MARINE STEAM



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MARINE STEAM



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