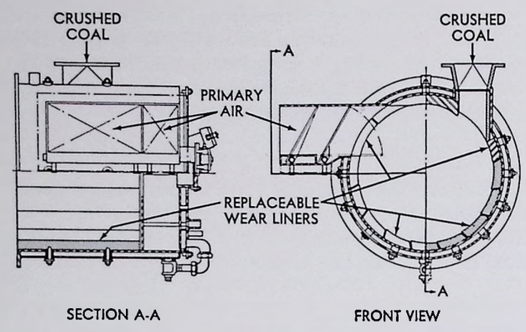


Fig. 17. *Burner of Cyclone Furnace showing replaceable wear liners*

The high availability and low maintenance of the Cyclone Furnace unit are evident from the outstanding record of the boiler shown in Fig. 18, installed in a Midwestern utility in late 1951. During the years 1952 through 1955, this unit had an availability of 98.4%, which included regularly scheduled annual outages.

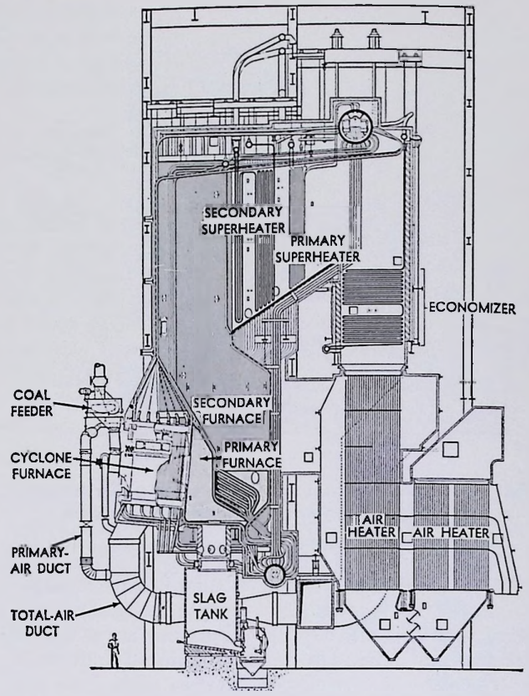


Fig. 18. *Cyclone Furnace unit with screened boiler furnace. Maximum continuous steam output 600,000 lb per hr, 960 F, 1525 psi design pressure. Three 8-ft-diameter Cyclone Furnaces per boiler*

Summary

The Cyclone Furnace has made a significant contribution to the steam-power industry by improving combustion and furnace performance, by permitting the economic utilization of the lower grades and ranks of coal, and by alleviating the problems caused by coal ash. Because of this, it has gained wide acceptance in our country and abroad. This is shown by the fact that in 1960 there were in service or under construction in the United States 371 Cyclone Furnaces serving 102 boilers with a combined capacity of 94 million lb of steam per hr. The corresponding figures for Canada, England, France, Germany, and Japan combined were 169 Cyclone Furnaces, 89 boilers, and 30.5 million lb of steam per hr. Although the Cyclone Furnace is an important advance in the art of fuel burning, engineers are continuing their efforts to improve its performance and to develop new and even better methods of burning fuel.

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MARINE APPLICATIONS

SINCE the Industrial Revolution, the ingenuity of many inventors and engineers has been devoted to the application of steam boilers and engines to ship propulsion. At first, because of low efficiency, steam was only a supplemental form of power used in dead-calm seas or in entering and leaving port, but continued experimentation with design and production techniques led to improvements which brought new economy and ultimately doomed the sailing vessel. Today the American Merchant Marine, faced with low-cost foreign-flag competition, must operate larger, more reliable, and more fully automated ships capable of high sustained speeds regardless of weather (Fig. 1). Modern marine-boiler designs, by providing efficiency and dependable power, make these ships possible.

Historical Development

Through the years marine boilers have evolved from units which were little more than closed containers holding water to the modern header type and the integral-furnace two-drum type of boiler almost universally used today.

By World War II, most new merchant vessels were equipped with boilers operating at pressures of 450 to 600 psi and temperatures of 750 to 825 F. Following the war, conditions in world shipping placed a greater emphasis on speed and efficiency, and the tendency has been to increase both steam pressure and steam temperature to provide power more economically. This is accomplished through the use of the two-drum integral-furnace type of boiler (Fig. 2).

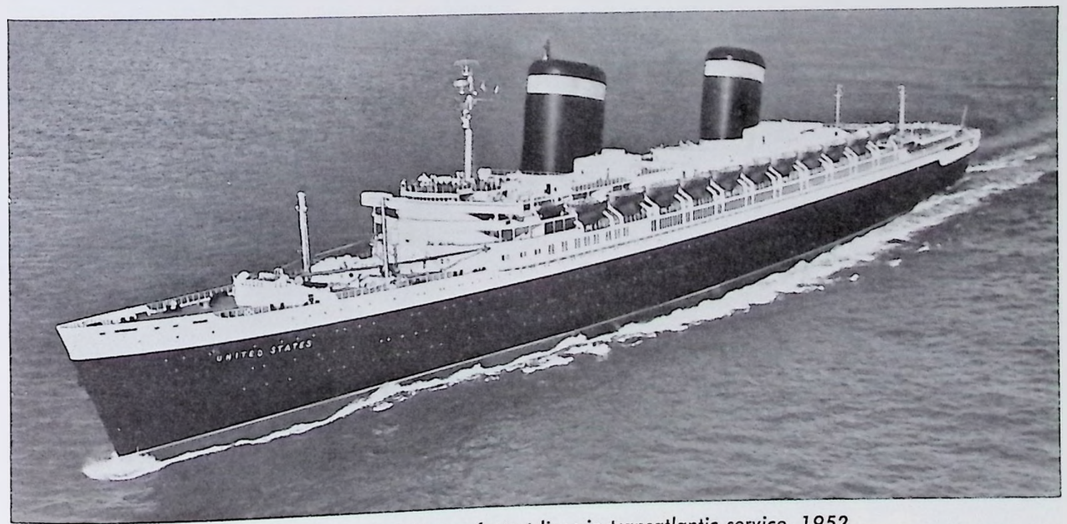


Fig. 1. *S.S. United States, fastest liner in transatlantic service, 1952*

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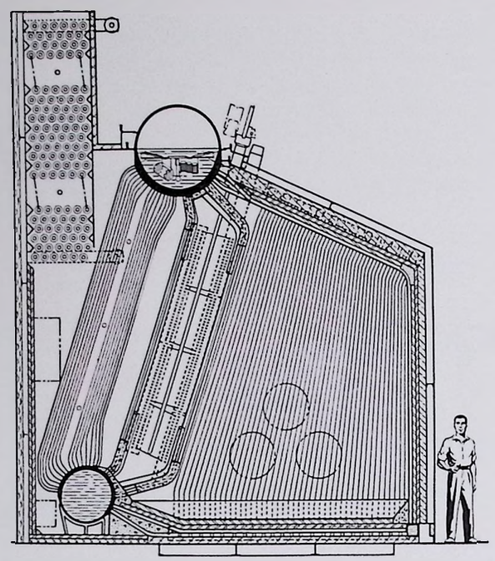


Fig. 2. *Two-drum integral-furnace marine boiler*

The trend to higher steam temperatures, coupled with the firing of residual fuel oils rich in slag- and corrosion-promoting constituents, resulted in design changes that provided better access to superheaters and thereby reduced the time required for cleaning and maintenance. B&W pioneered the use of an access cavity in the superheater bank and of superheater tube supports that are replaceable without removal of superheater tubes.

Despite this advanced design, slag deposition caused an excessive drop in average steam temperature between cleanings. To overcome this, mass-action retractable soot blowers, which had enjoyed great success in the stationary-boiler field, were first installed in 1952, and later an additional cavity was added to increase accessibility to the superheater.

Modern merchant ships (Fig. 3) with sectional-header or two-drum boilers are designed for high efficiency in order to minimize fuel costs. Consequently, almost all installations have auxiliary heat exchangers, such as economizers, steam air heaters, and gas air heaters. In addition, automatic feedwater controls and automatic combustion controls regulating wide-range oil burners have permitted reductions in operating personnel and labor costs, the other big factor in direct operating expense.

General Design Considerations

The selection and design of suitable steam-generating equipment for marine service follow the principles and considerations enumerated for stationary units, with some modification to meet the specific requirements for operation at sea.

Reliability is foremost in the list, since a ship at sea has none of the system reserve which backs up the usual land installation. Relatively simple steam cycles permit the use, in a limited space, of equipment that is readily accessible for operation and maintenance. It must also be exceptionally rugged to absorb vibration and the dynamic forces resulting from rolling and pitching in heavy seas or shocks which may result from accidental causes such as groundings and collisions. In the case of naval vessels, the effect of the detonation of explosives must also be considered.

Additional reliability is achieved through the careful selection of materials to withstand the high temperatures encountered in highly rated marine boilers and the corrosive atmosphere of the ocean environment in which they operate. High-strength materials

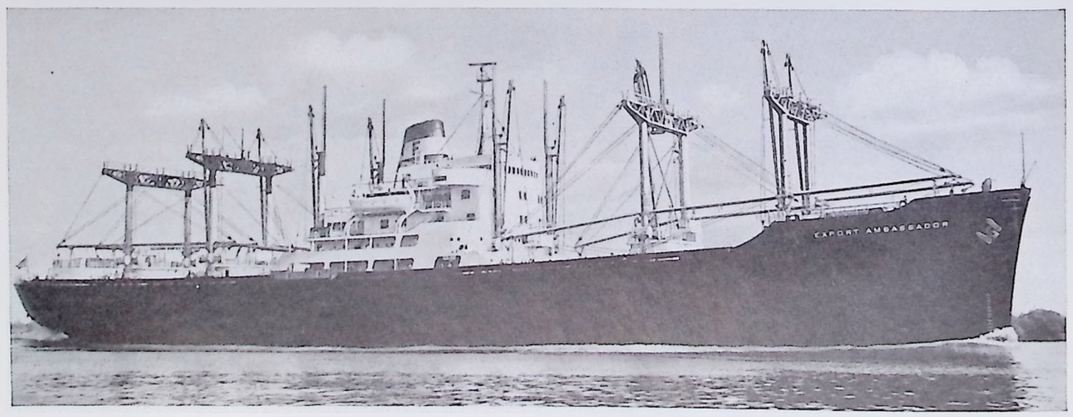


Fig. 3. *S.S. Export Ambassador, a post-WW II high-speed cargo ship*

provide light-weight construction with the desired ruggedness.

The ability to change power output rapidly over a wide range of load and to operate for long periods at high rating is another important requirement. Maneuvering a ship into and out of docks or in restricted waters may require frequent reversals of power with attendant fluctuations in boiler steam flow, oil flow, steam pressures and temperatures, and drum water levels. On the other hand, sustained high power outputs with high efficiency may be required on long passages by cargo and bulk-carrier vessels (see Fig. 4). To satisfy these requirements the marine engineer must consider 1) the performance characteristics of oil burners and furnace to assure complete and smokeless combustion, 2) boiler circulation and steam-separator performance to provide dry steam, and 3) auxiliary heat exchangers, such as economizers and air heaters, to give the desired efficiency.

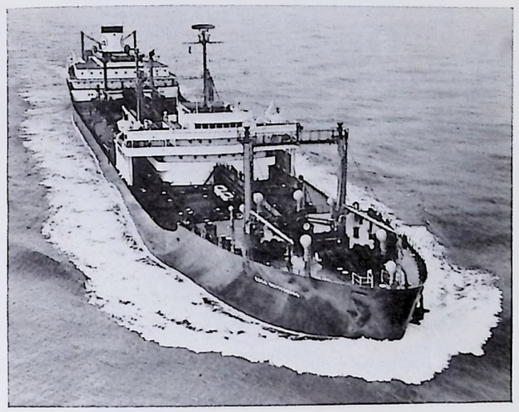


Fig. 4. *S.S. Esso Gettysburg, cargo and bulk carrier*

Naval combat vessels are usually designed for maximum efficiency at a rate corresponding to cruising horsepower (about 35% of rated full power), since they cruise at this power about 85% of their service life. Even so, naval boilers are capable of sustained operation at 120% of full-power rating without undue maintenance.

Boilers and related equipment for marine service are, in general, constructed in accordance with the rules and regulations of the codes set up by several different authorities, such as the U. S. Navy, the U. S. Coast Guard and Maritime Commission of the Department of Commerce, the American Bureau of Ships, and Lloyd's Register of Shipping. Boilers for nonself-propelled vessels may also have to conform to the ASME *Boiler and Pressure Vessel Code*. In addition to these general specifications, the design and construction of marine boilers must satisfy any special specification prepared by the owner and his architect.

Many years of shipbuilding have established the general requirements for the main propulsion plant and auxiliary equipment. Operating experience has established the fuel-input rates, steam flows and temperatures, pressure-part details, materials, and design arrangements as well as the operating procedures for satisfactory service.

The boilers of almost all ocean-going steam vessels built in the U.S. since World War I have been designed for oil firing. The few boilers designed for coal firing are usually larger and heavier than corresponding units burning oil. Larger furnaces are required to provide sufficient grate area and heating surface, and hoppers must be arranged to handle fly-ash. The use of heavy residual fuel oils, with ash rich in corrosive slag-forming constituents, has led to the establishment of minimum clearances that must be maintained between the oil burners and the adjacent furnace surfaces to assure complete combustion and to avoid deterioration of refractory. Their use has also necessitated limitations on maximum metal temperatures of superheater tubes and supports to avoid vanadium corrosion at elevated temperatures. Likewise, the minimum temperatures of cold-end heat-exchanger surface in economizers and air heaters must be limited to avoid sulfur corrosion at or below the dew point (see Fig. 16, Chapter 11). Slag and soot accumulations also impose limitations on tube spacings. However, if distillate fuels are used, these considerations become less important, and it is possible to decrease the over-all size of the boiler provided other design criteria are not violated.

The restricted space in marine service has made it necessary to use small tubes and relatively high rates of heat input. Consequently good feedwater is essential, and evaporated and deaerated make-up feedwater, with careful control of waterside chemistry in the boiler, is essential for most marine installations. Where it is impossible to provide these safeguards, it is necessary to design for a more conservative rating and to operate the boiler accordingly.

Steam temperature can be controlled through inherent design characteristics of the superheater and through the use of control devices, such as attemperators (desuperheaters) or gas by-pass arrangements. Steam for auxiliary services, such as galley, space, and cargo heating, deck machinery, salt-water evaporators, and electric-generator-plant requirements, may be obtained from auxiliary surface-type desuperheaters installed in the steam or water drum or from spray or bucket types external to the boiler.

Although some smaller-capacity low-rating boilers still continue to operate with natural draft, most units are now fitted with forced- or induced-draft fans, or

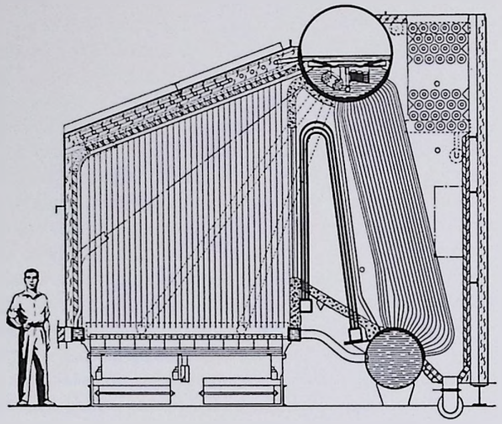
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both, to reduce size and weight. Insulated double casings, now the rule in all modern constructions, prevent flue gas leakage into the fireroom and provide lower outer-casing temperatures and smaller radiation losses than were previously obtained with single-cased boilers.

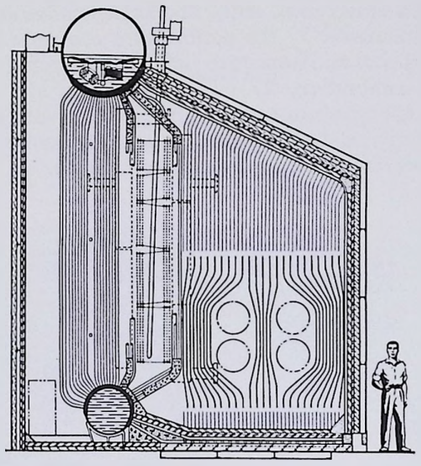
A few of the outstanding characteristics of representative boiler types currently installed in new ships are given below. In most instances, modifications of the basic design concepts can be readily made to satisfy the space or operating requirements of a particular vessel design.

Two-Drum Marine Boiler, Merchant Type

Oil and Coal Fired



COAL FIRED
VERTICAL SUPERHEATER



OIL FIRED
HORIZONTAL SUPERHEATER

General Description

This integral-furnace type of boiler has a single gas pass and an insulated welded double casing for pressurized operation. The furnace is water cooled by closely spaced 2-in. OD tubes at the side, roof, and rear walls and may also be equipped with a water-cooled burner wall at the owner's option. Supply tubes cool the floor. Steam-drum diameters range from 48 to 54 in. and water drums from 30 to 36 in.

An inclined or vertical saturated bank, composed of 2 to 4 rows of 2-in. OD screen tubes and 17 to 24 rows of 1¾-in. OD generating tubes, may be used, depending upon space and performance considerations. The superheater consists of either horizontal or vertical 1¾-in. OD tubes and may be arranged with one or two walk-in access cavities. The unit can be operated alone or, as is usually the case, in conjunction with a steam air heater and an economizer or air heater.

Range in Size, Steam Output
Lb per hr, to 200,000 in no fixed increments

Design Pressure
Lb per sq in., 150 to 1250

Operating Pressure
Lb per sq in., 100 to safety-valve setting

Steam Temperature
Saturation to 950 F

Fuel
Oil, light fractions to heavy residuals; coal, bituminous, spreader- or underfeed-stoker fired

Operational Control
Manual, or completely automatic, including feedwater flow, combustion, and steam temperature

Draft Loss at Maximum Output
In. of water, from 15 to 20 through all components

Dimensions Outside Setting, Approximate
Vary widely, from smallest, 7′-8″ high × 5′-5″ wide × 5′-6″ deep, to largest, 31′-5″ high × 23′-6″ wide × 18′-6″ deep

Indicated Field of Application

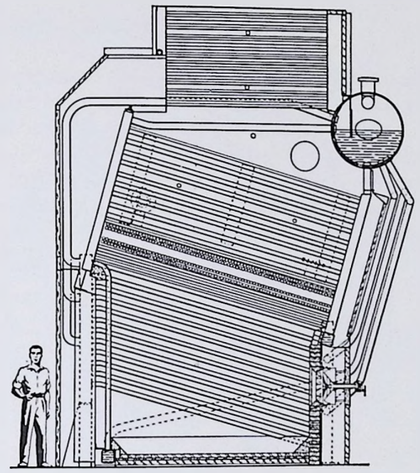
1. Chiefly for propulsion power and attendant requirements, although also used without a superheater for auxiliary or heating service in the smaller sizes.
2. Where space is at a premium and a compact light-weight design is needed.
3. Where feedwater is of good quality and it is unnecessary to inspect tubes visually.
4. With single superheater cavity when the slagging characteristics of Bunker C oil are average, two cavities when more severe.
5. Where fly-ash carry-over from spreader or underfeed stoker is average.

General Comments

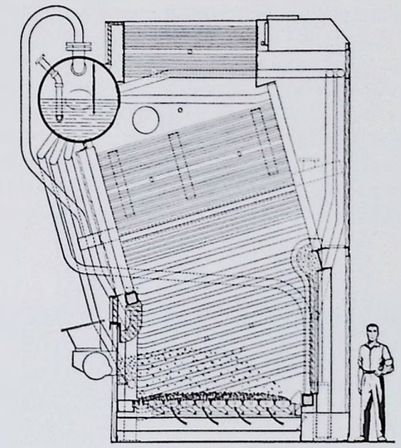
This design is standard for vessels in which large power plants must be installed in a minimum of space and where weight-saving is a vital consideration. Many possible variations in configuration permit application under difficult space conditions. Mass-action retractable soot blowers are recommended for the superheater zones of oil-fired units. The additional access cavity (when incorporated) facilitates rapid water-washing, cleaning, inspection, and maintenance of the superheater and the boiler bank.

Sectional-Header Boiler, Merchant Type

Oil and Coal Fired



OIL FIRED



COAL FIRED

General Description

Standard straight-tube units are available from 7 to 35 sections in width, with 2-in. OD screen tubes ahead of the superheater arranged in clusters of 2 or 4 tubes and clusters of nine 1¾-in. OD tubes in the saturated bank. Superheaters have 1¾-in. OD tubes arranged in 2 to 8 rows. Gas flow is upward through a single gas pass. Steam drums are normally 48-in. OD. Furnace side walls are cooled by spaced inclined 3¼-in. tubes. Boilers are usually double cased to the top of the superheater and single cased above. Casing is all welded. For increased efficiency, air heaters or economizers may be included. Boilers may be installed with steam drums either athwartship or fore-and-aft, as space conditions permit.

Range in Size, Steam Output
Lb per hr, to 150,000 in no fixed increments

Design Pressure
Lb per sq in., 150 to 965

Operating Pressure
Lb per sq in., 100 to safety-valve setting

Steam Temperature
Saturation to 910 F

Fuel
Oil, light fraction to heavy residuals; coal, bituminous, spreader- or underfeed-stoker fired

Operational Control
Manual, or completely automatic, including feedwater flow, combustion, and steam temperature

Draft Loss at Maximum Output
In. of water, from 10 to 15 through all components

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Dimensions Outside Setting, Approximate
Smallest, 11′-11″ high × 7′-10″ wide × 12′-4″ deep
Largest, 17′-10″ high × 23′-10″ wide × 18′-7″ deep

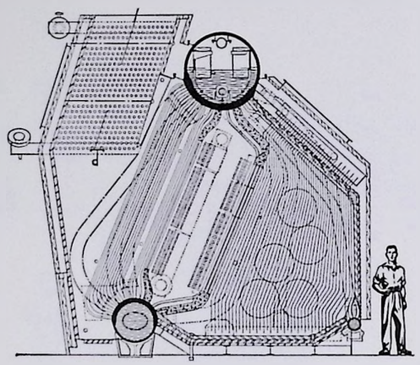
Indicated Field of Application

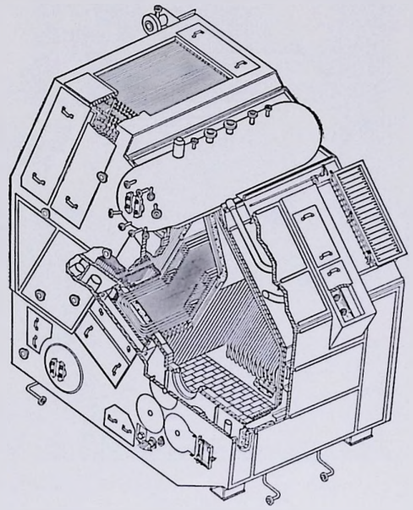
1. Chiefly used for propulsion power and attendant requirements but may be installed for auxiliary or heating service.
2. Particularly well suited for vessels equipped with steam engines or those which operate with large percentages of make-up feedwater, since oil or solid deposits can be observed and removed readily.

General Comments

The header-type boiler has certain fundamental advantages over other types. The generating tubes are straight and of few sizes, which reduces the number of spares that must be carried. Each tube can be cleaned and inspected easily and, if necessary, removed and replaced independently of any other without having to enter the drum. The time and cost of erecting and maintaining this unit are much less than required for a drum-type boiler. It is easy to make an adequate check on the condition of the equipment or the work done, since it is unnecessary to enter the drum.

Two-Drum Boiler, Naval Type





General Description

This integral-furnace type of boiler is fitted with high-strength low-weight stainless or low-alloy steel casings, either welded or bolted. Furnace roof, side, and rear walls are 2-in. OD touching tubes. Front (burner) wall and floor are refractory. Steam-drum diameters range from 46 to 56 in. and water drums from 27 to 30 in. Two to four rows of 2-in. OD tubes form the superheater screen. Cavity-type superheater has 1-in. OD tubes up to 8 rows deep arranged for complete drainability. Saturated boiler bank is usually inclined and has from 18 to 23 rows of 1-in. OD tubes. For maximum efficiency, boiler is fitted with stud-tube economizer and, in some cases, steam air heaters.

Range in Size, Steam Output
Lb per hr, to 350,000 in no fixed increments
Design Pressure
Lb per sq in., up to 1400
Operating Pressure
Lb per sq in., up to 1200
Steam Temperature
Saturation to 1000 F
Fuel
Oil, Navy Special Type
Operational Control
Manual to complete automatic combustion and feedwater regulation
Draft Loss at Maximum Output
In. of water, about 75 through all components
Dimensions Outside Setting, Approximate
Smallest, 14′-2″ high × 15′-1″ wide × 10′-6″ deep
Largest, 21′-5″ high × 18′-11″ wide × 16′-0″ deep

Indicated Field of Application

1. Combatant naval vessels or specialized installations.

General Comments

The nature of the construction and the rating at which these units operate limit their use to naval or high-speed naval auxiliary vessels requiring maximum power in minimum space. They are designed for maximum efficiency at cruising speed. Some attainable efficiency is sacrificed to develop high power-to-weight and power-to-cubic ratios. Construction utilizes light-weight tubes, high-tensile drum plate, and other expensive special features to minimize weight.

Marine Boiler Design

The techniques used in the design of marine boilers are similar to those applied to stationary units. Reference should be made to *Boiler Design*, Chapter 11, page 11-10, and *Selection of Steam-Producing Equipment*, Chapter 23, for an indication of general design requirements. To these must be added specific requirements, such as space conditions including major fireroom interferences and preferred boiler, burner, and cold-end heat-exchanger types.

The type of fuel can have considerable bearing on the design. Regardless of whether oil or coal is used, its analysis and ash characteristics are particularly important and should be specified so that the selected arrangements of heat transfer surface will minimize ash and slag deposition and corrosion.

The selection of a suitable boiler for a particular application starts with the determination of the dimensions and arrangement of a furnace within which the desired fuel-burning equipment can be installed and the required fuel can be burned with good combustion efficiency, without exceeding allowable furnace heat-absorption rates. This is particularly necessary for high-speed naval vessels, such as the frigate shown in Fig. 5.

The furnace arrangement and dimensions generally establish the over-all size of the boiler tube bank, so the first step in designing a new boiler is to make a preliminary layout of a suitable furnace. To do this, it is first necessary to ascertain the amount of fuel to be fired. In the usual case, with oil as the fuel, it is then possible to assume the type and number of burners to be used, the possible draft loss allowable through them for good combustion conditions, the

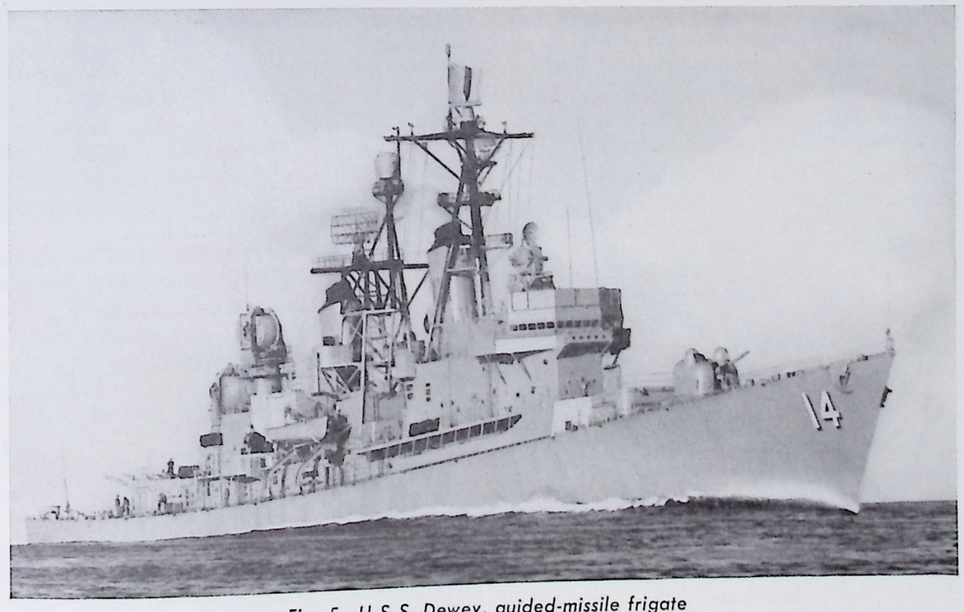


Fig. 5. U.S.S. Dewey, guided-missile frigate

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size and shape of the furnace, and the allowable furnace heat-absorption rate.

Boiler efficiency, which determines the fuel input rate, is defined as the ratio of the heat absorbed by water and steam in the boiler, superheater, and economizer, and by air in the air heater to the total heat input. Boiler efficiency may also be defined as the heat input minus the various losses, all expressed in per cent. The heat input is the higher heating value of the as-fired fuel plus any externally added heat in the air above the base temperature of 100 F, which would be available, for instance, if a steam air heater were used to raise combustion-air temperature above the ambient. Ambient air is usually considered to be at 100 F, 40% relative humidity, and 29.921 in. Hg. Oil-fired marine boilers are usually designed to operate with 115% total air at the uptake. Radiation and unaccounted-for losses and the manufacturer's margin are generally specified as 1.5%. Additional information on this subject may be found in *Principles of Combustion,* Chapter 4, pages 4-6 to 4-12, as well as in the Appendix, pages 4-A7 and 4-A8.

Fuel input rates are determined from the relationship:

$$w_F = \frac{w(h - h_w) + w_1(h_1 - h_w)}{Q_H \times \text{Eff}}$$

where:

w_F = weight of fuel, lb/hr
w = weight of superheated steam, lb/hr
w_1 = weight of desuperheated steam, lb/hr
h = enthalpy of superheated steam, Btu/lb
h_1 = enthalpy of desuperheated steam, Btu/lb
h_w = enthalpy of feedwater, Btu/lb
Q_H = high heat value of fuel, Btu/lb
Eff = boiler efficiency, %

The type and the number of oil burners selected depend upon the fuel rate and the allowable draft loss. The number is usually not less than two, so that

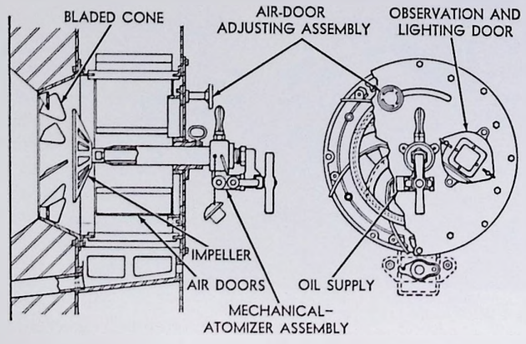


Fig. 6. *B&W Decagon mechanical-atomizing oil burner*

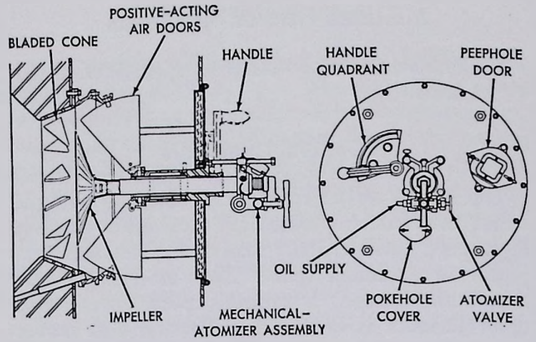


Fig. 7. *B&W Iowa mechanical-atomizing oil burner*

at least one burner operates at all times, even when changing sprayer plates or cleaning atomizers of the idle burners. While there are many oil burners available, two types are sufficient for most marine applications. The B&W Decagon type, Fig. 6, is used for oil capacities up to 1700 lb/burner, hr in boilers of relatively low capacity, and the B&W Iowa type, Fig. 7, is for the higher-powered merchant and naval vessels requiring up to 3100 lb/burner, hr. Either of these burners can be equipped with atomizers of the straight-mechanical, return-flow, or steam-atomizing variety, depending upon the owner's preference and the turn-down desired without changing sprayer plates. While most vessels use only mechanical atomizers, there is a trend toward the installation of dual systems to permit the use of wide-range steam atomizers during maneuvering and straight-mechanical atomizers during steady runs at sea, where the saving of evaporated water may be important. For extremely wide-range operation with a minimum of attention, the Navy employs the return-flow type of mechanical atomizer, with oil supply pressure in the range of 600 to 1000 psi and a variable return pressure to regulate the amount of fuel fired.

When burning heavy residual fuel oil, clearances are required around the burners to prevent flame impingement and carbon deposits (see Fig. 8). Due

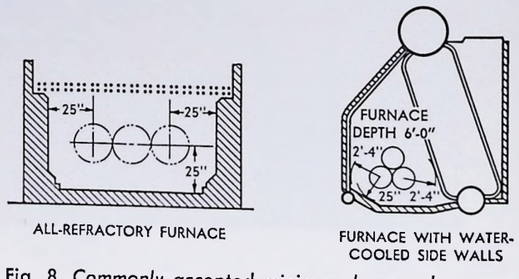


Fig. 8. *Commonly accepted minimum burner clearances for two types of furnaces when burning Bunker C oil*

consideration must also be given to arrangement if the front wall is to be water cooled, since the distance between burners may have to be increased to provide space for the wall tubes. In all boilers the burners should be conveniently located for ease of operation (Fig. 9). To assure adequate flame travel and complete combustion of the fuel, the furnace depth of both header- and drum-type boilers is usually limited to a minimum of 6 ft, although both Diesel and bunker oils have been burned successfully in furnaces 5 ft or less in depth.

With the minimum width and height fixed by the number of burners required and a selected minimum depth, the preliminary design of the furnace is set,

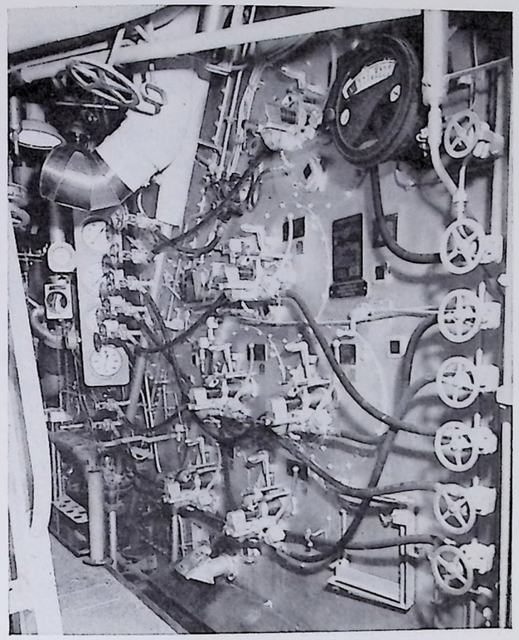


Fig. 9. *Burner front of two-drum boiler unit equipped with high-pressure return-flow atomizers*

provided that the heat absorption rates and the temperature of the gas leaving the furnace do not exceed acceptable values and any specified arbitrary criterion, such as the heat release rate per cu ft of furnace volume.

In the marine field the most widely used criterion for comparing boilers, without regard to their similarity, is the hourly heat release per cu ft of furnace volume. This ratio, which is not an important design criterion, owes its wide popularity to its simplicity and ease of application and is of value only when used in comparing similar furnace designs. In a very general way it reflects the relative time of retention of the fuel in the furnace or the time required for the products of combustion to pass through the furnace.

It is important to realize that two boilers may be designed for identical heat release rates per cu ft of furnace volume and yet have such entirely different furnace contours and arrangements that the respective heat-absorbing surfaces may be in the ratio of 2 to 1 or, for that matter, almost any ratio. Since it is the installed heating surface and not the furnace volume which absorbs heat, furnace gas temperatures and furnace tube temperatures are more closely related to surface than to volume. A practical criterion of furnace release rate per cu ft is that the firing equipment maintain satisfactory combustion conditions over the entire operating range. The availability of suitable oil burners has permitted the installation (in Naval vessels) of high-capacity light-weight boiler units with hourly heat releases of 200,000 Btu/cu ft of furnace volume at cruising rates and over 1,000,000 at maximum evaporative rates with most satisfactory results. For merchant ships, arbitrary limits of 65,000 to 75,000 Btu/cu ft, hr are commonly specified.

A more meaningful criterion, and one to which marine engineers are currently paying more heed, is the actual heat absorbed by the cold surfaces of the furnace, expressed as the heat absorbed per sq ft of radiant heat-absorbing surface. The considerations enumerated under *Furnaces,* Chapter 7, pages 7-18 through 7-20, also apply to the marine furnace, with particular emphasis on the determination of the furnace exit gas temperature to assure satisfactory superheater design and to establish the heat absorption by the furnace wall tubes for adequate circulation margins.

The marine boiler operates with higher heat-input rates per sq ft of both saturated and superheated surfaces than does the corresponding stationary boiler designed for the same steam outputs. It has less heating surface and smaller over-all dimensions. Hence, by the general equation for heat flow, $q = US \triangle t_m$ (see Chapter 7, Eq 1), the requisite heat transfer is obtained in a marine boiler by increasing the combined conductance, U, and the mean temperature difference, $\triangle t_m$. Flame temperatures more closely approach the adiabatic temperature, and furnace exit gas temperature is considerably higher than that encountered in larger boilers. Fig. 10a indicates the effect of excess air on furnace temperature. Comparison of Fig. 10b with Fig. 19 of Chapter 7 shows, in a relative way, that the average marine heat absorption rates are usually considerably above those of stationary practice.

The effectiveness of the water-cooled surface in the furnace is determined by applying the factors shown in Fig. 16, Chapter 7, to the flat projected areas of the furnace walls and tube banks facing the furnace. Bare tangent tubes are used wherever possible in furnace water walls. Where tubes must be spaced to

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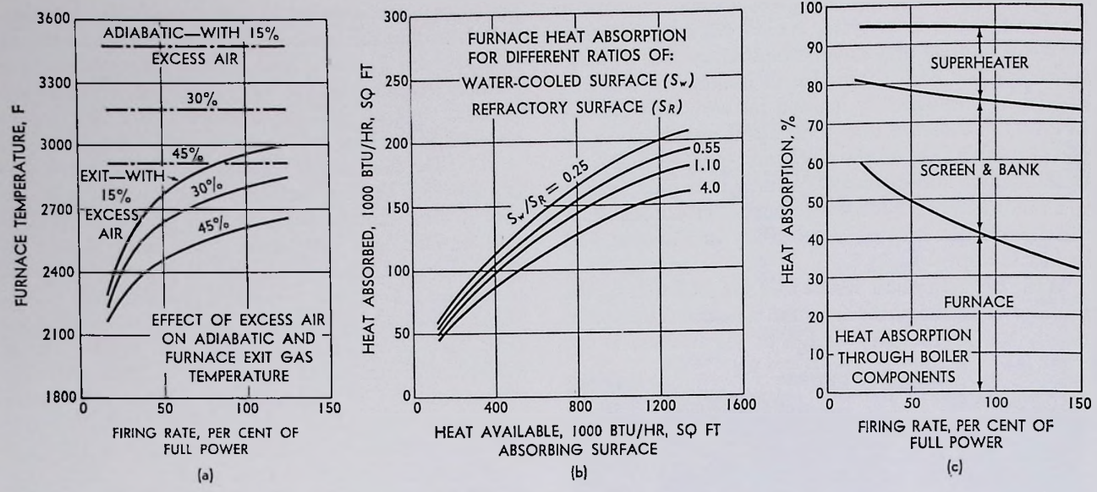


Fig. 10. *General effect of excess air (a), ratio of heat-absorbing to refractory surface (b), and firing rate (c) on furnace heat absorption and temperature, based on 18,500 Btu per lb fuel oil*

facilitate construction, it is preferable to limit the width of the exposed refractory areas between tubes to one inch or less to avoid wastage resulting from sodium compounds in the oil ash. The effect that tube spacing, or the ratio of water-cooled surface to refractory surface, has on furnace heat absorption is shown in Fig. 10b. The heat absorption rate in the furnace increases with increased firing rate. However, furnace absorption as a percentage of the total boiler absorption decreases with increased firing rate, Fig. 10c. At low firing rates, as much as 50% of the total heat available may be absorbed in the furnace, while at higher rates this may be reduced to as little as 15%.

Tube Temperatures

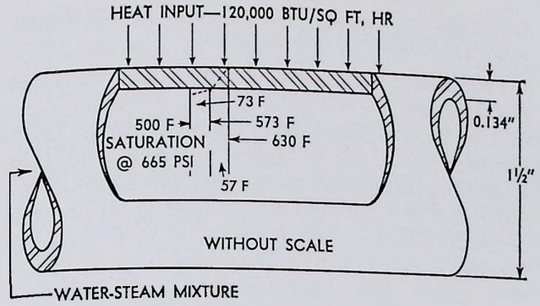
The wall temperatures of saturated boiler tubes must be determined to provide adequate circulating margins and to establish the tube-wall thicknesses required by the boiler design pressure. In most instances an estimate will suffice, since the tube-wall thickness is usually such that the temperature drop through it will not exceed 50 to 70 F. The heat transfer rate (coefficient) across the boiling-water film in saturated boiler tubes may be of the order of 20,000 Btu/sq ft, hr, F. However, in estimating tube temperature it is good practice to use 2000 Btu/sq ft, hr, F, since the use of a lower-than-actual heat transfer coefficient across the film yields a somewhat higher estimated tube temperature and results in the selection of more conservative tube thicknesses.

A typical example of normal saturated tube temperatures is indicated in Fig. 11, where, with a furnace-row screen tube operating at 665 psi and an absorption rate of 120,000 Btu/sq ft, hr, the use of a film coefficient of 2000 will give a calculated drop of 73 F through the water film. Consequently, with a saturated water temperature of 500 F, the temperature of the inside tube wall will be 573 F. The drop through the tube wall will be 57 F, which is within the range of 50 to 70 F previously noted. The outside wall temperature for this example is 630 F. It is readily apparent that there is considerable margin between the normal outside wall temperature of tubes without scale and the 950 F oxidation limit for carbon steel. However, a very thin scale deposit on the inside will cause the outside metal temperature to rise rapidly and lead to early failure unless prompt measures are taken to improve the boiler-water conditions and eliminate scale. With adequate boiler circulation and proper boiler-water conditions, heat input to furnace tubes is limited only by the tube-metal temperatures and thermal and pressure stresses.

Boiler Tube Banks

Based on the design and operating characteristics of the furnace, the saturated bank, superheater, and cold-end heat exchangers are derived through use of the techniques in *Temperature, Surface, and Draft Loss Calculations,* Chapter 11, pages 11-18 ff.

The saturated boiler tube bank, composed of multiple rows of tubes in which steam is generated, generally comprises two sections, a screen ahead of the superheater and a so-called saturated bank behind it. Both sections generate saturated steam but are con-



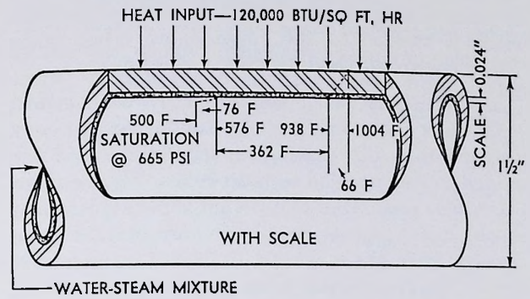


Fig. 11. *Diagrams showing how very thin scale on inner surface of tube causes seriously elevated tube-metal temperatures*

sidered separately, since the tubes in the furnace rows absorb proportionally more heat than the tubes at a greater distance from the furnace. Their total heat input is the sum of the heat transferred by radiation and convection, where the convection to screen tubes may be only 5 to 20% of the radiant transfer. Consequently, to assure an adequate flow of water, the furnace-row tubes are made larger in diameter, and it is for this reason that marine boilers are designed with two sizes of steam-generating tubes. Generally, in drum-type boilers the diameters of screen tubes are 2 in. while those of the other generating tubes are 1 or 1¼ in. In header-type boilers the furnace-row tubes are usually 2 in. and the other generating tubes are 1¼ in. Circulation and the amount of heating surface required to obtain the desired gas temperature leaving the tube bank are the chief factors in determining the tube size and the number of tube rows to be installed, although resistance to gas flow is also a factor.

In the design of the superheater screen, due consideration must be given to the effect of tube spacing and the number of rows upon the desired steam temperature and the maximum allowable superheater tube-metal temperatures. It is customary to establish the spacing and the number of rows so that the gas temperature entering the superheater and the radiant penetration from the furnace will provide the desired steam temperature with a superheater of reasonable size and arrangement. Fig. 12 shows the effect of radiant penetration on the performance of the superheater over the designed load range of the boiler.

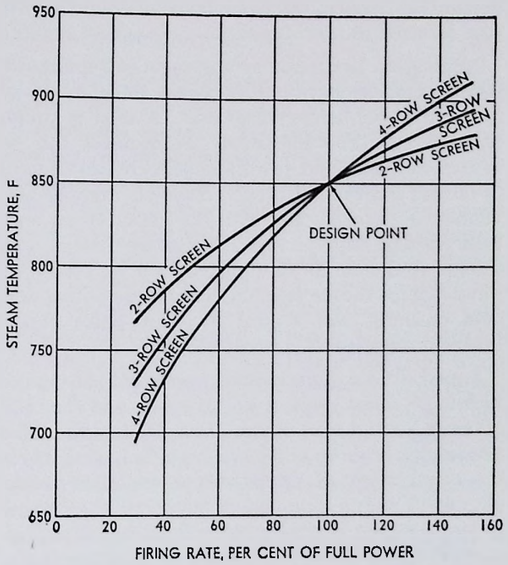


Fig. 12. *Typical curves showing effect of superheater location and firing rate on superheat*

The number of rows in the saturated bank is usually established by an analysis of the economic advantages of economizers and air heaters as compared with boiler surface. The proper proportions of boiler surface and additional heat-reclaiming surface beyond the boiler bank are based on the efficiency desired.

Superheaters

Superheaters of the convection type are generally used in marine boilers. While some radiant superheaters have been used, the difficulty of providing adequate cooling during fast start-ups and under maneuvering conditions has severely limited their application. In the convection superheater there is usually enough absorption by radiant penetration of

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the screen to give a flatter steam-temperature characteristic than would be obtainable by convection alone. The characteristic steam temperature curves of radiant and convection superheaters are shown in Fig. 1, Chapter 12.

With a properly designed screen between the convection superheater and the furnace it is possible to maintain a high temperature differential between the gas and the steam. The effect of this is to minimize the amount of heating surface necessary to obtain the desired steam temperature and to reduce the size and weight of the superheater and, consequently, the boiler unit as a whole. Superheater heating surface usually consists of "hairpin," or U-shaped, tubes connecting the headers, but continuous-tube superheaters may be used in special designs.

Although a horizontal arrangement of superheater tubes is mandatory in header boilers, either a vertical or a horizontal arrangement may be used in drum-type boilers. The horizontal arrangement can be vented, drained, and readily cleaned by mechanical or chemical means. However, complete draining and venting is possible only with the vessel on an even keel. Vertical superheaters have inverted loops and, though drainable at all times, cannot be vented—a disadvantage during start-up, hydrostatic testing, and acid cleaning. For typical two-drum-boiler superheater arrangements, see page 29-4.

Superheater surface is usually arranged in from one to four or more loops of in-line tubes, and the most commonly used tube size is 1¼-in. OD. As the tubes are relatively short, adequate strength is available in the usual range of thicknesses to maintain proper alignment with a minimum of supports. The number of steam passes is selected to provide sufficient pressure drop to obtain proper steam distribution and to assure satisfactory tube-metal temperatures.

Good practice requires that the number of lanes or gaps between passes be minimal and that all gas and steam maldistribution be limited, insofar as possible, by the design of the superheater. However, with various fuels and operating demands it is almost impossible to maintain the conditions necessary for perfect distribution. To provide a satisfactory margin when calculating steam and tube temperatures, the average gas-side heat transfer rate is increased and the steam-side rate is reduced, to allow for probable maldistribution of steam, which is a function of the tube to header pressure-drop ratio. The resulting "upsets," as they are called, give the maximum calculated steam and tube-metal temperatures for tubes with less than average steam flow and more than average gas flow. Tube materials are selected on the basis of the calculated maximum mean wall temperature anticipated in each pass, whereas header materials are selected for the maximum steam temperature to which they are exposed.

To minimize leakage and eliminate delay and maintenance costs, it is the usual practice to expand tubes into the headers when steam temperatures are not in excess of 800 F and to expand and seal weld when steam temperatures are greater than this figure. Above 875 F use of rolled and seal-welded joints or joints of the stub-welded type is necessary to provide satisfactory service. Use of stub-welded joints is limited to single- or double-loop superheaters by the space requirements for welding. Where dissimilar metals are used for tubes and headers, the tubes must be "safe-ended" at the place of manufacture with a short length of tubing, identical with the header material, which is welded to the superheater tubes under closely controlled shop-welding procedures.

In addition to being supported by the headers, superheater tubes may also be supported at one or more points by alloy castings dovetailed into brackets welded to water-cooled support tubes. Fig. 13 shows two widely used types that permit rapid replacement should high-temperature oil-ash corrosion be experienced. With these designs it is not necessary to remove the superheater tubes to effect replacement. The number of supports exposed to the effects of the gas can be substantially reduced, or in some cases eliminated entirely, by locating them within the double casing.

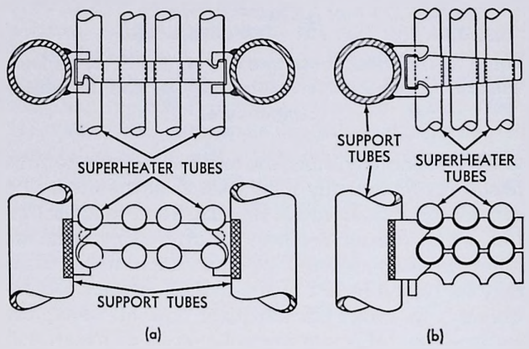


Fig. 13. *Two widely used types of tube supports for horizontal marine superheaters*

In two-drum boilers, superheaters of the inverted-loop vertical type have their headers beneath the superheater, whereas the headers of the horizontal design are at the front or the rear of the boiler, depending upon the tube-renewal space available in the fireroom. In header boilers, superheater headers must be located at the boiler side, with tube-renewal space provided on the opposite side of the unit.

Superheater Slagging and High-Temperature Corrosion

The use of higher steam temperatures and the nature of the residual fuel oils burned in recent years have resulted, in many cases, in rapid slagging of superheater surface and corrosion and wastage of superheater supports. The heavy slag deposits come from burning oils with an ash quantity of 0.05 to 0.20% or more by weight and containing, as major constituents, vanadium, sodium, and sulfur. While the exact mechanism of deposit formation is a matter of conjecture, experience indicates that heavy deposits occur only with certain combinations of gas and heat-receiving-surface temperatures. To some extent deposits are a function of the chemical analysis of the ash, which determines its melting point. Generally speaking, the melting temperatures average about 1650 F but, depending on chemical composition, can be as low as 1000 F.

The heaviest slag accumulations occur on the furnace side of a superheater, where the temperatures of the gas and heat-receiving surface are both high, while less slag forms at the bank side, where the gas temperatures are much lower and the receiving-surface temperatures are practically unchanged. In the case of the saturated screen between the furnace and the superheater, the gas temperature entering the screen may be nearly 500 F higher than the gas temperature entering the superheater, yet, with a receiving-surface temperature about 400 F lower than that of the superheater, very little, if any, slag is deposited.

Satisfactory control of superheater slagging could be achieved if it were possible to regulate closely the gas and receiving-surface temperatures, thereby controlling the temperature of the ash particles themselves. If an impinging slag particle is sufficiently cooled to be in a solid state, it rebounds from the superheater surface, but if it remains in a plastic or semifluid state it has a much greater affinity for the receiving surface, and rapid build-up becomes a distinct possibility. Unfortunately, the usual space allocated for a marine boiler does not provide the latitude in design needed to obtain the gas and receiving-surface temperatures required to control the ash temperature. Any possible physical changes to furnace, screen, or superheater do not result in temperatures sufficiently low to solidify the ash before it enters the superheater. Consequently, the design must have a high degree of tolerance for slag.

This is best achieved as follows: design for tube-metal temperatures that will minimize high-temperature vanadium corrosion; use expendable and replaceable superheater tube supports wherever supports cannot be eliminated; install long retractable mass-action soot blowers for in-service cleaning; and provide access cavities to facilitate out-of-service water-washing. These features will reduce maintenance costs and extend periods between outages. Obviously, if the ash content of the oil were eliminated, slag would not be a problem. To this end, experimental work is under way in the marine field on systems designed to remove ash from the fuel prior to its introduction into the furnace. Also, the development of suitable additives to neutralize the harmful effects of the ash may someday eliminate the slag problem and, as a consequence, reduce, if not eliminate, high-temperature vanadium corrosion. If and when this is accomplished, development of even smaller and higher-duty superheaters will be possible.

Steam-Temperature Control

In the marine power plant, just as in the stationary power plant, accurate control of steam temperature is sometimes desirable. The trend toward fast high-powered ships, operating with improved cycle efficiencies, has resulted in higher steam temperatures and greater changes in rating between the full-power (normal design conditions) and the maximum rates of operation. In some cases steam-temperature controls may be required to assure that the allowable metal temperatures of the main steam piping and the main turbine or engine are not exceeded. Close control of the maximum steam temperature permits the use of less expensive alloys in the superheater and piping. To alter the characteristic steam-temperature curve of a particular boiler design, a single method or a combination of the methods may be employed, as shown in Chapter 13, *Steam-Temperature Adjustment and Control.*

Gas by-passing has been widely used, chiefly in two-furnace boiler designs (Fig. 7, page 13-6) in which regulation of the firing rates in each of the furnaces provides control of steam temperature over a wide range of load by changing the superheater absorption. By firing more or less oil in the superheater furnace, the quantity and temperature of the gases flowing across the superheater are varied until the desired steam temperature is obtained. At the same time, the firing rate in the saturated furnace is adjusted to hold the desired steam pressure. This is the preferred method of supplying the close control required in very highly rated units generating steam at temperatures of 950 F or above, where it is extremely important to hold tube-metal temperatures at satisfactorily low values to minimize high-temperature oil-ash corrosion.

In two-drum or header-type boilers, steam temperature is controlled, where required, through the use of a drum-type surface attemperator or "desuperheater." With a control of this type, all of the gas passes over the superheater at all times, but a portion

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of the steam is passed through the attemperator, which has boiler water as the coolant. Fig. 14 shows, schematically, satisfactory piping arrangements for control attemperators.

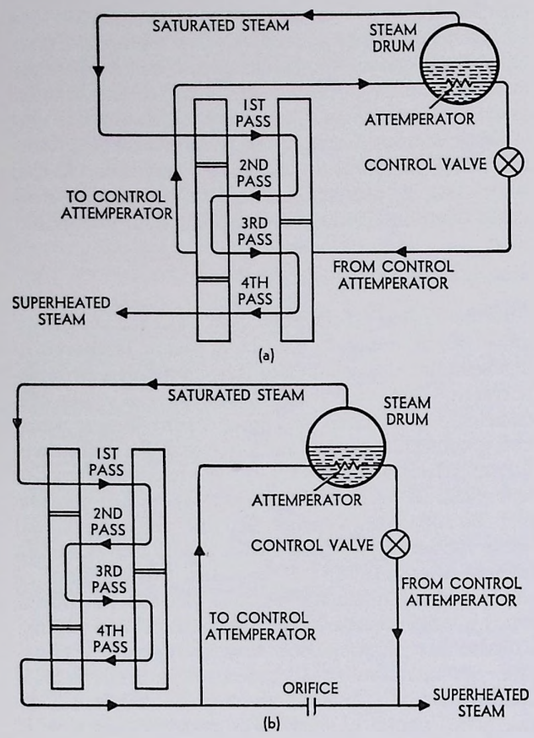


Fig. 14. *Internal attemperators—(a) interpass control, (b) afterpass control*

Steam for auxiliary use, such as fire, feed, and ballast pumps, electric generators, cargo heating, and tank cleaning, may be required at a lower temperature than that delivered by the superheater to the main engine. To satisfy these demands, auxiliary desuperheaters are used, often in conjunction with pressure-reducing stations. These auxiliary desuperheaters may be either the internal surface type installed in steam or water drums or the external spray type, depending upon the quantity of steam required, the availability of low-solids water (for the spray type), and the owner's preference.

Calculations on superheaters are illustrated in *Temperature, Surface, and Draft Loss Calculations*, Chapter 11. For additional general information on superheaters and superheat control, refer to Chapter 12, *Superheated Steam; Superheaters and Reheaters*, and Chapter 13, *Steam-Temperature Adjustment and Control.*

Air Heaters and Economizers

Air heaters or economizers, and in some instances both, are necessary if high boiler efficiency is to be obtained. The temperature of flue gas leaving the saturated boiler bank at full power (the design rate) is a function of the saturation temperature corresponding to the drum pressure at which the unit is operating and would only approach saturation temperature as a limit if an infinitely large boiler bank were used. As this is a physical impossibility, space, weight, and economic considerations usually result in a bank sized to reduce the exit gas temperature to within approximately 50 to 100 F above saturation temperature. Gas temperatures in the range of 550 to 600 F are typical in merchant units operating at about 600 psi at the superheater outlet. To reduce the uptake temperature sufficiently to obtain acceptable values of efficiency, air heaters or economizers may be added, and whether either or both are used depends upon the design of the power plant and its desired performance characteristics.

Where the design includes a deaerating feedwater heater and perhaps one stage of feed heating to supply water at 240 to 280 F, an economizer can be used, either alone or in conjunction with a steam air heater, to provide reasonably high boiler efficiency. In this range of feed temperature an economizer can be economically designed to reduce the products of combustion to within 50 F of the inlet water temperature at the normal rating. Thus, with 280 F feedwater a boiler efficiency of 87.8% can be obtained with an economizer alone and about 88.0% if a steam air heater is added.

Where the feedwater temperature to the economizer is higher, the maximum efficiency is limited, since the temperature of the gas leaving the economizer cannot be lowered below that of the inlet feedwater. Consequently, when additional stages of regenerative feed heating (turbine bleeding) are used, the inlet feedwater may be at a temperature of 300 to 450 F, and it would generally not be economical to use an economizer unless it were followed by an air heater, or an air heater alone can be used.

In either case, the cycle efficiency is increased approximately 1% for each 100 F rise in feedwater temperature through regenerative feed heating. The attractiveness of the air heater stems from the additional efficiency contributed by it when the maximum practical amount of regenerative feedwater heating is used. At normal operating rates and an inlet air temperature of 100 F, exit gas temperatures of 300 to 320 F are readily obtained, corresponding to 88.5 to 88.0% efficiency. Air temperatures leaving air heaters are normally in a range of 300 to 450 F.

Air Heaters

The air heater is usually the least expensive form of heat-recovery equipment that can be selected. The air heater, unlike the economizer, is not a pressure vessel; consequently it can be relatively simple in design and light in construction. Most air heaters in marine service are of the tubular recuperative type, although the rotary regenerative type (see Chapter 11, page 11-18) has been applied successfully in several instances and will become economically more attractive when boiler efficiencies of 88.5% or higher are required. Because of size and weight, regenerative air heaters must be installed in the fiddley over the boilers, and suitable connecting ductwork is required. These factors affect the size of hull structural members, the over-all draft loss, and forced-draft fan specifications.

Tubular air heaters in marine service are usually of the type indicated in Fig. 20, page 11-17, in which the tubes are horizontal, with air passing through them and hot gases over them. They are supported on boiler structural members and are integral with the boiler casings. Air heaters of the vertical type, in which gas passes up through the tubes and air crosses the outside, are seldom used in marine practice because of their large space (height) requirements.

In-line tube patterns increase soot blower effectiveness and facilitate water-washing, which more than offset the slight advantage in heat transfer of staggered arrangements. Tube diameters range from 1½ to 2½ in. The common sizes are 1½ and 2 in., the selection depending upon draft loss and heat absorption requirements. Large tubes reduce internal flow resistance; small tubes provide more surface in a limited space.

Corrosion of tubes and tube sheets may occur if the metal temperatures of these parts fall below the dew point of sulfur trioxide (SO_3)-bearing flue gas. Even small quantities of sulfur present in the fuel can produce troublesome deposits and sulfuric acid (H_2SO_4) of sufficient strength to attack the steel. Much effort has been devoted to the development of practical means to overcome this difficulty, which would not exist if it were possible to keep metal temperatures above the dew point at all rates of operation.

Air heater corrosion can be reduced or practically eliminated by a combination of several means. Initially, the basic design should be selected so that, even with probable air and gas maldistribution, metal temperatures are as high as possible. If the fuel is expected to have a medium to high sulfur content, a material more resistant than steel should be selected for the tubes. Low-alloy copper-bearing steels, originally developed for high strength under atmospheric corrosion conditions, have been found to outlast ordinary steels by a factor of 2½ to 1 and can be used at somewhat greater initial cost. Increasing the tube-wall thickness will also provide longer life at slightly higher cost and with increased weight and fan requirements.

Another factor to be considered in reducing corrosion is the air-flow pattern entering the tubes. Because of turbulence, the local heat transfer rates at tube inlets may actually be greater than the average and may upset rates used in the basic design, with the result that in operation the actual tube temperatures are frequently below satisfactory values. When this occurs, rapid failure of the tubes adjacent to the tube sheet may result. To avoid such failures, inserts are installed in the inlet ends of the tubes to provide dead-air spaces, which reduce heat transfer in the critical zone and raise tube-metal temperatures to safe values. Insulation applied to the tube sheet reduces conduction of heat from the tube ends and helps maintain satisfactory metal temperatures of both the tubes and the tube sheet.

A by-pass damper may be used to control tube temperatures and minimize corrosion either during low-load operation or when low ambient-air temperatures are encountered (see horizontal air heater, Fig. 20, page 11-17). By manipulating the dampers so that a major portion of the combustion air by-passes the air heater during lighting-off and low-load operation, the tube temperatures approach the temperature of the surrounding gas.

While tube-metal temperatures could be increased by recirculating a part of the hot air leaving the air heater to the entering air pass to raise the inlet air temperature, the space required for suitable air ducts and the increased fan power required make this means generally unfeasible in marine service.

The installation of a steam air heater provides a more satisfactory way of increasing inlet air temperature. With hotter air entering the air heater, some performance is sacrificed, since the uptake gas temperature increases and boiler efficiency decreases as the inlet air temperature is raised. However, this loss may be more than offset, or at the very least minimized, if bleed steam or exhaust steam is used for heating. This, coupled with savings from reduced maintenance costs, may justify continuous use of steam air heaters for tube-metal temperature-control purposes. Where the economics are not quite so favorable, the steam air heater is used during periods of low load or low ambient-air temperature when fuels of high sulfur content are burned.

Economizers

Two general classes of economizers are used in marine service. They are nonsteaming and are almost

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always arranged for counterflow of water and flue gas to obtain the best possible heat transfer characteristics. The first, a bare-tube type, is usually used where high feedwater temperature makes the application of both an economizer and air heater desirable. Bare-tube economizers operate in a relatively high gas-temperature zone and are normally designed to reduce the gas temperature about 100 F, with the remainder of the drop necessary to achieve the desired efficiency obtained with an air heater. The second and more important type employs extended surface, either in the form of cast-iron or aluminum gill rings or oval-shaped studs made of steel or low-alloy material welded to the tubes, and the entire gas temperature drop for the required efficiency is usually obtained without a following air heater.

Bare-tube economizers utilize ½-in. to 2-in. OD tubes arranged in hairpin loops or continuous loops. In the hairpin design, single or multiple rows of U-shaped tubes are welded or expanded and seal welded into two or more headers. With continuous-loop tubes, the number of header connections is greatly reduced, since several runs of straight tubing are connected by a series of U bends and only two headers are required.

Extended-surface economizers also employ continuous tubes joined by welding at the return bends and use only two headers. The tube-to-header joints are either welded or expanded and seal welded to eliminate seat leakage. There is a definite trend toward the elimination of handhole fittings, except those required for inspection, by externally welding the tubes in sockets or stubs on the headers. The use of suitable extended surface results in more compact economizers for a given performance than would be possible with bare tubes. A typical arrangement of an extended-surface economizer with studded tubes is shown in Fig. 15.

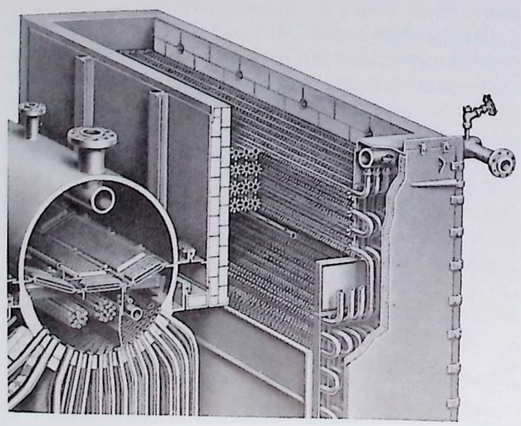


Fig. 15. *Extended-surface stud-tube economizer*

Economizers are generally designed to reduce gas temperatures by 200 to 300 F. Since in oil-fired boilers the rise in temperature of the water passing through the economizer is about ⅓ of the drop in temperature of the gas passing over the economizer, the rise in water temperature varies between 70 and 100 F. If the inlet water temperature is 280 F, the exit water temperatures normally will range from 350 to 365 F, which is well below saturation temperature at a pressure of 600 psi.

The possibility of sulfuric acid corrosion of economizer tubes must also be considered. The tube-metal temperature of bare-tube economizers is essentially the same as that of the water within the tube, since the temperature drop through the tube wall is small. This is also true for the tube metal of extended-surface elements, but the tip temperature of the studs or the gills is considerably higher. It is possible, with a sufficiently high incoming feedwater temperature, to avoid sulfuric acid deposits on the surface and thereby eliminate tube-metal wastage. Where the combination of the type of fuel to be fired and the selected feedwater temperature indicates that corrosion is a distinct possibility, cast-iron fin-type extended surface on steel tubes provides satisfactory life, as does the use of low-alloy copper-bearing steels for both tube and stud materials of stud-tube economizers.

Frequently a by-pass line is provided for operation of the boiler with the economizer out of service. During such operation it is necessary to fire more fuel to maintain the required evaporative rating (because of decreased efficiency), and proportionally more air must be supplied at an increased static pressure. When the economizer is by-passed, more heat is transferred to the steam because of the increase in the ratio of gas to steam weight. This increase in steam temperature is of primary importance and requires more attemperation (if an attemperator is provided) or a reduction in rating to prevent overheating of superheater tubes or turbine. There is little danger of metal oxidation in the economizer during by-pass operation, since the gas temperatures entering the economizer are usually less than 900 F.

With economizers, deaerating heaters should be installed to remove all traces of oxygen from the feedwater. If the latter contains dissolved oxygen, internal corrosion of tubes and headers may be expected, since oxygen is released within the economizer as heat is applied.

All marine economizers are designed to be drainable and ventable. This coupled with the wide acceptance of chemical cleaning for internal surfaces of boilers and superheaters (see Chapter 21), has made it possible to eliminate the cleanout fittings formerly required at each return bend to allow mechanical

cleaning. The elimination of these fittings, which were often leaky, and the use of all-welded continuous-tube designs with a minimum number of handhole fittings have effectively reduced maintenance costs and provided increased reliability to the shipboard power plant.

As is the case with air heaters, care must be taken to prevent soot fires in economizers. This is accomplished by the reduction of soot deposits through correct operating procedures, by the installation and use of suitable soot blowers, and by the establishment of water-washing schedules to clean the unit thoroughly at periodic intervals. Further information on economizers will be found in Chapter 11, *Boilers, Economizers, and Air Heaters,* pages 11-11 through 11-14.

Boiler Settings, Insulation, and Casing

As defined in Chapter 14, the "setting" of a boiler is the enclosure that surrounds the pressure parts and furnace. In marine practice this definition is extended to include economizers and air heaters, and the "setting" is more usually referred to simply as "casing and insulation."

To provide a comfortable fireroom, an outer-casing temperature of 130 F is usually specified and is achieved through the selection and use of suitable insulation and casing arrangements in conjunction with suitable fireroom ventilation. A typical double-casing arrangement of furnace refractory and insulation at both boiler bank and uptake areas is shown in Fig. 16. The air space between the inner and the outer casing is pressurized with air at or near the forced-draft-fan discharge pressure, to prevent leakage of combustion gases into the fireroom. In addition, the air space assists in maintaining the desired outer-casing temperature.

For general considerations of the subject as well as calculation procedures, see Chapter 7, *Heat Transfer,* and Chapter 14, *Boiler Settings, Insulation, and Casing.*

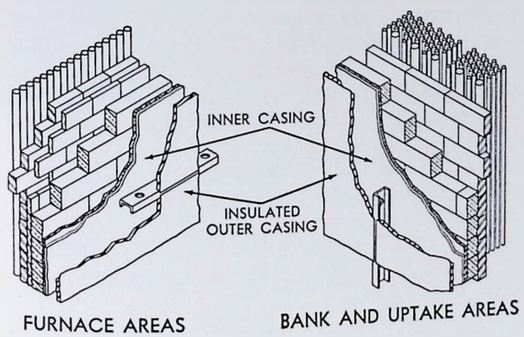


Fig. 16. *Casing constructions*

Circulation and Steam Separation

Thoroughly satisfactory circulation characteristics and efficient steam separation are of prime importance in the successful operation of any boiler. In a marine boiler rolling and pitching in a seaway or undergoing rapid load variations, these factors take on increased importance. In analyzing circulation, the procedures outlined in *Principles of Natural Circulation,* Chapter 9, pages 9-5 through 9-8, are applied to the design at the maximum anticipated rate of evaporation to provide the necessary margins.

Generally, steam-drum diameters range from 48 to 54 in. for merchant vessels and from 46 to 56 in. for naval units. The drum must accommodate the desired number of tube rows and provide sufficient space for steam separation, feedwater introduction, distribution, and treatment, and water-level fluctuation caused by sudden load variations.

Steam-drum baffling is usually kept simple in construction and arrangement. Most header-type boilers achieve primary steam separation by gravity behind a vertical baffle (Fig. 17a) or an inclined baffle (Fig. 17b), sized to maintain steam velocities below the critical velocity at which steam entrains water particles. A V-type baffle using a triple layer of perforated steel plates is used in moderately rated drum-type boilers. With relatively small clearances between the plates it is installed (Fig. 17c) just below the normal water level to break up steam-and-water jets issuing

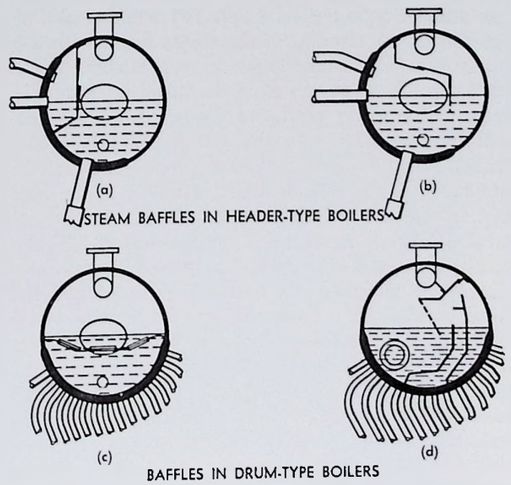


Fig. 17. *Several arrangements of steam-drum baffling in header- and drum-types of marine boilers*

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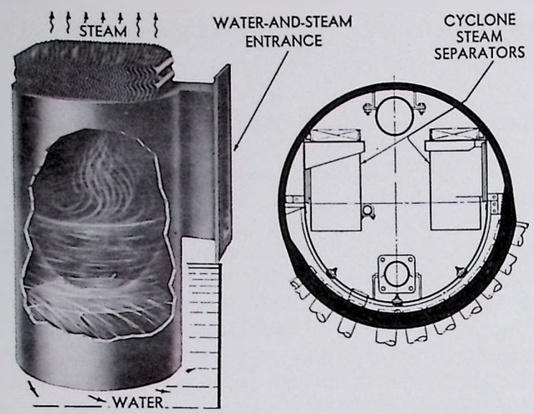


Fig. 18. *Cyclone steam separators—recommended for high-rating drum-type marine boilers*

from the riser circuits. In more highly rated two-drum boilers the duplex compartment baffle (Fig. 17d) is more effective, since it separates the high steam-by-volume mixture leaving furnace and screen circuits from the low steam-by-volume discharge of the boiler bank.

Cyclone steam separators (Fig. 18) are used in boilers of high rating, where moisture carry-over is liable to be encountered because of fluctuating water levels or high boiler-water solids concentrations (see Chapter 9). While the resistance to flow is increased by either duplex compartment baffles or cyclone steam separators, circulation is not impeded. Rather, the circulating head is increased, because the water discharged from these separators has a greater density than that beneath plate baffles.

In header-type boilers (Fig. 19) external downcomers and risers connect the steam drum with the supply and the uptake headers, respectively, of the side walls. Downcomers from the steam drum supply water to the lower header of the rear furnace water wall for distribution to the generating tubes. The steam-water mixture from these tubes is discharged into the boiler uptake headers, where it mixes with that from the screen and boiler banks. Horizontal circulator tubes connect the uptake headers with the steam drum and convey the combined steam-water mixtures of the rear wall and the boiler bank to the drum. Each bank section consists of a downtake and an uptake header joined by boiler generating tubes. Water is supplied from the steam drum to each downtake header by tube nipples joining them. A horizontal header, or mud drum, connects the lower ends of the downtake headers to equalize flow and to provide a place for the accumulation and subsequent removal of solids.

In most drum-type boilers, furnace water-wall supply headers receive water from the water drum through horizontal supply tubes located below the furnace floor. These tubes are spaced along the lengths of the water-wall headers to assure even distribution of water to the high-duty furnace tubes and, at the same time, to provide cooling for the furnace-floor refractory, thereby reducing brickwork maintenance. Naval and other high-rated special designs have external downcomer tubes from the steam drum to the lower wall headers. These are necessitated by increased head requirements resulting from space and weight restrictions, which often limit the length of steam drums and, as a consequence, the number of cyclone separators that can be installed.

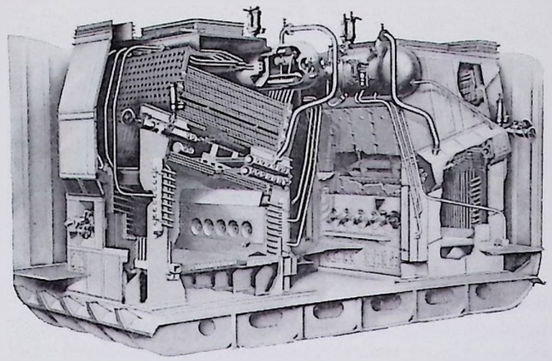


Fig. 19. *Compact arrangement of header boilers in a typical fireroom*

Furnace side-wall tubes discharge directly into the steam drum, since the tubes are arranged to form the furnace roof. Rear water-wall tubes either terminate in a header connected to the steam drum by riser tubes or go directly into the steam drum. Similar arrangements are used where front water walls are installed.

In the normal conservatively rated drum-type boilers, external downcomers are not required, since adequate circulation can be maintained by the rear rows of saturated boiler-bank tubes, where the gas temperature is 850 F or less. At low rates the first several rows of tubes act as risers, and the remaining tubes serve as downcomers. As the firing rate increases, the high-temperature gas zone moves deeper into the tube bank, and the number of tubes acting as risers increases, with a corresponding decrease in the number of downcomer tubes. Excessive firing rates and rates beyond those contemplated in the design may reduce the number of downcomers below the minimum requirements. For highly rated units it is necessary to install external or internal unheated downcomers to supplement the number of bank tubes acting as downcomers.

Circulation tends to be benefited by interposing a convection superheater in the boiler bank, since the heat absorbed by the superheater reduces the temperature of the gas flowing over the boiler tubes beyond it and more boiler tubes act as downcomers over a wide operating range. External downcomers are required in conservatively rated drum-type boilers only in those parts where the tubes cannot perform this function. Where downcomers are required to maintain adequate circulation in the boiler banks it is customary to locate them out of the gas pass between the inner and the outer casings, where they require no insulation.

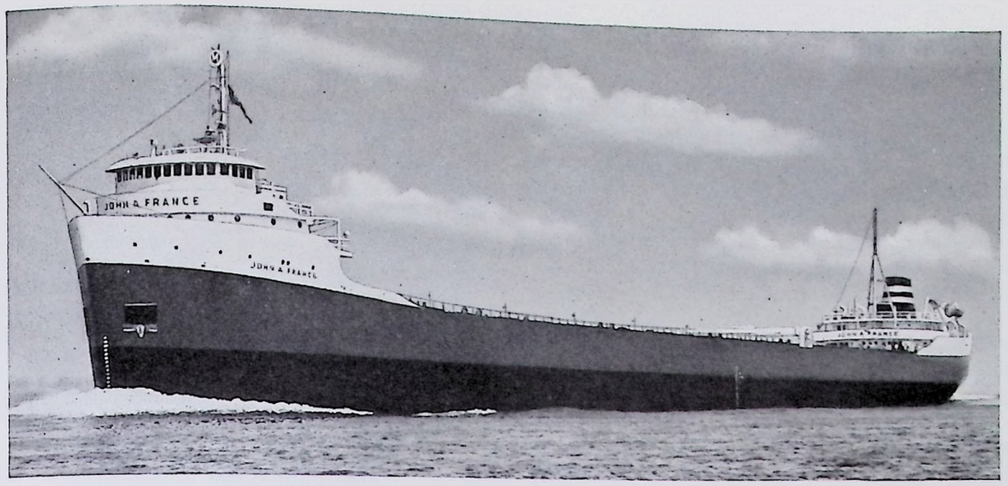


Fig. 20. *S.S. John A. France, bulk carrier*

Review of Boiler Components

Upon completion of the preliminary design phase, the estimated performance of all boiler components must be reviewed and correlated by the designer. It is at this time, while assessing the performance of the various components, that it may be possible to reduce the size and weight of a particular design or to increase its reliability and efficiency by making suitable adjustments. Following this it is possible to develop final design details to assure optimum performance for merchant and naval vessels, such as those shown in Figs. 20, 21, and 23.

To produce the best possible design for each installation, the designer must be free, within the limitations imposed by the intended service, to apply his knowledge and accumulated experience. The task of developing thoroughly satisfactory and economical boiler units is relatively easy to accomplish when the ship owner, his naval architect, and the boiler designer have a complete and mutual understanding of all the problems involved.

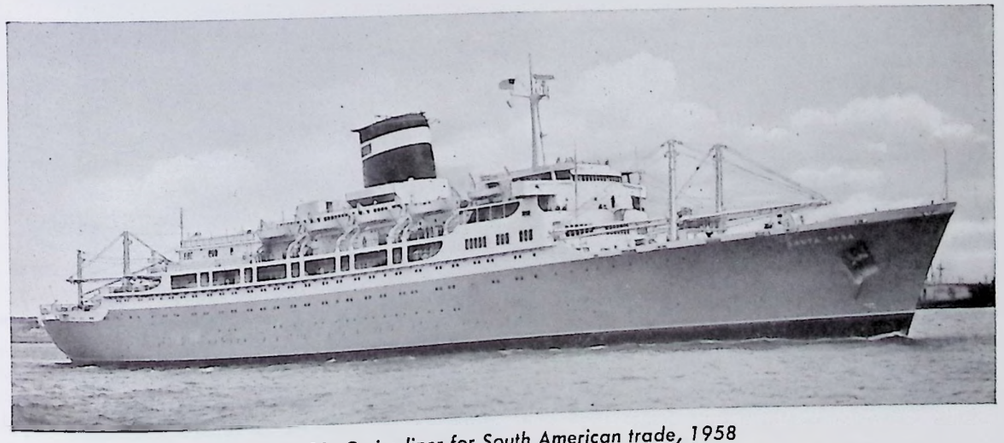


Fig. 21. *Cruise liner for South American trade, 1958*

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The Future

Over the years, B&W marine boilers installed in all types of ships have established enviable records of reliability and efficiency. In World War II, boilers of the two-furnace single-uptake type, Fig. 22, were furnished to the Navy for all but two of the battleships, all the cruisers both heavy and light, all of the aircraft carriers except some of the smaller escort class, and 90% of the destroyers. Two-drum single-furnace boilers were furnished for 33% of the destroyer escorts and numerous miscellaneous small craft. The Company also manufactured the boilers for 30% of the merchant vessels and designed the boiler units installed in the 2500 Liberty ships. This wartime record is being extended through strict adherence to sound principles of design and high standards of workmanship in the modern two-drum and header-type boilers.

In both naval and merchant service, the emphasis on higher-powered vessels, capable of carrying more passengers, cargo, or equipment at increased speeds, has necessitated a complete review and thorough appraisal of possible energy cycles and machinery plants by the marine engineer. The success of the nuclear submarine propulsion plant has led to the construction of nuclear-propelled surface vessels, such as the *N.S. Savannah* (Chapter 27, page 8). The nuclear cycle, with its many safety and operational requirements, appears at present to be limited in application to specialized installations. Other cycles,

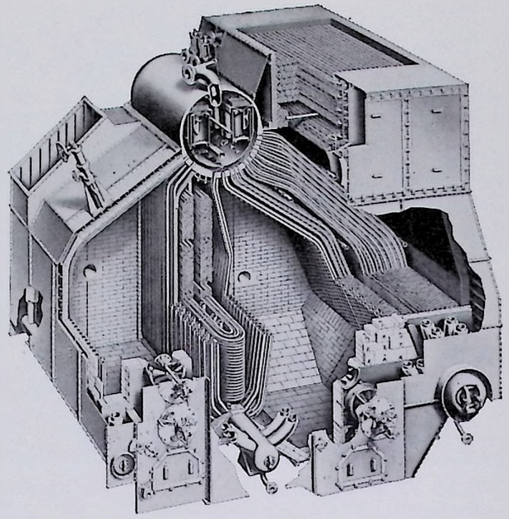


Fig. 22. *Two-furnace single-uptake (controlled-superheat) boiler*

such as those using Diesel or free-piston engines and gas turbines, cannot at present utilize as low-quality untreated fuels nor can they operate as dependably with a minimum of maintenance over long periods as does the steam plant employing boilers and steam turbines. In the foreseeable future, the steam plant will continue to be called upon to meet the major requirements of America's shipping.

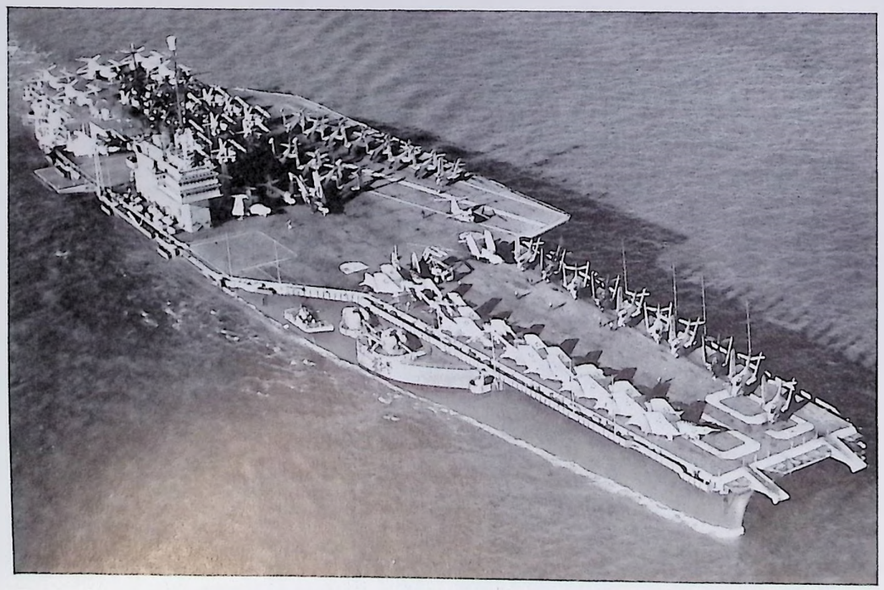


Fig. 23. *U.S.S. Saratoga, aircraft carrier*

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INSTRUMENTS AND CONTROLS FOR STEAM POWER PLANTS

INSTRUMENTS and controls are an essential part of all boiler installations. They serve to assure safe, economic, and reliable operation of the equipment. They range from the simplest manual devices to the complete automatic control of the boiler and all associated apparatus. The subject of instrumentation and control is consequently so diverse and extensive that any brief treatment, as in this chapter, must be limited to basic concepts and underlying fundamentals.

Test instrumentation is normally of a portable nature and is employed in the early investigation of the exact flow, pressures, and temperatures required to satisfy the customer and equipment supplier that the conditions of design and operation have been met. These instruments require skilled technical operators, careful handling, and frequent calibration. They are generally not suitable for long-term continuous commercial operation.

Commercial instruments are those permanently installed and are expected to give satisfactory accuracy for extended periods. The emphasis is on dependability and repeatability. This often demands some compromise in absolute accuracy. Commercial instruments used in the power-plant field are the subject of this chapter.

Why Measurements are Required

The boiler operator must be advised continually and correctly about many factors, so that he can adjust the firing rates and the distribution of fuel and air to the burners to effect the necessary balance between input and output. In the last 50 years, progress in instrumentation has resulted in a more accurate and thorough presentation of information to the operating personnel. Early in this century, boilers were operated almost without benefit of instruments. Units were small, pressures and temperatures were low, and firing was mainly by hand or by stokers. The development and availability of more and better instrumentation not only paralleled the needs of the boiler industry but also stimulated the development of larger units. The rising cost of steam production has caused both user and supplier to redouble their efforts to improve the efficiency of the conversion of thermal energy to power.



Fig. 1. *Control panel for two medium-size industrial boilers, with instruments included for indicating and recording draft, steam pressure, water level, steam flow, and operating temperatures*

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Requirements of Boiler Instrumentation

All steam boilers should be provided with instruments (see Fig. 1) to indicate or record the following variables:

Steam pressure
Steam temperature (if other than saturated)
Water level
Feedwater pressure
Furnace draft or pressure

In addition to the above, units producing more than 10,000 lb steam/hr should also monitor:

Steam flow
Feedwater flow
Combustion-air flow
Component drafts or pressures
Feedwater temperature
Flue gas temperature
Fuel flow (if possible)
Fuel pressures (if involved)
Fuel temperatures (if involved)
Speed and amperage indicators for fans, pumps, feeders, and other power-consuming auxiliaries

It is often desirable to include flue gas analysis and smoke-density determination, as well as conductivity measurement and hydrogen ion concentration in the boiler water. The completeness of instrumentation is generally proportional to the size and complexity of the unit (see Fig. 2). A thorough study is necessary to assure proper selection of the equipment needed to indicate or record the essential data.

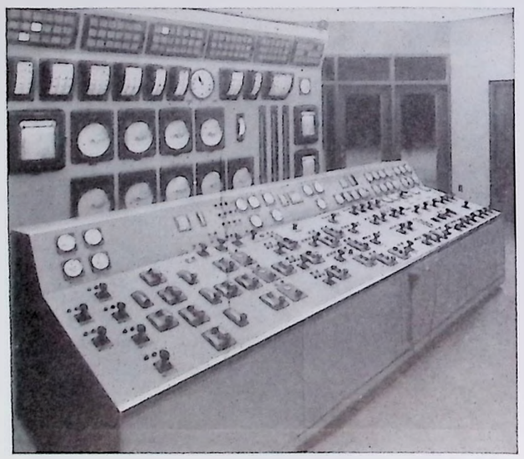


Fig. 2. *Control console and vertical board for a central station, with extensive instrumentation for manual, supervisory, and automatic operation of equipment*

Benefits from Proper Instrumentation

The advantages of good instrumentation are:

1. Adequate presentation of all key factors indicates readily any malfunctioning of the unit. Operators can then make necessary adjustments before damage to equipment or injury to personnel occurs.
2. Accurate presentation of information allows results engineers to detect deterioration of unit performance and to determine the maintenance or operation procedures necessary to rectify the conditions. Records of load and fuel consumption facilitate the efficient distribution of steam load and monitor the extent of capacity utilization.
3. Combustion guides enable personnel to maintain optimum fuel-air ratios and minimize fuel consumption.

Why Boiler Controls are Required

While it is possible to operate a steam boiler with indicators and recorders alone, few units can be run efficiently in this manner. Many of the dampers and valves that control the flow of air, fuel, and water demand substantial forces for their rapid manipulation. Controls relieve the boiler attendant of these tasks and permit him to devote his time to more rewarding problems, such as burner pattern. Manual operation is seldom relied upon because of the extremely flexible control components and systems available.

Items Controlled

While each steam-boiler installation should be analyzed for specific requirements, regulation is normally provided for:

Fuel flow
Combustion-air flow
Feedwater flow
Furnace draft or pressure
Steam temperature

Regulation is added, in some installations for auxiliary firing, attemperation, gas recirculation, and air heater protection.

Benefits Resulting from Boiler Controls

The advantages obtained by proper application of controls are:

1. The ability to position remotely any of the regulating devices in the system. It is difficult to maintain an adequate operating force to manipulate satisfactorily all of the variables when changing-load demands are to be met or when malfunctioning occurs.
2. A more systematic method of operation for the entire unit through the use of control centers.
3. The reduction of production costs through lower maintenance.
4. Better supervision of operation.
5. The continuous maintenance of optimum plant efficiency.
6. The detection, through system sensitivity, of off-normal conditions that are difficult or impossible with visual observation. Automatic correction refines the accuracy of adjustment and improves both efficiency and safety.
7. The release of operating personnel for observations and decisions rather than for routine adjustments.

Boiler Instrumentation

Instrumentation components as applied to the boiler must be designed for maximum dependability and repeatability. Equipment, used generally in the industry, is as follows:

Temperature Measurement

Temperature measurement is of prime significance on all modern large high-pressure boilers. Units are designed to operate close to the maximum allowable temperature limits of construction materials. Minor temperature elevation above these limits can result in severe damage to the equipment. It is necessary to obtain accurate flue gas, feedwater, and air temperatures for evaluation of boiler performance.

Mechanical Temperature Indicators and Recorders

The gas-filled bulb and capillary system is highly reliable and low in cost. A hollow bulb is filled with nitrogen or helium and connected by a capillary to a pressure-sensing device, such as a Bourdon tube or pressure helix. Movement of this sensor positions an indicating pointer or recording mechanism. The bulb is inserted at the location of the desired temperature measurement. Any temperature variation in the bulb changes the pressure of the gas-filled system. A pressure change is reflected by an unwinding or recoiling of the pressure-sensing unit in the receiver. This, in turn, is calibrated to indicate actual temperature.

Electrical Temperature Indicators and Recorders

An electrical resistance element, made of platinum wire wound around an insulator and contained in a protective case, is placed in a thermometer well and measures temperature by virtue of the change in the electrical resistance of the wire.

A second device is the thermocouple, which consists of two electrical conductors of dissimilar materials joined at their extremities to form a circuit. One junction, known as the "hot junction," is placed at the point of temperature measurement while the other is maintained at a fixed lower temperature, such as an ice bath. The temperature difference creates a voltage recorded on a d-c instrument, which, in turn, actuates a pen on a temperature indicator or chart.

Mechanical- and Electrical-Sensing Devices Compared

A careful study of accuracy, speed of response, degree of reliability, and stability is essential for the proper application of sensing devices. In general, the mechanical devices demand less maintenance and service and are less expensive than the electrical systems, which are more sensitive, more accurate, and respond more rapidly to temperature changes. The latter usually require more skillful maintenance.

Pressure-Measuring Devices

Required pressure measurements include high pressures, such as those encountered in the steam lines, and extremely low or negative pressures, such as those experienced in a furnace or in a condenser.

Higher pressures are measured by a Bourdon tube or a pressure helix while lower pressures utilize a diaphragm-operated unit in which the movement of the diaphragm is opposed by a calibrated spring. The readings often appear as multipointer indications on the control panel.

Flow Measurement

The measurements of water flow to and steam flow from the boiler are important and difficult to obtain. These measurements are secured with a primary element, such as an orifice or nozzle, installed in the pipeline (Figs. 3 and 4). The installation must be made carefully and a number of precautions observed to assure accuracy. The primary element, installed between flanges or welded in the pipe itself, furnishes a differential pressure which is a function of the flow rate.

A wide variety of indicating and recording devices is available, but steam-boiler users normally choose between the Ledoux Bell and a float-operated mechanism.

The Ledoux Bell (see Fig. 5), a mercury-sealed device employing a parabolic bell, enables a direct reading to be made of the flow rate, which is proportional to the square root of the differential pressure. It is one of the most stable instruments ever developed.

A float-operated device, basically a modified mercury manometer using a metallic float to determine level changes, indicates the differential pressure head.

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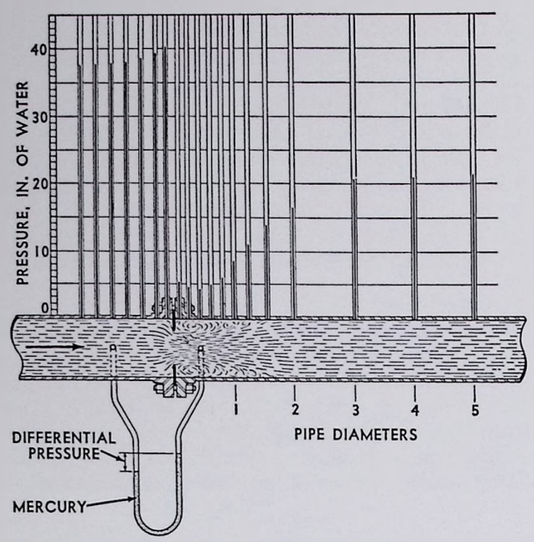


Fig. 3. *Fluid flow through an orifice creates a pressure differential used as an index of flow rate*

To convert this to flow rate, additional equipment for performing this calculation is required or a non-uniform chart must be used.

As the complexity of instrumentation grows, other secondary indicating, transmitting, and recording devices are required.

Level-Measuring Devices

A knowledge of the exact level of the water in the boiler drum is essential to the safe operation of the unit. It is important to keep this level within narrow limits 1) to maintain correct circulation within the boiler, 2) to prevent carry-over of water and solids with the steam, and 3) to avoid operation with a deficiency of water that might uncover metal surfaces normally cooled by water.

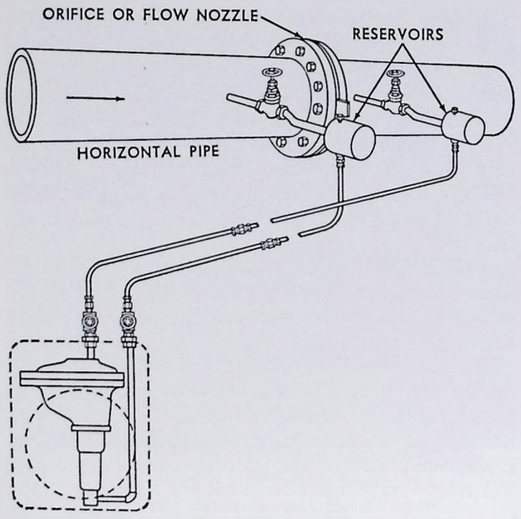


Fig. 4. *Flowmeter installation*

In addition to the gage glass, always provided on the boiler drum, it is often desirable to have some remote indication of the water level. Two connections are made to the end of the drum, one located at the minimum level of measurement and the second at the maximum. Connecting lines are run from these points to a remote differential device, either float or bell operated and mercury sealed. A recorder is used with the differential device to produce a chart record.

The required calibration of any remotely located instrument is a funtion of the location of the boiler drum connections and the pressure in the drum itself. Careful calculations are required to assure correct calibration of any remote differential device. The calibration is influenced by the density of the water inside the drum, which in turn is affected by the saturation temperature at the drum pressure. Installations often utilize automatic compensation on this differential device so that the correct reading is obtained at any operating pressure.

Combustion Guides

It is necessary to provide the operating personnel with a device to allow manual or automatic proportioning of the amount of air to the amount of fuel. Extra fuel will be consumed, increasing the cost of operation, if proper care is not taken in this one item alone.

Excess air is the amount of air supplied over and above that required for theoretically perfect combustion. It is always necessary to supply some excess air to assure complete combustion of all the fuel. Any excess air not actually required constitutes a substantial loss, thus a higher fuel bill. Consumption of additional fuel also results in higher fuel-handling costs, higher refuse-handling costs if coal is the fuel, and more maintenance and servicing on the equipment involved.

Major Combustion Guides

Three basic types of combustion guides are applied in the steam-generating industry. They are, in order of frequency of application, steam-flow air-flow, fuel-flow air-flow, and gas analysis. Each has distinct advantages. The individual steam boiler and the fuel properties must be studied to choose intelligently the best guide for a particular unit. A combination is often incorporated in the instrumentation system.

Steam-Flow Air-Flow

To obtain an instantaneous indication of the efficiency of fuel consumption, engineers apply the steam-flow air-flow concept (Fig. 5) to this problem. Since the boiler efficiency at a specific load remains essentially constant for a given excess air, the fuel consumption can be determined from steam flow, which is established by the use of standard flow-measuring devices.

The air-flow indication or index can be established by primary elements located in either the clean-air side or the flue gas side.

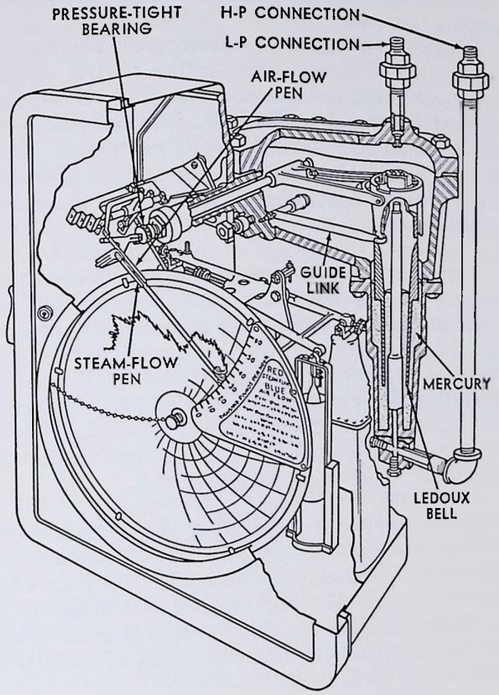


Fig. 5. *A steam-flow air-flow combustion guide for maintaining desired excess air. Fuel-flow air-flow device is similar but may have different flow-measuring mechanism*

Air-Flow Measurement. The measurement of combustion air supplied to a boiler is made difficult by several factors:

1. A restricting device, similar to the primary elements used in other flow measurements, must be installed to meter the volume of combustion air. The unrestored head loss results in added fan power.
2. On large and complicated units it is difficult to find one duct through which all of the combustion air passes. Metering equipment capable of accurately totalizing several flows must then be employed.
3. Air density must be considered in both the calibration and operation of any air flowmeter. Either manual or automatic temperature compensation must be made to the air-flow indication for it to reflect the mass rate of air flow.

Connections on Flue Gas Side. In many applications it is satisfactory to make connections on the flue gas side. Connections are located so that they measure differences in pressures resulting from the flow of combustion products through a section of the main boiler or air heater. Since the products of combustion at a given flue gas temperature are proportional to the amount of air supplied, connections on the flue gas side can serve as an effective index of air flow.

The steam-flow air-flow meter is calibrated after the unit is placed in operation and is based on results obtained from a number of combustion tests. Proper calibration enables a visual presentation of the relative correctness of the fuel-air proportioning. The operators cannot read the excess air directly but are able to achieve optimum firing conditions by maintaining the correct relative positions of two measurement pens.

Fuel-Flow Air-Flow

Combustion-control engineers prefer to measure the rate of admission of fuel and air prior to combustion. Whenever this is possible, a fuel-flow air-flow ratio guide can be established.

Fuel-Flow Measurement. The measurement of gaseous fuels requires consideration of the physical properties of the gas, particularly temperature, pressure, and specific gravity. Where widely changing properties are encountered, automatic compensation should be provided. The user must be willing to install a primary element in the gas supply line so that a differential can be produced to actuate the recording meter. This can result in some unrestored head loss with additional pumping costs. Less complex installations can sometimes use burner pressure as an indication of flow.

Metering of some liquid fuels is complicated by a tendency toward solidification of the fuel in the meter connecting lines and body, adding complexity and cost to the installation.

When burning solid fuel, such as coal on stokers, a measurement of the instantaneous rate of fuel consumption is impossible. Where substantial quantities of green fuel are present in or near the combustion zone, an index or measurement of the instantaneous rate of fuel consumption by fuel measurements cannot be obtained. Steam-flow air-flow is consequently used.

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Gas Analysis

Gas analyzers may be used to ascertain correct fuel-air relationships. Representative sample-point locations are determined by study of a continuous sample of products withdrawn from each position. Either chemical or electronic analysis of the gases is performed, resulting in chart records of the oxygen and combustibles present in the products of combustion. Since there is an accurate correlation between the percentage of oxygen in the flue gases and the quantity of excess air supplied to the combustion zone, the operator is furnished with a continuous and direct reading of combustion efficiency. Several sample points may be utilized and periodically analyzed so that a good over-all average is maintained.

Gas analyzers are useful and applicable on all boilers, particularly on large furnaces with complicated burner arrangements and on divided furnaces.

Comparison of Combustion Guides

No combustion guide is infallible. Each has particular merits. The informed combustion engineer utilizes the advantages of one to overcome the disadvantages of another.

The fuel-flow air-flow ratio guide proportions fuel and air continuously during severe load swings. It is, therefore, popular on gas- and oil-fired units. It has the disadvantage of being in error for wide variations in heating value of the fuel. When the heating value of gas changes, the theoretically correct fuel-air relationship also changes. Hence, to maintain a given excess air the proportions of air and fuel must also be changed.

On major load changes, the steam-flow air-flow device is momentarily in error, because the fuel consumption is normally higher or lower than indicated by the steam-flow load index. This guide is also affected by changing feedwater temperatures, since a variation of feedwater temperature demands more or less transfer of heat to each pound of steam produced at the specified pressure and temperature. It is necessary, when substantial changes are anticipated, to provide manual or automatic compensation.

A gas analyzer gives true excess-air determinations. Some delay is introduced, since combustion must be completed before a correct sample can be obtained. If the sample is withdrawn close to the combustion zone, this delay is not usually objectionable.

Considerable study may be required to locate sample points that will give correct average excess-air values. The dirty, hot, and corrosive conditions of the flue gases at these points make periodic maintenance necessary for continuous dependable sampling.

Types of Control Systems

Pneumatic, electric or electronic, and hydraulic-actuated systems are all available.

Pneumatic Systems. During the last twenty-five years, pneumatic systems have been used in the majority of installations. The pneumatic systems are flexible in application, serviceable by mechanical skills, and are competitive with others in cost. For all power-plant applications, the speed of response is adequate and unaffected by moisture and dirt in the surrounding atmosphere. Pneumatic systems can be serviced with a minimum number of test devices, easily understood and adjusted by instrument technicians, without personal hazards.

Pneumatic transmission of information is not recommended for distances of more than 1,000 ft. Control distances should not be more than 400 ft. Exceptions to these limits may be made if additional time delay can be tolerated. Pneumatic switching devices allow direct connection of transmitter and final control element without passing the signal through the control console.

Electric or Electronic Systems. Electric or electronic control has the advantage of unlimited length of transmission for control lines, and a supply of compressed air is not required. Components, while more complex, are also more compact. Although specially trained service personnel are needed and initial costs are higher, this type of control is selected by boiler operators who consider the advantages sufficiently important to outweigh the initial cost.

Hydraulic Systems. Hydraulic or liquid-operated systems find favor on small-size units. They are simple, reliable, and easy to understand. They are competitive in cost but do not offer the flexibility of multielement-system operation. System leaks cannot be tolerated, and care must be taken in installation and service procedures.

Requirements of a Steam-Boiler Control System

The main requirements for a satisfactory boiler control system are:

1. Adjustment of the furnace heat release to the steam demand. Control components, that receive their instructions from steam pressure, are called upon to vary the firing rate. When boilers must respond to rapid or major load changes, either steam flow or first-stage turbine pressure may be used in the control system so that the rate of firing is proportioned to the load demand.
2. Regulation of the ratio of air to fuel for optimum combustion efficiency. Too much excess air raises the stack losses and results in lower combustion efficiency with increased fan power.
3. Maintenance of a constant furnace draft in the combustion zone (except in the case of pressurized units), thus avoiding changes in the air infiltration and upset of the combustion-guide calibration.
4. Regulation of the supply of feedwater for a predetermined water level.
5. Maintenance of the final and reheat steam temperatures within prescribed limits.
6. Provision of flexibility during operation, with convenient "switch to manual" operation when occasion demands.
7. Provision of interlock devices for protection against major-equipment damage. Flame-failure devices and purge interlocks are examples of auxiliary equipment that contribute to safe operation.

Controls for Supercritical-Pressure Operation. The above requirements also apply to Universal-Pressure boilers for both subcritical and supercritical operation, although the method of control is different. The rate of combustion is governed by steam temperature, and the feedwater-control system maintains boiler pressure. The absence of a steam drum eliminates the need for water-level measurement.

Discussion of Control Components

While available boiler controls include pneumatic, electric or electronic, and hydraulic-actuated systems, the equipment performs the same functions, which are as follows:

Measure and Transmit. These devices, such as steam-pressure recorders, convert a primary measurement into an indication of the actual value. They also convert this reading to a signal that is understood by the other components of the system. Fig. 6 gives a diagram of a simplified combustion-control system for a medium-size boiler. The steam-pressure transmitter establishes a pneumatic signal proportional to the actual steam pressure.

Perform Control Actions. This group is referred to as the "calculating" or "control-action" units. They include control actions of proportional band (gain), reset (integrating), and derivative (rate). They also allow the addition, subtraction, and averaging of control signals.

Proportional band or gain relates to the sensitivity of the initial control response. Pneumatic relays or electronic operational amplifiers are generally employed to allow adjustment of the control system's response, as dictated by the individual boiler requirements.

Reset or integrating action is generally used in conjunction with proportional band and is included in the same control-action unit. The outgoing control signal is so modified that the controlled variable is maintained at a constant value. This combination is referred to as "proportional plus reset" and is almost always applied to boiler control systems.

Rate or derivative action can be compared to an effort to anticipate. Where long measurement or process delays are encountered, it may be employed so that the final control element (a valve or a control drive) overcorrects for a short period of time. This allows the variable being controlled to return to the desired value sooner.

Enable Centralized Control. These are the panel-mounted components that permit the operator to supervise performance. Transfer from manual to automatic operation, and vice versa, is possible. Operators may change conditions, such as steam pressure or steam temperature, and alter the load distribution among units.

Operate Dampers and Valves. The final element is generally either a power-operated valve or a control drive. In Fig. 6, an air-operated drive, receiving its signal from the selector station, remotely adjusts the flow of fuel to the furnace. A second drive adjusts the amount of air supplied to the furnace.

Combustion-Control Systems

There are almost as many combustion-control systems as there are steam-generating units. Load demands, operating philosophy, plant layout, and types of firing must be considered before the ultimate selection of a system is made. The following examples show a few of the systems and serve to demonstrate some possible approaches to the problem. They do not imply any particular recommendation for a contemplated installation. Equipment advances, instrument improvements, and individual needs call for constant study and revision in application.

Combustion Control for Stoker-Fired Boilers. Stoker-fired boilers are regulated by positioning fuel and combustion air from changes in steam pressure. The fuel or air may also be automatically readjusted to maintain optimum excess air. Fig. 6 illustrates an application to a spreader stoker.

A change in steam demand initiates a signal from the steam-pressure controller that causes an increase or decrease of both fuel and air. A combustion guide, in this case steam-flow air-flow, compares the index of the rate of fuel being burned (proportional to steam flow) to a calibrated air-flow index and initiates any required readjustments to the air damper so that optimum firing conditions are maintained.

Furnace draft is regulated separately through the

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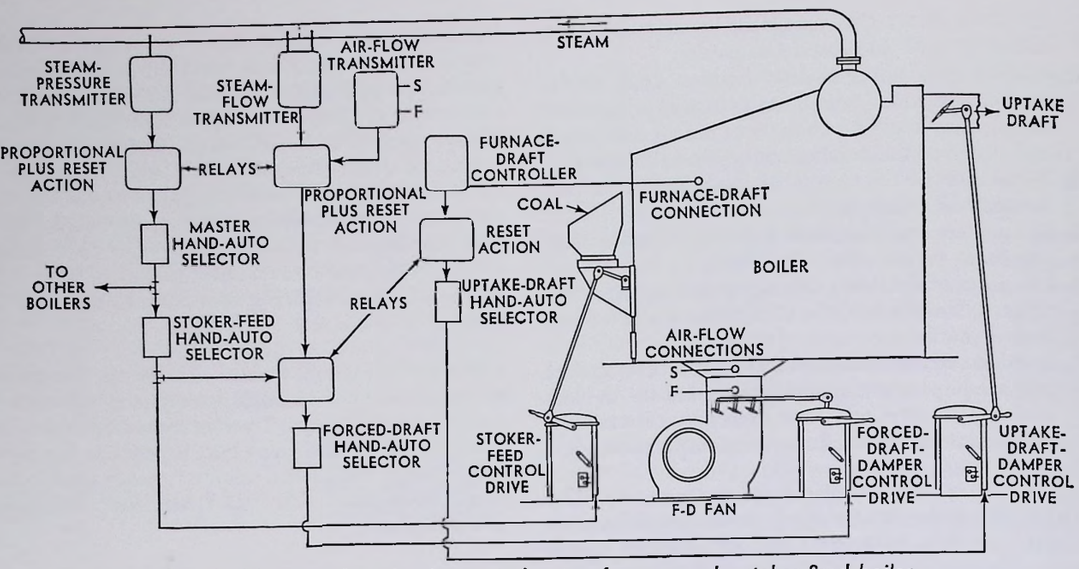


Fig. 6. *Diagram of combustion-control system for a spreader-stoker-fired boiler*

use of a furnace-draft controller and a power operator that positions the uptake damper.

Combustion Control for Pulverized-Coal-Fired Boilers. Steam pressure is used to initiate the master control signal and operates through the other components to adjust fuel and air in parallel. The efficiency of the combustion process is monitored by the use of a steam-flow air-flow ratio controller or an oxygen analyzer, or both, and the readjusting impulses to the air or fuel automatically maintain optimum excess air (see Fig. 7).

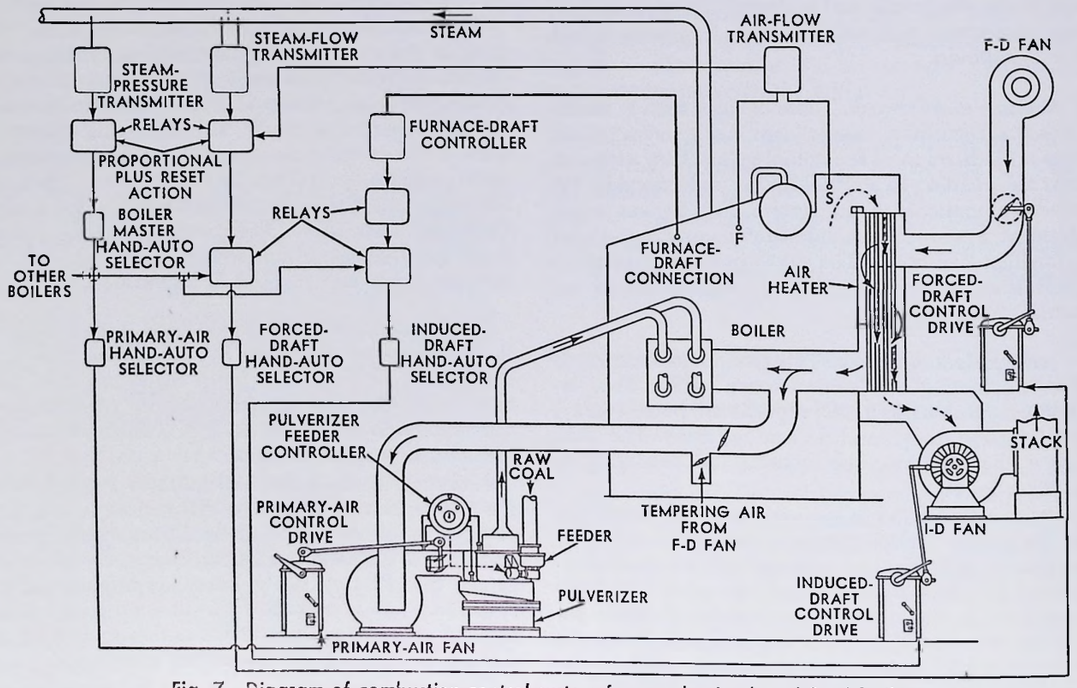


Fig. 7. *Diagram of combustion-control system for a pulverized-coal-fired boiler*

Combustion Control for Cyclone Furnaces. The principles involved in controlling Cyclone Furnaces are illustrated by Fig. 8. Actual installations frequently include additional major and auxiliary equipment for safe and satisfactory operation.

The steam-pressure measurement initiates a master signal which controls cyclone fuel flow (coal-feeder speed) and cyclone air flows, simultaneously in parallel, to maintain required steam pressure and approximate fuel-air ratio under any boiler load.

Each feeder drive is adjusted so that all feeders run at the same speed for the same master signal. On multicyclone installations, individual cyclone air flows are automatically compensated for temperature. A predetermined air flow is attained for each specific value of the master signal. Proper fuel-air relationship is maintained in each Cyclone Furnace. Each cyclone air flow is continuously readjusted from total-steam-flow total-air-flow to maintain the desired total-fuel total-air ratio.

Forced draft is controlled primarily from air-duct pressure. Since the duct-pressure set point is a function of the cyclone load, it is readjusted from the master signal to hold the desired pressure at the cyclone secondary-air dampers. On balanced-draft furnaces, induced draft is controlled from furnace pressure to hold a constant draft in the secondary furnace.

Additional features are often incorporated, such as a bias loader on duct-pressure control, automatic or manual compensation for the number of cyclones in service, and an oxygen and combustible analyzer-recorder to provide the operator with a check on the fuel-air ratio of the individual Cyclone Furnaces. The gas analyzer is an essential component of the instrumentation system.

Combustion Control for Gas- and Oil-Fired Boilers. Fig. 9 illustrates typical systems as applied to the burning of gas and oil, separately or together. A study of the diagram shows the similarities and differences when compared with the other methods of firing.

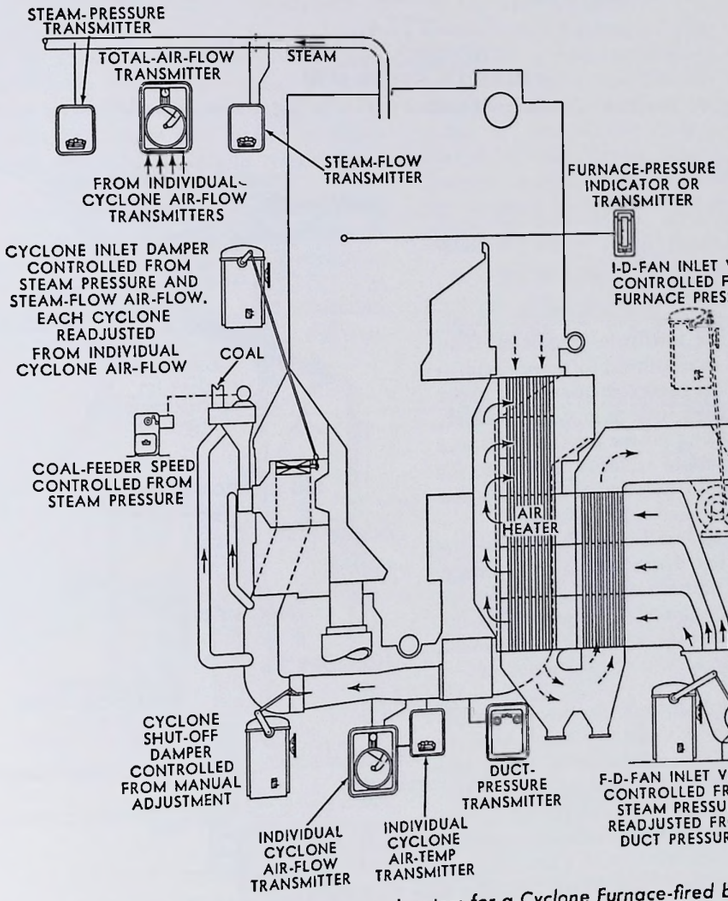


Fig. 8. *Diagram of combustion-control system for a Cyclone Furnace-fired boiler*

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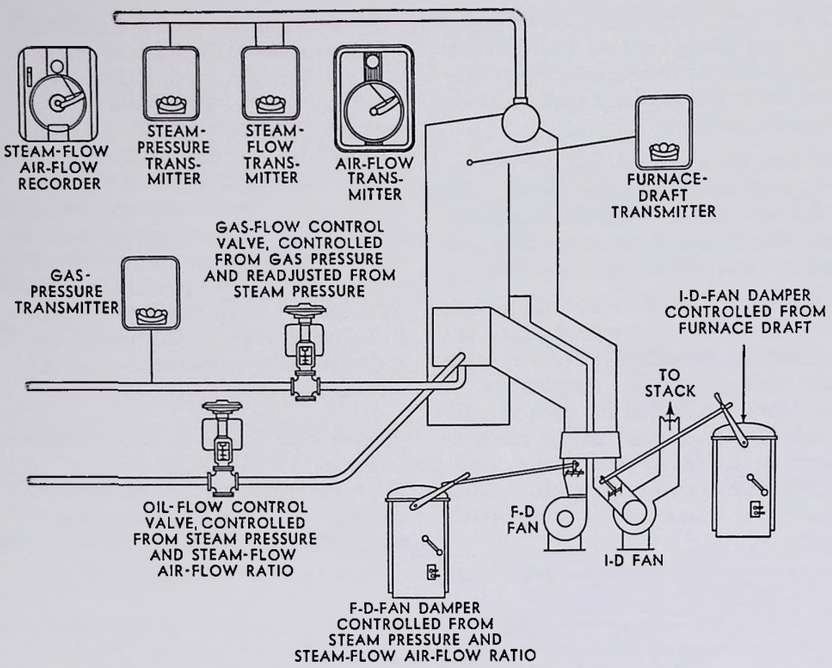


Fig. 9. *Diagram of combustion-control system for gas- and oil-fired boilers*

Feedwater-Control Systems

Feedwater-control systems are divided into four major classifications, defined by the number of measurements utilized in the over-all control action.

Single-Element, Self-Operated. Fig. 10 illustrates one style of thermohydraulic feedwater regulator. Two connections are made in the drum to allow installation of a "generator." This generator adjusts the vapor pressure within the closed hydraulic system, which, in turn, furnishes operating power for the feedwater valve. The lower the level of water in the drum, the higher the resulting vapor pressure and the further the feedwater valve is opened.

This type of regulator establishes a relation between water level in the drum and valve opening. Hence, for each rate of steam flow, a slightly different water level will be maintained. This control is not applied to large high-capacity boilers or to those with severely and frequently changing load demands.

Single-Element, Air or Electrically Operated. Considerable improvement can be made in a single-element system if compressed air or electronic control is employed. A level recorder is used, adding to it a transmitting device and the components of a control system (Fig. 11). The water flow into the boiler is regulated from water-level measurement only. A cam in a positioner mounted on the feedwater valve provides a

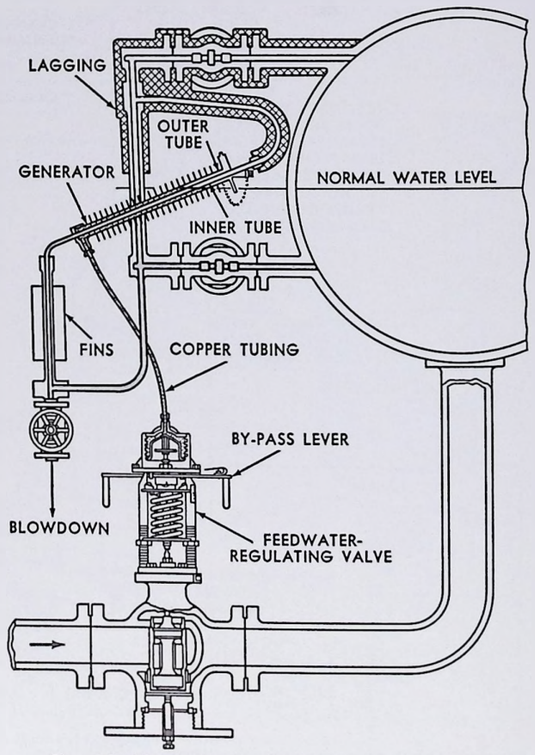


Fig. 10. *Thermohydraulic feedwater regulator*

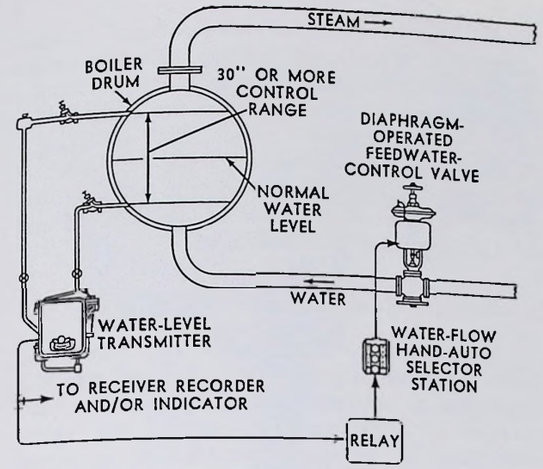


Fig. 11. *Single-element air-operated feedwater-control system*

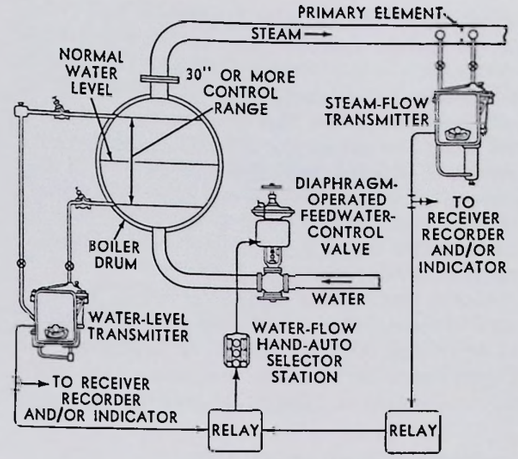


Fig. 12. *Two-element air-operated feedwater-control system*

means for matching the valve-flow characteristic to the system requirement.

This system has several advantages over the self-operated type. It holds a constant level at all loads, provides a means for remote manual control of level, and may be converted into a two-element or three-element system if operating conditions require it.

Two-Element Feedwater-Control System. This system functions primarily from steam-flow measurements but is readjusted from water-level measurement to assure maintenance of the desired level (Fig. 12). The steam flow anticipates changes in level and opens the feedwater valve proportionately. A cam, in the positioning relay mounted on the feedwater valve, provides a means for matching the valve-flow characteristic to the system requirement.

This system minimizes the effect of swell and shrinkage within the drum. It responds quickly since it does not wait for level to indicate a need for water-flow change. Any boiler installation which has or anticipates frequent changes in load should incorporate the steam-flow element in the feedwater-control system.

Three-Element Feedwater-Control System. This system functions primarily from steam-flow and water-flow measurements but is readjusted from the water-level

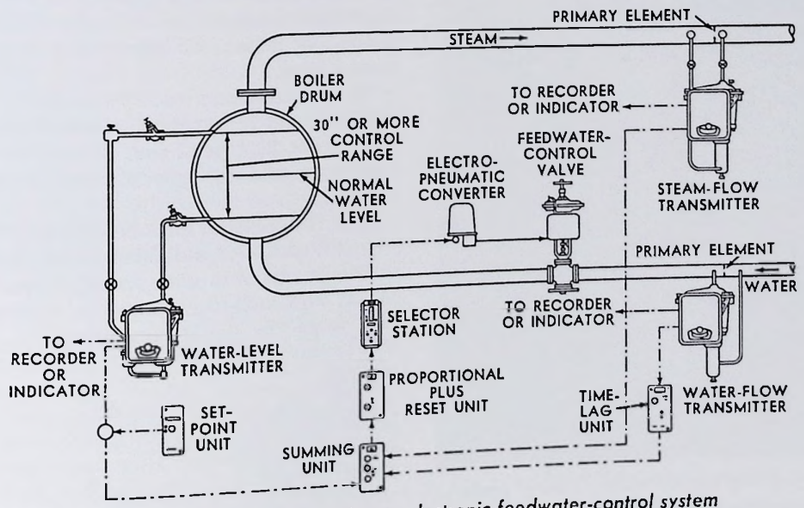


Fig. 13. *Three-element solid-state electronic feedwater-control system*

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measurement to make certain that the level is held at the desired location (Fig. 13). The steam-flow element acts as a continuous load index and positions the control valve immediately on load change. To assure feedwater flow that is always proportional to the steam-flow output, the water-flow measurement is compared with the steam-flow measurement. If pressure variations ahead of the regulating valve cause a change in water flow, the flow ratio repositions the control valve to maintain the proper flow before any effect on level is experienced.

This system may be adjusted to permit the water level in the drum to change during normal operation in accordance with the "swell" or "shrinkage" corresponding to the load. This maintains a more nearly constant weight of water in the boiler.

Steam-Temperature Control

The subject of steam-temperature adjustment and control is covered in Chapter 13.

Package-Boiler Control

Control systems are often provided with shop-assembled boilers of capacities up to 100,000 lb steam/hr, as illustrated by Fig. 14.

Users should be certain that the system provided is adequate. Many of these units are purchased by small industrial plants and must perform with a minimum amount of attention. This accentuates the importance of the automatic-control system and accompanying safety devices. Improper or inadequate control can eliminate savings that resulted from the selection of the packaged unit. Emphasis on initial cost only may result in less efficient operation of the installed unit and a consequent greater total cost.

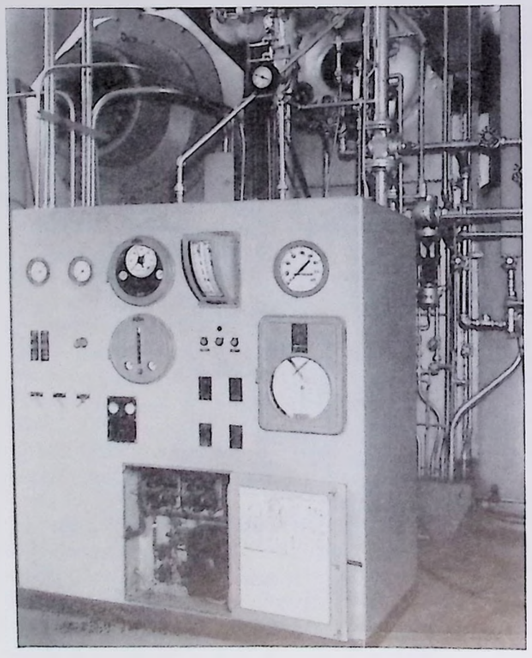


Fig. 14. *Instrument and control panel for B&W package boiler, with complete interlock protection equipment*

Installation and Service Requirements

It is important to make certain that meter and control equipment is properly installed and serviced. Supervision of installation and the required adjustments should be performed by qualified and experienced personnel. Adequate time for tune up should be provided.

Experienced field engineers, employed by control manufacturers, are available to assist customers. The field adjustment of a complicated control system is a job for an expert. Once done, maintenance can be performed by trained plant personnel. A planned maintenance program should be developed.

A boiler interlock may be defined as "a device that is interrelated with another and has the facility of initiating a control, trip, or alarm action should the other device malfunction." Typical factors requiring interlocks on a boiler are forced- and induced-draft fans, water level, furnace pressure, furnace combustibles, fuel pressures, pilot and main flames, and boiler feed pumps.

The problems of engineering an interlock system fall into three categories: 1) What factors must be interlocked? 2) How should they be interlocked? and 3) What equipment is required to accomplish the desired results? The equipment is of the utmost importance, since the effectiveness of any system can be seriously impaired if improper or unreliable equipment is used.

An adequate interlock system for a boiler must consider many factors, some of which are 1) type and size of unit, 2) type of fuel, 3) number and type of auxiliaries, and 4) type of installation—power or process.

Since many factors must be considered in engineering each system, it is difficult to "standardize." Each installation requires some specific considerations. No matter how well-engineered the system may be or how large the dollar investment, its effectiveness will be directly related to its reliability and its maintenance.

Until very recently, interlock systems for large boilers were deficient in one important aspect—flame-monitoring equipment. However, reliable flame detectors are now available, designed to monitor radiation in the ultraviolet portion of the spectrum (Fig. 15).

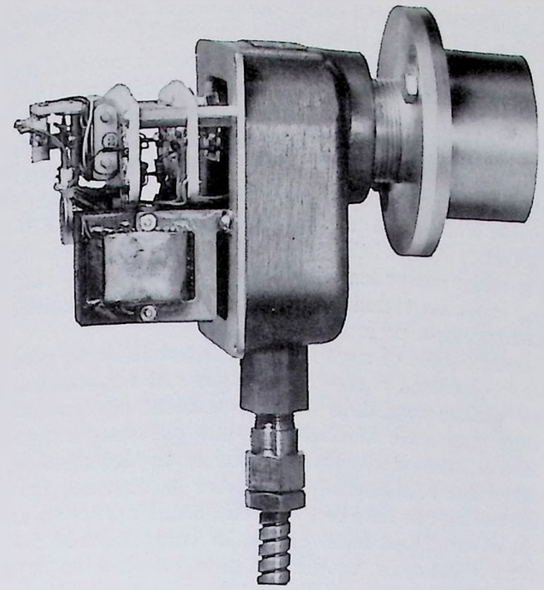


Fig. 15. *Flāmon detector for monitoring individual burner of a boiler*

The development of this new flame detector portends great advances in boiler automation, because automation could not be realized until means were available to assure the presence or absence of flame on each individual burner, regardless of the type of fuel being burned.

The Flāmon detector has been satisfactorily applied to almost every type of gas and oil burner. Initial results on pulverized-coal applications have been remarkably successful and indicate that, within a short period of time, this detector will be available for pulverized-coal burners. It should be emphasized that these detectors cannot be applied or installed indiscriminately. They require adequate application engineering and considerable care in installation.

Manually or automatically initiated remote burner-light-off-sequence control is an important consideration today. Single-burner package boilers have utilized a complete light-off and trip-sequence control for years. Automatic ignition systems are currently being engineered for gas- and oil-fired installations on large utility boilers.

Recent Developments in Instrumentation

Nearly all power-plant control centers consist of selected analog instrumentation. Analog instrumentation assigns a receiver to each transmitter or measuring element, and it continuously receives the measured signal. The records or indications are continuous.

In recent years, digital instrumentation has been developed and made available for application to the power industry. In a digital system, analog signals representing temperatures, pressures, and other measurements are converted to a pulse code, similar to the Morse telegraph code. Thus the value of the variable is represented by a number, the digits of which are in pulse code. It is a discontinuous system in which common equipment is sequentially connected to different measuring elements. Individual measurements are sequentially converted to pulses, stored, and later manipulated as necessary. While digital instrumentation is expensive and complex, it is extremely versatile and particularly well-suited to processing the large masses of data obtained in power-plant operation.

Simplifying Information Display

A large boiler-turbine unit is normally equipped with instruments that present more than 1000 pieces of information. These are in the form of indications, analog records, signal lights, annunciator points, and position indicators. It is beyond the operator's ability to use this vast quantity of information intelligently

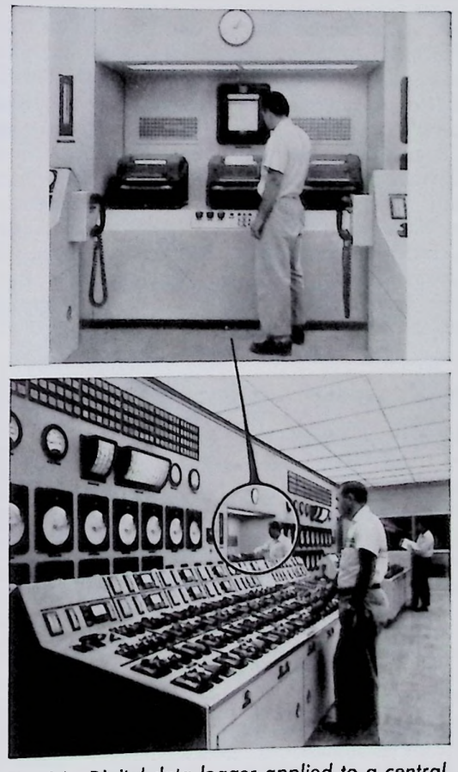


Fig. 16. *Digital data logger applied to a central station unit*

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or to identify important and key values under emergency conditions. Digital scanning-logging equipment can simplify the presentation of information and make it more easily understood. Simple automatic computations can reduce the quantity of data presented to the operator to a few key variables essential for efficient operation. Variable set points in the alarm circuitry minimize the amount of numerical information provided the operator. In many areas he needs only to know whether or not the unit is within safe limits.

Much of the data displayed in the control rooms need not appear at all but can be received and acted upon entirely within the information system. Some of the decisions formerly made by operators are being assigned to logic units within information and computing systems.

Fault Monitors and Alarms

Fault monitoring can be made extremely flexible by the continuous scanning of many variables and their comparison with established limits or set points. As an example, set points can be established as functions of one or more variables in the system, again reducing the mental analysis that the operator must perform. When the variables exceed these limits, the data-acquisition program can be automatically modified to route all associated key readings to special read-out or recording devices. This presents a concise and logical format of data which alerts the operator to deviations in accepted performance requirements. Alarms may be incorporated to accentuate the deviations. Variables still within safe limits and not directly associated with the existing malfunction continue to be scanned within the system. The operator's attention is therefore focused only on the existing malfunctions.

Interlock and Limit Circuitry

Recent advances in the art of solid-state switching permit the extensive use of interlock and limit circuitry. IHuman errors and misjudgment presently contribute to considerable equipment damage. Logic and sequence circuitry can be employed so that the proper decisions may be automatically made under emergency conditions.

Performance Computations

The continuing growth in size and complexity of units has made the frequent analysis of performance not only desirable but necessary. Many operating companies are analyzing their performance-evaluation programs and developing continuous-monitoring techniques that will allow detection of malfunctions before substantial losses have occurred.

A number of performance-evaluation procedures are practiced today. In some power companies, the performance of each unit is checked periodically by extensive heat-rate tests. Techniques and procedures for performing such evaluations are covered under ASME Power Test Codes. These tests require large crews and test instrumentation to obtain useful information. Other utility companies periodically check the major components that make up the cycle, using less expensive techniques to discover any deterioration of component performance.

An example of such techniques would be to construct expected curves for flue gas exit temperature for specific conditions of fuel, ambient air temperature, and load. Deviation of flue gas temperature from the expected values can be easily detected by observation. Advisability of further and more extensive testing can then be evaluated. Similar procedures can be employed for every major component of the entire steam cycle. Curves can be so constructed that they will indicate directly the existing loss in efficiency or increase in heat rate.

An analog computer applied to a coal-fired unit is illustrated in Fig. 17. Continuous computation of boiler efficiency, gross and net heat rate, as well as other items are either indicated or recorded. Correction of malfunctions or improper operation that contribute to inefficiency is made by the operating personnel.

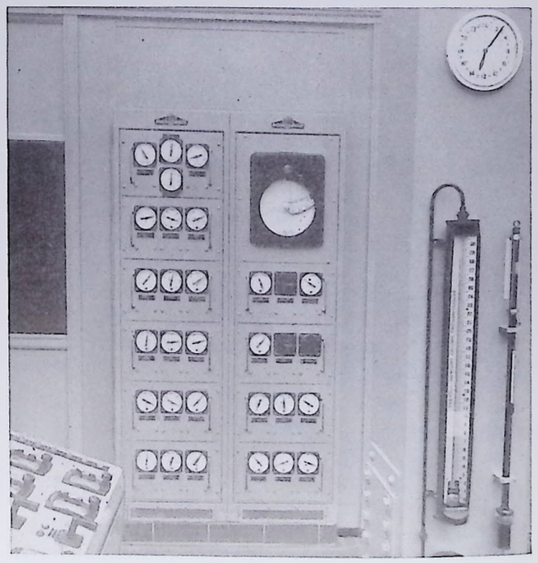


Fig. 17. *Analog performance monitor installation for determination of boiler efficiency and cycle heat rate*

The Automatic Power Plant

The great majority of power plants is now essentially automatic in operation between one-third and full load. This is provided by the conventional equipment discussed earlier in the chapter and does not require the application of any of the more recent developments in instrumentation. Complete automation of a unit requires the inclusion of instrumentation in areas previously considered to be the exclusive responsibility of the human operator.

It is first necessary to develop reliable sequence control subloops for all auxiliaries. For example, a complete gas- or oil-burner control can be achieved from a single push button, with all sequence operations and interlock monitoring handled automatically. The possibility of improper action under emergency and start-up conditions is therefore minimized, and many of the benefits of full plant automation are realized. The operator is still required to initiate a specific subloop operation. The final integration of the subloop controls results in the completely automatic plant.

The problems encountered are often unique to an individual unit. The advantages to be realized by their solution are many:

1. Improved safety of equipment and personnel through more complete instrumentation, simplified display of information, and more extensive and thorough supervisory control.
2. Reduced outages, reduced maintenance, and longer equipment life.
3. More uniform start-up and shutdown procedures, increasing both component life and safety.
4. Decreased operating costs through continuous computation of plant performance and correction of malfunctions soon after their occurrence.
5. More efficient use of available manpower during both start-up and continuous operation.
6. Reduction of over-all system generating costs through the ability to load units on the basis of current heat-rate information.

The steps needed for the realization of the above advantages are given below:

Steps to Full Automation

As shown in Fig. 18, step No. 1, Analog Display-Control, is essentially contained in the conventional control installation. The additional steps necessary for complete automation are:

1. Centralize and simplify information display; use analog and digital techniques to clarify information, aid operator understanding, improve reliability of interpretation, and save space (add Section 2, Fig. 18).
2. Extend use of interlock and limit circuitry; use digital solid-state components to extend super-

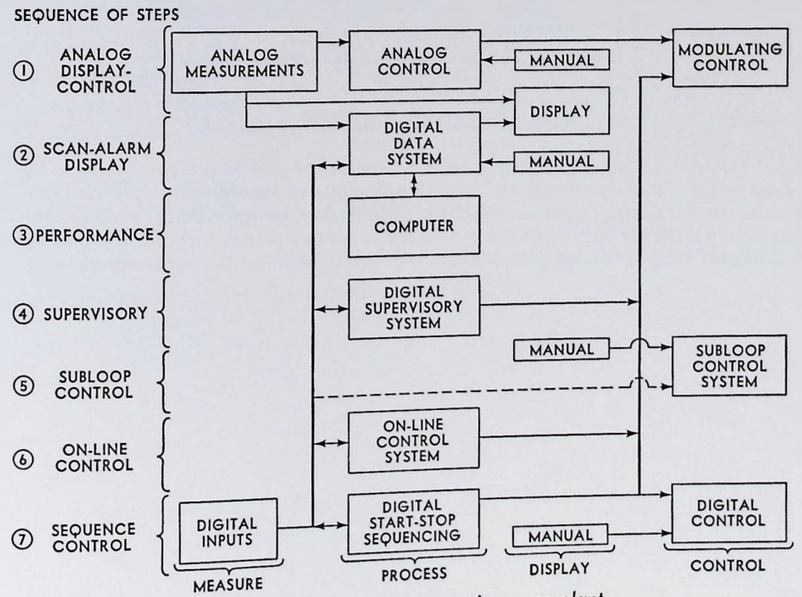


Fig. 18. *Steps to the automatic power plant*

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visory controls; minimize effects of human error (add Section 3, Fig. 18).

3. Provide performance monitoring; add computing facilities to provide up-to-date calculations for operation improvement and maintenance scheduling (add Section 4, Fig. 18).
4. Increase use of automatic subloops; simplify and standardize the starting, controlling, and stopping of major plant components and subsystems (add Section 5, Fig. 18).
5. Extend on-line controls; integrate subloops, interlocks, and limit controls with conventional controls to secure automatic operation over full range, once unit is on the line (add Section 6, Fig. 18).
6. Integrate all systems elements; add start-stop control to provide full automation, the ultimate goal (add Section 7, Fig. 18).

The trend of boiler automation is clear, and the future scope can be predicted with considerable confidence. The achievement of these goals will be difficult. There are operating problems to be solved, traditions to be overcome, and people to be trained to accept new philosophies and approaches. Reliable and adequate equipment is already available, Fig. 19. Its application, through the cooperation of manufacturers and users, will make the automatic power plant a complete reality.

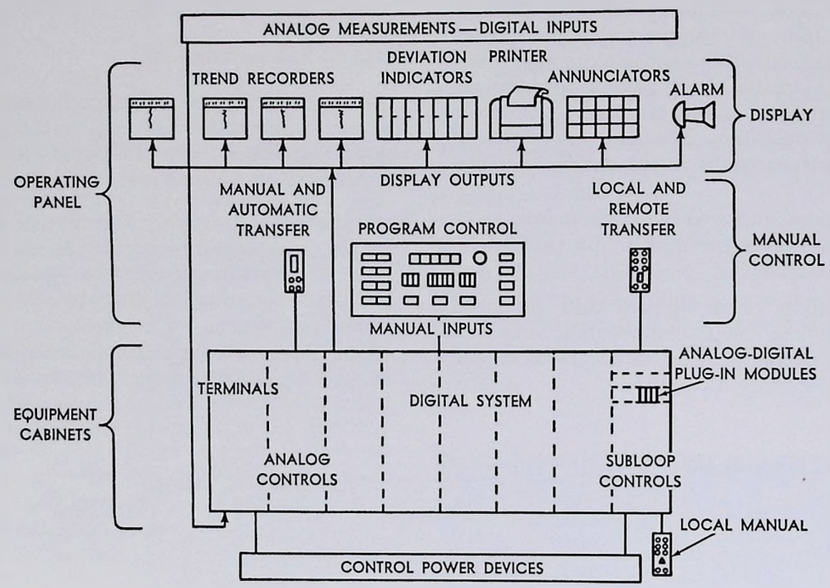


Fig. 19. *Two major areas of a fully automated system are shown here as the operating panel and the equipment cabinets. Simplified-display and manual-control devices are located on the operating panel, which is in a centralized control room. The equipment cabinets, which may be outside the control room, provide plug-in racks for the analog and digital solid-state electronic modules required to build up the necessary circuitry*

APPENDIX

Chapter 2. Source of Heat Energy—Coal

The following ASTM standards for testing coal, specifications, and definitions of terms are generally used and should be followed in detail to produce comparable results.

ASTM Standards for Testing Coal

D 492 Sampling Coals classed According to Ash Content
D 271 Laboratory Sampling and Analysis of Coal and Coke
D 1412 Test for Equilibrium Moisture of Coal at 96 to 97 per cent Relative Humidity and 30 C
D 197 Sampling and Fineness Test of Powdered Coal
D 980 Sampling and Analysis of Coal for Volatile Matter Determination in Connection with Smoke Ordinances
D 409 Test for Grindability of Coal by the Hardgrove Machine Method
D 440 Drop Shatter Test for Coal
D 441 Tumbler Test for Coal
D 410 Test for Screen Analysis of Coal
°D 431 Designating the Size of Coal from Its Screen Analysis
D 310 Test for Size of Anthracite
D 311 Test for Sieve Analysis of Crushed Bituminous Coal
D 291 Test for Cubic Weight of Crushed Bituminous Coal
D 547 Test for Index of Dustiness of Coal and Coke
D 720 Test for Free-Swelling Index of Coal, Test for Agglutinating Value of Coal (Proposed Draft)

Specifications

°D 388 Classification of Coals by Rank
°D 389 Classification of Coals by Grade
D 166 Gas and Coking Coals
°E 11 Sieves for Testing Purposes (Wire Cloth Sieves, Round-Hole and Square-Hole Screens or Sieves)

Definitions of Terms

D 121 Relating to Coal and Coke
°D 407 Gross Calorific Value and Net Calorific Value of Fuels
°D 493 Commercial Varieties of Bituminous and Subbituminous Coals

°Approved as American Standard by the American Standards Association.

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TABLE 12

INTERNATIONAL CLASSIFICATION OF HARD COALS BY TYPE

GROUPS (DETERMINED BY CAKING PROPERTIES)			CODE NUMBERS										SUBGROUPS (DETERMINED BY COKING PROPERTIES)		
GROUP NUMBER	ALTERNATIVE GROUP PARAMETERS		THE FIRST FIGURE OF THE CODE NUMBER INDICATES THE CLASS OF THE COAL, DETERMINED BY VOLATILE-MATTER CONTENT UP TO 33% VOLATILE MATTER AND BY CALORIFIC PARAMETER ABOVE 33% VOLATILE MATTER. THE SECOND FIGURE INDICATES THE GROUP OF COAL, DETERMINED BY CAKING PROPERTIES. THE THIRD FIGURE INDICATES THE SUBGROUP, DETERMINED BY COKING PROPERTIES.										SUBGROUP NUMBER	ALTERNATIVE SUBGROUP PARAMETERS	
	FREE-SWELLING INDEX (CRUCIBLE-SWELLING NUMBER)	ROGA INDEX												DILATOMETER	GRAY-KING
3	>4	>45					435	535	635				5	>140	>G_8
						334	434	534	634				4	>50-140	G_5-G_8
						333	433	533	633	733			3	>0-50	G_1-G_4
						332a / 332b	432	532	632	732	832		2	≷0	E-G
2	$2\frac{1}{2}$-4	>20-45				323	423	523	623	723	823		3	>0-50	G_1-G_4
						322	422	522	622	722	822		2	≷0	E-G
						321	421	521	621	721	821		1	CONTRACTION ONLY	B-D
1	1-2	>5-20			212	312	412	512	612	712	812		2	≷0	E-G
					211	311	411	511	611	711	811		1	CONTRACTION ONLY	B-D
0	0-½	0-5		100 A / B	200	300	400	500	600	700	800	900	0	NONSOFTENING	A
CLASS NUMBER →			0	1	2	3	4	5	6	7	8	9	AS AN INDICATION, THE FOLLOWING CLASSES HAVE AN APPROXIMATE VOLATILE-MATTER CONTENT OF:		
CLASS PARAMETERS	VOLATILE MATTER (DRY, ASH-FREE) →		0-3	>3-10 (>3-6.5 / >6.5-10)	>10-14	>14-20	>20-28	>28-33	>33	>33	>33	>33	CLASS 6 33-41% VOLATILE MATTER CLASS 7 33-44% VOLATILE MATTER CLASS 8 35-50% VOLATILE MATTER CLASS 9 42-50% VOLATILE MATTER		
	CALORIFIC PARAMETER* →		—	—	—	—	—	—	>13,950	>12,960-13,950	>10,980-12,960	>10,260-10,980			
CLASSES (DETERMINED BY VOLATILE MATTER UP TO 33% VOLATILE MATTER AND BY CALORIFIC PARAMETER ABOVE 33% VOLATILE MATTER)															

NOTES: (I) WHERE THE ASH CONTENT OF COAL IS TOO HIGH TO ALLOW CLASSIFICATION ACCORDING TO THE PRESENT SYSTEMS, IT MUST BE REDUCED BY LABORATORY FLOAT-AND-SINK METHOD (OR ANY OTHER APPROPRIATE MEANS). THE SPECIFIC GRAVITY SELECTED FOR FLOTATION SHOULD ALLOW A MAXIMUM YIELD OF COAL WITH 5-10% OF ASH.

(II) 332a >14-16% VOLATILE MATTER.
332b >16-20% VOLATILE MATTER.

* GROSS CALORIFIC VALUE ON MOIST, ASH-FREE BASIS (86 F/96% RELATIVE HUMIDITY) BTU/LB.

SOURCE: BUREAU OF MINES REPORT OF INVESTIGATIONS 5435, 1958

TABLE 12 (Continued)

STATISTICAL GROUPING OF HARD COALS

GROUPS (DETERMINED BY CAKING PROPERTIES)			CODE NUMBERS										SUBGROUPS (DETERMINED BY COKING PROPERTIES)		
GROUP NUMBER	ALTERNATIVE GROUP PARAMETERS		THE FIRST FIGURE OF THE CODE NUMBER INDICATES THE CLASS OF THE COAL, DETERMINED BY VOLATILE-MATTER CONTENT UP TO 33% VOLATILE MATTER AND BY CALORIFIC PARAMETER ABOVE 33% VOLATILE MATTER. THE SECOND FIGURE INDICATES THE GROUP OF COAL, DETERMINED BY CAKING PROPERTIES. THE THIRD FIGURE INDICATES THE SUBGROUP, DETERMINED BY COKING PROPERTIES.										SUBGROUP NUMBER	ALTERNATIVE SUBGROUP PARAMETERS	
	FREE-SWELLING INDEX (CRUCIBLE-SWELLING NUMBER)	ROGA INDEX												DILATOMETER	GRAY-KING
3	>4	>45					435	535 (V$_C$)	635				5	>140	>G_8
						334 (V$_A$)	434 (V$_B$)	534	634				4	>50-140	G_5-G_8
						333	433	533	633	733 (V$_D$)			3	>0-50	G_1-G_4
						332a / 332b	432	532	632	732	832		2	≷0	E-G
2	$2\frac{1}{2}$-4	>20-45				323	423	523	623	723 (VI$_A$)	823		3	>0-50	G_1-G_4
						322 (IV)	422	522	622	722	822		2	≷0	E-G
						321	421	521	621	721 (VI$_B$)	821		1	CONTRACTION ONLY	B-D
1	1-2	>5-20			212	312	412	512	612	712	812		2	≷0	E-G
					211	311 (III)	411	511	611	711 (VII)	811		1	CONTRACTION ONLY	B-D
0	0-½	0-5		100 A (I$_A$) / B (I$_B$)	200 (II)	300	400	500	600	700	800	900	0	NONSOFTENING	A
CLASS NUMBER →			0	1	2	3	4	5	6	7	8	9	AS AN INDICATION, THE FOLLOWING CLASSES HAVE AN APPROXIMATE VOLATILE-MATTER CONTENT OF:		
CLASS PARAMETERS	VOLATILE MATTER (DRY, ASH-FREE) →		0-3	>3-10 (>3-6.5 / >6.5-10)	>10-14	>14-20	>20-28	>28-33	>33	>33	>33	>33	CLASS 6 33-41% VOLATILE MATTER CLASS 7 33-44% VOLATILE MATTER CLASS 8 35-50% VOLATILE MATTER CLASS 9 42-50% VOLATILE MATTER		
	CALORIFIC PARAMETER* →		—	—	—	—	—	—	>13,950	>12,960-13,950	>10,980-12,960	>10,260-10,980			
CLASSES (DETERMINED BY VOLATILE MATTER UP TO 33% VOLATILE MATTER AND BY CALORIFIC PARAMETER ABOVE 33% VOLATILE MATTER)															

NOTES: (I) WHERE THE ASH CONTENT OF COAL IS TOO HIGH TO ALLOW CLASSIFICATION ACCORDING TO THE PRESENT SYSTEMS, IT MUST BE REDUCED BY LABORATORY FLOAT-AND-SINK METHOD (OR ANY OTHER APPROPRIATE MEANS). THE SPECIFIC GRAVITY SELECTED FOR FLOTATION SHOULD ALLOW A MAXIMUM YIELD OF COAL WITH 5-10% OF ASH.

(II) 332a >14-16% VOLATILE MATTER.
332b >16-20% VOLATILE MATTER.

* GROSS CALORIFIC VALUE ON MOIST, ASH-FREE BASIS (86 F/96% RELATIVE HUMIDITY) BTU/LB.

SOURCE: BUREAU OF MINES REPORT OF INVESTIGATIONS 5435, 1958

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Table 12 (Continued)

GROUPING OF AMERICAN COALS IN INTERNATIONAL SYSTEM OF CLASSIFICATION

GROUPS (DETERMINED BY CAKING PROPERTIES)			CODE NUMBERS										SUBGROUPS (DETERMINED BY COKING PROPERTIES)		
GROUP NUMBER	ALTERNATIVE GROUP PARAMETERS: FREE-SWELLING INDEX (CRUCIBLE-SWELLING NUMBER)	ALTERNATIVE GROUP PARAMETERS: ROGA INDEX	THE FIRST FIGURE OF THE CODE NUMBER INDICATES THE CLASS OF THE COAL, DETERMINED BY VOLATILE-MATTER CONTENT UP TO 33% VOLATILE MATTER AND BY CALORIFIC PARAMETER ABOVE 33% VOLATILE MATTER. THE SECOND FIGURE INDICATES THE GROUP OF COAL, DETERMINED BY CAKING PROPERTIES. THE THIRD FIGURE INDICATES THE SUBGROUP, DETERMINED BY COKING PROPERTIES.										SUBGROUP NUMBER	ALTERNATIVE SUBGROUP PARAMETERS: DILATOMETER	ALTERNATIVE SUBGROUP PARAMETERS: GRAY-KING
3	>4	>45					435	535	635				5	>140	>G_8
						334	434	534	634	734			4	>50-140	G_5-G_8
						333	433	533	633	733			3	>0-50	G_1-G_4
						332 a, 332 b	432	532	632	732	832		2	≷0	E-G
2	2½-4	>20-45				323	423	523	623	723	823		3	>0-50	G_1-G_4
						322	422	522	622	722	822		2	≷0	E-G
						321	421	521	621	721	821		1	CONTRACTION ONLY	B-D
1	1-2	>5-20			212	312	412	512	612	712	812		2	≷0	E-G
					211	311	411	511	611	711	811		1	CONTRACTION ONLY	B-D
0	0-½	0-5	000	100 (A | B)	200	300	400	500	600	700	800	900	0	NONSOFTENING	A
CLASS NUMBER →			0	1	2	3	4	5	6	7	8	9	AS AN INDICATION, THE FOLLOWING CLASSES HAVE AN APPROXIMATE VOLATILE-MATTER CONTENT OF: CLASS 6 33-41% VOLATILE MATTER; CLASS 7 33-44% VOLATILE MATTER; CLASS 8 35-50% VOLATILE MATTER; CLASS 9 42-50% VOLATILE MATTER		
CLASS PARAMETERS	VOLATILE MATTER (DRY, ASH-FREE) →		0-3	>3-10 (>3-6.5 | >6.5-10)	>10-14	>14-20	>20-28	>28-33	>33	>33	>33	>33			
	CALORIFIC PARAMETER* →		—	—	—	—	—	—	>13,950	>12,960-13,950	>10,980-12,960	>10,260-10,980			
CLASSES (DETERMINED BY VOLATILE MATTER UP TO 33% VOLATILE MATTER AND BY CALORIFIC PARAMETER ABOVE 33% VOLATILE MATTER)															

NOTES: (I) WHERE THE ASH CONTENT OF COAL IS TOO HIGH TO ALLOW CLASSIFICATION ACCORDING TO THE PRESENT SYSTEMS, IT MUST BE REDUCED BY LABORATORY FLOAT-AND-SINK METHOD (OR ANY OTHER APPROPRIATE MEANS). THE SPECIFIC GRAVITY SELECTED FOR FLOTATION SHOULD ALLOW A MAXIMUM YIELD OF COAL WITH 5-10% OF ASH.

(II) 332a >14-16% VOLATILE MATTER.
332b >16-20% VOLATILE MATTER.

(III) FROM INFORMATION AVAILABLE, AMERICAN COALS ARE REPRESENTED BY OPEN-FACED BLOCKS. THERE ARE PROBABLY NO COMMERCIAL DEPOSITS OF AMERICAN COALS REPRESENTED BY DOTTED BLOCKS.

* GROSS CALORIFIC VALUE ON MOIST, ASH-FREE BASIS (86 F/96% RELATIVE HUMIDITY) BTU/LB.

SOURCE: BUREAU OF MINES REPORT OF INVESTIGATIONS 5435, 1958

Chapter 3. Sources of Heat Energy—Other than Coal

Detailed Requirements for Fuel Oils[a]

Commercial Standard, CS12–48, U.S. Dept of Commerce

Grade of fuel oil[b] (No.)*	1[c]	2	4	5	6
Flash point (min), F	100 or legal	100 or legal	130 or legal	130 or legal	150 or legal
Pour point (max), F	0	20[d]	20	—	—
Water and sediment (max), %	trace	0.10	0.50	1.00	2.00[e]
Carbon residue on 10% residuum (max), %	0.15	0.35	—	—	—
Ash (max), %	—	—	0.10	0.10	—
Distillation temp (max), F					
10% point	420	[f]	—	—	—
90% point	—	675	—	—	—
End point	625	—	—	—	—
Viscosity					
Saybolt					
Universal @ 100 F (max)	—	40	125	—	—
Universal @ 100 F (min)	—	—	45	150	—
Furol @ 122 F (max)	—	—	—	40	300
Furol @ 122 F (min)	—	—	—	—	45
Kinematic centistokes					
At 100 F (max)	2.2	(4.3)	(26.4)	—	—
At 100 F (min)	1.4	—	(5.8)	(32.1)	—
At 122 F (max)	—	—	—	(81)	(638)
At 122 F (min)	—	—	—	—	(92)
Gravity, deg API (min)	35	26	—	—	—

*Grade of Fuel Oil

No. 1. Distillate oil intended for vaporizing pot-type burners requiring this grade.

No. 2. Distillate oil for general-purpose domestic heating for use in burners not requiring No. 1.

No. 4. Oil for burner installations not equipped with preheating facilities.

No. 5. Residual-type oil for burner installations equipped with preheating facilities.

No. 6. Oil for use in burners equipped with preheaters permitting a high-viscosity oil.

Notes
(See reference to these notes in Table)

a. Recognizing the necessity for low-sulfur fuel oils used in connection with heat-treatment, nonferrous-metal, glass, and ceramic furnaces, and other special uses, a sulfur requirement may be specified in accordance with the following table:

Grade of fuel oil	Sulfur, max, %
No. 1	0.5
No. 2	1.0
Nos. 4, 5, and 6	No limit

Other sulfur limits may be specified only by mutual agreement between buyer and seller.

b. It is the intent of these classifications that failure to meet any requirement of a given grade does not automatically place an oil in the next lower grade unless in fact it meets all requirements of the lower grade.

c. Number 1 oil shall be tested for corrosion in accordance with Par. 15 for 3 hours at 122 F. The exposed copper strip shall show no gray or black deposit.

d. Lower or higher pour points may be specified whenever required by conditions of storage or use. However, these specifications shall not require a pour point lower than zero F under any conditions.

e. The amount of water by distillation plus the sediment by extraction shall not exceed 2.00%. The amount of sediment by extraction shall not exceed 0.50%. A deduction in quantity shall be made for all water and sediment in excess of 1.00%.

f. The 10% point may be specified at 440 F. Maximum for use in other than atomizing burners.

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Typical Analyses and Relative Cost of Fuel Oils

Grade of Fuel Oil	No. 1	No. 2	No. 4	No. 5	No. 6
Weight, per cent					
Sulfur	0.1	0.3	0.8	1.0	2.3
Hydrogen	13.8	12.5	—	—	9.7
Carbon	86.1	87.2	—	—	85.6
Nitrogen	—	0.02	—	—	2.0
Oxygen	Nil	Nil	—	—	
Ash	Nil	Nil	0.03	0.03	0.12
Gravity					
Deg API	42	32	20	19	—
Specific	0.815	0.865	0.934	0.940	—
Lb per gal	6.79	7.21	7.78	7.83	—
Pour point, F	−35	−5	+20	+30	—
Viscosity					
Centistokes @ 100 F	1.8	2.4	27.5	130	—
SSU @ 100 F	—	34	130	—	—
SSF @ 122 F	—	—	—	30	—
Water & sediment, vol %	Nil	Nil	0.2	0.3	0.74
Heating value					
Btu per lb, gross	19,810	19,430	18,860	18,760	18,300°
Relative cost per Btu	118	100	76	60	51

°Bomb calorimeter determination.

Range of Analyses of Fuel Oils

Grade of Fuel Oil	No. 1	No. 2	No. 4	No. 5	No. 6
Weight, per cent					
Sulfur	0.01-0.5	0.05-1.0	0.2-2.0	0.5-3.0	0.7-3.5
Hydrogen	13.3-14.1	11.8-13.9	(10.6-13.0)°	(10.5-12.0)°	(9.5-12.0)°
Carbon	85.9-86.7	86.1-88.2	(86.5-89.2)°	(86.5-89.2)°	(86.5-90.2)°
Nitrogen	Nil-0.1	Nil-0.1	—	—	—
Oxygen	—	—	—	—	—
Ash	—	—	0-0.1	0-0.1	0.01-0.5
Gravity					
Deg API	40-44	28-40	15-30	14-22	7-22
Specific	0.825-0.806	0.887-0.825	0.966-0.876	0.972-0.922	1.022-0.922
Lb per gal	6.87-6.71	7.39-6.87	8.04-7.30	8.10-7.68	8.51-7.68
Pour point, F	0 to −50	0 to −40	−10 to +50	−10 to +80	+15 to +85
Viscosity					
Centistokes @ 100 F	1.4-2.2	1.9-3.0	10.5-65	65-200	260-750
SSU @ 100 F	—	32-38	60-300	—	—
SSF @ 122 F	—	—	—	20-40	45-300
Water & sediment, vol %	—	0-0.1	tr to 1.0	0.05-1.0	0.05-2.0
Heating value					
Btu per lb, gross (calculated)	19,670-19,860	19,170-19,750	18,280-19,400	18,100-19,020	17,410-18,990

°Estimated.

Analyses of Natural Gas from United States Fields

Source of Gas		Pa.	So. Cal.	Ohio	La.	Okla.
Analyses						
Constituents, % by vol						
H_2	Hydrogen	—	—	1.82	—	—
CH_4	Methane	83.40	84.00	93.33	90.00	84.10
C_2H_4	Ethylene	—	—	0.25	—	—
C_2H_6	Ethane	15.80	14.80	—	5.00	6.70
CO	Carbon monoxide	—	—	0.45	—	—
CO_2	Carbon dioxide	—	0.70	0.22	—	0.80
N_2	Nitrogen	0.80	0.50	3.40	5.00	8.40
O_2	Oxygen	—	—	0.35	—	—
H_2S	Hydrogen sulfide	—	—	0.18	—	—
Ultimate, % by wt						
S	Sulfur	—	—	0.34	—	—
H_2	Hydrogen	23.53	23.30	23.20	22.68	20.85
C	Carbon	75.25	74.72	69.12	69.26	64.84
N_2	Nitrogen	1.22	0.76	5.76	8.06	12.90
O_2	Oxygen	—	1.22	1.58	—	1.41
Specific gravity (rel to air)		0.636	0.636	0.567	0.600	0.630
Higher heat value						
Btu/cu ft @ 60 F & 30 in. Hg		1,129	1,116	964	1,002	974
Btu/lb of fuel		23,170	22,904	22,077	21,824	20,160

Properties of Colorado Oil Shale*

Green River Formation

Properties of Raw Shale		Shale Ash Analysis	
Specific gravity, 60 F/60 F	2.16	SiO_2, %	42.7
Heating value, Btu/lb	2590	Al_2O_3, %	13.1
Ash, %	65.7	Fe_2O_3, %	4.6
Mineral CO_2, %	18.9	CaO, %	23.3
Organic carbon, %	12.4	MgO, %	10.0
Hydrogen, %	1.8	SO_3, %	1.8
Nitrogen, %	0.4	Na_2O, %	3.1
Sulfur, %	0.6	K_2O, %	2.3
Modified Fisher Assay		Properties of Shale Oil	
Oil, %	10.6	Specific gravity, 60 F/60 F	0.916
Water, %	0.7	Heating value, Btu/lb	18,510
Spent shale, %	86.6	Carbon, %	84.6
Gas and loss, %	2.1	Hydrogen, %	11.5
Oil, gal/ton	27.7	Nitrogen, %	2.0
Water, gal/ton	1.7	Sulfur, %	0.6

° Bureau of Mines.

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Analyses of Wood and Wood Ash

Column No.*		1	2	3	4	5	6
Wood Analyses (Dry Basis), % by wt							
Proximate							
Volatile matter		72.9	76.0	74.7	74.3	69.6	72.6
Fixed carbon		24.2	18.7	23.3	24.0	26.6	27.0
Ash		2.9	5.3	2.0	1.7	3.8	0.4
Ultimate							
H_2	Hydrogen	5.6	5.4	5.7	5.8	5.7	5.1
C	Carbon	53.4	49.7	53.9	52.2	51.8	51.9
S	Sulfur	0.1	0.1	trace	trace	0.1	trace
N_2+O_2	Nitrogen + Oxygen	38.0	39.5	38.4	40.3	38.6	42.6
A	Ash	2.9	5.3	2.0	1.7	3.8	0.4
Heating value, Btu/lb		9030	8370	9120	8810	8740	8350
Ash Analyses, % by wt							
Silica	as SiO_2	–	–	–	1.7	32.0	14.3
Iron	as Fe_2O_3	–	–	–	3.2	6.4	3.5
Titanium	as TiO_2	–	–	–	0.0	0.8	0.3
Aluminum	as Al_2O_3	–	–	–	3.2	11.0	4.0
Manganese	as Mn_3O_4	–	–	–	3.9	1.5	0.1
Calcium	as CaO	–	–	–	60.8	25.3	6.0
Magnesium	as MgO	–	–	–	3.0	4.1	6.6
Alkalies	as Na_2O	–	–	–	10.4	10.4	25.0
Sulfate	as SO_3	–	–	–	3.0	2.1	7.4
Chloride	as Cl	–	–	–	0.4	trace	18.4
Carbonate	as CO_2	–	–	–	11.5	7.0	14.0
Undetermined		–	–	–	–	–	–
Ash-fusion temperatures		†	‡	–	–	–	–

°Col No.	Kind of Wood
1	Southern pine bark
2	Oak bark
3	Oregon wood, hogged
4	White-fir bark, salt-water stored
5	Spruce bark, salt-water stored
6	Redwood bark, salt-water stored

Ash-Fusion Temperatures, F

	Reducing		Oxidizing	
	†	‡	†	‡
Initial Deform.	2180	2690	2180	2680
Softening	2220	2720	2220	2730
Fluid	2340	2740	2470	2750

Analyses—Typical Bagasse, Coke Breeze, and Fluidized-Bed Char

Analyses (As Fired), % by wt		Bagasse	Coke Breeze	Fluidized-Bed Char
Proximate				
Moisture		52.0	7.3	0.7
Volatile matter		40.2	2.3	14.7
Fixed carbon		6.1	79.4	70.4
Ash		1.7	11.0	14.2
Ultimate				
H_2	Hydrogen	2.8	0.3	
C	Carbon	23.4	80.0	
S	Sulfur	trace	0.6	4.1
N_2+O_2	Nitrogen + Oxygen	20.1	0.8	
H_2O	Moisture	52.0	7.3	
A	Ash	1.7	11.0	
Heating value, Btu/lb		4000	11,670	12,100

Analyses of Typical Gaseous Fuels Derived from Coal

Column No.*		1	2	3	4	5	6
Analyses, % by vol							
H_2	Hydrogen	57.4	47.9	2.4	2.4	34.0	14.0
CH_4	Methane	28.5	33.9	0.1	0.1	15.5	3.0
C_2H_4	Ethylene	2.9	5.2	–	–	4.7	–
C_2H_6	Ethane	–	–	–	–	–	–
CO	Carbon monoxide	5.1	6.1	23.3	26.6	32.0	27.0
CO_2	Carbon dioxide	1.4	2.6	14.4	11.1	4.3	4.5
N_2	Nitrogen	4.2	3.7	56.4	56.4	6.5	50.9
O_2	Oxygen	0.5	0.6	–	–	0.7	0.6
C_6H_6	Benzene	–	–	–	–	2.3	–
H_2O	Water	–	–	3.4	3.4	–	–
Specific gravity (rel to air)		0.344	0.413	1.015	1.000	0.666	0.857
Higher heat value							
Btu/cu ft, gross @ 60 F & 30 in. Hg		528	590	–	–	534	163
Btu/cu ft, sat. @ 80 F & 30 in. Hg		–	–	83.8	94.3	–	–

°Col No.	Kind of Gas
1 and 2	Coke-oven gas
3	Blast-furnace gas (lean)
4	Blast-furnace gas (rich)
5	Carbureted water gas
6	Producer gas

Analyses of Typical Solid Fuels Derived from Oil

Analyses (Dry Basis), % by wt	Delayed Coke		Fluid Coke	
Proximate				
Volatile matter	10.8	9.0	6.0	6.7
Fixed carbon	88.5	90.9	93.7	93.2
Ash	0.7	0.1	0.3	0.1
Ultimate				
Sulfur	9.9	1.5	4.7	5.7
Heating value, Btu/lb	14,700	15,700	14,160	14,290

Elemental Analysis of Black Liquor*

Element	Weight Percentage (Dry Basis)
Na	18.3
S	3.6
H_2	3.6
C	42.6
N_2	0.2
O_2	31.7
Heating value, Btu/lb	6600

*Average for Sulfate (Kraft) Process.

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Chapter 4. Principles of Combustion

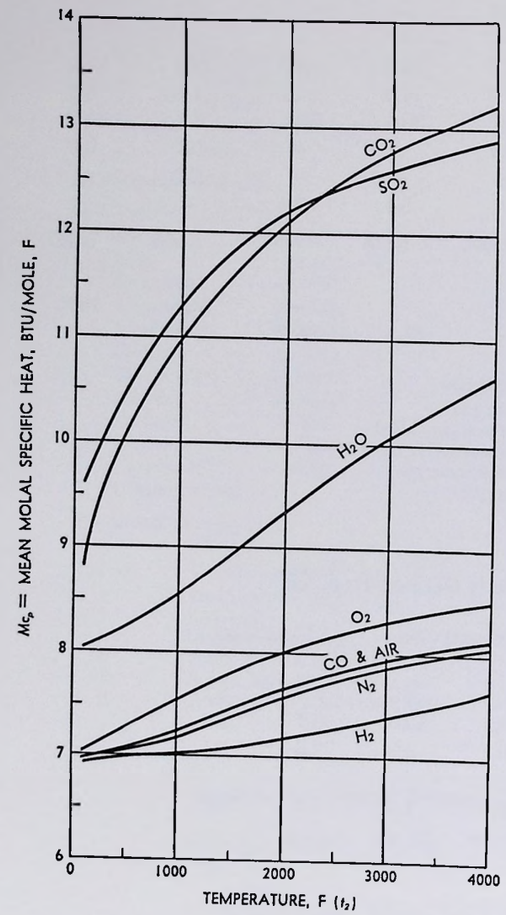


Fig. 1. *Mean molal specific heat of gases between final temperature (t_2) and 80 F at std atmospheric pressure*

Table 5

Theoretical Air, Fuel, and Resulting Moisture Per 10,000 Btu As Fired

Fuel	Theoretical Air,° lb/10kB	Fuel, lb/10kB	Moisture, lb/10kB
Fuel oil	7.46	0.544	0.51
Natural gas	7.10	0.496	0.93
Coal (prox anal.) See Fig. 4		—	—
Coal (ult anal.) See Table 6		—	—

°Dry air. To obtain wt of wet air required, moisture in air at standard conditions (0.0132 lb per lb dry air @ 60% relative humidity and 80 F dry bulb) must be added.

Table 6

Formula for Calculating Theoretical Air°

In lb per 10,000 Btu of Fuel as Fired

Ultimate Analysis of Fuel on As-Fired Basis, Per Cent by Weight

C = Carbon
H_2 = Hydrogen
O_2 = Oxygen
S = Sulfur
Btu/lb = Heat value of fuel

$$\text{Theoretical Air,† lb (per 10kB)} = 144 \times \frac{8C + 24\left\{H_2 - \frac{O_2}{8}\right\} + 3S}{\text{Btu/lb}}$$

°This formula should be used only when the exact ultimate analysis and the correct heating value are given for the fuel.

†Dry air. To obtain wt of wet air required, moisture in air at standard conditions (0.0132 lb per lb dry air @ 60% relative humidity and 80 F dry bulb) must be added.

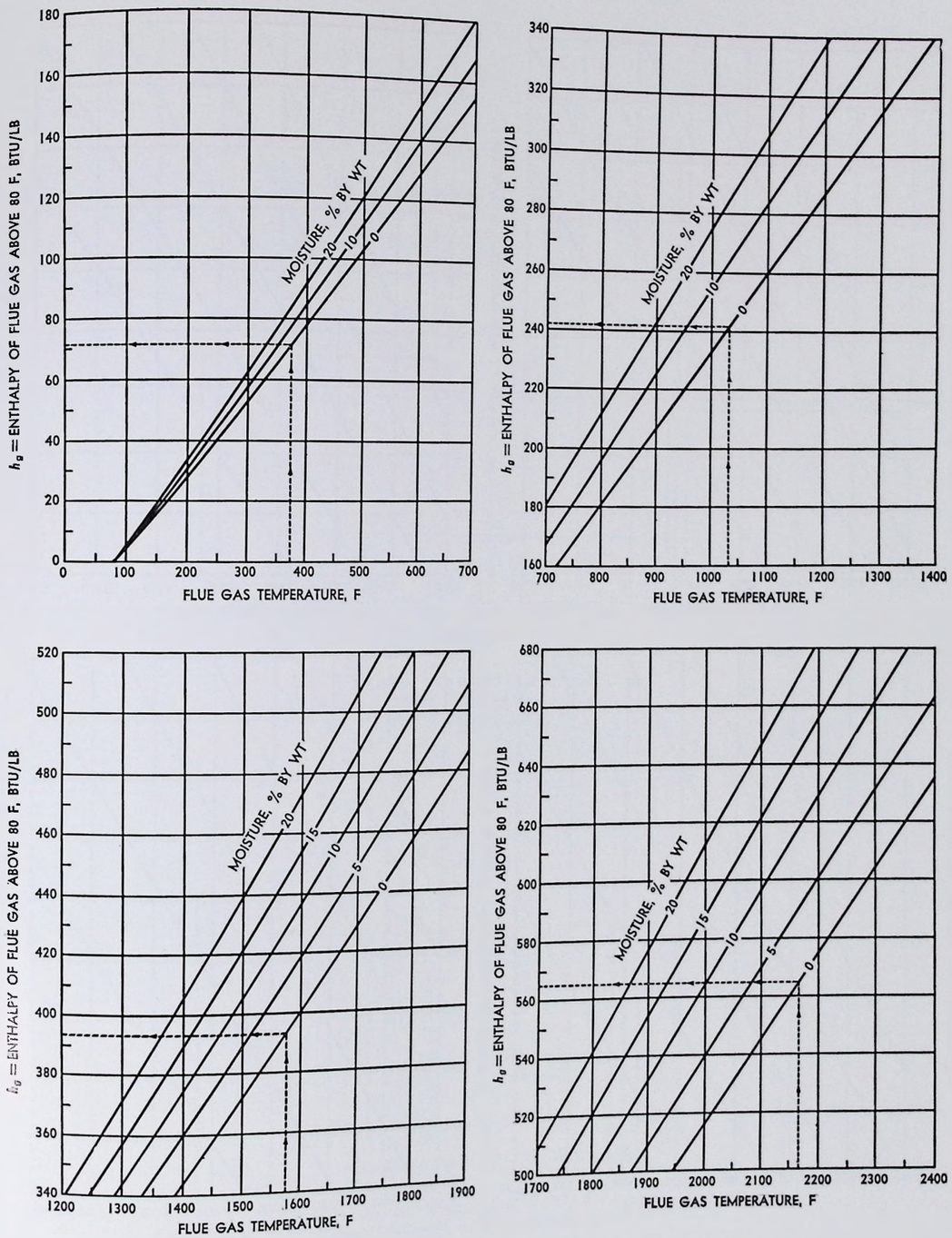
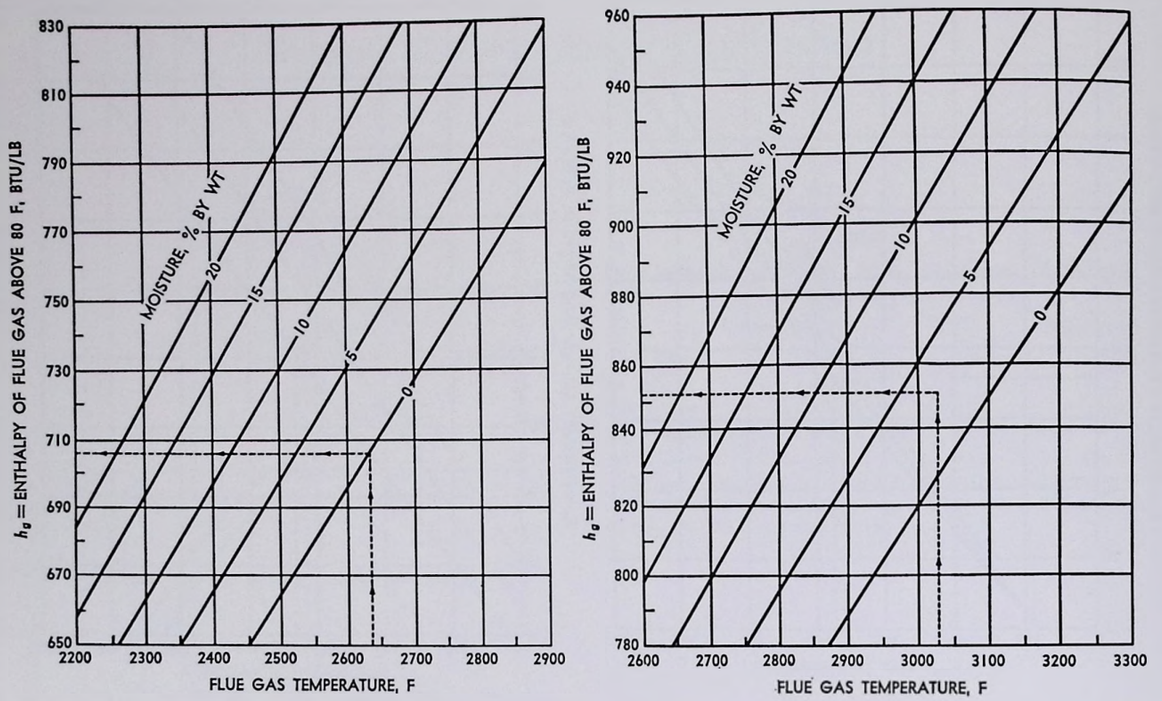


Fig. 2. *Enthalpy of flue gas above 80 F at 30 in. Hg, Btu per lb*



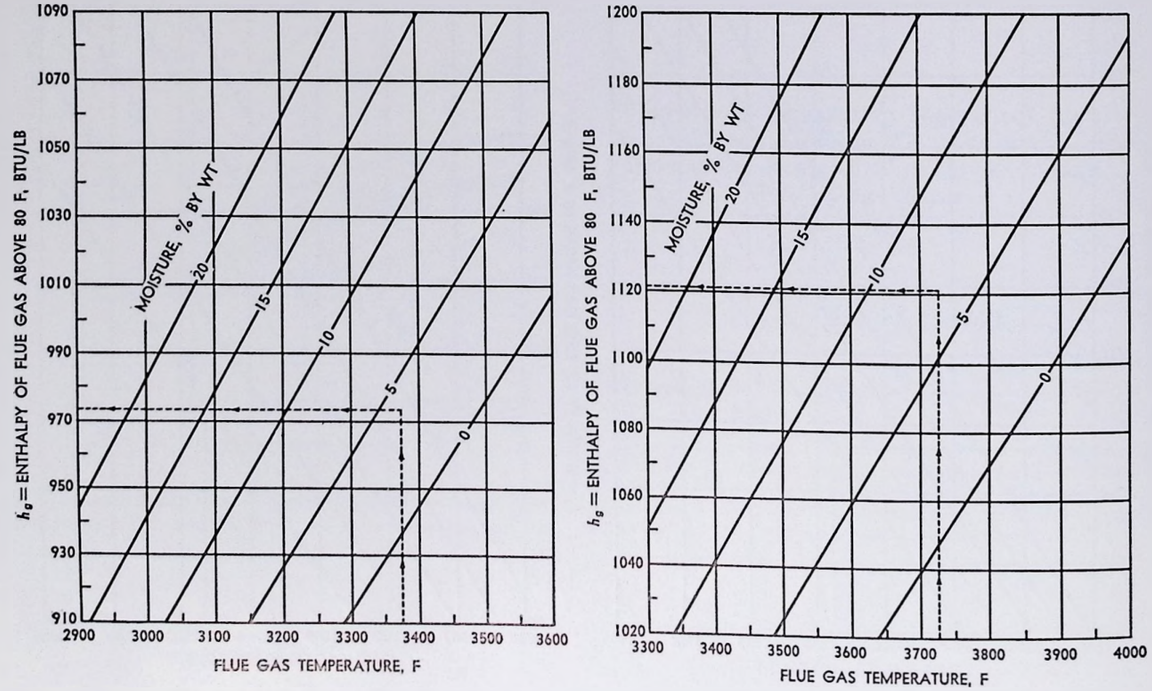


Fig. 2. (Cont'd) *Enthalpy of flue gas above 80 F at 30 in. Hg, Btu per lb*

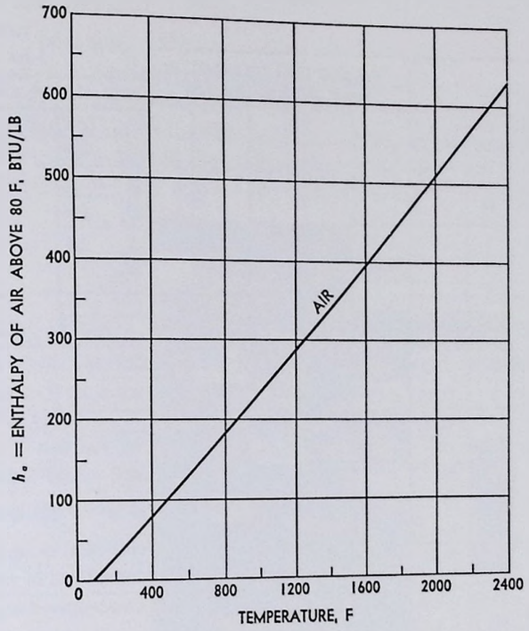


Fig. 3. *Enthalpy (above 80 F) of air (0.987 lb dry air plus 0.013 lb water vapor per lb mixture) at 30 in. Hg, Btu per lb*

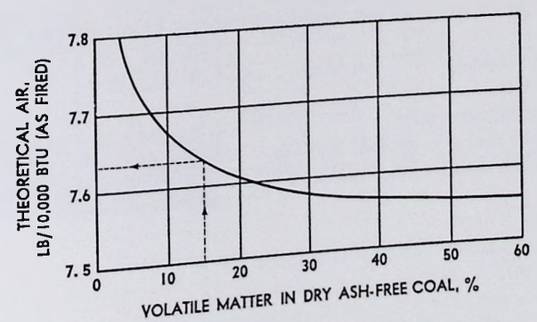


Fig. 4. *Theoretical air in lb per 10,000 Btu heat value of coal with a range of volatile*

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COMBUSTION CALCULATIONS—MOLAL BASIS

CONDITIONS—ASSIGNED OR OBSERVED AND MISCELLANEOUS

DATE

FUEL *Bituminous Coal*

SOURCE *Ohio*

FUEL UNIT { 100 LB, SOLID OR LIQUID FUELS / 100 MOLES, GASEOUS FUELS

FUEL, O_2, AND AIR PER UNIT OF FUEL

LINE	FUEL CONSTITUENT	PER FUEL UNIT, LB	MOL. WT DIVISOR	MOLES FUEL CONSTITUENT	O_2 MULTIPLIER	O_2 MOLES THEO REQD
1	C TO CO_2	*72.0*	12	*6.00*	1	*6.00*
2	C TO CO		12		.5	
3	CO TO CO_2		28		.5	
4	C UNBURNED, LINE k	*0.0*	12	*0.00*		
5	H_2	*4.4*	2	*2.20*	.5	*1.10*
6	S	*1.6*	32	*0.05*	1	*0.05*
7	O_2 (DEDUCT)	*3.6*	32	*0.11*	1	*−0.11*
8	N_2	*1.4*	28	*0.05*		0
9	CO_2		44			0
10	H_2O	*8.0*	18	*0.44*		0
11	ASH	*9.0*				0
12	SUM	*100.0*		*8.85*		*7.04*

O_2 AND AIR, MOLES FOR TOTAL AIR = *120* % (SEE LINE d AT RIGHT)

LINE		
13	O_2 (THEO) REQD = O_2, LINE 12	*7.04*
14	O_2 (EXCESS) = $\frac{\text{T.A.}-100}{100} \times O_2$, LINE 12	*1.41*
15	O_2 (TOTAL) SUPPLIED = LINES 13 + 14	*8.45*
16	N_2 SUPPLIED = 3.76 × O_2, LINE 15	*31.77*
17	AIR (DRY) SUPPLIED = $O_2 + N_2$	*40.22*
18	H_2O IN AIR = MOLES DRY AIR × $\frac{A}{B-A}$ *	*0.85*
19	AIR (WET) SUPPLIED = LINES 17 + 18	*41.07*

FLUE GAS (F.G.) COMPOSITION MOLES PER FUEL UNIT (AF)

LINE	$CO_2 + SO_2$	O_2	N_2	H_2O	CO
1	*6.00*				
5				*2.20*	
6	*0.05*				
8			*0.05*		
10				*0.44*	
13		*1.41*			
15			*31.77*		
18				*0.85*	
20	*6.05*	*1.41*	*31.82*	*3.49*	

	TOTAL MOLES	WET FLUE GAS	DRY FLUE GAS
s		*42.77*	*39.28*

FUEL ANAL. AS FIRED (AF), % BY WT OR VOL

C	*72.0*
H_2	*4.4*
S	*1.6*
O_2	*3.6*
N_2	*1.4*
H_2O	*8.0*
Ash	*9.0*
	100.0

CO_2 *15.4* O_2 *3.6* CO *0.0* N_2 *81.0* %†

TOTAL AIR (T.A.) ASSIGNED or by ORSAT *120* %

LINES f, g, h FOR GASEOUS FUELS

WT, FUEL UNIT = Σ(MOLES EACH × MOL. WT) LB

MOL. WT OF FUEL = LINE f ÷ 100

SP WT OF FUEL @ 80 F & 29.9" = $\frac{\text{LINE g}}{394}$ $\frac{\text{LB}}{\text{CU FT}}$

FUEL HEAT VALUE, BTU/LB *12,800* /CU FT

COMBUSTIBLE IN REFUSE, %"C" *0.0* %

CARBON UNBURNED, LB/100 LB FUEL = % ASH IN FUEL × $\frac{\%\text{ "C"}}{100 - \%\text{ "C"}}$ *0.0*

EXIT TEMP OF FLUE GAS, t_2 *345* F

DRY-BULB (AMBIENT) TEMP, t_1' *80* F

WET-BULB TEMP F

REL HUMID. (PSYCHROMETRIC CHART) %

B*, BAROMETRIC PRESSURE, IN. HG

SAT. PRESS. H_2O AT AMB TEMP, IN. HG

A*, PRESS. H_2O IN AIR, LINES (o × q), IN. HG

*NOTE—FOR AIR AT 80 F AND 60% RELATIVE HUMIDITY, $\frac{A}{B-A}$ = 0.0212 IS OFTEN USED AS STANDARD.

DETERMINATION OF FLUE GAS AND COMBUSTIBLE LOSSES IN BTU PER FUEL UNIT (AF)

LINE	FLUE GAS CONSTITUENTS	CO_2+SO_2	O_2	N_2	H_2O	CO	TOTAL
22		*9.7* *10.2*	*7.15*	*7.0*	*8.12*		
23	Mc_p, MEAN, t_2 to t_1' (FOR t_1' = 80 F, SEE FIG. 1)						
24	IN DRY FLUE GAS = MOLES EACH, LINE 20 × Mc_p × ($t_2 - t_1'$)	*15,560*	*2,670*	*59,000*			*77,230*
25	IN H_2O IN AIR = MOLES H_2O, LINE 18 × Mc_p × ($t_2 - t_1'$)				*1,830*		*1,830*
26	IN SENS HEAT, H_2O IN FUEL = MOLES LINES (5 + 10) × Mc_p × ($t_2 - t_1'$)				*5,680*		*5,680*
27	IN LATENT HEAT, H_2O IN FUEL = MOLES LINES (5 + 10) × 1040 × 18				*49,400*		*49,400*
28	TOTAL IN WET FLUE GASES						*134,140*
29	DUE TO CARBON IN REFUSE = LINE k × 14,550						*0*
30	DUE TO UNBURNED CO IN FLUE GAS = MOLES C TO CO × 12 × 10,200						*0*
31	TOTAL FLUE GAS LOSSES + UNBURNED COMBUSTIBLE = LINES 28+29+30					TOTAL	*134,140*
32	HEAT VALUE OF FUEL UNIT = 100 × LINE i FOR SOLID AND LIQUID FUELS / 394 × LINE i × 100, FOR GASEOUS FUELS						*1,280,000*
33	STACK AND COMBUSTIBLE LOSS, % OF HEAT INPUT = 100 × LINE 31 ÷ LINE 32						*10.48* %

†NOTE: FLUE GAS ANALYSIS BY ORSAT. IF CO IS PRESENT IN FLUE GASES, A CARBON BALANCE IS USED TO DETERMINE DISTRIBUTION OF C, THUS:
ALL C IN FUEL = C IN FLUE GAS CONSTITUENTS + C IN REFUSE. MOLES C IN FUEL = % C BY ANAL. ÷ 12.
MOLES C IN REFUSE = LINE k ÷ 12. MOLES C IN CO_2 = (MOLES C IN FUEL − MOLES C IN REFUSE) × % CO_2 BY ORSAT ÷ % CO_2 + CO BY ORSAT.
MOLES C IN CO = MOLES C IN FUEL − MOLES C IN REFUSE − MOLES C IN CO_2.

Conversion from Molal to Pound Units

SINCE the weight of a mole in lb is equal to the molecular weight of the element or compound, and the volume in cu ft of a mole of any gas under the same condition of temperature and pressure is approximately constant, the combustion calculations on the molal basis can be readily converted to lb units.

An example of conversion to lb units for the combustion calculations on the opposite page is given below. Reference in this form to gaseous fuels applies to combustion calculations for natural and blast-furnace gases on following pages.

MOLAL QUANTITIES—CONVERTED TO LB UNITS

SEE COMBUSTION CALCULATIONS—MOLAL BASIS FOR LINE REFERENCES

LINE			
a'	AIR (DRY) SUPPLIED PER UNIT OF FUEL = MOLES AIR, LINE 17 × 29, (MOL. WT AIR)	*1166.4* LB	a'
b'	H_2O IN AIR SUPPLIED PER UNIT OF FUEL = MOLES H_2O, LINE 18 × 18, (MOL. WT H_2O)	*15.3* LB	b'
c'	WET AIR SUPPLIED PER UNIT OF FUEL = LINES a' + b'	*1181.7* LB	c'
d'	WET AIR SUPPLIED PER LB OF FUEL { LINE c' ÷ 100 FOR SOLID OR LIQUID FUELS / LINE c' ÷ LINE f FOR GASEOUS FUELS	*11.52* LB / LB	d'

LINE	FLUE GAS CONSTITUENT AND MOL. WT	$CO_2 + SO_2$ 44 64	O_2 32	N_2 28	H_2O 18	CO 28	
e'							e'
f'	WET F.G., % BY VOL = 100 × LINE 20 ÷ LINE s (WET)*	*14.14*	*3.30*	*74.40*	*8.16*		f'
g'	DRY F.G., % BY VOL = 100 × LINE 20 ÷ LINE s (DRY)*	*15.40*	*3.59*	*81.01*			g'
h'	WT OF EACH CONSTIT = LINE 20 × MOL. WT OF EACH, LB	*264.0*	*3.2*	*45.1*	*891.0*	*62.3*	h'

LINE				
i'	TOTAL WT FLUE GAS = Σ (LINE h' WET) AND Σ (LINE h' DRY)	WET WT *1266.1* LB	DRY WT *1203.3* LB	i'
j'	WET FLUE GAS PER LB FUEL { WET WT, LINE i' ÷ 100, SOLID AND LIQUID FUELS / WET WT, LINE i' ÷ LINE f, GASEOUS FUELS		*1.2.66* LB / LB	j'
k'	DRY FLUE GAS PER LB FUEL { DRY WT, LINE i' ÷ 100, SOLID AND LIQUID FUELS / DRY WT, LINE i' ÷ LINE f, GASEOUS FUELS		*1.2.03* LB / LB	k'
l'	MOLECULAR WT OF WET FLUE GAS = WET WT, LINE i' ÷ WET MOLES, LINE s		*29.60*	l'
m'	MOLECULAR WT OF DRY FLUE GAS = DRY WT, LINE i' ÷ DRY MOLES, LINE s		*30.63*	m'
n'	SP WT OF WET FLUE GAS = LINE l' ÷ 394 † (AT 80 F AND 29.92 IN. HG)		*0.0751* LB/CU FT	n'
o'	SP WT OF DRY FLUE GAS = LINE m' ÷ 394 † (AT 80 F AND 29.92 IN. HG)		*0.0778* LB/CU FT	o'

* FLUE GAS ANALYSIS AS CALCULATED, BASED ON TOTAL AIR, LINE d, ASSIGNED OR OBSERVED. THE CALCULATED VALUES, LINE g', SHOULD AGREE SUBSTANTIALLY WITH ORSAT ANALYSIS WHEN FLUE GAS IS ANALYZED.

† VOLUME OF 1 MOLE OF ANY GAS AT 80 F AND 29.92 IN. HG = 394 CU FT.

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COMBUSTION CALCULATIONS—MOLAL BASIS

CONDITIONS—ASSIGNED OR OBSERVED AND MISCELLANEOUS

DATE

FUEL *Bituminous Coal*
SOURCE *Ohio*
FUEL UNIT { 100 LB, SOLID OR LIQUID FUELS / 100 MOLES, GASEOUS FUELS

LINE	FUEL, O_2, AND AIR PER UNIT OF FUEL — FUEL CONSTITUENT	PER FUEL UNIT, LB	MOL. WT DIVISOR	MOLES FUEL CONSTITUENT	O_2 MULTIPLIER	O_2 MOLES THEO REQD	FLUE GAS (F.G.) COMPOSITION MOLES PER FUEL UNIT (AF) $CO_2 + SO_2$	O_2	N_2	H_2O	CO
1	C TO CO_2	*69.6*	12	*5.80*	1	*5.80*	*5.80*				
2	C TO CO	*1.3*	12	*0.11*	.5	*0.06*					*0.11*
3	CO TO CO_2		28		.5						
4	C UNBURNED, LINE k	*1.1*	12	*0.09*							
5	H_2	*4.4*	2	*2.20*	.5	*1.10*				*2.20*	
6	S	*1.6*	32	*0.05*	1	*0.05*	*0.05*				
7	O_2 (DEDUCT)	*3.6*	32	*0.11*	1	*−0.11*					
8	N_2	*1.4*	28	*0.05*		O			*0.05*		
9	CO_2		44			O					
10	H_2O	*8.0*	18	*0.44*		O				*0.44*	
11	ASH	*9.0*				O					
12	SUM	*100.0*		*8.85*		*6.90*					

O_2 AND AIR, MOLES FOR TOTAL AIR = *120* % (SEE LINE d AT RIGHT)

13	O_2 (THEO) REQD = O_2, LINE 12	*6.90*				
14	O_2 (EXCESS) = $\frac{TA.-100}{100} \times O_2$, LINE 12	*1.38*		*1.38*		
15	O_2 (TOTAL) SUPPLIED = LINES 13 + 14	*8.28*				
16	N_2 SUPPLIED = 3.76 × O_2, LINE 15	*31.13*			*31.13*	
17	AIR (DRY) SUPPLIED = $O_2 + N_2$	*39.41*				
18	H_2O IN AIR = MOLES DRY AIR × $\frac{A}{B-A}$ *	*0.84*				*0.84*
19	AIR (WET) SUPPLIED = LINES 17 + 18	*40.25*				
20	FLUE GAS CONSTITUENTS = LINES 1 TO 18, TOTAL		*5.85*	*1.38*	*31.18*	*3.48*	*0.11*

	TOTAL MOLES	WET FLUE GAS	DRY FLUE GAS
		42.00	*38.52*

21 *NOTE—FOR AIR AT 80 F AND 60% RELATIVE HUMIDITY, $\frac{A}{B-A}$ = 0.0212 IS OFTEN USED AS STANDARD.

FUEL ANAL. AS FIRED (AF), % BY WT OR VOL

C	72.0
H_2	4.4
S	1.6
O_2	3.6
N_2	1.4
H_2O	8.0
Ash	9.0
	100.0

Moles C/100 lb fuel

C in fuel, 72 ÷ 12 = 6.00
C in refuse, 1.1 ÷ 12 = 0.09
C in CO_2 + CO, 6.00 − 0.09 = 5.91
C in CO_2, 5.91 × 15.2 ÷ 15.5 = 5.80
C in CO, 5.91 − 5.80 = 0.11

CO_2 *15.2* O_2 *3.6* CO *0.3* N_2 *80.9* % †

TOTAL AIR (T.A.) ASSIGNED or by ORSAT *120* %

LINES f, g, h FOR GASEOUS FUELS

WT, FUEL UNIT = Σ(MOLES EACH × MOL. WT) LB

MOL. WT OF FUEL = LINE f ÷ 100

SP WT OF FUEL @ 80 F & 29.9" = $\frac{\text{LINE g}}{394}$ $\frac{\text{LB}}{\text{CU FT}}$

FUEL HEAT VALUE, BTU/LB *12,800* / CU FT

COMBUSTIBLE IN REFUSE, %"C" *10.6* %

CARBON UNBURNED, LB/100 LB FUEL = % ASH IN FUEL × $\frac{\%\text{ "C"}}{100 - \%\text{ "C"}}$ *1.07 (1.1)*

EXIT TEMP OF FLUE GAS, t_2 *345* F

DRY-BULB (AMBIENT) TEMP, t_1' *80* F

WET-BULB TEMP F

REL HUMID. (PSYCHROMETRIC CHART) %

B*, BAROMETRIC PRESSURE, IN. HG

SAT. PRESS. H_2O AT AMB TEMP, IN. HG

A^* PRESS. H_2O IN AIR, LINES (o × q), IN. HG

DETERMINATION OF FLUE GAS AND COMBUSTIBLE LOSSES IN BTU PER FUEL UNIT (AF)

		CO_2+SO_2	O_2	N_2	H_2O	CO	TOTAL
22	FLUE GAS CONSTITUENTS	*9.7 10.2*	*7.15*	*7.0*	*8.12*	*7.0*	
23	Mc_p, MEAN, t_2 TO t_1' (FOR t_1' = 80 F, SEE FIG. 1)	*15,040*	*2,620*	*57,840*		*204*	*75,704*
24	IN DRY FLUE GAS = MOLES EACH LINE 20 × Mc_p × ($t_2 - t_1'$)				*1,810*		*1,810*
25	IN H_2O IN AIR = MOLES H_2O, LINE 18 × Mc_p × ($t_2 - t_1'$)				*5,680*		*5,680*
26	IN SENS HEAT, H_2O IN FUEL = MOLES LINES (5 + 10) × Mc_p × ($t_2 - t_1'$)				*49,420*		*49,420*
27	IN LATENT HEAT, H_2O IN FUEL = MOLES LINES (5 + 10) × 1040 × 18						*132,614*
28	TOTAL IN WET FLUE GASES						*15,570*
29	DUE TO CARBON IN REFUSE = LINE k × 14,550						*13,460*
30	DUE TO UNBURNED CO IN FLUE GAS = MOLES C TO CO × 12 × 10,200					TOTAL	*161,644*
31	TOTAL FLUE GAS LOSSES + UNBURNED COMBUSTIBLE = LINES 28+29+30						*1,280,000*
32	HEAT VALUE OF FUEL UNIT = 100 × LINE i FOR SOLID AND LIQUID FUELS / 394 × LINE i × 100, FOR GASEOUS FUELS						*12.63* %
33	STACK AND COMBUSTIBLE LOSS, % OF HEAT INPUT = 100 × LINE 31 ÷ LINE 32						

† NOTE: FLUE GAS ANALYSIS BY ORSAT. IF CO IS PRESENT IN FLUE GASES, A CARBON BALANCE IS USED TO DETERMINE DISTRIBUTION OF C, THUS:
ALL C IN FUEL = C IN FLUE GAS CONSTITUENTS + C IN REFUSE. MOLES C IN FUEL = % C BY ANAL. ÷ 12.
MOLES C IN REFUSE = LINE k ÷ 12. MOLES C IN CO_2 = (MOLES C IN FUEL − MOLES C IN REFUSE) × % CO_2 BY ORSAT ÷ % CO_2 + CO BY ORSAT.
MOLES C IN CO = MOLES C IN FUEL − MOLES C IN REFUSE − MOLES C IN CO_2.

COMBUSTION CALCULATIONS—MOLAL BASIS

CONDITIONS—ASSIGNED OR OBSERVED AND MISCELLANEOUS

DATE

FUEL *Fuel Oil, Bunker "C"*
SOURCE *California*
FUEL UNIT { 100 LB, SOLID OR LIQUID FUELS / 100 MOLES, GASEOUS FUELS

LINE	FUEL, O_2, AND AIR PER UNIT OF FUEL — FUEL CONSTITUENT	PER FUEL UNIT, LB	MOL. WT DIVISOR	MOLES FUEL CONSTITUENT	O_2 MULTIPLIER	O_2 MOLES THEO REQD	FLUE GAS (F.G.) COMPOSITION MOLES PER FUEL UNIT (AF) $CO_2 + SO_2$	O_2	N_2	H_2O	CO
1	C TO CO_2	*87.9*	12	*7.32*	1	*7.32*	*7.32*				
2	C TO CO		12		.5						
3	CO TO CO_2		28		.5						
4	C UNBURNED, LINE k		12								
5	H_2	*10.3*	2	*5.15*	.5	*2.58*				*5.15*	
6	S	*1.2*	32	*0.04*	1	*0.04*	*0.04*				
7	O_2 (DEDUCT)	*0.5*	32	*0.02*	1	*−0.02*					
8	N_2	*0.1*	28	*0.00*		O			*0.00*		
9	CO_2		44			O					
10	H_2O		18			O					
11	ASH					O					
12	SUM	*100 0*		*12.53*		*9.92*					

O_2 AND AIR, MOLES FOR TOTAL AIR = *119* % (SEE LINE d AT RIGHT)

13	O_2 (THEO) REQD = O_2, LINE 12	*9.92*				
14	O_2 (EXCESS) = $\frac{T.A.-100}{100} \times O_2$, LINE 12	*1.89*		*1.89*		
15	O_2 (TOTAL) SUPPLIED = LINES 13 + 14	*11.81*				
16	N_2 SUPPLIED = 3.76 × O_2, LINE 15	*44.41*			*44.41*	
17	AIR (DRY) SUPPLIED = $O_2 + N_2$	*56.22*				
18	H_2O IN AIR = MOLES DRY AIR × $\frac{A}{B-A}$ *	*1.19*				*1.19*
19	AIR (WET) SUPPLIED = LINES 17 + 18	*57.41*				
20	FLUE GAS CONSTITUENTS = LINES 1 TO 18, TOTAL		*7.36*	*1.89*	*44.41*	*6.34*	

	TOTAL MOLES	WET FLUE GAS	DRY FLUE GAS
		60.00	*53.66*

21 *NOTE—FOR AIR AT 80 F AND 60% RELATIVE HUMIDITY, $\frac{A}{B-A}$ = 0.0212 IS OFTEN USED AS STANDARD.

FUEL ANAL. AS FIRED (AF), % BY WT OR VOL

C	87.9
H_2	10.3
S	1.2
O_2	0.5
N_2	0.1
	100.0

CO O_2 CO N_2 % †

TOTAL AIR (T.A.) ASSIGNED or by ORSAT *119* %

LINES f, g, h FOR GASEOUS FUELS

WT, FUEL UNIT = Σ(MOLES EACH × MOL. WT) LB

MOL. WT OF FUEL = LINE f ÷ 100

SP WT OF FUEL @ 80 F & 29.9" = $\frac{\text{LINE g}}{394}$ $\frac{\text{LB}}{\text{CU FT}}$

FUEL HEAT VALUE, BTU/LB *18,400* / CU FT

COMBUSTIBLE IN REFUSE, %"C" %

CARBON UNBURNED, LB/100 LB FUEL = % ASH IN FUEL × $\frac{\%\text{ "C"}}{100 - \%\text{ "C"}}$ *0.0*

EXIT TEMP OF FLUE GAS, t_2 *303* F

DRY-BULB (AMBIENT) TEMP, t_1' *80* F

WET-BULB TEMP F

REL HUMID. (PSYCHROMETRIC CHART) %

B*, BAROMETRIC PRESSURE, IN. HG

SAT. PRESS. H_2O AT AMB TEMP, IN. HG

A^* PRESS. H_2O IN AIR, LINES (o × q), IN. HG

DETERMINATION OF FLUE GAS AND COMBUSTIBLE LOSSES IN BTU PER FUEL UNIT (AF)

		CO_2+SO_2	O_2	N_2	H_2O	CO	TOTAL
22	FLUE GAS CONSTITUENTS	*9.5 10.1*	*7.1*	*7.0*	*8.1*		
23	Mc_p, MEAN, t_2 TO t_1' (FOR t_1' = 80 F, SEE FIG. 1)	*15,600*	*2,990*	*69,330*			*87,920*
24	IN DRY FLUE GAS = MOLES EACH LINE 20 × Mc_p × ($t_2 - t_1'$)				*2,160*		*2,160*
25	IN H_2O IN AIR = MOLES H_2O, LINE 18 × Mc_p × ($t_2 - t_1'$)				*9,300*		*9,300*
26	IN SENS HEAT, H_2O IN FUEL = MOLES LINES (5 + 10) × Mc_p × ($t_2 - t_1'$)				*96,400*		*96,400*
27	IN LATENT HEAT, H_2O IN FUEL = MOLES LINES (5 + 10) × 1040 × 18						*195,780*
28	TOTAL IN WET FLUE GASES						*0*
29	DUE TO CARBON IN REFUSE = LINE k × 14,550						*0*
30	DUE TO UNBURNED CO IN FLUE GAS = MOLES C TO CO × 12 × 10,200					TOTAL	*195,780*
31	TOTAL FLUE GAS LOSSES + UNBURNED COMBUSTIBLE = LINES 28+29+30						*1,840,000*
32	HEAT VALUE OF FUEL UNIT = 100 × LINE i FOR SOLID AND LIQUID FUELS / 394 × LINE i × 100, FOR GASEOUS FUELS						*10.64* %
33	STACK AND COMBUSTIBLE LOSS, % OF HEAT INPUT = 100 × LINE 31 ÷ LINE 32						

† NOTE: FLUE GAS ANALYSIS BY ORSAT. IF CO IS PRESENT IN FLUE GASES, A CARBON BALANCE IS USED TO DETERMINE DISTRIBUTION OF C, THUS:
ALL C IN FUEL = C IN FLUE GAS CONSTITUENTS + C IN REFUSE. MOLES C IN FUEL = % C BY ANAL. ÷ 12.
MOLES C IN REFUSE = LINE k ÷ 12. MOLES C IN CO_2 = (MOLES C IN FUEL − MOLES C IN REFUSE) × % CO_2 BY ORSAT ÷ % CO_2 + CO BY ORSAT.
MOLES C IN CO = MOLES C IN FUEL − MOLES C IN REFUSE − MOLES C IN CO_2.

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COMBUSTION CALCULATIONS—MOLAL BASIS

CONDITIONS—ASSIGNED OR OBSERVED AND MISCELLANEOUS

DATE

FUEL *Natural Gas*

SOURCE *California*

FUEL UNIT { 100 LB, SOLID OR LIQUID FUELS / 100 MOLES, GASEOUS FUELS

FUEL ANAL. AS FIRED (AF), % BY WT OR VOL: CH_4 85.3; C_2H_6 12.6; O_2 0.3; N_2 1.7; CO_2 0.1; 100.0

Moles C & H_2/100 Moles fuel

	C	H_2
In CH_4	85.3	170.6
In C_2H_6	25.2	37.8
Totals	110.5	208.4

No unburned fuel

LINE	FUEL, O_2, AND AIR PER UNIT OF FUEL — FUEL CONSTITUENT	PER FUEL UNIT, LB	MOL. WT DIVISOR	MOLES FUEL CONSTITUENT	O_2 MULTIPLIER	O_2 MOLES THEO REQD	FLUE GAS (F.G.) COMPOSITION MOLES PER FUEL UNIT (AF): CO_2+SO_2	O_2	N_2	H_2O	CO
1	C TO CO_2	1326.0	12	110.5	1	110.5	110.5				
2	C TO CO		12		.5						
3	CO TO CO_2		28		.5						
4	C UNBURNED, LINE k		12								
5	H_2	416.8	2	208.4	.5	104.2				208.4	
6	S		32		1						
7	O_2 (DEDUCT)	9.6	32	0.3	1	− 0.3					
8	N_2	47.6	28	1.7		0			1.7		
9	CO_2	4.4	44	0.1		0	0.1				
10	H_2O		18			0					
11	ASH					0					
12	SUM	1804.4		321.0		214.4					
	O_2 AND AIR, MOLES FOR TOTAL AIR = 114 % (SEE LINE d AT RIGHT)										
13	O_2 (THEO) REQD = O_2, LINE 12					214.4					
14	O_2 (EXCESS) = $\frac{\text{T.A.}-100}{100} \times O_2$, LINE 12					30.0		30.0			
15	O_2 (TOTAL) SUPPLIED = LINES 13 + 14					244.4					
16	N_2 SUPPLIED = 3.76 × O_2, LINE 15					918.9			918.9		
17	AIR (DRY) SUPPLIED = $O_2 + N_2$					1163.3					
18	H_2O IN AIR = MOLES DRY AIR × $\frac{A}{B-A}$ *					24.7				24.7	
19	AIR (WET) SUPPLIED = LINES 17 + 18					1188.0					
20	FLUE GAS CONSTITUENTS = LINES 1 TO 18, TOTAL						110.6	30.0	920.6	233.1	

	TOTAL MOLES	WET FLUE GAS	DRY FLUE GAS
21		1294.3	1061.2

*NOTE—FOR AIR AT 80 F AND 60% RELATIVE HUMIDITY, $\frac{A}{B-A}$ = 0.0212 IS OFTEN USED AS STANDARD.

LINE		
	CO_2 O_2 CO N_2 % †	c
	TOTAL AIR (T.A.) ASSIGNED or by ORSAT 114 %	d
	LINES f, g, h FOR GASEOUS FUELS	e
	WT, FUEL UNIT = Σ(MOLES EACH × MOL. WT) LB	f
	MOL. WT OF FUEL = LINE f ÷ 100 18.04	g
	SP WT OF FUEL @ 80 F & 29.9" = $\frac{\text{LINE g}}{394}$ LB/CU FT	h
	FUEL HEAT VALUE, BTU/LB /CU FT 1092	i
	COMBUSTIBLE IN REFUSE, %"C" 0.0 %	j
	CARBON UNBURNED, LB/100 LB FUEL = % ASH IN FUEL × $\frac{\%\text{ "C"}}{100-\%\text{ "C"}}$ 0.0	k
	EXIT TEMP OF FLUE GAS, t_2 304 F	l
	DRY-BULB (AMBIENT) TEMP, t_1' 80 F	m
	WET-BULB TEMP F	n
	REL HUMID. (PSYCHROMETRIC CHART) %	o
	B*, BAROMETRIC PRESSURE, IN. HG	p
	SAT. PRESS. H_2O AT AMB TEMP, IN. HG	q
	A*, PRESS. H_2O IN AIR, LINES (o × q), IN. HG	r

DETERMINATION OF FLUE GAS AND COMBUSTIBLE LOSSES IN BTU PER FUEL UNIT (AF)

LINE	FLUE GAS CONSTITUENTS	CO_2+SO_2	O_2	N_2	H_2O	CO	TOTAL
22		9.55	7.12	7.0	8.1		
23	Mc_p, MEAN, t_2 TO t_1' (FOR t_1' = 80 F, SEE FIG. 1)						
24	IN DRY FLUE GAS = MOLES EACH LINE 20 × Mc_p × ($t_2 - t_1'$)	236,600	47,800	1,443,500			1,727,900
25	IN H_2O IN AIR = MOLES H_2O, LINE 18 × Mc_p × ($t_2 - t_1'$)				44,800		44,800
26	IN SENS HEAT, H_2O IN FUEL = MOLES LINES (5 + 10) × Mc_p × ($t_2 - t_1'$)				378,100		378,100
27	IN LATENT HEAT, H_2O IN FUEL = MOLES LINES (5 + 10) × 1040 × 18				3,901,300		3,901,300
28	TOTAL IN WET FLUE GASES						6,052,100
29	DUE TO CARBON IN REFUSE = LINE k × 14,550						0
30	DUE TO UNBURNED CO IN FLUE GAS = MOLES C TO CO × 12 × 10,200						0
31	TOTAL FLUE GAS LOSSES + UNBURNED COMBUSTIBLE = LINES 28+29+30					TOTAL	6,052,100
32	HEAT VALUE OF FUEL UNIT = 100 × LINE i FOR SOLID AND LIQUID FUELS / 394 × LINE i × 100, FOR GASEOUS FUELS						43,024,800
33	STACK AND COMBUSTIBLE LOSS, % OF HEAT INPUT = 100 × LINE 31 ÷ LINE 32						14.07 %

† NOTE: FLUE GAS ANALYSIS BY ORSAT. IF CO IS PRESENT IN FLUE GASES, A CARBON BALANCE IS USED TO DETERMINE DISTRIBUTION OF C, THUS:
ALL C IN FUEL = C IN FLUE GAS CONSTITUENTS + C IN REFUSE. MOLES C IN FUEL = % C BY ANAL. ÷ 12.
MOLES C IN REFUSE = LINE k ÷ 12. MOLES C IN CO_2 = (MOLES C IN FUEL − MOLES C IN REFUSE) × % CO_2 BY ORSAT ÷ % CO_2 + CO BY ORSAT.
MOLES C IN CO = MOLES C IN FUEL − MOLES C IN REFUSE − MOLES C IN CO_2.

COMBUSTION CALCULATIONS—MOLAL BASIS

CONDITIONS—ASSIGNED OR OBSERVED AND MISCELLANEOUS

DATE

FUEL *Blast Furnace Gas*

SOURCE *Wierton, W. Va.*

FUEL UNIT { 100 LB, SOLID OR LIQUID FUELS / 100 MOLES, GASEOUS FUELS

FUEL ANAL. AS FIRED (AF), % BY WT OR VOL: H_2 3.0; CO 25.5; CO_2 14.5; N_2 57.0; ..... ; 100.0

LINE	FUEL, O_2, AND AIR PER UNIT OF FUEL — FUEL CONSTITUENT	PER FUEL UNIT, LB	MOL. WT DIVISOR	MOLES FUEL CONSTITUENT	O_2 MULTIPLIER	O_2 MOLES THEO REQD	FLUE GAS (F.G.) COMPOSITION MOLES PER FUEL UNIT (AF): CO_2+SO_2	O_2	N_2	H_2O	CO
1	C TO CO_2		12		1						
2	C TO CO		12		.5						
3	CO TO CO_2	714	28	25.5	.5	12.75	25.5				
4	C UNBURNED, LINE k		12								
5	H_2	6	2	3.0	.5	1.50				3.0	
6	S		32		1						
7	O_2 (DEDUCT)		32		1	−					
8	N_2	1596	28	57.0		0			57.0		
9	CO_2	638	44	14.5		0	14.5				
10	H_2O		18			0					
11	ASH					0					
12	SUM	2954		100.0		14.25					
	O_2 AND AIR, MOLES FOR TOTAL AIR = 125 % (SEE LINE d AT RIGHT)										
13	O_2 (THEO) REQD = O_2, LINE 12					14.25					
14	O_2 (EXCESS) = $\frac{\text{T.A.}-100}{100} \times O_2$, LINE 12					3.56		3.56			
15	O_2 (TOTAL) SUPPLIED = LINES 13 + 14					17.81					
16	N_2 SUPPLIED = 3.76 × O_2, LINE 15					66.97			67.0		
17	AIR (DRY) SUPPLIED = $O_2 + N_2$					84.78					
18	H_2O IN AIR = MOLES DRY AIR × $\frac{A}{B-A}$ *					1.80				1.80	
19	AIR (WET) SUPPLIED = LINES 17 + 18					86.58					
20	FLUE GAS CONSTITUENTS = LINES 1 TO 18, TOTAL						40.0	3.56	124.0	4.80	

	TOTAL MOLES	WET FLUE GAS	DRY FLUE GAS
21		172.36	167.56

*NOTE—FOR AIR AT 80 F AND 60% RELATIVE HUMIDITY, $\frac{A}{B-A}$ = 0.0212 IS OFTEN USED AS STANDARD.

LINE		
	CO_2 O_2 CO N_2 % †	c
	TOTAL AIR (T.A.) ASSIGNED or by ORSAT 125 %	d
	LINES f, g, h FOR GASEOUS FUELS	e
	WT, FUEL UNIT = Σ(MOLES EACH × MOL. WT) 954 LB	f
	MOL. WT OF FUEL = LINE f ÷ 100 29.54	g
	SP WT OF FUEL @ 80 F & 29.9" = $\frac{\text{LINE g}}{394}$ .075 LB/CU FT	h
	FUEL HEAT VALUE, BTU/LB /CU FT 90.0	i
	COMBUSTIBLE IN REFUSE, %"C" 0.0 %	j
	CARBON UNBURNED, LB/100 LB FUEL = % ASH IN FUEL × $\frac{\%\text{ "C"}}{100-\%\text{ "C"}}$ 0.0	k
	EXIT TEMP OF FLUE GAS, t_2 525 F	l
	DRY-BULB (AMBIENT) TEMP, t_1' 50 F	m
	WET-BULB TEMP F	n
	REL HUMID. (PSYCHROMETRIC CHART) %	o
	B*, BAROMETRIC PRESSURE, IN. HG	p
	SAT. PRESS. H_2O AT AMB TEMP, IN. HG	q
	A*, PRESS. H_2O IN AIR, LINES (o × q), IN. HG	r

DETERMINATION OF FLUE GAS AND COMBUSTIBLE LOSSES IN BTU PER FUEL UNIT (AF)

LINE	FLUE GAS CONSTITUENTS	CO_2+SO_2	O_2	N_2	H_2O	CO	TOTAL
22		10.1	7.22	7.05	8.20		
23	Mc_p, MEAN, t_2 TO t_1' (FOR t_1' = 80 F, SEE FIG. 1)						
24	IN DRY FLUE GAS = MOLES EACH, LINE 20 × Mc_p × ($t_2 - t_1'$)	179,800	11,400	389,000			580,200
25	IN H_2O IN AIR = MOLES H_2O, LINE 18 × Mc_p × ($t_2 - t_1'$)				6,570		6,570
26	IN SENS HEAT, H_2O IN FUEL = MOLES LINES (5 + 10) × Mc_p × ($t_2 - t_1'$)				10,950		10,950
27	IN LATENT HEAT, H_2O IN FUEL = MOLES LINES (5 + 10) × 1040 × 18				56,160		56,160
28	TOTAL IN WET FLUE GASES						653,880
29	DUE TO CARBON IN REFUSE = LINE k × 14,550						0
30	DUE TO UNBURNED CO IN FLUE GAS = MOLES C TO CO × 12 × 10,200						0
31	TOTAL FLUE GAS LOSSES + UNBURNED COMBUSTIBLE = LINES 28+29+30					TOTAL	653,880
32	HEAT VALUE OF FUEL UNIT = 100 × LINE i FOR SOLID AND LIQUID FUELS / 394 × LINE i × 100, FOR GASEOUS FUELS						3,546,000
33	STACK AND COMBUSTIBLE LOSS, % OF HEAT INPUT = 100 × LINE 31 ÷ LINE 32						18.44 %

† NOTE: FLUE GAS ANALYSIS BY ORSAT. IF CO IS PRESENT IN FLUE GASES, A CARBON BALANCE IS USED TO DETERMINE DISTRIBUTION OF C, THUS:
ALL C IN FUEL = C IN FLUE GAS CONSTITUENTS + C IN REFUSE. MOLES C IN FUEL = % C BY ANAL. ÷ 12.
MOLES C IN REFUSE = LINE k ÷ 12. MOLES C IN CO_2 = (MOLES C IN FUEL − MOLES C IN REFUSE) × % CO_2 BY ORSAT ÷ % CO_2 + CO BY ORSAT.
MOLES C IN CO = MOLES C IN FUEL − MOLES C IN REFUSE − MOLES C IN CO_2.

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COMBUSTION CALCULATIONS

BASED ON QUANTITIES PER 10,000 BTU FUEL INPUT

FUEL— *Bituminous Coal, Virginia*

ANALYSIS AS FIRED

ULTIMATE, % BY WT		PROXIMATE, % BY WT	
C	80.31	MOISTURE	2.90
H_2	4.47	VOLATILE	22.05
S	1.54	FIXED CARBON	68.50
O_2	2.85	ASH	6.55
N_2	1.38		100.00
H_2O	2.90		
ASH	6.55		
	100.00		

BTU PER LB, AS FIRED, *14,100*

	CONDITIONS BY SPECIFICATION		DATE	
a				a
b	TOTAL AIR	%	120	b
c	AIR TEMPERATURE TO HEATER	F	80	c
d	AIR TEMPERATURE FROM HEATER	F	350	d
e	FLUE GAS TEMPERATURE LEAVING UNIT	F	280	e
f	H_2O PER LB DRY AIR	LB	0.0132	f
g				g
h	UNBURNED FUEL LOSS	%	0.4	h
i	UNACCOUNTED LOSS & MFR MARGIN	%	1.5	i
j	RADIATION LOSS (ABMA), FIG. 20, CHAPTER 7	%	0.8	j
k				k

13	**QUANTITIES PER 10,000 BTU FUEL INPUT**			13
14	FUEL BURNED, 10,000 ÷ LINE 12	LB	0.71	14
15	TOTAL AIR REQUIRED, LINE b ÷ 100 × VALUE FROM FIG. 4 OR TABLE 5 OR 6 = *1.2 × 7.62*	LB	9.14	15
16	H_2O IN AIR, LINE 15 × LINE f = *9.14 × 0.0132*	LB	0.12	16
17	WET GAS, TOTAL, LINES (14 + 15 + 16)	LB	9.97	17
18	H_2O IN FUEL, (LINE 5 ÷ 100) × LINE 14 × 8.94 + (LINE 9 ÷ 100) × LINE 14; OR FROM TABLE 5	LB	0.30	18
19	H_2O IN FLUE GAS, TOTAL, LINE 16 + LINE 18	LB	0.42	19
20	H_2O IN FLUE GAS, TOTAL, IN PER CENT, (LINE 19 ÷ LINE 17) × 100	%	4.21	20
21	DRY GAS, TOTAL, LINE 17—LINE 19	LB	9.55	21
22	**LOSSES PER 10,000 BTU FUEL INPUT**			22
23	UNBURNED FUEL, 10,000 × LINE h ÷ 100	BTU	40	23
24	UNACCOUNTED & MFR MARGIN, 10,000 × LINE i ÷ 100	BTU	150	24
25	RADIATION, 10,000 × LINE j ÷ 100	BTU	80	25
26	LATENT HEAT, H_2O IN FUEL, 1040 × LINE 18	BTU	312	26
27	SENSIBLE HEAT, FLUE GAS, LINE 17 × BTU FROM FIG. 2 @ LINE e AND LINE 20 = *9.97 × 50*	BTU	499	27
28	TOTAL LOSSES, LINES (23 + 24 + 25 + 26 + 27)	BTU	1081	28
29	TOTAL LOSSES IN PER CENT, LINE (28 ÷ 10,000) × 100	%	10.8	29
30	EFFICIENCY, BY DIFFERENCE, 100—LINE 29	%	89.2	30
31	**QUANTITIES PER 10,000 BTU FUEL INPUT COMBUSTION TEMPERATURE, ADIABATIC**			31
32	HEAT INPUT FROM FUEL	BTU	10,000	32
33	HEAT INPUT FROM AIR, LINES (15 + 16) × BTU FROM FIG. 3 @ LINE d TEMP = *9.26 × 66*	BTU	612	33
34	HEAT INPUT, TOTAL, LINES (32 + 33)	BTU	10,612	34
35	LESS LATENT HEAT LOSS, H_2O IN FUEL, LINE 26	BTU	−312	35
36	HEAT AVAILABLE, MAXIMUM	BTU	10,300	36
37	LESS LINES (24 + 25) × 0.5*	BTU	− 115	37
38	HEAT AVAILABLE, LINE 36—LINE 37	BTU	10,185	38
39	HEAT AVAILABLE PER LB OF FLUE GAS, LINE 38 ÷ LINE 17	BTU	1021	39
40	ADIABATIC TEMPERATURE, FROM FIG. 2 FOR LINES 20 & 39	F	3513	40

* NOTE: IT IS CUSTOMARY TO REDUCE THE MAXIMUM HEAT AVAILABLE, LINE 36, BY FROM ⅓ TO ½ OF THE UNACCOUNTED & MFR MARGIN PLUS RADIATION LOSSES, ON THE ASSUMPTION THAT A PORTION OF THESE LOSSES OCCURS IN THE COMBUSTION ZONE.

COMBUSTION CALCULATIONS

BASED ON QUANTITIES PER 10,000 BTU FUEL INPUT

FUEL— *Fuel Oil, Texas, Heavy*

ANALYSIS AS FIRED

ULTIMATE, % BY WT		PROXIMATE, % BY WT	
C	*Unknown*	MOISTURE	
H_2		VOLATILE	
S		FIXED CARBON	
O_2		ASH	
N_2			
H_2O			
ASH			

BTU PER LB, AS FIRED, *18,500*

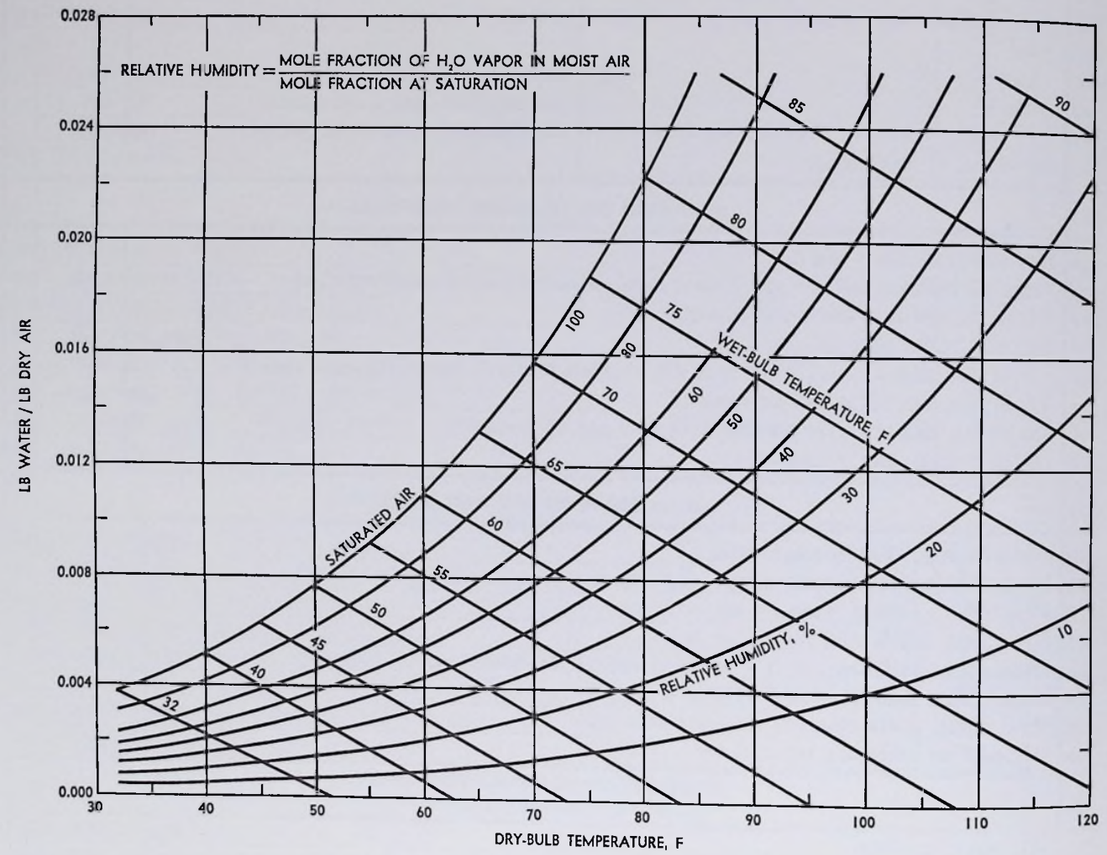
	CONDITIONS BY SPECIFICATION		DATE	
a				a
b	TOTAL AIR	%	120	b
c	AIR TEMPERATURE TO HEATER	F	80	c
d	AIR TEMPERATURE FROM HEATER	F	500	d
e	FLUE GAS TEMPERATURE LEAVING UNIT	F	350	e
f	H_2O PER LB DRY AIR	LB	0.0132	f
g				g
h	UNBURNED FUEL LOSS	%	0.00	h
i	UNACCOUNTED LOSS & MFR MARGIN	%	1.5	i
j	RADIATION LOSS (ABMA), FIG. 20, CHAPTER 7	%	1.0	j
k				k

13	**QUANTITIES PER 10,000 BTU FUEL INPUT**			13
14	FUEL BURNED, 10,000 ÷ LINE 12	LB	0.54	14
15	TOTAL AIR REQUIRED, LINE b ÷ 100 × VALUE FROM FIG. 4 OR TABLE 5 OR 6 = *1.2 × 7.46*	LB	8.95	15
16	H_2O IN AIR, LINE 15 × LINE f = *8.95 × 0.0132*	LB	0.12	16
17	WET GAS, TOTAL, LINES (14 + 15 + 16)	LB	9.61	17
18	H_2O IN FUEL, (LINE 5 ÷ 100) × LINE 14 × 8.94 + (LINE 9 ÷ 100) × LINE 14; OR FROM TABLE 5	LB	0.51	18
19	H_2O IN FLUE GAS, TOTAL, LINE 16 + LINE 18	LB	0.63	19
20	H_2O IN FLUE GAS, TOTAL, IN PER CENT, (LINE 19 ÷ LINE 17) × 100	%	6.55	20
21	DRY GAS, TOTAL, LINE 17—LINE 19	LB	8.98	21
22	**LOSSES PER 10,000 BTU FUEL INPUT**			22
23	UNBURNED FUEL, 10,000 × LINE h ÷ 100	BTU	0	23
24	UNACCOUNTED & MFR MARGIN, 10,000 × LINE i ÷ 100	BTU	150	24
25	RADIATION, 10,000 × LINE j ÷ 100	BTU	100	25
26	LATENT HEAT, H_2O IN FUEL, 1040 × LINE 18	BTU	530	26
27	SENSIBLE HEAT, FLUE GAS, LINE 17 × BTU FROM FIG. 2 @ LINE e AND LINE 20 = *9.61 × 70*	BTU	673	27
28	TOTAL LOSSES, LINES (23 + 24 + 25 + 26 + 27)	BTU	1453	28
29	TOTAL LOSSES IN PER CENT, LINE (28 ÷ 10,000) × 100	%	14.5	29
30	EFFICIENCY, BY DIFFERENCE, 100—LINE 29	%	85.5	30
31	**QUANTITIES PER 10,000 BTU FUEL INPUT COMBUSTION TEMPERATURE, ADIABATIC**			31
32	HEAT INPUT FROM FUEL	BTU	10,000	32
33	HEAT INPUT FROM AIR, LINES (15 + 16) × BTU FROM FIG. 3 @ LINE d TEMP = *9.07 × 104*	BTU	943	33
34	HEAT INPUT, TOTAL, LINES (32 + 33)	BTU	10,943	34
35	LESS LATENT HEAT LOSS, H_2O IN FUEL, LINE 26	BTU	−530	35
36	HEAT AVAILABLE, MAXIMUM	BTU	10,413	36
37	LESS LINES (24 + 25) × 0.5*	BTU	−125	37
38	HEAT AVAILABLE, LINE 36—LINE 37	BTU	10,288	38
39	HEAT AVAILABLE PER LB OF FLUE GAS, LINE 38 ÷ LINE 17	BTU	1071	39
40	ADIABATIC TEMPERATURE, FROM FIG. 2 FOR LINES 20 & 39	F	3600	40

* NOTE: IT IS CUSTOMARY TO REDUCE THE MAXIMUM HEAT AVAILABLE, LINE 36, BY FROM ⅓ TO ½ OF THE UNACCOUNTED & MFR MARGIN PLUS RADIATION LOSSES, ON THE ASSUMPTION THAT A PORTION OF THESE LOSSES OCCURS IN THE COMBUSTION ZONE.

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PSYCHROMETRIC CHART



Chapter 6. Temperature Measurement

Temperature-emf relationship of standardized thermocouple wires as established by Leeds & Northrup for four thermocouple pairs

90% Platinum-10% Rhodium (+) to Platinum (−) Thermocouple

Degrees F — Reference Junction, Zero F

Millivolts

F	0	100	200	300	400	500	600	700	800	900	1000	1100	1200	1300	1400	1500
0	0.000	0.313	0.687	1.108	1.565	2.048	2.549	3.067	3.597	4.136	4.686	5.247	5.817	6.399	6.990	7.592
5	0.014	0.331	0.707	1.130	1.589	2.072	2.574	3.093	3.624	4.163	4.713	5.275	5.846	6.428	7.020	7.622
10	0.028	0.349	0.727	1.152	1.613	2.097	2.600	3.120	3.651	4.190	4.741	5.303	5.875	6.458	7.050	7.653
15	0.043	0.366	0.747	1.174	1.636	2.122	2.626	3.146	3.677	4.217	4.769	5.331	5.904	6.487	7.079	7.683
20	0.057	0.384	0.768	1.197	1.660	2.147	2.652	3.173	3.704	4.245	4.797	5.360	5.933	6.516	7.109	7.713
25	0.071	0.402	0.788	1.219	1.684	2.172	2.677	3.199	3.731	4.272	4.825	5.388	5.961	6.545	7.139	7.743
30	0.086	0.420	0.809	1.242	1.708	2.197	2.703	3.225	3.759	4.300	4.853	5.416	5.990	6.575	7.169	7.774
35	0.101	0.438	0.829	1.265	1.732	2.222	2.729	3.251	3.786	4.327	4.881	5.444	6.019	6.604	7.199	7.805
40	0.116	0.457	0.850	1.288	1.756	2.247	2.755	3.278	3.812	4.355	4.909	5.473	6.048	6.634	7.229	7.836
45	0.132	0.475	0.871	1.310	1.780	2.272	2.781	3.304	3.839	4.382	4.937	5.501	6.077	6.663	7.259	7.866
50	0.148	0.493	0.892	1.333	1.804	2.297	2.807	3.331	3.866	4.410	4.965	5.530	6.107	6.693	7.290	7.897
55	0.164	0.512	0.913	1.356	1.828	2.322	2.833	3.358	3.893	4.437	4.993	5.558	6.136	6.722	7.320	7.927
60	0.180	0.531	0.935	1.379	1.852	2.347	2.859	3.385	3.920	4.465	5.021	5.587	6.165	6.752	7.350	7.958
65	0.196	0.550	0.956	1.402	1.876	2.372	2.885	3.411	3.947	4.492	5.049	5.616	6.194	6.782	7.380	7.989
70	0.212	0.569	0.978	1.425	1.900	2.397	2.911	3.438	3.974	4.520	5.077	5.645	6.224	6.812	7.410	8.020
75	0.229	0.588	0.999	1.448	1.924	2.422	2.937	3.464	4.001	4.547	5.105	5.673	6.263	6.841	7.440	8.050
80	0.245	0.608	1.021	1.472	1.949	2.448	2.963	3.491	4.028	4.575	5.134	5.702	6.282	6.871	7.471	8.081
85	0.262	0.627	1.042	1.495	1.973	2.473	2.989	3.517	4.055	4.603	5.162	5.730	6.311	6.900	7.501	8.112
90	0.279	0.647	1.064	1.518	1.998	2.498	3.015	3.544	4.082	4.631	5.190	5.759	6.341	6.930	7.531	8.143
95	0.296	0.667	1.086	1.541	2.023	2.523	3.041	3.570	4.109	4.658	5.218	5.788	6.370	6.960	7.561	8.173
100	0.313	0.687	1.108	1.565	2.048	2.549	3.067	3.597	4.136	4.686	5.247	5.817	6.399	6.990	7.592	8.204
*	0.00313	0.00374	0.00421	0.00457	0.00483	0.00501	0.00518	0.00530	0.00539	0.00550	0.00561	0.00570	0.00582	0.00591	0.00602	0.00612

F	1600	1700	1800	1900	2000	2100	2200	2300	2400	2500	2600	2700	2800	2900	3000	3100
							Millivolts									
0	8.204	8.826	9.457	10.099	10.749	11.408	12.069	12.734	13.397	14.060	14.721	15.380	16.035	16.688	17.339	17.984
5	8.235	8.857	9.489	10.131	10.782	11.441	12.102	12.767	13.431	14.093	14.754	15.413	16.068	16.721	17.371	18.017
10	8.266	8.888	9.521	10.163	10.815	11.474	12.135	12.800	13.464	14.126	14.787	15.445	16.101	16.753	17.403	18.049
15	8.297	8.919	9.553	10.195	10.848	11.507	12.168	12.833	13.497	14.159	14.820	15.478	16.133	16.786	17.436	18.081
20	8.328	8.951	9.585	10.228	10.881	11.540	12.202	12.867	13.530	14.192	14.853	15.510	16.166	16.818	17.468	18.113
25	8.359	8.982	9.617	10.260	10.914	11.573	12.235	12.900	13.564	14.225	14.886	15.543	16.199	16.851	17.500	—
30	8.390	9.014	9.649	10.293	10.947	11.606	12.269	12.933	13.597	14.258	14.918	15.576	16.231	16.883	17.532	—
35	8.421	9.045	9.681	10.325	10.979	11.639	12.302	12.966	13.630	14.292	14.951	15.609	16.264	16.916	17.565	—
40	8.452	9.077	9.713	10.358	11.012	11.672	12.335	12.999	13.663	14.325	14.984	15.642	16.297	16.948	17.597	—
45	8.483	9.108	9.745	10.390	11.045	11.705	12.368	13.033	13.696	14.358	15.017	15.675	16.329	16.981	17.630	—
50	8.514	9.140	9.777	10.423	11.078	11.738	12.402	13.066	13.729	14.391	15.050	15.707	16.362	17.014	17.662	—
55	8.545	9.171	9.809	10.455	11.111	11.771	12.435	13.099	13.762	14.424	15.083	15.740	16.395	17.047	17.694	—
60	8.576	9.203	9.841	10.488	11.144	11.804	12.468	13.132	13.795	14.457	15.116	15.772	16.427	17.079	17.726	—
65	8.607	9.235	9.873	10.520	11.177	11.837	12.501	13.166	13.829	14.490	15.149	15.805	16.460	17.112	17.759	—
70	8.638	9.267	9.905	10.553	11.210	11.870	12.534	13.199	13.862	14.523	15.182	15.838	16.493	17.144	17.791	—
75	8.669	9.298	9.937	10.585	11.243	11.903	12.567	13.232	13.895	14.556	15.215	15.871	16.525	17.177	17.823	—
80	8.701	9.330	9.969	10.618	11.276	11.936	12.601	13.265	13.928	14.589	15.248	15.904	16.558	17.209	17.855	—
85	8.732	9.362	10.002	10.650	11.309	11.969	12.634	13.298	13.961	14.622	15.281	15.937	16.590	17.242	17.888	—
90	8.763	9.394	10.034	10.683	11.342	12.003	12.667	13.331	13.994	14.655	15.314	15.970	16.623	17.274	17.920	—
95	8.794	9.425	10.066	10.716	11.375	12.036	12.700	13.364	14.027	14.688	15.347	16.003	16.656	17.307	17.952	—
100	8.826	9.457	10.099	10.749	11.408	12.069	12.734	13.397	14.060	14.721	15.380	16.035	16.688	17.339	17.984	—
*	0.00622	0.00631	0.00642	0.00650	0.00659	0.00661	0.00665	0.00663	0.00663	0.00661	0.00659	0.00655	0.00653	0.00651	0.00615	0.00645

* Average millivolts per degree F.

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Copper (+) to Constantan (−) Thermocouple

Degrees F — Reference Junction, Zero F

F	mv	F	mv	F	mv	F	mv	F	mv	F	mv	F	mv
0	0.00	100	2.19	200	4.64	300	7.31	400	10.19	500	13.24	600	16.44
2	0.04	102	2.23	202	4.69	302	7.37	402	10.25	502	13.30	602	16.50
4	0.08	104	2.28	204	4.74	304	7.43	404	10.31	504	13.37	604	16.57
6	0.12	106	2.33	206	4.79	306	7.48	406	10.37	506	13.43	606	16.63
8	0.16	108	2.37	208	4.84	308	7.54	408	10.43	508	13.49	608	16.70
10	0.21	110	2.42	210	4.89	310	7.59	410	10.49	510	13.56	610	16.77
12	0.25	112	2.47	212	4.95	312	7.65	412	10.55	512	13.62	612	16.83
14	0.29	114	2.51	214	5.00	314	7.71	414	10.61	514	13.68	614	16.90
16	0.33	116	2.56	216	5.05	316	7.76	416	10.67	516	13.74	616	16.96
18	0.37	118	2.61	218	5.10	318	7.82	418	10.73	518	13.81	618	17.03
20	0.42	120	2.66	220	5.15	320	7.88	420	10.79	520	13.87	620	17.09
22	0.46	122	2.70	222	5.21	322	7.93	422	10.85	522	13.93	622	17.16
24	0.50	124	2.75	224	5.26	324	7.99	424	10.91	524	14.00	624	17.23
26	0.54	126	2.80	226	5.31	326	8.05	426	10.97	526	14.06	626	17.29
28	0.58	128	2.85	228	5.36	328	8.10	428	11.03	528	14.12	628	17.36
30	0.63	130	2.90	230	5.42	330	8.16	430	11.09	530	14.19	630	17.42
32	0.67	132	2.94	232	5.47	332	8.22	432	11.15	532	14.25	632	17.49
34	0.71	134	2.99	234	5.52	334	8.27	434	11.21	534	14.31	634	17.56
36	0.76	136	3.04	236	5.58	336	8.33	436	11.27	536	14.38	636	17.62
38	0.80	138	3.09	238	5.63	338	8.39	438	11.33	538	14.44	638	17.69
40	0.84	140	3.14	240	5.68	340	8.44	440	11.39	540	14.50	640	17.75
42	0.89	142	3.19	242	5.74	342	8.50	442	11.45	542	14.57	642	17.82
44	0.93	144	3.23	244	5.79	344	8.56	444	11.51	544	14.63	644	17.89
46	0.97	146	3.28	246	5.84	346	8.62	446	11.57	546	14.69	646	17.95
48	1.02	148	3.33	248	5.90	348	8.67	448	11.64	548	14.76	648	18.02
50	1.06	150	3.38	250	5.95	350	8.73	450	11.70	550	14.82	650	18.09
52	1.10	152	3.43	252	6.00	352	8.79	452	11.76	552	14.89	652	18.15
54	1.15	154	3.48	254	6.06	354	8.85	454	11.82	554	14.95	654	18.22
56	1.19	156	3.53	256	6.11	356	8.90	456	11.88	556	15.01	656	18.29
58	1.24	158	3.58	258	6.16	358	8.96	458	11.94	558	15.08	658	18.35
60	1.28	160	3.63	260	6.22	360	9.02	460	12.00	560	15.14	660	18.42
62	1.32	162	3.68	262	6.27	362	9.08	462	12.06	562	15.21	662	18.49
64	1.37	164	3.73	264	6.33	364	9.14	464	12.12	564	15.27	664	18.55
66	1.41	166	3.78	266	6.38	366	9.19	466	12.19	566	15.33	666	18.62
68	1.46	168	3.83	268	6.43	368	9.25	468	12.25	568	15.40	668	18.69
70	1.50	170	3.88	270	6.49	370	9.31	470	12.31	570	15.46	670	18.75
72	1.55	172	3.93	272	6.54	372	9.37	472	12.37	572	15.53	672	18.82
74	1.59	174	3.98	274	6.60	374	9.43	474	12.43	574	15.59	674	18.89
76	1.64	176	4.03	276	6.65	376	9.48	476	12.49	576	15.66	676	18.95
78	1.68	178	4.08	278	6.71	378	9.54	478	12.56	578	15.72	678	19.02
80	1.73	180	4.13	280	6.76	380	9.60	480	12.62	580	15.79	680	19.09
82	1.77	182	4.18	282	6.82	382	9.66	482	12.68	582	15.85	682	19.16
84	1.82	184	4.23	284	6.87	384	9.72	484	12.74	584	15.92	684	19.22
86	1.86	186	4.28	286	6.93	386	9.78	486	12.80	586	15.98	686	19.29
88	1.91	188	4.33	288	6.98	388	9.84	488	12.87	588	16.05	688	19.36
90	1.96	190	4.38	290	7.04	390	9.90	490	12.93	590	16.11	690	19.42
92	2.00	192	4.43	292	7.09	392	9.96	492	12.99	592	16.18	692	19.49
94	2.05	194	4.48	294	7.15	394	10.01	494	13.05	594	16.24	694	19.56
96	2.09	196	4.53	296	7.20	396	10.07	496	13.12	596	16.31	696	19.63
98	2.14	198	4.58	298	7.26	398	10.13	498	13.18	598	16.37	698	19.69
100	2.19	200	4.64	300	7.31	400	10.19	500	13.24	600	16.44	700	19.76
*	0.022		0.025		0.027		0.029		0.031		0.032		0.033

*Average millivolts per degree F.

Chromel (+) to Alumel (−) Thermocouple

Degress F — Reference Junction, Zero F

F	0	100	200	300	400	500	600	700	800	900	1000	1100	1200
	Millivolts												
0	0.00	2.20	4.50	6.77	8.99	11.24	13.53	15.86	18.20	20.56	22.93	25.30	27.66
5	0.10	2.31	4.61	6.88	9.10	11.35	13.64	15.97	18.31	20.68	23.05	25.42	27.78
10	0.21	2.42	4.73	6.99	9.21	11.47	13.76	16.09	18.43	20.80	23.17	25.53	27.89
15	0.31	2.53	4.84	7.10	9.32	11.58	13.87	16.20	18.55	20.92	23.28	25.65	28.01
20	0.42	2.65	4.96	7.21	9.44	11.70	13.99	16.32	18.67	21.04	23.40	25.77	28.13
25	0.53	2.76	5.07	7.32	9.55	11.81	14.11	16.44	18.78	21.15	23.52	25.89	28.25
30	0.64	2.88	5.19	7.43	9.66	11.93	14.23	16.56	18.90	21.27	23.64	26.01	28.36
35	0.75	2.99	5.30	7.54	9.77	12.04	14.35	16.68	19.02	21.39	23.76	26.13	28.48
40	0.86	3.11	5.42	7.66	9.88	12.15	14.46	16.79	19.14	21.51	23.88	26.25	28.60
45	0.97	3.22	5.55	7.77	9.99	12.26	14.57	16.91	19.26	21.63	24.00	26.37	28.72
50	1.08	3.34	5.65	7.88	10.11	12.38	14.69	17.03	19.38	21.75	24.11	26.48	28.83
55	1.19	3.45	5.76	7.99	10.23	12.49	14.80	17.15	19.49	21.86	24.23	26.60	28.95
60	1.30	3.57	5.87	8.10	10.34	12.61	14.92	17.26	19.61	21.98	24.35	26.72	29.07
65	1.41	3.68	5.98	8.21	10.45	12.72	15.04	17.38	19.73	22.10	24.47	26.84	29.19
70	1.52	3.80	6.10	8.32	10.56	12.84	15.16	17.50	19.85	22.22	24.59	26.95	29.30
75	1.63	3.92	6.21	8.43	10.67	12.95	15.28	17.61	19.97	22.34	24.70	27.07	29.42
80	1.74	4.04	6.32	8.55	10.79	13.07	15.39	17.73	20.09	22.46	24.82	27.19	29.54
85	1.85	4.16	6.43	8.66	10.90	13.18	15.51	17.85	20.20	22.57	24.94	27.31	29.66
90	1.97	4.27	6.55	8.77	11.01	13.30	15.62	17.97	20.32	22.69	25.06	27.42	29.77
95	2.08	4.38	6.66	8.88	11.12	13.41	15.74	18.08	20.44	22.81	25.18	27.54	29.89
100	2.20	4.50	6.77	8.99	11.24	13.53	15.86	18.20	20.56	22.93	25.30	27.66	30.01
*	0.022	0.023	0.0227	0.0222	0.0225	0.0229	0.0233	0.0234	0.0236	0.0237	0.0237	0.0236	0.0235

F	1300	1400	1500	1600	1700	1800	1900	2000	2100	2200	2300	2400
	Millivolts											
0	30.01	32.33	34.62	36.88	39.11	41.30	43.45	45.57	47.65	49.69	51.68	53.63
5	30.13	32.45	34.74	36.99	39.22	41.41	43.56	45.68	47.76	49.79	51.78	53.73
10	30.24	32.56	34.85	37.10	39.33	41.51	43.66	45.78	47.86	49.89	51.88	53.82
15	30.36	32.68	34.97	37.22	39.44	41.62	43.77	45.88	47.96	49.99	51.98	53.92
20	30.47	32.79	35.08	37.33	39.55	41.73	43.88	45.99	48.06	50.09	52.07	54.01
25	30.59	32.91	35.19	37.44	39.66	41.84	43.99	46.09	48.17	50.19	52.17	54.11
30	30.70	33.02	35.30	37.55	39.77	41.95	44.09	46.20	48.27	50.29	52.27	54.20
35	30.82	33.14	35.42	37.67	39.88	42.06	44.20	46.30	48.37	50.39	52.37	54.30
40	30.94	33.25	35.53	37.78	39.99	42.16	44.30	46.41	48.47	50.49	52.46	54.39
45	31.06	33.37	35.65	37.89	40.10	42.27	44.41	46.51	48.57	50.59	52.56	54.49
50	31.17	33.48	35.76	38.00	40.21	42.38	44.51	46.61	48.67	50.69	52.66	54.58
55	31.29	33.60	35.87	38.11	40.32	42.49	44.62	46.72	48.78	50.79	52.76	54.68
60	31.40	33.71	35.98	38.22	40.43	42.59	44.72	46.82	48.88	50.89	52.85	54.77
65	31.52	33.83	36.10	38.33	40.54	42.70	44.83	46.93	48.98	50.99	52.95	54.87
70	31.64	33.94	36.21	38.44	40.64	42.81	44.94	47.03	49.08	51.09	53.05	54.96
75	31.76	34.06	36.32	38.56	40.75	42.92	45.05	47.14	49.19	51.19	53.15	55.06
80	31.87	34.17	36.43	38.67	40.86	43.02	45.15	47.24	49.29	51.29	53.24	55.15
85	31.99	34.28	36.55	38.78	40.97	43.13	45.26	47.34	49.39	51.39	53.34	55.25
90	32.10	34.39	36.66	38.89	41.08	43.24	45.36	47.44	49.49	51.48	53.43	55.34
95	32.22	34.51	36.77	39.00	41.19	43.35	45.47	47.55	49.59	51.58	53.53	55.44
100	32.33	34.62	36.88	39.11	41.30	43.45	45.57	47.65	49.69	51.68	53.63	55.53
*	0.0232	0.0229	0.0226	0.0223	0.0219	0.0215	0.0212	0.0208	0.0204	0.0199	0.0195	0.0190

*Average millivolts per degree F.

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Iron (+) to Constantan (−) Thermocouple

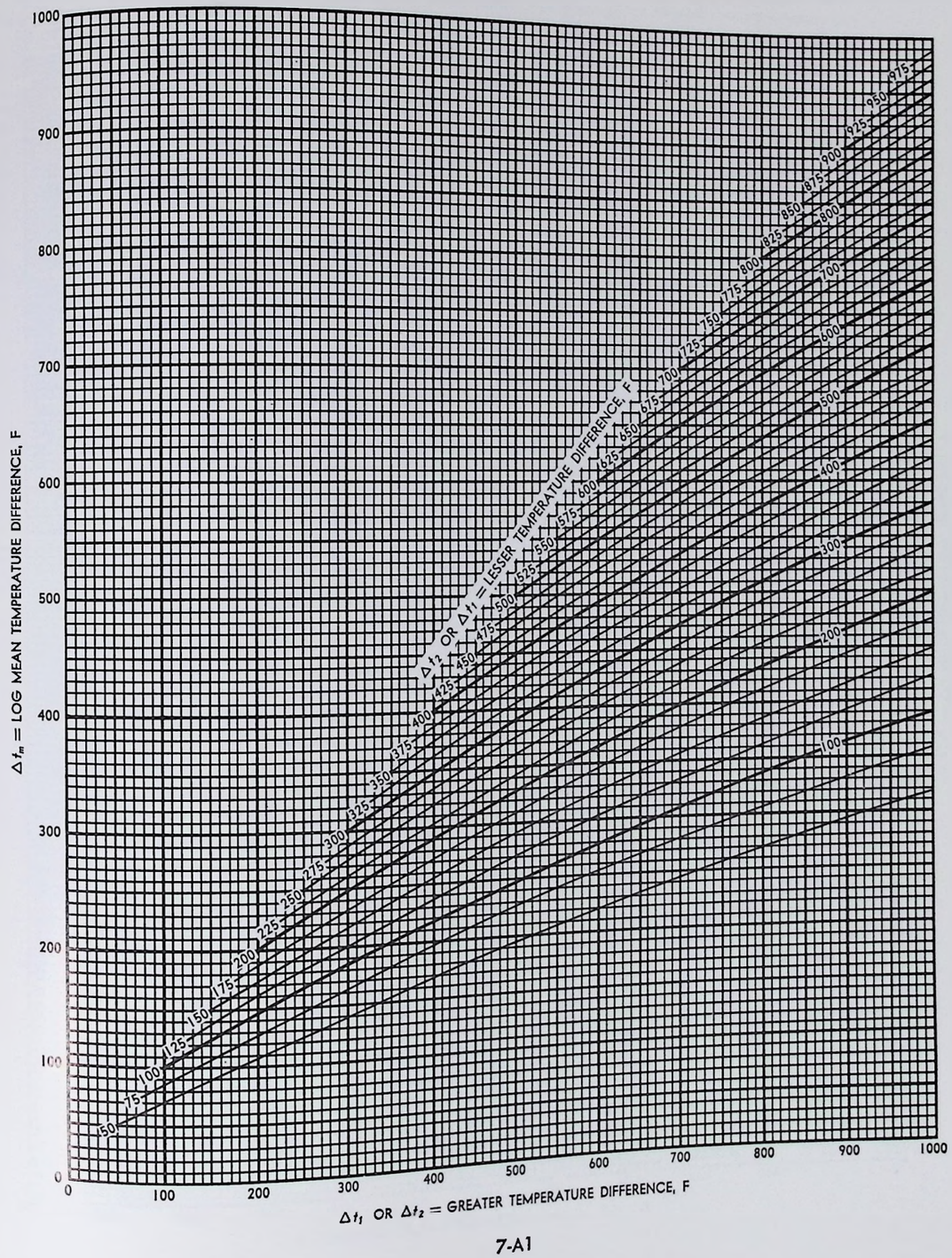
Degrees F Reference Junction, Zero F

F	0	100	200	300	400	500	600	700	800	900	1000	1100
						Millivolts						
0	0.000	2.88	5.84	8.87	11.94	15.01	18.08	21.15	24.23	27.33	30.46	33.66
5	0.144	3.03	5.99	9.02	12.09	15.16	18.23	21.30	24.38	27.49	30.62	33.82
10	0.288	3.18	6.14	9.18	12.24	15.32	18.39	21.46	24.54	27.64	30.78	33.99
15	0.432	3.32	6.29	9.33	12.40	15.47	18.54	21.61	24.69	27.80	30.94	34.15
20	0.576	3.47	6.45	9.48	12.55	15.62	18.69	21.77	24.85	27.96	31.10	34.32
25	0.720	3.62	6.60	9.64	12.71	15.78	18.85	21.92	25.00	28.11	31.26	34.48
30	0.864	3.77	6.75	9.79	12.86	15.93	19.00	22.07	25.16	28.27	31.42	34.65
35	1.01	3.92	6.90	9.94	13.01	16.08	19.15	22.23	25.31	28.42	31.58	34.81
40	1.15	4.06	7.05	10.10	13.17	16.24	19.31	22.38	25.47	28.58	31.74	34.98
45	1.30	4.21	7.20	10.25	13.32	16.39	19.46	22.54	25.62	28.74	31.90	35.14
50	1.44	4.36	7.36	10.40	13.47	16.54	19.61	22.69	25.78	28.89	32.06	35.31
55	1.58	4.51	7.50	10.56	13.63	16.70	19.77	22.84	25.93	29.05	32.22	35.47
60	1.73	4.66	7.66	10.71	13.78	16.85	19.92	23.00	26.09	29.21	32.38	35.64
65	1.87	4.80	7.81	10.86	13.94	17.00	20.07	23.15	26.24	29.36	32.54	35.80
70	2.02	4.95	7.96	11.02	14.09	17.16	20.23	23.31	26.40	29.52	32.70	35.97
75	2.16	5.10	8.11	11.17	14.24	17.31	20.38	23.46	26.55	29.68	32.86	36.13
80	2.30	5.25	8.26	11.32	14.40	17.47	20.54	23.61	26.71	29.83	33.02	36.30
85	2.45	5.40	8.42	11.47	14.55	17.62	20.69	23.77	26.86	29.99	33.18	36.46
90	2.59	5.54	8.57	11.63	14.70	17.77	20.84	23.92	27.02	30.15	33.34	36.63
95	2.74	5.69	8.72	11.78	14.86	17.93	21.00	24.08	27.17	30.30	33.50	36.79
100	2.88	5.84	8.87	11.94	15.01	18.08	21.15	24.23	27.33	30.46	33.66	36.96
*	0.0288	0.0296	0.0303	0.0307	0.0307	0.0307	0.0307	0.0308	0.0310	0.0313	0.0320	0.0330

F	1200	1300	1400	1500	1600	1700	1800	1900	2000	2100
					Millivolts					
0	36.96	40.38	43.88	47.40	50.92	54.44	57.96	61.48	65.00	68.52
5	37.13	40.56	44.06	47.58	51.10	54.62	58.14	61.66	65.18	68.70
10	37.30	40.73	44.23	47.75	51.27	54.79	58.31	61.83	65.35	68.87
15	37.47	40.90	44.41	47.93	51.45	54.97	58.49	62.01	65.53	69.05
20	37.64	41.08	44.58	48.10	51.62	55.14	58.66	62.18	65.70	69.22
25	37.81	41.26	44.76	48.28	51.80	55.32	58.84	62.36	65.88	69.40
30	37.99	41.43	44.94	48.46	51.98	55.50	59.02	62.54	66.06	69.58
35	38.16	41.60	45.11	48.63	52.15	55.67	59.19	62.71	66.23	69.75
40	38.33	41.78	45.29	48.81	52.33	55.85	59.37	62.89	66.41	69.93
45	38.50	41.95	45.46	48.98	52.50	56.02	59.54	63.06	66.58	70.10
50	38.67	42.13	45.64	49.16	52.68	56.20	59.72	63.24	66.76	70.28
55	38.84	42.30	45.82	49.34	52.86	56.38	59.90	63.42	66.94	70.46
60	39.01	42.48	45.99	49.51	53.03	56.55	60.07	63.59	67.11	70.63
65	39.18	42.65	46.17	49.69	53.21	56.73	60.25	63.77	67.29	70.81
70	39.35	42.83	46.34	49.86	53.38	56.90	60.42	63.94	67.46	70.98
75	39.52	43.00	46.52	50.04	53.56	57.08	60.60	64.12	67.64	71.16
80	39.70	43.18	46.70	50.22	53.74	57.26	60.78	64.30	67.82	71.34
85	39.87	43.35	46.87	50.39	53.91	57.43	60.95	64.47	67.99	71.51
90	40.04	43.53	47.05	50.57	54.09	57.61	61.13	64.65	68.17	71.69
95	40.21	43.70	47.22	50.74	54.26	57.78	61.30	64.82	68.34	71.86
100	40.38	43.88	47.40	50.92	54.44	57.96	61.48	65.00	68.52	72.04
*	0.0342	0.0350	0.0352	0.0352	0.0352	0.0352	0.0352	0.0352	0.0352	0.0352

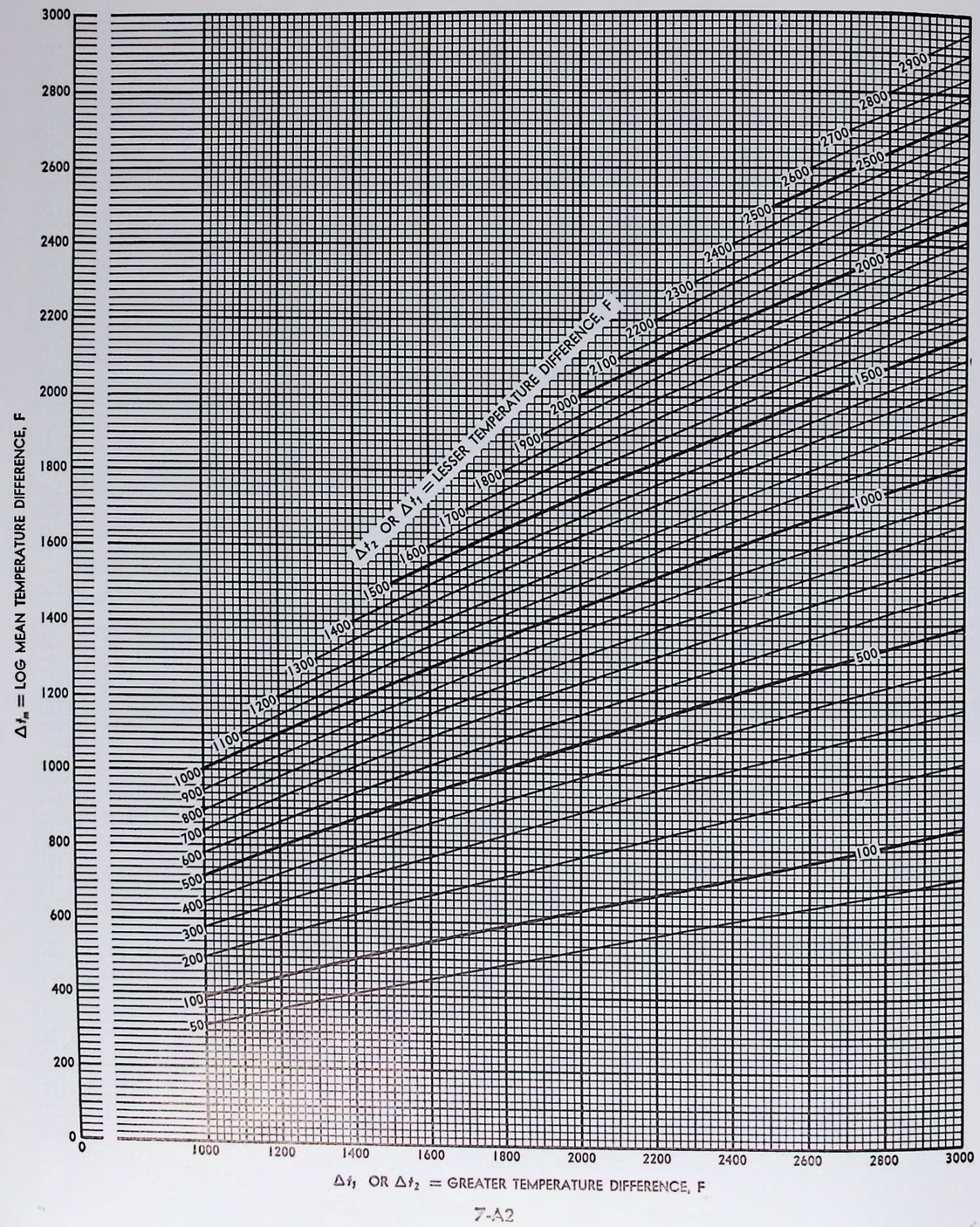
*Average millivolts per degree F.

LOGARITHMIC MEAN TEMPERATURE DIFFERENCE



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LOGARITHMIC MEAN TEMPERATURE DIFFERENCE



7-A2

Chapter 10. Steam and Other Energy Cycles

Note: The following steam tables and Mollier chart have been abstracted from *Thermodynamic Properties of Steam* (Copyright, 1936, by Keenan and Keyes) published by John Wiley & Sons, Inc. Reference to this book is necessary for values outside the range of the abstract and for other data.

TABLE 1. SATURATION: TEMPERATURES

Temp F t	Abs Press. Lb/Sq In. p	Specific Volume Sat. Liquid v_f	Specific Volume Evap v_{fg}	Specific Volume Sat. Vapor v_g	Enthalpy Sat. Liquid h_f	Enthalpy Evap h_{fg}	Enthalpy Sat. Vapor h_g	Entropy Sat. Liquid s_f	Entropy Evap s_{fg}	Entropy Sat. Vapor s_g	Temp F t
32	0.08854	0.01602	3306	3306	0.00	1075.8	1075.8	0.0000	2.1877	2.1877	32
35	0.09995	0.01602	2947	2947	3.02	1074.1	1077.1	0.0061	2.1709	2.1770	35
40	0.12170	0.01602	2444	2444	8.05	1071.3	1079.3	0.0162	2.1435	2.1597	40
45	0.14752	0.01602	2036.4	2036.4	13.06	1068.4	1081.5	0.0262	2.1167	2.1429	45
50	0.17811	0.01603	1703.2	1703.2	18.07	1065.6	1083.7	0.0361	2.0903	2.1264	50
60	0.2563	0.01604	1206.6	1206.7	28.06	1059.9	1088.0	0.0555	2.0393	2.0948	60
70	0.3631	0.01606	867.8	867.9	38.04	1054.3	1092.3	0.0745	1.9902	2.0647	70
80	0.5069	0.01608	633.1	633.1	48.02	1048.6	1096.6	0.0932	1.9428	2.0360	80
90	0.6982	0.01610	468.0	468.0	57.99	1042.9	1100.9	0.1115	1.8972	2.0087	90
100	0.9492	0.01613	350.3	350.4	67.97	1037.2	1105.2	0.1295	1.8531	1.9826	100
110	1.2748	0.01617	265.3	265.4	77.94	1031.6	1109.5	0.1471	1.8106	1.9577	110
120	1.6924	0.01620	203.25	203.27	87.92	1025.8	1113.7	0.1645	1.7694	1.9339	120
130	2.2225	0.01625	157.32	157.34	97.90	1020.0	1117.9	0.1816	1.7296	1.9112	130
140	2.8886	0.01629	122.99	123.01	107.89	1014.1	1122.0	0.1984	1.6910	1.8894	140
150	3.718	0.01634	97.06	97.07	117.89	1008.2	1126.1	0.2149	1.6537	1.8685	150
160	4.741	0.01639	77.27	77.29	127.89	1002.3	1130.2	0.2311	1.6174	1.8485	160
170	5.992	0.01645	62.04	62.06	137.90	996.3	1134.2	0.2472	1.5822	1.8293	170
180	7.510	0.01651	50.21	50.23	147.92	990.2	1138.1	0.2630	1.5480	1.8109	180
190	9.339	0.01657	40.94	40.96	157.95	984.1	1142.0	0.2785	1.5147	1.7932	190
200	11.526	0.01663	33.62	33.64	167.99	977.9	1145.9	0.2938	1.4824	1.7762	200
210	14.123	0.01670	27.80	27.82	178.05	971.6	1149.7	0.3090	1.4508	1.7598	210
212	14.696	0.01672	26.78	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566	212
220	17.186	0.01677	23.13	23.15	188.13	965.2	1153.4	0.3239	1.4201	1.7440	220
230	20.780	0.01684	19.365	19.382	198.23	958.8	1157.0	0.3387	1.3901	1.7288	230
240	24.969	0.01692	16.306	16.323	208.34	952.2	1160.5	0.3531	1.3609	1.7140	240
250	29.825	0.01700	13.804	13.821	218.48	945.5	1164.0	0.3675	1.3323	1.6998	250
260	35.429	0.01709	11.746	11.763	228.64	938.7	1167.3	0.3817	1.3043	1.6860	260
270	41.858	0.01717	10.044	10.061	238.84	931.8	1170.6	0.3958	1.2769	1.6727	270
280	49.203	0.01726	8.628	8.645	249.06	924.7	1173.8	0.4096	1.2501	1.6597	280
290	57.556	0.01735	7.444	7.461	259.31	917.5	1176.8	0.4234	1.2238	1.6472	290
300	67.013	0.01745	6.449	6.466	269.59	910.1	1179.7	0.4369	1.1980	1.6350	300
320	89.66	0.01765	4.896	4.914	290.28	894.9	1185.2	0.4637	1.1478	1.6115	320
340	118.01	0.01787	3.770	3.788	311.13	879.0	1190.1	0.4900	1.0992	1.5891	340
360	153.04	0.01811	2.939	2.957	332.18	862.2	1194.4	0.5158	1.0519	1.5677	360
380	195.77	0.01836	2.317	2.335	353.45	844.6	1198.1	0.5413	1.0059	1.5471	380
400	247.31	0.01864	1.8447	1.8633	374.97	826.0	1201.0	0.5664	0.9608	1.5272	400
420	308.83	0.01894	1.4811	1.5000	396.77	806.3	1203.1	0.5912	0.9166	1.5078	420
440	381.59	0.01926	1.1979	1.2171	418.90	785.4	1204.3	0.6158	0.8730	1.4887	440
460	466.9	0.0196	0.9748	0.9944	441.4	763.2	1204.6	0.6402	0.8298	1.4700	460
480	566.1	0.0200	0.7972	0.8172	464.4	739.4	1203.7	0.6645	0.7868	1.4513	480
500	680.8	0.0204	0.6545	0.6749	487.8	713.9	1201.7	0.6887	0.7438	1.4325	500
520	812.4	0.0209	0.5385	0.5594	511.9	686.4	1198.2	0.7130	0.7006	1.4136	520
540	962.5	0.0215	0.4434	0.4649	536.6	656.6	1193.2	0.7374	0.6568	1.3942	540
560	1133.1	0.0221	0.3647	0.3868	562.2	624.2	1186.4	0.7621	0.6121	1.3742	560
580	1325.8	0.0228	0.2989	0.3217	588.9	588.4	1177.3	0.7872	0.5659	1.3532	580
600	1542.9	0.0236	0.2432	0.2668	617.0	548.5	1165.5	0.8131	0.5176	1.3307	600
620	1786.6	0.0247	0.1955	0.2201	646.7	503.6	1150.3	0.8398	0.4664	1.3062	620
640	2059.7	0.0260	0.1538	0.1798	678.6	452.0	1130.5	0.8679	0.4110	1.2789	640
660	2365.4	0.0278	0.1165	0.1442	714.2	390.2	1104.4	0.8987	0.3485	1.2472	660
680	2708.1	0.0305	0.0810	0.1115	757.3	309.9	1067.2	0.9351	0.2719	1.2071	680
700	3093.7	0.0369	0.0392	0.0761	823.3	172.1	995.4	0.9903	0.1484	1.1389	700
705.4	3206.2	0.0503	0	0.0503	902.7	0	902.7	1.0580	0	1.0580	705.4

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TABLE 2. SATURATION: PRESSURES

Abs Press. Lb/Sq In. p	Temp F t	Specific Volume Sat. Liquid v_f	Specific Volume Sat. Vapor v_g	Enthalpy Sat. Liquid h_f	Enthalpy Evap h_{fg}	Enthalpy Sat. Vapor h_g	Entropy Sat. Liquid s_f	Entropy Evap s_{fg}	Entropy Sat. Vapor s_g	Internal Energy Sat. Liquid u_f	Internal Energy Evap u_{fg}	Internal Energy Sat. Vapor u_g	Abs Press. Lb/Sq In. p
1.0	101.74	0.01614	333.6	69.70	1036.3	1106.0	0.1326	1.8456	1.9782	69.70	974.6	1044.3	1.0
2.0	126.08	0.01623	173.73	93.99	1022.2	1116.2	0.1749	1.7451	1.9200	93.98	957.9	1051.9	2.0
3.0	141.48	0.01630	118.71	109.37	1013.2	1122.6	0.2008	1.6855	1.8863	109.36	947.3	1056.7	3.0
4.0	152.97	0.01636	90.63	120.86	1006.4	1127.3	0.2198	1.6427	1.8625	120.85	939.3	1060.2	4.0
5.0	162.24	0.01640	73.52	130.13	1001.0	1131.1	0.2347	1.6094	1.8441	130.12	933.0	1063.1	5.0
6.0	170.06	0.01645	61.98	137.96	996.2	1134.2	0.2472	1.5820	1.8292	137.94	927.5	1065.4	6.0
7.0	176.85	0.01649	53.64	144.76	992.1	1136.9	0.2581	1.5586	1.8167	144.74	922.7	1067.4	7.0
8.0	182.86	0.01653	47.34	150.79	988.5	1139.3	0.2674	1.5383	1.8057	150.77	918.4	1069.2	8.0
9.0	188.28	0.01656	42.40	156.22	985.2	1141.4	0.2759	1.5203	1.7962	156.19	914.6	1070.8	9.0
10	193.21	0.01659	38.42	161.17	982.1	1143.3	0.2835	1.5041	1.7876	161.14	911.1	1072.2	10
14.696	212.00	0.01672	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566	180.02	897.5	1077.5	14.696
15	213.03	0.01672	26.29	181.11	969.7	1150.8	0.3135	1.4415	1.7549	181.06	896.7	1077.8	15
20	227.96	0.01683	20.089	196.16	960.1	1156.3	0.3356	1.3962	1.7319	196.10	885.8	1081.9	20
30	250.33	0.01701	13.746	218.82	945.3	1164.1	0.3680	1.3313	1.6993	218.73	869.1	1087.8	30
40	267.25	0.01715	10.498	236.03	933.7	1169.7	0.3919	1.2844	1.6763	235.90	856.1	1092.0	40
50	281.01	0.01727	8.515	250.09	924.0	1174.1	0.4110	1.2474	1.6585	249.93	845.4	1095.3	50
60	292.71	0.01738	7.175	262.09	915.5	1177.6	0.4270	1.2168	1.6438	261.90	836.0	1097.9	60
70	302.92	0.01748	6.206	272.61	907.9	1180.6	0.4409	1.1906	1.6315	272.38	827.8	1100.2	70
80	312.03	0.01757	5.472	282.02	901.1	1183.1	0.4531	1.1676	1.6207	281.76	820.3	1102.1	80
90	320.27	0.01766	4.896	290.56	894.7	1185.3	0.4641	1.1471	1.6112	290.27	813.4	1103.7	90
100	327.81	0.01774	4.432	298.40	888.8	1187.2	0.4740	1.1286	1.6026	298.08	807.1	1105.2	100
120	341.25	0.01789	3.728	312.44	877.9	1190.4	0.4916	1.0962	1.5878	312.05	795.6	1107.6	120
140	353.02	0.01802	3.220	324.82	868.2	1193.0	0.5069	1.0682	1.5751	324.35	785.2	1109.6	140
160	363.53	0.01815	2.834	335.93	859.2	1195.1	0.5204	1.0436	1.5640	335.39	775.8	1111.2	160
180	373.06	0.01827	2.532	346.03	850.8	1196.9	0.5325	1.0217	1.5542	345.42	767.1	1112.5	180
200	381.79	0.01839	2.288	355.36	843.0	1198.4	0.5435	1.0018	1.5453	354.68	759.0	1113.7	200
250	400.95	0.01865	1.8438	376.00	825.1	1201.1	0.5675	0.9588	1.5263	375.14	740.7	1115.8	250
300	417.33	0.01890	1.5433	393.84	809.0	1202.8	0.5879	0.9225	1.5104	392.79	724.3	1117.1	300
350	431.72	0.01913	1.3260	409.69	794.2	1203.9	0.6056	0.8910	1.4966	408.45	709.6	1118.0	350
400	444.59	0.0193	1.1613	424.0	780.5	1204.5	0.6214	0.8630	1.4844	422.6	695.9	1118.5	400
450	456.28	0.0195	1.0320	437.2	767.4	1204.6	0.6356	0.8378	1.4734	435.5	683.2	1118.7	450
500	467.01	0.0197	0.9278	449.4	755.0	1204.4	0.6487	0.8147	1.4634	447.6	671.0	1118.6	500
550	476.93	0.0199	0.8422	460.8	743.1	1203.9	0.6608	0.7934	1.4542	458.8	659.4	1118.2	550
600	486.21	0.0201	0.7698	471.6	731.6	1203.2	0.6720	0.7734	1.4454	469.4	648.3	1117.7	600
700	503.10	0.0205	0.6554	491.5	709.7	1201.2	0.6925	0.7371	1.4296	488.8	627.5	1116.3	700
800	518.23	0.0209	0.5687	509.7	688.9	1198.6	0.7108	0.7045	1.4153	506.6	607.8	1114.4	800
900	531.98	0.0212	0.5006	526.6	668.8	1195.4	0.7275	0.6744	1.4020	523.1	589.0	1112.1	900
1000	544.61	0.0216	0.4456	542.4	649.4	1191.8	0.7430	0.6467	1.3897	538.4	571.0	1109.4	1000
1100	556.31	0.0220	0.4001	557.4	630.4	1187.8	0.7575	0.6205	1.3780	552.9	553.5	1106.4	1100
1200	567.22	0.0223	0.3619	571.7	611.7	1183.4	0.7711	0.5956	1.3667	566.7	536.3	1103.0	1200
1300	577.46	0.0227	0.3293	585.4	593.2	1178.6	0.7840	0.5719	1.3559	580.0	519.4	1099.4	1300
1400	587.10	0.0231	0.3012	598.7	574.7	1173.4	0.7963	0.5491	1.3454	592.7	502.7	1095.4	1400
1500	596.23	0.0235	0.2765	611.6	556.3	1167.9	0.8082	0.5269	1.3351	605.1	486.1	1091.2	1500
2000	635.82	0.0257	0.1878	671.7	463.4	1135.1	0.8619	0.4230	1.2849	662.2	403.4	1065.6	2000
2500	668.13	0.0287	0.1307	730.6	360.5	1091.1	0.9126	0.3197	1.2322	717.3	313.3	1030.6	2500
3000	695.36	0.0346	0.0858	802.5	217.8	1020.3	0.9731	0.1885	1.1615	783.4	189.3	972.7	3000
3206.2	705.40	0.0503	0.0503	902.7	0	902.7	1.0580	0	1.0580	872.9	0	872.9	3206.2

TABLE 3. SUPERHEATED VAPOR

Abs Press. Lb/Sq In. (Sat. Temp)		Temperature, F 200	300	400	500	600	700	800	900	1000	1200	1400	1600
1	v	392.6	452.3	512.0	571.6	631.2	690.8	750.4	809.9	869.5	988.7	1107.8	1227.0
	h	1150.4	1195.8	1241.7	1288.3	1335.7	1383.8	1432.8	1482.7	1533.5	1637.7	1745.7	1857.5
(101.74)	s	2.0512	2.1153	2.1720	2.2233	2.2702	2.3137	2.3542	2.3923	2.4283	2.4952	2.5566	2.6137
5	v	78.16	90.25	102.26	114.22	126.16	138.10	150.03	161.95	173.87	197.71	221.6	245.4
	h	1148.8	1195.0	1241.2	1288.0	1335.4	1383.6	1432.7	1482.6	1533.4	1637.7	1745.7	1857.4
(162.24)	s	1.8718	1.9370	1.9942	2.0456	2.0927	2.1361	2.1767	2.2148	2.2509	2.3178	2.3792	2.4363
10	v	38.85	45.00	51.04	57.05	63.03	69.01	74.98	80.95	86.92	98.84	110.77	122.69
	h	1146.6	1193.9	1240.6	1287.5	1335.1	1383.4	1432.5	1482.4	1533.2	1637.6	1745.6	1857.3
(193.21)	s	1.7927	1.8595	1.9172	1.9689	2.0160	2.0596	2.1002	2.1383	2.1744	2.2413	2.3028	2.3598
14.696	v		30.53	34.68	38.78	42.86	46.94	51.00	55.07	59.13	67.25	75.37	83.48
	h		1192.8	1239.9	1287.1	1334.8	1383.2	1432.3	1482.3	1533.1	1637.5	1745.5	1857.3
(212.00)	s		1.8160	1.8743	1.9261	1.9734	2.0170	2.0576	2.0958	2.1319	2.1989	2.2603	2.3174
20	v		22.36	25.43	28.46	31.47	34.47	37.46	40.45	43.44	49.41	55.37	61.34
	h		1191.6	1239.2	1286.6	1334.4	1382.9	1432.1	1482.1	1533.0	1637.4	1745.4	1857.2
(227.96)	s		1.7808	1.8396	1.8918	1.9392	1.9829	2.0235	2.0618	2.0978	2.1648	2.2263	2.2834
40	v		11.040	12.628	14.168	15.688	17.198	18.702	20.20	21.70	24.69	27.68	30.66
	h		1186.8	1236.5	1284.8	1333.1	1381.9	1431.3	1481.4	1532.4	1637.0	1745.1	1857.0
(267.25)	s		1.6994	1.7608	1.8140	1.8619	1.9058	1.9467	1.9850	2.0212	2.0883	2.1498	2.2069
60	v		7.259	8.357	9.403	10.427	11.441	12.449	13.452	14.454	16.451	18.446	20.44
	h		1181.6	1233.6	1283.0	1331.8	1380.9	1430.5	1480.8	1531.9	1636.6	1744.8	1856.7
(292.71)	s		1.6492	1.7135	1.7678	1.8162	1.8605	1.9015	1.9400	1.9762	2.0434	2.1049	2.1621
80	v			6.220	7.020	7.797	8.562	9.322	10.077	10.830	12.332	13.830	15.325
	h			1230.7	1281.1	1330.5	1379.9	1429.7	1480.1	1531.3	1636.2	1744.5	1856.5
(312.03)	s			1.6791	1.7346	1.7836	1.8281	1.8694	1.9079	1.9442	2.0115	2.0731	2.1303
100	v			4.937	5.589	6.218	6.835	7.446	8.052	8.656	9.860	11.060	12.258
	h			1227.6	1279.1	1329.1	1378.9	1428.9	1479.5	1530.8	1635.7	1744.2	1856.2
(327.81)	s			1.6518	1.7085	1.7581	1.8029	1.8443	1.8829	1.9193	1.9867	2.0484	2.1056
120	v			4.081	4.636	5.165	5.683	6.195	6.702	7.207	8.212	9.214	10.213
	h			1224.4	1277.2	1327.7	1377.8	1428.1	1478.8	1530.2	1635.3	1743.9	1856.0
(341.25)	s			1.6287	1.6869	1.7370	1.7822	1.8237	1.8625	1.8990	1.9664	2.0281	2.0854
140	v			3.468	3.954	4.413	4.861	5.301	5.738	6.172	7.035	7.895	8.752
	h			1221.1	1275.2	1326.4	1376.8	1427.3	1478.2	1529.7	1634.9	1743.5	1855.7
(353.02)	s			1.6087	1.6683	1.7190	1.7645	1.8063	1.8451	1.8817	1.9493	2.0110	2.0683
160	v			3.008	3.443	3.849	4.244	4.631	5.015	5.396	6.152	6.906	7.656
	h			1217.6	1273.1	1325.0	1375.7	1426.4	1477.5	1529.1	1634.5	1743.2	1855.5
(363.53)	s			1.5908	1.6519	1.7033	1.7491	1.7911	1.8301	1.8667	1.9344	1.9962	2.0535
180	v			2.649	3.044	3.411	3.764	4.110	4.452	4.792	5.466	6.136	6.804
	h			1214.0	1271.0	1323.5	1374.7	1425.6	1476.8	1528.6	1634.1	1742.9	1855.2
(373.06)	s			1.5745	1.6373	1.6894	1.7355	1.7776	1.8167	1.8534	1.9212	1.9831	2.0404
200	v			2.361	2.726	3.060	3.380	3.693	4.002	4.309	4.917	5.521	6.123
	h			1210.3	1268.9	1322.1	1373.6	1424.8	1476.2	1528.0	1633.7	1742.6	1855.0
(381.79)	s			1.5594	1.6240	1.6767	1.7232	1.7655	1.8048	1.8415	1.9094	1.9713	2.0287
220	v			2.125	2.465	2.772	3.066	3.352	3.634	3.913	4.467	5.017	5.565
	h			1206.5	1266.7	1320.7	1372.6	1424.0	1475.5	1527.5	1633.3	1742.3	1854.7
(389.86)	s			1.5453	1.6117	1.6652	1.7120	1.7545	1.7939	1.8308	1.8987	1.9607	2.0181
240	v			1.9276	2.247	2.533	2.804	3.068	3.327	3.584	4.093	4.597	5.100
	h			1202.5	1264.5	1319.2	1371.5	1423.2	1474.8	1526.9	1632.9	1742.0	1854.5
(397.37)	s			1.5319	1.6003	1.6546	1.7017	1.7444	1.7839	1.8209	1.8889	1.9510	2.0084

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TABLE 3. SUPERHEATED VAPOR
(Continued)

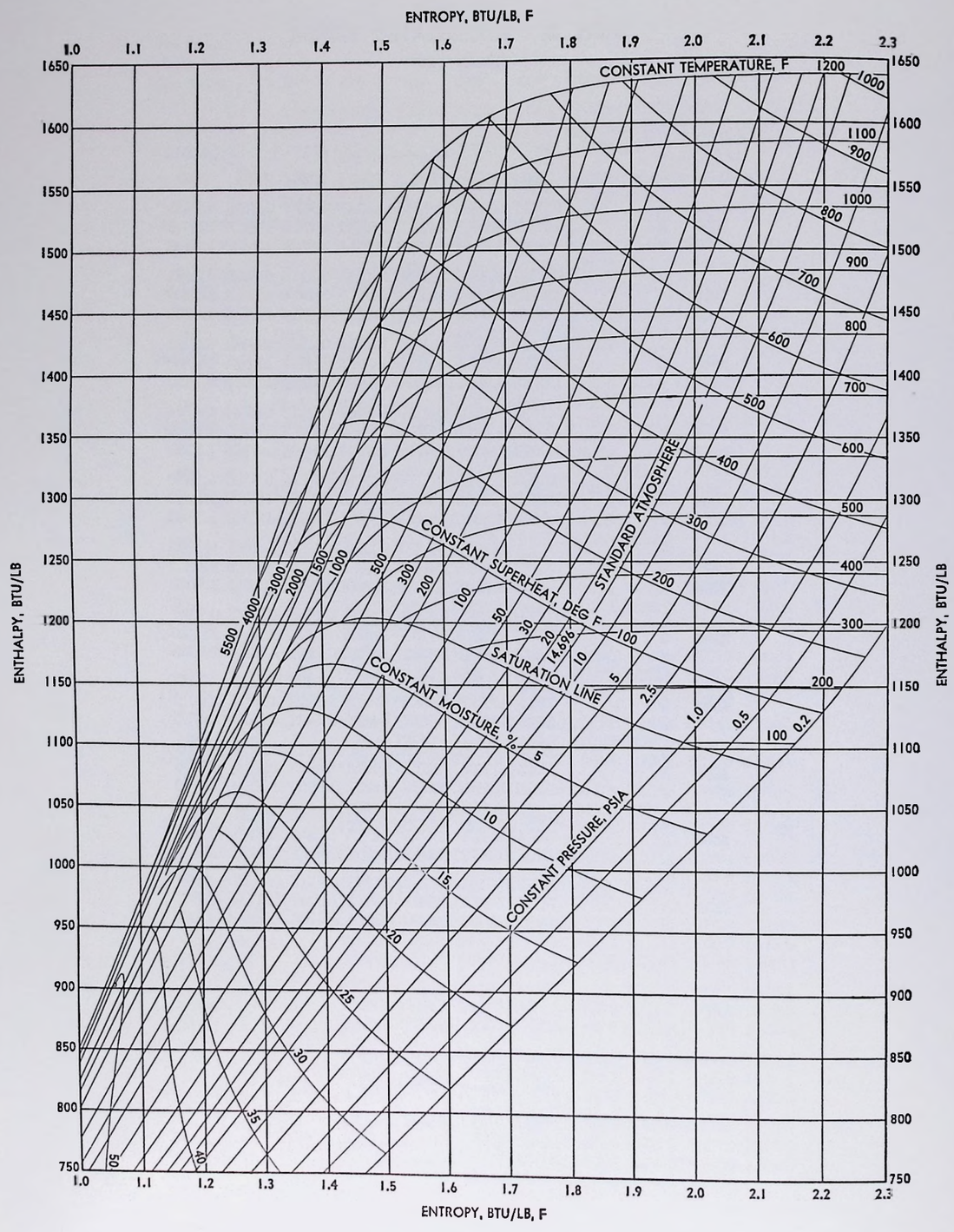
Abs Press. Lb/Sq In. (Sat. Temp)		Temperature, F 500	600	700	800	900	1000	1200	1400	1600
	v	2.063	2.330	2.582	2.827	3.067	3.305	3.776	4.242	4.707
260	h	1262.3	1317.7	1370.4	1422.3	1474.2	1526.3	1632.5	1741.7	1854.2
(404.42)	s	1.5897	1.6447	1.6922	1.7352	1.7748	1.8118	1.8799	1.9420	1.9995
	v	1.9047	2.156	2.392	2.621	2.845	3.066	3.504	3.938	4.370
280	h	1260.0	1316.2	1369.4	1421.5	1473.5	1525.8	1632.1	1741.4	1854.0
(411.05)	s	1.5796	1.6354	1.6834	1.7265	1.7662	1.8033	1.8716	1.9337	1.9912
	v	1.7675	2.005	2.227	2.442	2.652	2.859	3.269	3.674	4.078
300	h	1257.6	1314.7	1368.3	1420.6	1472.8	1525.2	1631.7	1741.0	1853.7
(417.33)	s	1.5701	1.6268	1.6751	1.7184	1.7582	1.7954	1.8638	1.9260	1.9835
	v	1.4923	1.7036	1.8980	2.084	2.266	2.445	2.798	3.147	3.493
350	h	1251.5	1310.9	1365.5	1418.5	1471.1	1523.8	1630.7	1740.3	1853.1
(431.72)	s	1.5481	1.6070	1.6563	1.7002	1.7403	1.7777	1.8463	1.9086	1.9663
	v	1.2851	1.4770	1.6508	1.8161	1.9767	2.134	2.445	2.751	3.055
400	h	1245.1	1306.9	1362.7	1416.4	1469.4	1522.4	1629.6	1739.5	1852.5
(444.59)	s	1.5281	1.5894	1.6398	1.6842	1.7247	1.7623	1.8311	1.8936	1.9513
	v	1.1231	1.3005	1.4584	1.6074	1.7516	1.8928	2.170	2.443	2.714
450	h	1238.4	1302.8	1359.9	1414.3	1467.7	1521.0	1628.6	1738.7	1851.9
(456.28)	s	1.5095	1.5735	1.6250	1.6699	1.7108	1.7486	1.8177	1.8803	1.9381
	v	0.9927	1.1591	1.3044	1.4405	1.5715	1.6996	1.9504	2.197	2.442
500	h	1231.3	1298.6	1357.0	1412.1	1466.0	1519.6	1627.6	1737.9	1851.3
(467.01)	s	1.4919	1.5588	1.6115	1.6571	1.6982	1.7363	1.8056	1.8683	1.9262
	v	0.8852	1.0431	1.1783	1.3038	1.4241	1.5414	1.7706	1.9957	2.219
550	h	1223.7	1294.3	1354.0	1409.9	1464.3	1518.2	1626.6	1737.1	1850.6
(476.94)	s	1.4751	1.5451	1.5991	1.6452	1.6868	1.7250	1.7946	1.8575	1.9155
	v	0.7947	0.9463	1.0732	1.1899	1.3013	1.4096	1.6208	1.8279	2.033
600	h	1215.7	1289.9	1351.1	1407.7	1462.5	1516.7	1625.5	1736.3	1850.0
(486.21)	s	1.4586	1.5323	1.5875	1.6343	1.6762	1.7147	1.7846	1.8476	1.9056
	v		0.7934	0.9077	1.0108	1.1082	1.2024	1.3853	1.5641	1.7405
700	h		1280.6	1345.0	1403.2	1459.0	1513.9	1623.5	1734.8	1848.8
(503.10)	s		1.5084	1.5665	1.6147	1.6573	1.6963	1.7666	1.8299	1.8881
	v		0.6779	0.7833	0.8763	0.9633	1.0470	1.2088	1.3662	1.5214
800	h		1270.7	1338.6	1398.6	1455.4	1511.0	1621.4	1733.2	1847.5
(518.23)	s		1.4863	1.5476	1.5972	1.6407	1.6801	1.7510	1.8146	1.8729
	v		0.5873	0.6863	0.7716	0.8506	0.9262	1.0714	1.2124	1.3509
900	h		1260.1	1332.1	1393.9	1451.8	1508.1	1619.3	1731.6	1846.3
(531.98)	s		1.4653	1.5303	1.5814	1.6257	1.6656	1.7371	1.8009	1.8595
	v		0.5140	0.6084	0.6878	0.7604	0.8294	0.9615	1.0893	1.2146
1000	h		1248.8	1325.3	1389.2	1448.2	1505.1	1617.3	1730.0	1845.0
(544.61)	s		1.4450	1.5141	1.5670	1.6121	1.6525	1.7245	1.7886	1.8474
	v		0.4532	0.5445	0.6191	0.6866	0.7503	0.8716	0.9885	1.1031
1100	h		1236.7	1318.3	1384.3	1444.5	1502.2	1615.2	1728.4	1843.8
(556.31)	s		1.4251	1.4989	1.5535	1.5995	1.6405	1.7130	1.7775	1.8363
	v		0.4016	0.4909	0.5617	0.6250	0.6843	0.7967	0.9046	1.0101
1200	h		1223.5	1311.0	1379.3	1440.7	1499.2	1613.1	1726.9	1842.5
(567.22)	s		1.4052	1.4843	1.5409	1.5879	1.6293	1.7025	1.7672	1.8263
	v		0.3174	0.4062	0.4714	0.5281	0.5805	0.6789	0.7727	0.8640
1400	h		1193.0	1295.5	1369.1	1433.1	1493.2	1608.9	1723.7	1840.0
(587.10)	s		1.3639	1.4567	1.5177	1.5666	1.6093	1.6836	1.7489	1.8083

TABLE 3. SUPERHEATED VAPOR
(Concluded)

Abs Press. Lb/Sq In. (Sat. Temp)		Temperature, F 700	800	900	1000	1200	1400	1600
	v	0.3417	0.4034	0.4553	0.5027	0.5906	0.6738	0.7545
1600	h	1278.7	1358.4	1425.3	1487.0	1604.6	1720.5	1837.5
(604.90)	s	1.4303	1.4964	1.5476	1.5914	1.6669	1.7328	1.7926
	v	0.2907	0.3502	0.3986	0.4421	0.5218	0.5968	0.6693
1800	h	1260.3	1347.2	1417.4	1480.8	1600.4	1717.3	1835.0
(621.03)	s	1.4044	1.4765	1.5301	1.5752	1.6520	1.7185	1.7786
	v	0.2489	0.3074	0.3532	0.3935	0.4668	0.5352	0.6011
2000	h	1240.0	1335.5	1409.2	1474.5	1596.1	1714.1	1832.5
(635.82)	s	1.3783	1.4576	1.5139	1.5603	1.6384	1.7055	1.7660
	v	0.1686	0.2294	0.2710	0.3061	0.3678	0.4244	0.4784
2500	h	1176.8	1303.6	1387.8	1458.4	1585.3	1706.1	1826.2
(668.13)	s	1.3073	1.4127	1.4772	1.5273	1.6088	1.6775	1.7389
	v	0.0984	0.1760	0.2159	0.2476	0.3018	0.3505	0.3966
3000	h	1060.7	1267.2	1365.0	1441.8	1574.3	1698.0	1819.9
(695.36)	s	1.1966	1.3690	1.4439	1.4984	1.5837	1.6540	1.7163
	v		0.1583	0.1981	0.2288	0.2806	0.3267	0.3703
3206.2	h		1250.5	1355.2	1434.7	1569.8	1694.6	1817.2
(705.40)	s		1.3508	1.4309	1.4874	1.5742	1.6452	1.7080
	v		0.0306	0.1364	0.1762	0.2058	0.2546	0.2977	0.3381
3500	h		780.5	1224.9	1340.7	1424.5	1563.3	1689.8	1813.6
	s		0.9515	1.3241	1.4127	1.4723	1.5615	1.6336	1.6968
	v		0.0287	0.1052	0.1462	0.1743	0.2192	0.2581	0.2943
4000	h		763.8	1174.8	1314.4	1406.8	1552.1	1681.7	1807.2
	s		0.9347	1.2757	1.3827	1.4482	1.5417	1.6154	1.6795
	v		0.0276	0.0798	0.1226	0.1500	0.1917	0.2273	0.2602
4500	h		753.5	1113.9	1286.5	1388.4	1540.8	1673.5	1800.9
	s		0.9235	1.2204	1.3529	1.4253	1.5235	1.5990	1.6640
	v		0.0268	0.0593	0.1036	0.1303	0.1696	0.2027	0.2329
5000	h		746.4	1047.1	1256.5	1369.5	1529.5	1665.3	1794.5
	s		0.9152	1.1622	1.3231	1.4034	1.5066	1.5839	1.6499
	v		0.0262	0.0463	0.0880	0.1143	0.1516	0.1825	0.2106
5500	h		741.3	985.0	1224.1	1349.3	1518.2	1657.0	1788.1
	s		0.9090	1.1093	1.2930	1.3821	1.4908	1.5699	1.6369

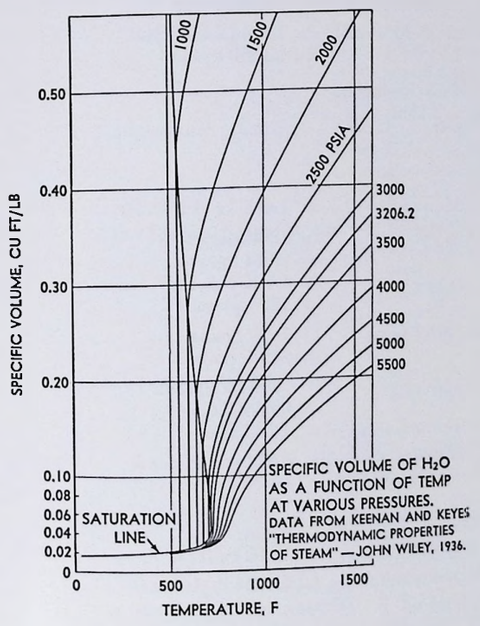
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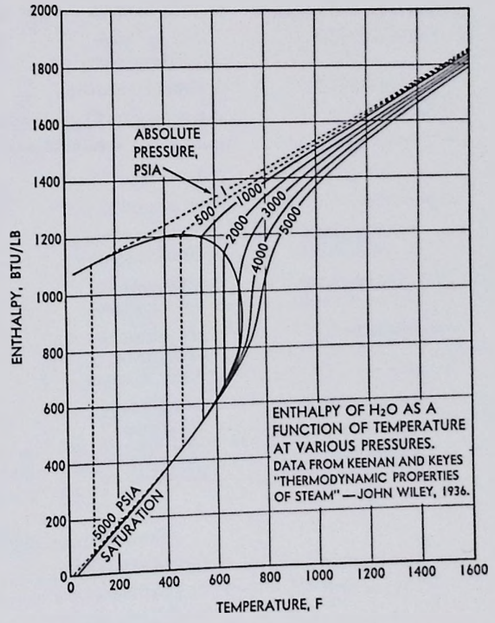
MOLLIER CHART



10-A6

Some Thermodynamic Properties of H_2O in the Subcritical and Supercritical Regions





10-A7

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Tables of Measurement

Length and Area

1 statute mile	= 1760 yards = 5280 feet = 1.609 kilometers	1 kilometer (km)	= 1000 meters = 0.621 statute mile
1 yard (yd)	= 3 feet = 0.914 meter	1 meter (m)	= 100 centimeters (cm) = 1000 millimeters = 1.094 yards = 3.281 feet = 39.37 inches
1 foot (ft)	= 12 inches = 30.48 centimeters		
1 inch (in.)	= 25.40 millimeters	1 micron	= 0.001 millimeter = 0.000039 inch
100 ft per min	= 0.508 meter per sec	1 meter per sec	= 196.9 ft per min
1 square mile	= 640 acres = 259 hectares	1 sq kilometer (km^2)	= 100 hectares = 0.3861 sq mile
1 acre	= 4840 sq yards = 0.4047 hectare	1 hectare (ha)	= 10,000 sq meters = 2.471 acres
1 square yard	= 9 sq feet = 0.836 sq meter	1 sq meter (m^2)	= 10,000 sq centimeters = 1.196 sq yards = 10.76 sq feet
1 square foot	= 144 sq inches = 0.0929 sq meter		
1 square inch	= 6.45 sq centimeters	1 sq centimeter (cm^2)	= 100 sq millimeters = 0.1550 sq inch

1 nautical mile = 6080 feet = 1.853 kilometers
1 nautical mile per hour = 1 knot

Weight

1 U.S. long ton	= 2240 pounds = 1016 kilograms	1 pound per foot	= 1.488 kg per meter
1 U.S. short ton	= 2000 pounds = 907 kilograms	1 metric ton	= 1000 kilograms = 0.984 long ton = 1.102 U.S. short tons = 2205 pounds
1 pound (lb)	= 16 ounces = 7000 grains = 0.454 kilogram	1 kilogram (kg)	= 1000 grams = 2.205 pounds
1 ounce (oz)	= 0.0625 pound = 28.35 grams	1 gram (g)	= 1000 milligrams = 0.03527 ounce = 15.43 grains
1 grain	= 64.8 milligrams = 0.0023 ounce	1 kg per meter	= 0.672 pound per ft

Density

1 cu ft per lb	= 0.0624 m^3 per kg	1 m^3 per kilogram	= 16.02 cu ft per lb
1 lb per cu ft	= 16.02 kg per m^3	1 kilogram per m^3	= 0.0624 lb per cu ft
1 grain per cu ft	= 2.288 grams per m^3	1 gram per m^3	= 0.437 grain per cu ft
1 grain per U.S. gallon	= 17.11 grams per m^3 = 17.11 milligrams/liter		= 0.0584 grain per U.S. gal
		1 gram per liter	= 58.4 grain per U.S. gal

Water at 62 F (16.7 C)

1 cubic foot = 62.3 lb 1 lb = 0.01604 cu ft 1 U.S. gal = 8.33 lb

Water at 39.2 F (4 C)

Maximum Density

1 cubic foot	= 62.4 pounds	1 pound	= 0.01602 cubic foot
1 cubic meter	= 1000 kilograms	1 liter	= 1.0 kilogram

1 kg per cubic meter = 1 gram per liter = 1 part per thousand
1 gram per cubic meter = 1 milligram per liter = 1 part per million

Volume

1 cubic yard	= 27 cubic feet = 0.765 cubic meter	1 cubic meter (m^3)	= 1000 liters = 1.308 cubic yards = 35.31 cubic feet
1 cubic foot	= 1728 cubic inches = 28.32 liters	1 liter	= 1000 cu centimeters = 0.2200 imperial gallon = 0.2642 U.S. gallon = 61.0 cubic inches
1 cubic inch	= 16.39 cu centimeters		
1 imperial gallon	= 277.4 cubic inches = 4.55 liters		
1 U.S. gallon	= 0.833 imperial gallon = 3.785 liters = 231 cubic inches	1 board foot	= 12 in. x 12 in. x 1 in. thick = 144 cubic inches
1 U.S. barrel (petroleum)	= 42 U.S. gallons = 35 imperial gallons	1 cu ft per min	= 1.699 m^3 per hour
		1 m^3 per hour	= 0.589 cu ft per min

Pressure

1 atmosphere	= 760 mm (29.92 in.) mercury with density 13.595 grams per cm^3 = 14.696 pounds per square inch = 1.033 kg per sq centimeter = 1013 millibars	1 lb per square inch	= 2.036 in. head of mercury = 2.309 ft head of water = 0.0703 kilogram per cm^2 = 0.0690 bar
		1 ton per square inch	= 1.406 kilograms per mm^2
		1 inch head of water	= 5.20 pounds per sq ft
1 atmosphere (metric)	= 1 kg per sq cm = 10,000 kg per m^2 = 10 m head of water = 14.22 lb per sq in.	1 foot head of water	= 0.433 pound per sq in.
		1 m head of water	= 0.1 kg per cm^2
		1 in. head of mercury	= 0.491 pound per sq in.
		1 m head of mercury	= 1.360 kg per cm^2 = 1333 millibars
1 bar (megadyne per sq cm)	= 1000 millibars = 750.1 mm head of mercury = 1.02 kg per sq centimeter = 14.50 lb per sq in.	1 kilogram per m^2	= 1 mm head of water = 0.2048 lb per sq ft
		1 kilogram per cm^2	= 735.5 mm of mercury = 0.981 bar = 14.22 lb per sq in.
1 lb per square foot	= 0.1924 in. of water = 4.88 kilograms per m^2	1 kg per mm^2	= 0.711 ton per sq in.

In these conversions, inches and feet of water are measured at 62 F (16.7 C), millimeters and meters of water at 39.2 F (4 C), and inches, millimeters, and meters of mercury at 32 F (0 C).

Heat, Work, and Power

The British thermal unit (Btu) or the kilogram-Calorie (Cal) were originally defined as the heat required to raise the temperature of 1 pound or 1 kilogram of water by 1 F or 1 C, respectively. Recently the International Calorie has been defined as 1/860 International kilowatthour. The Btu is derived from the International Calorie from the relation 1 Cal per kg = 1.8 Btu per lb. The differences are less than 0.05 of 1% and are therefore, for the purposes of these tables, negligible.

1 British thermal unit (Btu)	= 778.26 foot-pounds (778) = 107.6 kilogram-meters = 0.2520 Calorie
1 Btu per lb	= 0.556 Calorie per kg
1 Btu per cu ft	= 8.90 Cal per cu meter
1 Btu per sq ft	= 2.712 Cal per sq meter
1 Btu per sq ft, F	= 4.88 Cal per sq m, C
1 Btu/hr, sq ft, F/ft	= 1.488 Cal/hr, sq m, C/m

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Heat, Work, and Power (Cont'd)

1 Btu/hr, sq ft, F/in.	= 0.1240 Cal/hr, sq m, C/m
1 therm	= 100,000 Btu
1 Calorie (Cal)	= 3088 foot-pounds = 427 kilogram-meters = 3.968 Btu
1 Cal per kilogram	= 1.8 Btu per pound
1 Cal per cu meter	= 0.1124 Btu per cubic foot
1 Cal per sq meter	= 0.3687 Btu per square foot
1 Cal/sq meter, C	= 0.2048 Btu/sq ft, F
1 Cal/hr, sq m, C/m	= 0.672 Btu/hr, sq ft, F/ft = 8.06 Btu/hr, sq ft, F/in.
1 electrical unit	= 1 kilowatthour
1 foot-pound (ft-lb)	= 0.1383 kilogram-meter
1 kilogram-meter (kg-m)	= 7.23 foot-pounds
1 kilowatt (kw)	= 738 foot-pounds per sec = 102 kilogram-meters per sec = 1.341 horsepower = 1.360 metric horsepower
1 horsepower (hp)	= 33,000 foot-pounds per min = 550 foot-pounds per sec = 76.04 kilogram-meters per sec = 0.746 kilowatt = 1.014 metric horsepower
1 metric horsepower	= 32,550 foot-pounds per min = 542 foot-pounds per sec = 75 kilogram-meters per sec = 0.735 kilowatt = 0.986 horsepower
1 kilowatthour (kwhr)	= 3412.75 Btu (3413) = 860 Calories
1 horsepower-hour	= 2545.1 Btu (2545)
1 metric horsepower-hour	= 632 Calories
1 lb per horsepower-hour	= 0.447 kg per metric hp-hr
1 kg per metric hp-hour	= 2.235 lb per horsepower-hr
1 boiler horsepower	= 10 sq ft of boiler heating surface = 34.5 lb per hour evaporation from and at 212 F
100% boiler rating	= 3348 Btu (i.e., 3.45 lb evaporation from and at 212 F) per sq ft heating surface per hour

Chapter 18. Fuel Ash Effects on Boiler Design and Operation

Determination of Ash-Fusion Temperatures

The standard laboratory procedure for the determination of the fusibility of ash, quoted below, appears in Standards on Coal and Coke (D 271), published by the American Society for Testing Materials:

Fusibility of Ash

Apparatus

33. *(a) Furnace.*—Any gas or electric furnace conforming to the following requirements may be used:

1) Capable of maintaining a zone of uniform temperature in which to heat the ash cones,

2) Capable of maintaining a reducing atmosphere surrounding the ash cones during heating, in which the ratio by volume of reducing gases to oxidizing gases is as specified in Section 36 *(b)*,

3) Capable of regulation so that the rate of temperature rise shall not be less than 5 and not more than 10 C per min, and

4) Provided with means of observing the ash cones during heating.

(b) Cone Mold.—The cone mold shall be suitable for making ash cones ¾ in. in height and ¼ in. in width at each side of the base, which is an equilateral triangle. A suitable mold, made from metal, is shown in Fig. A.

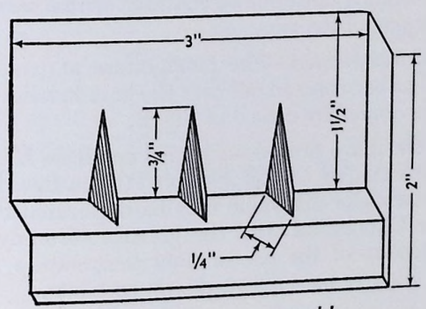


Fig. A. *Brass cone mold*

(c) Pyrometer.—Temperature measurements shall be made with a thermocouple of platinum and platinum-rhodium used in conjunction with a high-resistance millivoltmeter or potentiometer; or with an optical pyrometer of the disappearing-filament type, which has been calibrated to be accurate to 10 C up to 1400 C and to 15 C from 1400 to 1600 C. If a thermocouple is used it shall be protected from the furnace gases by a glazed porcelain tube and so placed in the furnace that the hot junction is in the immediate vicinity of the cones. The pyrometer equipment shall be checked frequently by mounting small pieces of pure gold and nickel in the same manner as the cones. With a strong reducing atmosphere, 1452 C shall be obtained for the melting point of nickel, and 1063 C for gold. The pyrometer equipment shall also be standardized from time to time through the temperature range for which it is used, by a suitably equipped standardizing laboratory, such as that of the National Bureau of Standards.

If temperature measurements are made by sighting an optical pyrometer through the glass window of the observation tube, the following corrections shall be added to the observed temperatures to correct for absorption of light by the glass window:

Observed Temperature, C	Correction, Deg C to be Added
800	5
1000	8
1200	10
1400	13
1600	16

Preparation of Ash

34. Spread out a 50 to 100-g portion of the coal or coke sample passing a No. 60 sieve on a 6-in. fireclay roasting dish. Completely convert the sample to ash in a muffle furnace at a temperature of 800 to 900 C. Transfer a 5 to 10-g portion of this ash to an agate mortar and grind it so that it will pass a No. 200 (74-micron) sieve. Then place the ash in a silica or porcelain capsule, ⅜ in. in depth and 1¾ in. in diameter, and ignite it for a period of 2 hr in a current of oxygen at a temperature of 800 to 850 C to insure complete and uniform oxidation of the ash.

Preparation of Cones

35. *(a)* Moisten the ignited ash with a solution of dextrin (10%) containing salicylic acid (0.1%) as a preservative and work it into a plastic mass with a spatula. Mold the plastic material into triangular pyramids (cones) ¾ in. in height and ¼ in. in width at the side of the base. One side of the cone shall be perpendicular to the base. Make the cones by firmly pressing the plastic material with a steel spatula into a mold of the dimensions specified. A suitable mold is shown in Fig. A. Strike off the surface of the material smooth, remove the cones from the mold by ap-

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plying pressure at the base, and place them in a suitable location to dry.

(b) When dry, mount the cones in a refractory base composed of a mixture of equal parts by weight of kaolin and calcined alumina, or of kaolin and alundum cement, or on a base of refractory brick. In using a mixture of kaolin and calcined alumina or of kaolin and alundum cement, moisten the mixture to make it workable. Spread part of the mixture on a sheet-iron plate, mount the cones to not over ⅜ in. in depth in vertical positions in small holes made in the base, and press the base material into the holes around the bottom of the cones to fill the crevices and to make the cones stand firmly. Usually a number of cones may be mounted in one base as shown in Fig. B. When a piece of refractory brick is used as the base, spread a layer of wet alundum cement about ⅛ in. in thickness on the base and push the cones down into it.

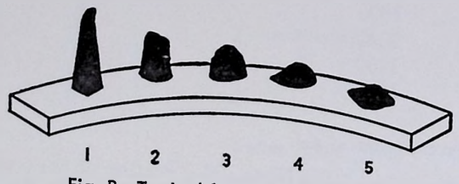


Fig. B. *Typical form of fused cones*

(c) Dry the mounted cones on a hot plate or in a drying oven, and ignite at a dull red heat in an open muffle furnace to remove the carbonaceous material.

Method of Heating

36. *(a)* The mounted test cones shall be placed in the furnace at a temperature of not over 800 C. If the furnace temperature is below 800 C, it may be rapidly increased to 800 C; then the rate of heating shall be controlled to give a rate of temperature increase of not less than 5 and not more than 10 C per min. This rate shall be maintained until the end of the test.

(b) A mildly reducing atmosphere shall be maintained surrounding the cones during the heating. The ratio by volume of reducing gases to oxidizing gases in this atmosphere shall be between the limits 20 to 80 and 80 to 20. Hydrogen, hydrocarbons, and carbon monoxide shall be considered as reducing gases, and oxygen, carbon dioxide, and water vapor shall be considered oxidizing gases. Nitrogen is inert. (For information concerning the effect of various atmospheres, see U.S. Bureau of Mines *Bulletin 129*, 1918.)

Softening Temperature

31. The softening temperature is defined as the temperature at which the cone has fused down to a spherical lump, as shown in cone 3 of Fig. B. Cone 2 has almost reached the softening temperature; cone 4 has passed the softening temperature.

Note: Critical points other than the softening temperature, that may be observed during the test, and that may be of value are as follows:

Initial-Deformation Temperature. The temperature at which the first rounding or bending of the apex of the cone takes place, as shown in cone 1 of Fig. B. Such bending must not be confused with a shrinking or warping of the cone.

Fluid Temperature. The temperature at which the cone has been spread out over the base in a flat layer, as represented by cone 5 of Fig. B.

In the above procedure, formal emphasis is placed on the softening temperature, and in practice the determination of this value may be duplicated by different laboratories with the greatest certainty. Determination of the last or fluid temperature is the least likely to be duplicated with certainty.

Chapter 22. Metallurgy and Properties of Materials

TABLE A

Specific Functions of Alloying Elements in Steel

(Abstract of data published by ASM)

Element	Solid Solubility In Alpha Iron	Solid Solubility In Gamma Iron	Principal Functions
Al	30% ±	1.1% (Increased by C)	(a) Deoxidizes efficiently. (b) Restricts grain growth by forming dispersed oxides or nitrides. (c) Alloying element in nitriding steel.
Cr	Unlimited	12.8% (20% with 0.5% C)	(a) Increases corrosion resistance and hardenability. (b) Adds some strength at high temp. (c) Resists abrasion and wear (with high C).
Co	80% ±	Unlimited	(a) Contributes to red-hardness by hardening ferrite.
Mn	15 to 18%	Unlimited	(a) Counteracts brittleness from sulfur. (b) Increases hardenability inexpensively.
Mo	32% (Less with lowered temperature)	3% (8% with 0.3% C)	(a) Raises coarsening temp of austenite. (b) Deepens hardening. (c) Raises hot and creep strengths, red-hardness. (d) Enhances corrosion resistance in stainless. (e) Forms abrasion-resisting particles.
Ni	25% ± (Irrespective of C)	Unlimited	(a) Strengthens unquenched or annealed steels. (b) Toughens pearlitic-ferritic steels (especially at low temperatures). (c) Renders high-chromium iron alloys austenitic.
P	2.5% (Irrespective of C)	0.5%	(a) Strengthens low-carbon steel. (b) Increases resistance to corrosion. (c) Improves machinability in free-cutting steels.
Si	18.5% (Not much changed by carbon)	2.0% ± (9% with 0.35% C)	(a) Used as general purpose deoxidizer. (b) Alloy for electrical and magnetic sheet. (c) Improves oxidation resistance. (d) Increases hardenability of steels carrying nongraphitizing elements. (e) Strengthens low-alloy steels.
Ti	6% ± (Less with lowered temperature)	0.75% (1% ± with 0.2% C)	(a) Fixes carbon as inert particles: (1) Reduces martensitic hardness and hardenability in medium-chromium steels. (2) Prevents formation of austenite in high-chromium steels. (3) Prevents localized depletion of chromium in stainless during long heating.
W	32% (Less with lowered temperature)	6% (11% with 0.25% C)	(a) Forms hard abrasion-resistant particles in tool steels. (b) Promotes red-hardness and hot strength.
V	Unlimited	1.5% ± (4% with 0.2% C)	(a) Elevates coarsening temp of austenite (promotes fine grain). (b) Increases hardenability (when dissolved). (c) Resists tempering and causes marked secondary hardening.

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Table B

Recommended Arc-Welding Procedures for Various Steel Combinations

Type of Steel	Plain Carbon	Carbon-Mo	Croloy ½	Croloy 1&1¼	Croloy 2	Croloy 2¼&3M	Croloy 5&5Ti	Croloy 5Si
Plain carbon	1,2-a-A	1,2-a-B	1,2-a-B	2-b-C	2-b-C	2-b-C	2-b-C	°
Carbon-molybdenum†	1,2-a-B	3,4-a-B	3,4-a-B	4-b-C	4-b-C	4-b-C	4-b-C	°
Croloy ½	1,2-a-B	3,4-a-B	3,4-a-B	5-b-C	5-b-C	5-b-C	5-b-C	°
Croloy 1	2-b-C	4-b-C	5-b-C	5-b-C	5-b-C	5-b-C	5-b-C	5-b-C
Croloy 1¼	2-b-C	4-b-C	5-b-C	5-b-C	5-b-C	5-b-C	5-b-C	5-b-C
Croloy 2	2-b-C	4-b-C	5-b-C	5-b-C	6-b-C	6-b-C	6-b-C	6-b-C
Croloy 2¼	2-b-C	4-b-C	5-b-C	5-b-C	6-b-C	7-b-C	7-b-C	7-b-C
Croloy 3M	2-b-C	4-b-C	5-b-C	5-b-C	6-b-C	7-b-C	7-b-C	7-b-C
Croloy 5	2-b-C	4-b-C	5-b-C	5-b-C	6-b-C	7-b-C	8-b-C	8-b-C
Croloy 5Ti	2-b-C	4-b-C	5-b-C	5-b-C	6-b-C	7-b-C	8-b-C	8-b-C
Croloy 5Si	°	°	°	5-b-C	6-b-C	7-b-C	8-b-C	9-b-C
Croloy 7	°	°	°	5-b-C	6-b-C	7-b-C	8-b-C	9-b-C
Croloy 9M	°	°	°	5-b-C	6-b-C	7-b-C	8-b-C	9-b-C
Croloy 18-8S	13,16-a-N	13,16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	°
Croloy 18-8Ti or CbTa	13,16-a-N	13,16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	°
Croloy 18-8Cb	13,16-a-N	13,16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	°
Croloy 18-8Si	13,16-a-N	13,16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	°
Croloy 16-13-3	13,16-a-N	13,16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	°
Croloy 25-20	13,16-a-N	13,16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	°

Type of Steel	Croloy 7	Croloy 9M	Croloy 18-8S	Croloy 18-8Ti or CbTa	Croloy 18-8Cb	Croloy 18-8Si	Croloy 16-13-3	Croloy 25-20
Plain carbon	°	°	13,16-a-N	13,16-a-N	13,16-a-N	13,16-a-N	13,16-a-N	13,16-a-N
Carbon-molybdenum†	°	°	13,16-a-N	13,16-a-N	13,16-a-N	13,16-a-N	13,16-a-N	13,16-a-N
Croloy ½	°	°	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N
Croloy 1	5-b-C	5-b-C	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N
Croloy 1¼	5-b-C	5-b-C	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N
Croloy 2	6-b-C	6-b-C	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N
Croloy 2¼	7-b-C	7-b-C	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N
Croloy 3M	7-b-C	7-b-C	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N
Croloy 5	8-b-C	8-b-C	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N
Croloy 5Ti	8-b-C	8-b-C	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N	16-a-N
Croloy 5Si	9-b-C	9-b-C	°	°	°	°	°	°
Croloy 7	10-b-C	10-b-C	°	°	°	°	°	°
Croloy 9M	10-b-C	11-b-C	°	°	°	°	°	°
Croloy 18-8S	°	°	12-a-O	12-a-O	12-a-O	12-a-O	12-a-O	13-a-O
Croloy 18-8Ti or CbTa	°	°	12-a-O	15-a-O	15-a-O	15-a-O	15-a-O	13-a-O
Croloy 18-8Cb	°	°	12-a-O	15-a-O	17-a-O	15-a-O	15-a-O	13-a-O
Croloy 18-8Si	°	°	12-a-O	15-a-O	15-a-O	12-a-O	12-a-O	13-a-O
Croloy 16-13-3	°	°	12-a-O	15-a-O	15-a-O	12-a-O	14-a-O	13-a-O
Croloy 25-20	°	°	13-a-O	13-a-O	13-a-O	13-a-O	13-a-O	13-a-O

° Joints marked "°" are not recommended, since heat-treatment may adversely affect the properties of one of the base metals involved.

† ½% molybdenum.

Key to Designations in Table B

Electrodes

1. E-6010 (SA-233)
2. E-6015 (SA-233)
3. E-7010 (SA-316)°
4. E-7015 (SA-316)°
5. Croloy 1A† (SA-316) E-7015
6. Croloy 2A† (SA-316) E-9015
7. Croloy 2¼A† (SA-316) E-9015
8. Croloy 5A† (E-502-15)
9. Croloy 5Si
10. Croloy 7
11. Croloy 9M
12. E-308-15 (SA-298)
13. E-310-15 (SA-298)
14. E-317-15 (SA-298)
15. E-347-15 (SA-298)
16. Inconel 132 or "A"
17. E-348-15 (SA-298) Cb Low Ta

Preheating

a. 150 F
b. 300 F

Postwelding Heat-Treatment

A. 1150 F
B. 1200 F
C. 1350 F
N. None
O. See note (4)

° Should contain ½% molybdenum.

† Croloy "A" special low-carbon electrodes available from Champion Rivet Co. or Metal & Thermit Corp. If higher-carbon commercial chromium-molybdenum electrodes are used, then the preheat temperature should be 100 F higher, and the welds should be stress-relieved immediately after welding and before the weld has cooled to room temperature.

Notes on Welding Procedures

Electrodes

Commercial equivalents of electrodes made of Croloys ½ to 9M inclusive should be of similar analysis, with a "15" type coating.

Preheating

The preheat temperatures listed are minimums for base materials to ⅜-inch thickness. Heavier sections or assemblies welded under restraint should be preheated up to 400 F or higher.

Postwelding Heat-Treatment

(1) The entire weldment, or a circumferential band 6 times the wall thickness on each side of the joint but in any case not less than 2 in. wide, should be heated to within plus or minus 25 F of the temperature listed, for 1 hr per inch of maximum section thickness, but not less than 30 minutes. To retard cooling and to assure a uniform rate of cooling, insulating wrappers should be applied to the joint.

(2) Weldments comprised of Croloys 5Si, 7, or 9M should be heated immediately after the weld has been made and before it has cooled below the minimum preheating temperature. These welds should be cooled, however, to the Mf temperature so as to permit the martensite transformation.

(3) (a) Postheating is not required for carbon-steel tubing or piping of less than ¾-inch wall thickness.

(b) Postheating is not required for carbon-molybdenum (C = 0.25%) tubing or piping of less than ½-inch wall thickness.

(4) Whether these weldments need heat-treatment depends upon the service intended and should be established accordingly. Where required, the recommended heat-treatment may be:

(a) A solution quench from within range 1900-2050 F.

(b) A stabilizing treatment consisting of heating to 1600-1650 F, followed by air cooling.

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Table C

Mechanical Properties of Steels Listed in Table B

Type of Steel	Ultimate Strength, psi	Yield Strength, psi	Elongation, % in 2 in.	Hardness, Brinell	Charpy, ft-lb
Plain carbon	59,750	42,450	46	117	46
Carbon-molybdenum*	63,800	49,000	47	141	53
Croloy ½	62,000	43,000	50	140	52
Croloy 1	69,200	34,600	40	130	46
Croloy 1¼	74,200	55,100	42	153	53
Croloy 2	69,450	46,200	44	140	52
Croloy 2¼	69,900	45,080	49	147	54
Croloy 3M	69,100	40,100	53	146	53
Croloy 5	69,800	38,600	47	139	46
Croloy 5Si	87,925	59,100	40	172	60
Croloy 9M	73,460	43,470	41	145	46
Croloy 18-8S	90,800	42,100	67	146	84
Croloy 18-8Ti	84,700	48,500	60	153	74
Croloy 18-8Cb	86,240	46,450	58	142	50
Croloy 16-13-3	84,500	43,660	70	142	78
Croloy 25-20	85,160	43,960	53	134	74

* ½% molybdenum.

Note: These properties are for fully annealed steels at room temperature.

Chemical Composition of Steels in Table D

Type of Steel	C	Mn	S	P	Si	Cr	Mo	Ni
Carbon steel, rimmed	0.10-0.20	0.80	0.045	0.04	trace	—	—	—
Carbon steel, killed	0.10-0.20	0.30-0.60	0.045	0.04	0.25	—	—	—
Carbon—0.50% Mo	0.10-0.20	0.30-0.80	0.045	0.04	0.10-0.50	—	0.45-0.65	—
Croloy ½	0.10-0.20	0.30-0.60	0.05	0.04	0.10-0.30	0.50-0.70	0.45-0.65	—
Croloy 1¼	0.15	0.30-0.60	0.03	0.03	0.50-1.00	1.00-1.50	0.45-0.65	—
Croloy 2	0.15	0.30-0.60	0.03	0.03	0.50	1.75-2.25	0.45-0.65	—
Croloy 2¼	0.15	0.30-0.60	0.03	0.03	0.50	2.00-2.50	0.90-1.10	—
Croloy 3M	0.15	0.30-0.60	0.03	0.03	0.50	2.75-3.25	0.80-1.00	—
Croloy 5	0.15	0.30-0.60	0.03	0.03	0.50	4.00-6.00	0.45-0.65	—
Croloy 5Si	0.15	0.30-0.60	0.03	0.03	1.00-2.00	4.00-6.00	0.45-0.65	—
Croloy 7	0.15	0.30-0.60	0.03	0.03	0.50-1.00	6.00-8.00	0.45-0.65	—
Croloy 9M	0.15	0.30-0.60	0.03	0.03	1.00	8.00-10.00	0.90-1.10	—
Croloy 18-8S	0.08	2.00	0.03	0.03	0.75	18.00-20.00	—	8.00-11.00
Croloy 16-13-3	0.08	2.00	0.03	0.03	0.75	16.00-18.00	2.00-3.00	11.00-14.00
Croloy 25-20	0.15	2.00	0.03	0.03	0.75	24.00-26.00	—	19.00-22.00

Note: Percentages are maximum unless range is given.

Table D

Creep Stress Data* for Tubular Materials in High-Temperature Service

Temp, F	Material	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O
800	1% Elong in 100,000 hr	12,000	19,500	33,500	—	—	—	—	—	—	—	—	—	—	—	—
900		6,300	11,000	24,500	35,000	35,000	28,000	40,500	19,500	17,900	—	14,500	33,250	24,000	—	—
1000		3,300	7,200	12,400	12,000	14,700	11,000	16,350	12,500	9,200	8,200	8,000	12,000	18,300	22,800	—
1100		1,000	—	—	3,600	5,500	5,875	9,200	8,000	4,800	4,400	4,400	6,950	11,550	13,800	—
1200		300	—	—	—	3,375	3,400	4,175	4,000	1,800	1,600	2,000	2,300	6,600	8,200	7,400
1350		—	—	—	—	—	—	—	—	—	—	—	—	2,500	4,200	3,300
1500		—	—	—	—	—	—	—	—	—	—	—	—	—	1,840	1,100
800	1% Elong in 10,000 hr	7,000	13,500	23,500	32,000	—	—	—	—	—	—	—	—	25,300	—	—
900		4,900	8,400	17,200	21,000	20,000	18,000	22,000	12,000	15,000	—	9,200	24,750	18,000	—	—
1000		2,000	2,150	5,200	5,600	8,200	6,300	7,600	7,000	7,200	5,700	5,000	8,500	11,500	14,800	—
1100		630	—	—	1,300	2,125	3,325	4,910	4,000	2,400	2,700	2,500	3,775	7,100	8,700	—
1200		70	—	—	—	910	1,075	2,510	1,550	900	850	1,300	1,600	4,250	5,000	5,400
1350		—	—	—	—	—	—	—	—	—	—	—	—	1,600	2,500	2,800
1500		—	—	—	—	—	—	—	—	—	—	—	—	—	930	800

* In psi for 1 per cent elongation in 10,000 and 100,000 hours.

Key to Materials

A—Carbon steel, rimmed
B—Carbon steel, killed
C—Carbon-0.50% Mo
D—Croloy ½
E—Croloy 1¼
F—Croloy 2
G—Croloy 2¼
H—Croloy 3M
I —Croloy 5
J —Croloy 5Si
K —Croloy 7
L —Croloy 9M
M—Croloy 18-8S
N—Croloy 16-13-3
O—Croloy 25-20

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TABLE E

Maximum Allowable Stress Values, Pounds per Square Inch

ASME Spec	Grade	Nominal Composition	Spec Min Tensile	Notes	For metal temperatures not exceeding deg F: −20 to 100	300	400	−20 to 400	500	650	700	800	900	1000	1100	1200	1300	1400	1500
PLATE STEELS																			
Carbon Steel																			
SA-30	Flange	—	55,000	(1)	—	—	—	13,750	13,750	13,750	13,250	10,200	—	—	—	—	—	—	—
SA-30	Firebox A	—	55,000	(1)	—	—	—	13,750	13,750	13,750	13,250	10,200	*5,000*	—	—	—	—	—	—
SA-30	Firebox B	—	48,000	(1)	—	—	—	12,000	12,000	12,000	11,650	9,000	*5,000*	—	—	—	—	—	—
SA-129	A, B, C	—	44,000	—	—	—	—	11,000	11,000	11,000	—	—	—	—	—	—	—	—	—
SA-201	A	C-Si	55,000	(1)	—	—	—	13,750	13,750	13,750	13,250	10,200	*5,000*	*1,500*	—	—	—	—	—
SA-201	B	C-Si	60,000	(1)	—	—	—	15,000	15,000	15,000	14,350	10,800	*5,000*	*1,500*	—	—	—	—	—
SA-212	A	C-Si	65,000	(1)	—	—	—	16,250	16,250	16,250	15,500	11,400	*5,000*	*1,500*	—	—	—	—	—
SA-212	B	C-Si	70,000	(1)	—	—	—	17,500	17,500	17,500	16,600	12,000	*5,000*	*1,500*	—	—	—	—	—
SA-285	A	—	45,000	(1) (6)	—	—	—	11,250	11,250	11,250	10,900	8,300	*5,000*	—	—	—	—	—	—
SA-285	B	—	50,000	(1) (6)	—	—	—	12,500	12,500	12,500	12,100	9,400	*5,000*	—	—	—	—	—	—
SA-285	C	—	55,000	(1) (6)	—	—	—	13,750	13,750	13,750	13,250	10,200	*5,000*	—	—	—	—	—	—
SA-299	—	C-Mn-Si	75,000	(1)	—	—	—	18,750	18,750	18,750	17,700	12,000	*5,000*	*1,500*	—	—	—	—	—
Low-Alloy Steel																			
SA-202	A	Cr-Mn-Si	75,000	—	—	—	—	18,750	18,750	18,750	17,700	12,000	*5,000*	*1,500*	—	—	—	—	—
SA-202	B	Cr-Mn-Si	85,000	—	—	—	—	21,250	21,250	21,250	19,800	12,000	*5,000*	*1,500*	—	—	—	—	—
SA-203	A, D	2½ & 3½Ni	65,000	—	—	—	—	16,250	16,250	16,250	15,500	11,400	*5,000*	*1,500*	—	—	—	—	—
SA-203	B, E	2½ & 3½Ni	70,000	—	—	—	—	17,500	17,500	17,500	16,600	12,000	*5,000*	*1,500*	—	—	—	—	—
SA-203	C	2½Ni	75,000	—	—	—	—	18,750	18,750	18,750	17,700	12,000	*5,000*	*1,500*	—	—	—	—	—
SA-204	A	C-½Mo	65,000	(2)	—	—	—	16,250	16,250	16,250	16,250	15,650	*12,500*	*5,500*	—	—	—	—	—
SA-204	B	C-½Mo	70,000	(2)	—	—	—	17,500	17,500	17,500	17,500	16,900	*12,750*	*5,500*	—	—	—	—	—
SA-204	C	C-½Mo	75,000	(2)	—	—	—	18,750	18,750	18,750	18,750	18,000	*13,000*	*5,500*	—	—	—	—	—
SA-225	A	Mn-V	70,000	(1)	—	—	—	17,500	17,500	17,500	16,600	12,000	*5,000*	*1,500*	—	—	—	—	—
SA-225	B	Mn-V	75,000	(1)	—	—	—	18,750	18,750	18,750	17,700	12,000	*5,000*	*1,500*	—	—	—	—	—
SA-302	A	½Cr-½Mo	65,000	—	—	—	—	16,250	16,250	16,250	16,250	15,650	*12,500*	*5,500*	—	—	—	—	—
SA-302	B	1Cr-½Mo	60,000	—	—	—	—	15,000	15,000	15,000	15,000	14,750	13,100	7,500	2,800	*1,000*	—	—	—
SA-387	A	Mn-½Mo	75,000	(2)	—	—	—	18,750	18,750	18,750	18,750	18,000	*13,000*	*5,500*	—	—	—	—	—
SA-387	B	Mn-½Mo	80,000	(2)	—	—	—	20,000	20,000	20,000	20,000	19,100	*13,250*	*5,500*	—	—	—	—	—
SA-357	—	5Cr-½Mo	60,000	—	—	—	—	15,000	14,500	13,700	13,400	12,800	11,500	7,300	3,300	1,500	—	—	—
High-Alloy Steel																			
SA-240	S	18Cr-8Ni	75,000	(7)	18,750	16,000	15,450	—	15,100	14,850	14,800	14,550	14,000	12,500	7,500	4,500	2,450	1,400	750
SA-240	S	18Cr-8Ni	75,000	—	18,750	15,000	13,650	—	12,500	11,200	10,800	10,000	9,400	8,800	7,500	4,500	2,450	1,400	750
SA-240	M	18Cr-10Ni-2Mo	75,000	—	18,750	17,900	17,500	—	17,200	17,050	17,000	16,750	16,000	14,000	10,400	6,800	4,000	2,000	1,000
SA-240	T	18Cr-8Ni-Ti	75,000	—	18,750	17,000	15,800	—	15,200	14,850	14,800	14,550	14,100	13,500	10,300	5,000	2,200	1,200	750
SA-240	C	18Cr-8Ni-Cb	75,000	—	18,750	17,000	15,800	—	15,200	14,850	14,800	14,550	14,100	13,500	10,300	5,000	2,200	1,200	750
PIPE & TUBES																			
Seamless Carbon Steel																			
SA-53	A	—	48,000	(1) (3)	—	—	—	12,000	12,000	12,000	11,650	9,000	*5,000*	—	—	—	—	—	—
SA-53	B	—	60,000	(1) (3)	—	—	—	15,000	15,000	15,000	14,350	10,800	*5,000*	—	—	—	—	—	—
SA-83	A	—	(47,000)	(1) (3) (12)	—	—	—	11,750	11,750	11,750	11,450	9,000	*5,000*	*1,500*	—	—	—	—	—
SA-83	B	—	(40,000)	(12)	—	—	—	10,000	10,000	10,000	*9,700*	—	—	—	—	—	—	—	—
SA-106	A	—	48,000	(1)	—	—	—	12,000	12,000	12,000	11,650	9,000	*5,000*	*1,500*	—	—	—	—	—
SA-106	B	—	60,000	(1)	—	—	—	15,000	15,000	15,000	14,350	10,800	*5,000*	*1,500*	—	—	—	—	—
SA-192	A	—	(47,000)	(1) (12)	—	—	—	11,750	11,750	11,750	11,450	9,000	*5,000*	*1,500*	—	—	—	—	—
SA-210	—	—	60,000	(1)	—	—	—	15,000	15,000	15,000	14,350	10,800	*5,000*	*1,500*	—	—	—	—	—

TABLE E (Continued)

Maximum Allowable Stress Values, Pounds per Square Inch

ASME Spec	Grade	Nominal Composition	Spec Min Tensile	Notes	For metal temperatures not exceeding deg F: −20 to 100	300	400	−20 to 400	500	650	700	800	900	1000	1100	1200	1300	1400	1500
Seamless Alloy Steel																			
SA-209	T1	C-½Mo	55,000	(2)	—	—	—	13,750	13,750	13,750	13,750	13,450	12,500	*5,500*	—	—	—	—	—
SA-209	T1a	C-½Mo	60,000	(2)	—	—	—	15,000	15,000	15,000	15,000	14,400	12,500	*5,500*	—	—	—	—	—
SA-209	T1b	C-½Mo	53,000	(2)	—	—	—	13,250	13,250	13,250	13,250	13,000	12,500	*5,500*	—	—	—	—	—
SA-213	T5	5Cr-½Mo	60,000	—	—	—	—	15,000	14,500	13,700	13,400	12,800	11,500	7,300	3,300	1,500	—	—	—
SA-213	T7	7Cr-½Mo	60,000	—	—	—	—	15,000	14,500	13,700	13,400	12,500	9,500	5,000	2,500	1,200	—	—	—
SA-213	T9	9Cr-1Mo	60,000	—	—	—	—	15,000	14,500	13,700	13,400	12,800	12,000	8,500	3,300	1,500	—	—	—
SA-213	T11	1¼Cr-½Mo	60,000	—	—	—	—	15,000	15,000	15,000	15,000	15,000	13,100	7,800	4,000	*1,200*	—	—	—
SA-213	T12	1Cr-½Mo	60,000	—	—	—	—	15,000	15,000	15,000	15,000	14,750	13,100	7,500	2,800	*1,000*	—	—	—
SA-213	T5b	5Cr-½Mo-Si	60,000	—	—	—	—	15,000	14,500	13,700	13,400	12,800	10,900	5,500	2,500	1,200	—	—	—
SA-213	T3b	2Cr-½Mo	60,000	—	—	—	—	15,000	15,000	15,000	15,000	14,700	12,500	6,200	2,750	1,200	—	—	—
SA-213	T21	3Cr-1Mo	60,000	—	—	—	—	15,000	15,000	15,000	14,800	13,900	12,000	7,000	4,000	1,500	—	—	—
SA-213	T22	2¼Cr-1Mo	60,000	—	—	—	—	15,000	15,000	15,000	15,000	15,000	13,100	7,800	4,200	2,000	—	—	—
SA-213	TP304	18Cr-8Ni	75,000	(7)	18,750	16,000	15,450	—	15,100	14,850	14,800	14,550	14,000	12,500	7,500	4,500	2,450	1,400	750
SA-213	TP304	18Cr-8Ni	75,000	—	18,750	15,000	13,650	—	12,500	11,200	10,800	10,000	9,400	8,800	7,500	4,500	2,450	1,400	750
SA-213	TP316	16Cr-13Ni-3Mo	75,000	—	18,750	17,900	17,500	—	17,200	17,050	17,000	16,750	16,000	14,000	10,400	6,800	4,000	2,000	1,000
SA-213	TP321	18Cr-8Ni-Ti	75,000	—	18,750	17,000	15,800	—	15,200	14,850	14,800	14,550	14,100	13,500	10,300	5,000	2,200	1,200	750
SA-213	TP347	18Cr-8Ni-Cb	75,000	—	18,750	17,000	15,800	—	15,200	14,850	14,800	14,550	14,100	13,500	10,300	5,000	2,200	1,200	750
SA-312	TP304	18Cr-8Ni	75,000	(7)	18,750	16,000	15,450	—	15,100	14,850	14,800	14,550	14,000	12,500	7,500	4,500	2,450	1,400	750
SA-312	TP304	18Cr-8Ni	75,000	—	18,750	15,000	13,650	—	12,500	11,200	10,800	10,000	9,400	8,800	7,500	4,500	2,450	1,400	750
SA-312	TP316	16Cr-13Ni-3Mo	75,000	—	18,750	17,900	17,500	—	17,200	17,050	17,000	16,750	16,000	14,000	10,400	6,800	4,000	2,000	1,000
SA-312	TP321	18Cr-8Ni-Ti	75,000	—	18,750	17,000	15,200	—	15,200	14,850	14,800	14,550	14,100	13,500	10,300	5,000	2,200	1,200	750
SA-312	TP347	18Cr-8Ni-Cb	75,000	—	18,750	17,000	15,800	—	15,200	14,850	14,800	14,550	14,100	13,500	10,300	5,000	2,200	1,200	750
SA-335	P1	C-½Mo	55,000	(2)	—	—	—	13,750	13,750	13,750	13,750	13,450	*12,500*	*5,500*	—	—	—	—	—
SA-335	P2	½Cr-½Mo	55,000	—	—	—	—	13,750	13,750	13,750	13,750	13,450	12,500	6,250	—	—	—	—	—
SA-335	P3b	2Cr-½Mo	60,000	—	—	—	—	15,000	15,000	15,000	15,000	14,700	12,500	6,200	2,750	1,200	—	—	—
SA-335	P5	5Cr-½Mo	60,000	—	—	—	—	15,000	14,500	13,700	13,400	12,800	11,500	7,300	3,300	1,500	—	—	—
SA-335	P5b	5Cr-½Mo-Si	60,000	—	—	—	—	15,000	14,500	13,700	13,400	12,800	10,900	5,500	2,500	1,200	—	—	—
SA-335	P11	1¼Cr-½Mo	60,000	—	—	—	—	15,000	15,000	15,000	15,000	15,000	13,100	7,800	4,000	*1,200*	—	—	—
SA-335	P12	1Cr-½Mo	60,000	—	—	—	—	15,000	15,000	15,000	15,000	14,750	13,100	7,500	2,800	*1,000*	—	—	—
SA-335	P7	7Cr-½Mo	60,000	—	—	—	—	15,000	14,500	13,700	13,400	12,500	9,500	5,000	2,500	1,200	—	—	—
SA-335	P9	9Cr-1Mo	60,000	—	—	—	—	15,000	14,500	13,700	13,400	12,800	12,000	8,500	3,300	1,500	—	—	—
SA-335	P21	3Cr-1Mo	60,000	—	—	—	—	15,000	15,000	15,000	14,800	13,900	12,000	7,000	4,000	1,500	—	—	—
SA-335	P22	2¼Cr-1Mo	60,000	—	—	—	—	15,000	15,000	15,000	15,000	15,000	13,100	7,800	4,200	2,000	—	—	—
Electric-Resistance-Welded Carbon Steel																			
SA-53	A	—	48,000	(1) (8)	—	—	—	10,200	10,200	10,200	9,900	7,650	*4,250*	—	—	—	—	—	—
SA-53	B	—	60,000	(1) (8)	—	—	—	12,750	12,750	12,750	12,200	9,200	*4,250*	—	—	—	—	—	—
SA-135	A	—	48,000	(1) (8)	—	—	—	10,200	10,200	10,200	9,900	7,650	*4,250*	—	—	—	—	—	—
SA-135	B	—	60,000	(1) (8)	—	—	—	12,750	12,750	12,750	12,200	9,200	*4,250*	—	—	—	—	—	—
SA-178	A	—	(47,000)	(1) (3) (11) (12)	—	—	—	11,750	11,750	11,750	11,450	7,650	*4,250*	*1,300*	—	—	—	—	—
SA-178	B	—	40,000	(8)	—	—	—	8,500	8,500	8,500	*8,200*	—	—	—	—	—	—	—	—
SA-178	C	—	60,000	(1) (3) (11)	—	—	—	15,000	15,000	15,000	14,350	9,200	*4,250*	*1,300*	—	—	—	—	—
SA-226	—	—	(47,000)	(1) (11) (12)	—	—	—	11,750	11,750	11,750	11,450	7,650	*4,250*	*1,300*	—	—	—	—	—
Electric-Resistance-Welded Alloy Steel																			
SA-249	TP304	18Cr-8Ni	75,000	(7) (8)	15,950	13,600	13,150	—	12,850	12,600	12,550	12,350	11,900	10,650	6,400	3,850	2,100	1,200	650
SA-249	TP304	18Cr-8Ni	75,000	(8)	15,950	12,750	11,600	—	10,650	9,500	9,200	8,500	8,000	7,500	6,400	3,850	2,100	1,200	650
SA-249	TP316	16Cr-13Ni-3Mo	75,000	(8)	15,950	15,200	14,900	—	14,600	14,500	14,450	14,250	13,600	11,900	8,850	5,800	3,400	1,700	850
SA-249	TP321	18Cr-8Ni-Ti	75,000	(8)	15,950	14,450	13,450	—	12,900	12,600	12,550	12,350	12,000	11,500	8,750	4,250	1,850	1,000	650
SA-249	TP347	18Cr-8Ni-Cb	75,000	(8)	15,950	14,450	13,450	—	12,900	12,600	12,550	12,350	12,000	11,500	8,750	4,250	1,850	1,000	650

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TABLE E (Continued)

Maximum Allowable Strees Values, Pounds per Square Inch

ASME Spec	Grade	Nominal Composition	Spec Min Tensile	Notes	For metal temperatures not exceeding deg F: −20 to 100	300	400	−20 to 400	500	650	700	800	900	1000	1100	1200	1300	1400	1500
Electric-Resistance-Welded Alloy Steel																			
SA-250	T1	C-½Mo	55,000	(2) (8)	—	—	—	13,750	13,750	13,750	13,750	13,450	12,500	*5,500*	—	—	—	—	—
SA-250	T1a	C-½Mo	60,000	(2) (8)	—	—	—	15,000	15,000	15,000	15,000	14,400	12,500	*5,500*	—	—	—	—	—
SA-250	T1b	C-½Mo	53,000	(2) (8)	—	—	—	13,250	13,250	13,250	13,250	13,000	12,500	*5,500*	—	—	—	—	—
Lap Welded																			
SA-53	steel	—	45,000	(9)	—	—	—	9,000	9,000	9,000	*8,750*	—	—	—	—	—	—	—	—
SA-72	wrought iron	—	40,000	(9)	—	—	—	8,000	8,000	8,000	*7,800*	—	—	—	—	—	—	—	—
SA-83	steel	—	(45,000)	(9) (12)	—	—	—	9,000	9,000	9,000	*8,750*	—	—	—	—	—	—	—	—
SA-83	iron	—	—	(9)	—	—	—	8,000	8,000	8,000	*7,800*	—	—	—	—	—	—	—	—
Butt Welded																			
SA-53	steel	—	45,000	(10)	—	—	—	6,750	6,750	6,750	*6,500*	—	—	—	—	—	—	—	—
SA-72	wrought iron	—	40,000	(10)	—	—	—	6,000	6,000	6,000	*5,800*	—	—	—	—	—	—	—	—
Automatically Welded Austenitic Stainless Steel																			
SA-312	TP304	18Cr-8Ni	75,000	(7) (8)	15,950	13,600	13,150	—	12,850	12,600	12,550	12,350	11,900	10,650	6,400	3,850	2,100	1,200	650
SA-312	TP304	18Cr-8Ni	75,000	(8)	15,950	12,750	11,600	—	10,650	9,500	9,200	8,500	8,000	7,500	6,400	3,850	2,100	1,200	650
SA-312	TP316	16Cr-13Ni-3Mo	75,000	(8)	15,950	15,200	14,900	—	14,600	14,500	14,450	14,250	13,600	11,900	8,850	5,800	3,400	1,700	850
SA-312	TP321	18Cr-8Ni-Ti	75,000	(8)	15,950	14,450	13,450	—	12,900	12,600	12,550	12,350	12,000	11,500	8,750	4,250	1,850	1,000	650
SA-312	TP347	18Cr-8Ni-Cb	75,000	(8)	15,950	14,450	13,450	—	12,900	12,600	12,550	12,350	12,000	11,500	8,750	4,250	1,850	1,000	650
FORGINGS																			
Carbon Steel																			
SA-105	I	—	60,000	(1) (3)	—	—	—	15,000	15,000	15,000	14,350	10,800	*5,000*	*1,500*	—	—	—	—	—
SA-105	II	—	70,000	(1) (3)	—	—	—	17,500	17,500	17,500	16,600	12,000	*5,000*	*1,500*	—	—	—	—	—
SA-181	I	—	60,000	(1) (3)	—	—	—	15,000	15,000	15,000	14,350	10,800	*5,000*	*1,500*	—	—	—	—	—
SA-181	II	—	70,000	(1) (3)	—	—	—	17,500	17,500	17,500	16,600	12,000	*5,000*	*1,500*	—	—	—	—	—
SA-266	I	—	60,000	(1)	—	—	—	15,000	15,000	15,000	14,350	10,800	*5,000*	*1,500*	—	—	—	—	—
SA-266	II	—	70,000	(1)	—	—	—	17,500	17,500	17,500	16,600	12,000	*5,000*	*1,500*	—	—	—	—	—
SA-266	III	—	75,000	(1)	—	—	—	18,750	18,750	18,750	17,700	12,000	*5,000*	*1,500*	—	—	—	—	—
Alloy Steel																			
SA-182	F1	C-½Mo	70,000	(2)	—	—	—	17,500	17,500	17,500	17,500	16,900	*12,750*	*5,500*	—	—	—	—	—
SA-182	F12	1Cr-½Mo	70,000	—	—	—	—	17,500	17,500	16,800	16,150	14,850	13,100	7,500	2,800	*1,000*	—	—	—
SA-182	F5	5Cr-½Mo	90,000	—	—	—	—	22,500	21,600	19,000	17,500	14,500	11,500	7,300	3,300	1,500	—	—	—
SA-182	F11	1¼Cr-½Mo	70,000	—	—	—	—	17,500	17,500	16,800	16,150	15,000	13,100	7,800	4,000	*1,200*	—	—	—
SA-182	F9	9Cr-1Mo	100,000	—	—	—	—	25,000	24,000	22,000	21,200	17,700	13,100	8,500	3,300	1,500	—	—	—
SA-182	F22	2¼Cr-1Mo	70,000	—	—	—	—	17,500	17,500	17,500	17,500	17,500	14,000	7,800	4,200	2,000	—	—	—
SA-182	F304	18Cr-8Ni	75,000	(7)	18,750	16,000	15,450	—	15,100	14,850	14,800	14,550	14,000	12,500	7,500	4,500	2,450	1,400	750
SA-182	F304	18Cr-8Ni	75,000	—	18,750	15,000	13,650	—	12,500	11,200	10,800	10,000	9,400	8,800	7,500	4,500	2,450	1,400	750
SA-182	F316	18Cr-8Ni-Mo	75,000	—	18,750	17,900	17,500	—	17,200	17,050	17,000	16,750	16,000	14,000	10,400	6,800	4,000	2,000	1,000
SA-182	F321	18Cr-8Ni-Ti	75,000	—	18,750	17,000	15,800	—	15,200	14,850	14,800	14,550	14,100	13,500	10,300	5,000	2,200	1,200	750
SA-182	F347	18Cr-8Ni-Cb	75,000	—	18,750	17,000	15,800	—	15,200	14,850	14,800	14,550	14,100	13,500	10,300	5,000	2,200	1,200	750
SA-336	F1	C-½Mo	70,000	(2)	—	—	—	17,500	17,500	17,500	17,500	16,900	*12,750*	*5,500*	—	—	—	—	—
SA-336	F2	1Cr-½Mo	70,000	—	—	—	—	17,500	17,500	16,800	16,150	14,850	13,100	7,500	2,800	*1,000*	—	—	—
SA-336	F5	5Cr-½Mo	80,000	—	—	—	—	20,000	19,200	17,300	16,500	14,500	11,500	7,300	3,300	1,500	—	—	—
SA-336	F22	2¼Cr-1Mo	80,000	—	—	—	—	20,000	20,000	20,000	20,000	18,000	14,000	7,800	4,200	2,000	—	—	—
SA-336	F8	18Cr-8Ni	75,000	(7)	18,750	16,000	15,450	—	15,100	14,850	14,800	14,550	14,000	12,500	7,500	4,500	2,450	1,400	750
SA-336	F8	18Cr-8Ni	75,000	—	18,750	15,000	13,650	—	12,500	11,200	10,800	10,000	9,400	8,800	7,500	4,500	2,450	1,400	750
SA-336	F8m	18Cr-8Ni-Mo	75,000	—	18,750	17,900	17,500	—	17,200	17,050	17,000	16,750	16,000	14,000	10,400	6,800	4,000	2,000	1,000
SA-336	F8t	18Cr-8Ni-Ti	75,000	—	18,750	17,000	15,800	—	15,200	14,850	14,800	14,550	14,100	13,500	10,300	5,000	2,200	1,200	750
SA-336	F8c	18Cr-8Ni-Cb	75,000	—	18,750	17,000	15,800	—	15,200	14,850	14,800	14,550	14,100	13,500	10,300	5,000	2,200	1,200	750

TABLE E (Concluded)

Maximum Allowable Stress Values, Pounds per Square Inch

ASME Spec	Grade	Nominal Composition	Spec Min Tensile	Notes	For metal temperatures not exceeding deg F: −20 to 100	300	400	−20 to 400	500	650	700	800	900	1000	1100	1200	1300	1400	1500
CASTINGS																			
Carbon Steel																			
SA-95	—	—	70,000	(1) (5)	—	—	—	17,500	17,500	17,500	16,600	12,000	*5,000*	*1,500*	—	—	—	—	—
SA-216	WCA	—	60,000	(1) (5)	—	—	—	15,000	15,000	15,000	14,350	10,800	*5,000*	*1,500*	—	—	—	—	—
SA-216	WCB	—	70,000	(1) (5)	—	—	—	17,500	17,500	17,500	16,600	12,000	*5,000*	*1,500*	—	—	—	—	—
Alloy Steel																			
SA-157	C9a	18Cr-8Ni	70,000	(5) (7)	17,500	15,600	15,000	—	14,600	14,200	14,050	13,600	13,000	12,100	7,500	4,500	2,450	1,400	750
SA-157	C9a	18Cr-8Ni	70,000	(5)	17,500	14,250	13,100	—	12,200	11,500	11,300	10,900	10,400	9,850	7,500	4,500	2,450	1,400	750
SA-217	WC1	C-½Mo	65,000	(2) (4) (5)	—	—	—	16,250	16,250	16,250	16,250	15,650	*12,500*	*5,500*	—	—	—	—	—
SA-217	WC4	Ni-Cr-½Mo	70,000	(4) (5)	—	—	—	17,500	17,500	17,500	17,500	17,000	14,000	6,250	—	—	—	—	—
SA-217	WC5	Ni-Cr-1Mo	70,000	(4) (5)	—	—	—	17,500	17,500	17,500	17,500	17,000	14,000	7,800	—	—	—	—	—
SA-217	WC6	1¼Cr-½Mo	70,000	(4) (5)	—	—	—	17,500	17,500	17,500	17,500	17,000	14,000	7,800	—	—	—	—	—
SA-217	WC9	2¼Cr-1Mo	70,000	(4) (5)	—	—	—	17,500	17,500	17,500	17,500	17,000	14,000	7,800	—	—	—	—	—
SA-217	C5	5Cr-½Mo	90,000	(4) (5)	—	—	—	22,500	22,500	22,500	21,600	19,000	13,600	7,300	3,300	1,500	—	—	—
SA-217	C12	9Cr-1Mo	90,000	(4) (5)	—	—	—	22,500	22,500	22,500	22,000	19,400	15,000	8,500	3,300	1,500	—	—	—
SA-351	CF8	18Cr-8Ni	70,000	(5) (7)	17,500	15,600	15,000	—	14,600	14,200	14,050	13,600	13,000	12,100	7,500	4,500	2,450	1,400	750
SA-351	CF8	18Cr-8Ni	70,000	(5)	17,500	14,250	13,100	—	12,200	11,500	11,300	10,900	10,400	9,850	7,500	4,500	2,450	1,400	750

GENERAL NOTE: Stress values shown in italic are permissible, but use of these materials at these temperatures is not current practice.

NOTES:

(1) Upon prolonged exposure to temperatures above about 800 F, the carbide phase of carbon steel may be converted to graphite.

(2) Upon prolonged exposure to temperatures above about 875 F, the carbide phase of carbon-molybdenum steel may be converted to graphite.

(3) Only killed steel shall be used above 900 F.

(4) These stress values apply to normalized and tempered material only.

(5) To these stress values a casting quality factor as specified in paragraph P-27, ASME Code, shall be applied.

(6) Flange quality in this specification not permitted above 850 F.

(7) See note 7, Table P-7, ASME Code.

(8) These stress values include a joint efficiency factor of 0.85.

(9) These stress values include a joint efficiency factor of 0.80.

(10) These stress values include a joint efficiency factor of 0.60.

(11) Above 700 F these stress values include a joint efficiency factor of 0.85. When material to this specification is used for pipe, multiply the stress values up to and including 700 F by a factor of 0.85. When material to this specification is used for boiler, water-wall, superheater, and economizer tubes that are enclosed within a setting, the stress values above 700 F up to and including 850 F may be divided by a factor of 0.85.

(12) Tensile value in parentheses is expected minimum.

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TABLE F

Index of Specifications for Ferrous Boiler Materials

Boiler Tubes

ASME Spec	Material	OD Min	OD Max	Wall Thk Min	Wall Thk Max
SA-83	Lap-welded steel	½″	6″	0.035″	0.320″
SA-83	Seamless steel	½″	6″	0.035″	0.320″
SA-83	Lap-welded iron	½″	6″	0.035″	0.320″
SA-178	ERW steel	½″	5″	0.035″	0.320″
SA-178	Open-hearth iron	½″	5″	0.035″	0.320″
SA-192	Seamless (h-p)	½″	7″	0.085″	1.000″

Boiler and Superheater Tubes

ASME Spec	Material	OD Min	OD Max	Wall Thk Min	Wall Thk Max
SA-209	Seamless C-Mo	½″	5″	0.035″	0.500″
SA-210	Med-C seamless	½″	5″	0.035″	0.500″
SA-213*	Seamless alloy	½″	5″	0.035″	0.500″
SA-226	ERW (h-p service)	½″	5″	0.085″	0.360″
SA-249	Welded alloy	½″	5″	0.035″	0.320″
SA-250	ERW C-Mo alloy	½″	5″	0.035″	0.320″

Pipe

ASME Spec	Material	Nom Size Min	Nom Size Max	Wall Thk Min	Wall Thk Max
SA-53	Welded & seamless	⅛″	8″	0.068″	0.875″
		—	12″	—	0.500″
SA-106	Seamless carbon (h-t)	⅛″	24″	0.068″	1.750″
SA-135	ERW steel	⅛″	30″	0.068″	—
SA-158*	Seamless alloy (h-t)	⅛″	24″	0.068″	1.750″
SA-206	Seamless C-Mo alloy (h-t)	⅛″	24″	0.068″	1.750″
SA-280†	Seamless Cr-Mo alloy (h-t)	½″	6″	0.187″	1.562″
SA-312	Austenitic stainless	⅛″	12″	0.049″	0.500″
SA-315‡	Seamless Cr-Mo alloy (h-t)	½″	16″	0.187″	1.562″
SA-335	Seamless ferritic (h-t)	½″	16″	0.187″	1.593″

ASME Grade	Corresponding B&W Grade
P11	Croloy 1¼
P3a	Croloy 1¾
P3b	Croloy 2
P5a*	Croloy 5
P5c	Croloy 5 (Ti or Cb)
P8a	Croloy 18-8S
P8b	Croloy 18-8Ti
†—	Croloy ½
‡—	Croloy 1

ASME Grade	Corresponding B&W Grade
T12	Croloy 1
T11	Croloy 1¼
T14	Croloy 2
T22	Croloy 2¼
T21	Croloy 3M
T5	Croloy 5
T13*	Croloy 5Si
T7	Croloy 7
T9	Croloy 9M
TP304	Croloy 18-8S
TP321	Croloy 18-8Ti
TP347	Croloy 18-8Cb
TP316	Croloy 16-13-3

Castings

ASME Spec	Material	Used For
SA-48	Gray iron	Iron castings
SA-95	Carbon steel	Valves, flanges, and fittings in h-t service
SA-157	Alloy steel	Valves, flanges, and fittings in h-t service
SA-216	Carbon steel	Fusion welding in high-temp service
SA-217	Alloy steel	Fusion welding in high-temp service
SA-351	Stainless steel	Stainless steel castings

Forgings

ASME Spec	Material	Used For
SA-105	Steel	Forged or rolled pipe flanges, forged fittings, valves, & parts for h-t
SA-182	Alloy steel	Forged or rolled pipe flanges, forged fittings, valves, & parts for h-t
SA-266	Carbon steel	Seamless drum forgings
SA-336	Alloy steel	Seamless drum forgings

Plates—Boilers & Other Pressure Vessels

ASME Spec	Material	Quality & Grade	Max Thk
SA-201	C-Si steel, fusion welded, usual tensile range	Flange, A, B	2″
		Firebox, A	12″
		Firebox, B	8″
SA-202	Cr-Mn-Si alloy	Firebox, all grades	2″
SA-203	Nickel steel	Flange, all grades	2″
		Firebox, A, B	6″
		Firebox, C, D, E	4″
SA-204	Molybdenum steel	Flange, all grades	2″
		Firebox, A, B	6″
		Firebox, C	4″
SA-212	High-tensile C-Si	Flange, all grades	2″
		Firebox, all grades	6″
SA-285	Carbon steel, low & inter tensile	Flange, Firebox, all grades	2″
SA-299	High-tensile C-Mn-Si	Flange, Firebox	2″
SA-301	Cr-molybdenum	Flange, all grades	2″
		Firebox, all grades	6″
SA-302	Mn-molybdenum	Flange, A, B	2″
		Firebox, A, B	4″

Bolting and Nuts

ASME Spec	Material	Used For
SA-193	Alloy steel	Bolting in h-t service
SA-194	Carbon & alloy steel	Nuts for bolts in high-press. & temp service
SA-306	Med-carbon steel	Bars subject to mechanical property requirements

GLOSSARY

Symbols and Abbreviations

SYMBOLS

a — acceleration, ft/sec² or ft/hr²
A — area, usually cross-sectional, sq ft

b — width dimension, in. or ft
B — barometer reading, in. of mercury

c — specific heat, Btu/lb, F — hotter fluid
c' — specific heat, Btu/lb, F — colder fluid
c_p — specific heat, Btu/lb, F — at constant pressure
c_v — specific heat, Btu/lb, F — at constant volume
C — a coefficient
C_c — coefficient of contraction
C_d — drag coefficient
C_q — coefficient of discharge, $= C_c \times C_v$
C_{qo} — coefficient of discharge, with zero approach velocity
C_v — coefficient of velocity

d — diameter, in.; depth of duct, in.; differential operator
d_e — equivalent diameter, in.
d_i — internal diameter, in.
d_o — external diameter, in.
d_t — throat or orifice diameter, in.
D — diameter, ft
D_e — equivalent diameter, ft
D_i — internal diameter, ft
D_o — external diameter, ft

E — energy, Btu or ft-lb

f — friction factor (Moody) $= 4f'$
f' — friction factor (Fanning)
F — force, lb; also used for factor
F_a — arrangement factor, depending upon geometry
F_c — correction factor
F_d — depth factor
F_ϵ — emissivity factor
F_{LF} — laminar flow factor
F_{pp} — physical properties factor
F_s — surface factor
F_T — temperature factor

g — gravitational acceleration, 32.2 ft/sec² or 4.17×10^8 ft/hr²
G — mass flow, lb/sq ft, hr $= w/A = \rho V$

h — enthalpy of superheated steam, Btu/lb
h_a — enthalpy of air, Btu/lb
h_f — enthalpy of saturated liquid, Btu/lb
h_{fg} — enthalpy of evaporation, Btu/lb
h_g — enthalpy of saturated vapor or flue gas, Btu/lb
h_w — enthalpy of feedwater, Btu/lb

h_{ac} — head lost accelerating fluid, ft-lb/lb = ft
h_b — head lost in bends, ft-lb/lb = ft
h_{en} — head lost at entrance, ft-lb/lb = ft
h_{ex} — head lost at exit, ft-lb/lb = ft
h_f — head lost in friction, ft-lb/lb = ft
h_k — head lost at change of section, ft-lb/lb = ft
h_p — pressure head $= pv$, ft-lb/lb = ft
h_s — head lost through separator or baffle, ft-lb/lb = ft
h_v — velocity head $= V^2/2g$, ft-lb/lb = ft
h_z — elevation head $= Z$, ft
$h_{\Sigma d}$ — sum of downcomer head losses, ft-lb/lb = ft
$h_{\Sigma r}$ — sum of riser head losses, ft-lb/lb = ft
H — net head causing flow $= \Delta h_p + \Delta h_v$, ft-lb/lb = ft
H_{ft} — total head of fluid flowing $= h_p + h_v$, ft-lb/lb = ft
H_{SP} — static pressure head as measured by Pitot-static tube, in. of water
H_{TP} — total pressure head as measured by Pitot tube, in. of water
H_w — head as measured by manometer, in. of water

k — conductivity, Btu/sq ft, hr, F/in.; reactivity, % of multiplication factor (nuclear)
K — a factor
K_f — fouling factor
K_{Re} — Reynolds number factor

l — length, in.
$l_\perp$ — tube pitch transverse to flow stream, in.
$l_\parallel$ — tube pitch longitudinal to flow stream, in.
L — length, ft

m — mass $= W_L/g$, slugs
M — molecular weight of element or compound

N — a number
N_b — number of velocity heads in bends
N_e — number of velocity heads for entrance and exit
N_{Re} — Reynolds number $= \rho VD/\mu = GD/\mu$

p — pressure, lb/sq in. or lb/sq ft (absolute)
p' — pressure, in. of mercury (absolute)
p_c — critical pressure of expansive fluids, lb/sq in. or lb/sq ft (absolute)
p_r — partial pressure, atmospheres
Δp — pressure drop, lb/sq in. or lb/sq ft
Δp_b — pressure drop in bends, lb/sq in. or lb/sq ft
Δp_e — pressure drop from entrance and exit, lb/sq in. or lb/sq ft
Δp_f — pressure drop from friction, lb/sq in. or lb/sq ft
P — contact perimeter, in. or ft

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SYMBOLS

q — rate of heat flow, Btu/hr
Q — heat in fluid, Btu/lb; heat of chemical reaction; thermal current, Btu/sq ft, hr; volume rate of flow, cu ft/sec, cu ft/min, or cu ft/hr
Q_H — high heat value of fuel, Btu/lb
Q_L — low heat value of fuel, Btu/lb

r — radius, in. or ft
R — over-all or combined resistance of heat flow path $= 1/U$, sq ft, hr, F/Btu
R_f — heat flow resistance caused by fouling, sq ft, hr, F/Btu
$\boldsymbol{R}$ — universal gas constant = 1544
$\boldsymbol{R}_g$ — specific gas constant $= 1544/M$

s — entropy, Btu/lb, F
S — maximum allowable stress values, lb/sq in.; surface area, sq ft

t — temperature, F — of hotter fluid, or at a point
t' — temperature, F — of colder fluid
Δt — temperature difference, F
Δt_m — logarithmic mean temperature difference, F
T — absolute temperature, R = F + 460

u — internal energy of fluid, Btu/lb
U — over-all or combined conductance $= 1/R$, Btu/sq ft, hr, F
U_c — convection conductance general, Btu/sq ft, hr, F
U_{ca} — convection conductance of air, Btu/sq ft, hr, F
U_{cc} — convection conductance in crossflow, Btu/sq ft, hr, F
U_{cg} — convection conductance of gas, Btu/sq ft, hr, F
U_{cl} — convection conductance in longitudinal flow, Btu/sq ft, hr, F
U_{cs} — convection conductance of colder fluid, Btu/sq ft, hr, F
U_d — conduction conductance, Btu/sq ft, hr, F
U_{fc} — free-convection conductance, Btu/sq ft, hr, F
U_g — over-all gas-side conductance $= U_{rg} + U_{cg}$, Btu/sq ft, hr, F
U_r — radiant conductance, Btu/sq ft, hr, F
U_{rg} — radiant conductance of gas, Btu/sq ft, hr, F

v — specific volume, cu ft/lb
V — velocity, ft/sec, ft/min, or ft/hr
V_L — total volume, cu ft

w — lb water formed in burning one lb fuel
w — weight rate of flow, lb/sec or lb/hr
W — radiant flux density, Btu/sq ft, hr
W_{PP} — feed-pump work, ft-lb
W_k — work, ft-lb
W_L — total weight, lb

x — length dimension (usually horizontal), in. or ft

y — depth dimension (usually vertical), in. or ft

Z — elevation above datum, ft

α — absorptivity; area ratio — small to large

β — degrees of included angle

γ — specific weight, lb/cu ft $= 1/v = \rho g$

Δ — a difference

ϵ — average height of roughness projections, in.; emissivity of surface, between 0 and 1.0

κ — ratio $= c_p/c_v$

μ — absolute viscosity of fluid, lb (mass)/ft, hr also lb, sec/sq ft

π — 3.1416

2π — number of steradians in a hemispherical solid angle

ρ — mass density, lb (mass)/cu ft = slugs/cu ft; reflectivity

σ — Stefan-Boltzmann constant, 1.73×10^{-9} Btu/sq ft, hr, T^4

τ — shear stress; transmissivity

ϕ — a function

ω — solid angle intercepted by one body as seen from another, measured in steradians

SUBSCRIPTS

a — actual; air; arrangement

b — bank; bend; boiler; bulk
B — bottom

c — contraction; convection; critical or acoustic
ca — convection air
cc — convection crossflow
cg — convection gas
ci — convection inside
cl — convection longitudinal flow
co — convection outside
cs — convection colder fluid
CF — colder fluid

d — conduction; depth; downcomer; drag
D — developed

SUBSCRIPTS

e — enlargement; entrance and exit; equivalent
en — entrance
ex — exit

f — friction; resistance
f — film; fouling; saturated liquid
fc — free convection
ff — fluid flowing
fg — evaporation
F — fuel
FP — feed pump

g — gas; sat. vapor

H — high heat value
HF — hotter fluid

i — ideal; inner; inside; internal
I — inertial

k — change of section; work

L — low heat value; total
LF — laminar flow

m — logarithmic mean
mf — manometer fluid
M — molecular weight

o — external; outer; outside

p — particle; pressure; projected
pc — pure coal
pp — physical properties

q — discharge
qo — discharge with zero approach velocity

r — partial; radiation; riser
rg — radiation gas
R — refractory
Re — Reynolds

s — separator; steam; surface or colder fluid
sf — superposed fluid
SP — static pressure

t — orifice; throat
tw — tube wall
T — temperature; top
TP — total pressure

v — velocity; volume

w — feedwater; water

z — elevation

ϵ — emissivity

Σ — summation
Σd — summation downcomer
Σr — summation riser

$\perp$ — perpendicular; transverse

$\parallel$ — parallel

ABBREVIATIONS

A — ash
ABMA — American Boiler Manufacturers Association
abs — absolute
a-c — alternating-current
AEC — Atomic Energy Commission
AF — as-fired
a.m. — ante meridiem
amb — ambient
amp — ampere
anal. — analysis
API — American Petroleum Institute
approx — approximately
ASA — American Standards Association
ASME — American Society of Mechanical Engineers
ASSN — association
ASTM — American Society for Testing Materials
atm — atmosphere
attemp — attemperator
aux — auxiliary
avail. — available
avg — average

B&S — Brown & Sharpe wire gage
B&W — Babcock & Wilcox
bbl — barrel
BCC — body-centered cubic
bd — board
bet. — between
bhp — brake horsepower
bitum — bituminous
blr — boiler
blr-hp — boiler horsepower
blr-hp$_D$ — developed boiler horsepower
Btu — British thermal unit
BWG — Birmingham Wire Gage

C — carbon; centigrade degrees
C-A — chromel to alumel
Cal — calorie
cc — cubic centimeters
cf — compare
CF — colder fluid
cfm — cubic feet per minute
cfs — cubic feet per second

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ABBREVIATIONS

cir — circular
℄ — center line
cm — centimeter
cnds — condensate
Co. — company
coef — coefficient
col — column
comp — compound
cond — condenser
conn — connection
constit — constituent
cont — continuous
cont'd — continued
corp — corporation
cu — cubic
cyl — cylinder

d-c — direct-current
def — definition
deg — degree
den. — density
dept — department
dev — developed
diag — diagram
diam — diameter
diff — difference

e — Napierian log base = 2.718
ea — each
econ — economizer
Ed. — editor
eff — efficiency
e.g. — for example
elong — elongation
emf — electromotive force
eng — engine
ent — entering
Eq — equation
equiv — equivalent
ERW — electric-resistance-welded (tubes)
evap — evaporation; evaporator

F — fahrenheit degrees
FB — firebrick
FC — fixed carbon
FCC — face-centered cubic; Federal Commerce Commission
f'ce — furnace
F-D — forced-draft
ff — following (pages)
F.G. — flue gas
Fig. — figure
fob — free on board
fpm — feet per minute
fps — feet per second
ft — feet
ft-lb — foot-pound
F.T. — fluid temperature (ash)

g — gram
ga — gage
gal — gallon
gen — generator
gpm — gallons per minute
gr — gravity
guar — guaranteed

ha — hectare
HF — hotter fluid
Hg — mercury
hp — horsepower
h-p — high-pressure
hp-hr — horsepower-hour
hr — hour
HR — heat rate
HRT — horizontal-return-tubular (boiler)
ht — heat
h-t — high-temperature
humid. — humidity
HVT — high-velocity thermocouple

I-C — iron to constantan
ID — inside diameter
I.D. — initial-deformation temperature (ash)
I-D — induced-draft
i.e. — that is
IFB — insulating firebrick
illus — illustration
in. — inch
Inc — Incorporated
incl — include
insp — inspection
inter — intermediate
IPS — international pipe size

JR — junior

k — kilo; thousand
K — Kelvin degrees
kB — thousand British thermal units
kg — kilogram
km — kilometer
kw — kilowatt
kwhr — kilowatthour
kwt — kilowatt thermal

lb — pound
lg — length; long
liq — liquid
log — logarithm
l-p — low-pressure
l-t — low-temperature
Ltd — Limited
lv — leaving

m — meter
M — moisture; thousand
mag — magnetic

ABBREVIATIONS

max — maximum
med — medium
MEP — mean effective pressure
Mev — million electron volts
MHVT — multiple-shield high-velocity thermocouple
min — minimum; minute
MkB — million British thermal units
Mm — mineral matter
mol. wt — molecular weight
mp — melting point
mph — miles per hour
mr — milliroentgen
M.T. — million tons
mult — multiply
mv — millivolt
mw — megawatt
mwd — megawatt days

nat — natural
natl — national
No. — number
nom — nominal
NP — National Pipe (thread)
NRTS — National Reactor Testing Station
N.S. — nuclear ship
NSSC — Neutral Sulfite Semichemical
nv — thermal flux (nuclear)

OD — outside diameter
OP — Open-Pass (boiler)
oz — ounce

PC — pulverized coal
p-c — pulverized-coal
PCE — pyrometric cone equivalent
PCO — plastic chrome ore
pH — negative log of hydrogen ion concentration
p.m. — post meridiem
pp — pages
ppm — parts per million
press. — pressure
pri — primary
proj — projected
prox — proximate
psi — gage pressure
psia — absolute pressure
PWR — pressurized-water reactor

R — Rankine degrees
R_b — boiler rating
RB — Radiant boiler
recirc — recirculating
reg — registered; regulating
rel — relative
rem — roentgen equivalent, man
reqd — required
rev — revolutions
rpm — revolutions per minute

S. — south
sat. — saturated; saturation
SBV — steam by volume
SBW — steam by weight
sec — second; secondary
sect — section
sens — sensible
SH — superheater
shp — shaft horsepower
spec — specification
sp gr — specific gravity
sp ht — specific heat
sp vol — specific volume
sp wt — specific weight
sq — square
S.S. — steamship
SSCR — spectral-shift control reactor
SSF — Seconds Saybolt Furol
SST — stainless steel
SSU — Seconds Saybolt Universal
S.T. — softening temperature (ash)
std — standard
SX — Sectional Express (boiler)

T.A. — total air
TC — thermocouple
temp — temperature
theo — theoretical
thk — thick; thickness
thru — through
tot — total
tr — trace
TS — tensile strength
TTD — terminal temperature difference

ult — ultimate
undeter. — undetermined
UP — Universal-Pressure (boiler)
U.S. — United States
U.S.S. — United States Ship
U.S.S.R. — Union of Soviet Socialist Republics

vel — velocity
VM — volatile matter
vol — volume
vs — versus

whp — water horsepower
WR — water rate
wt — weight

yd — yard
YP — yield point
yr — year
YS — yield strength

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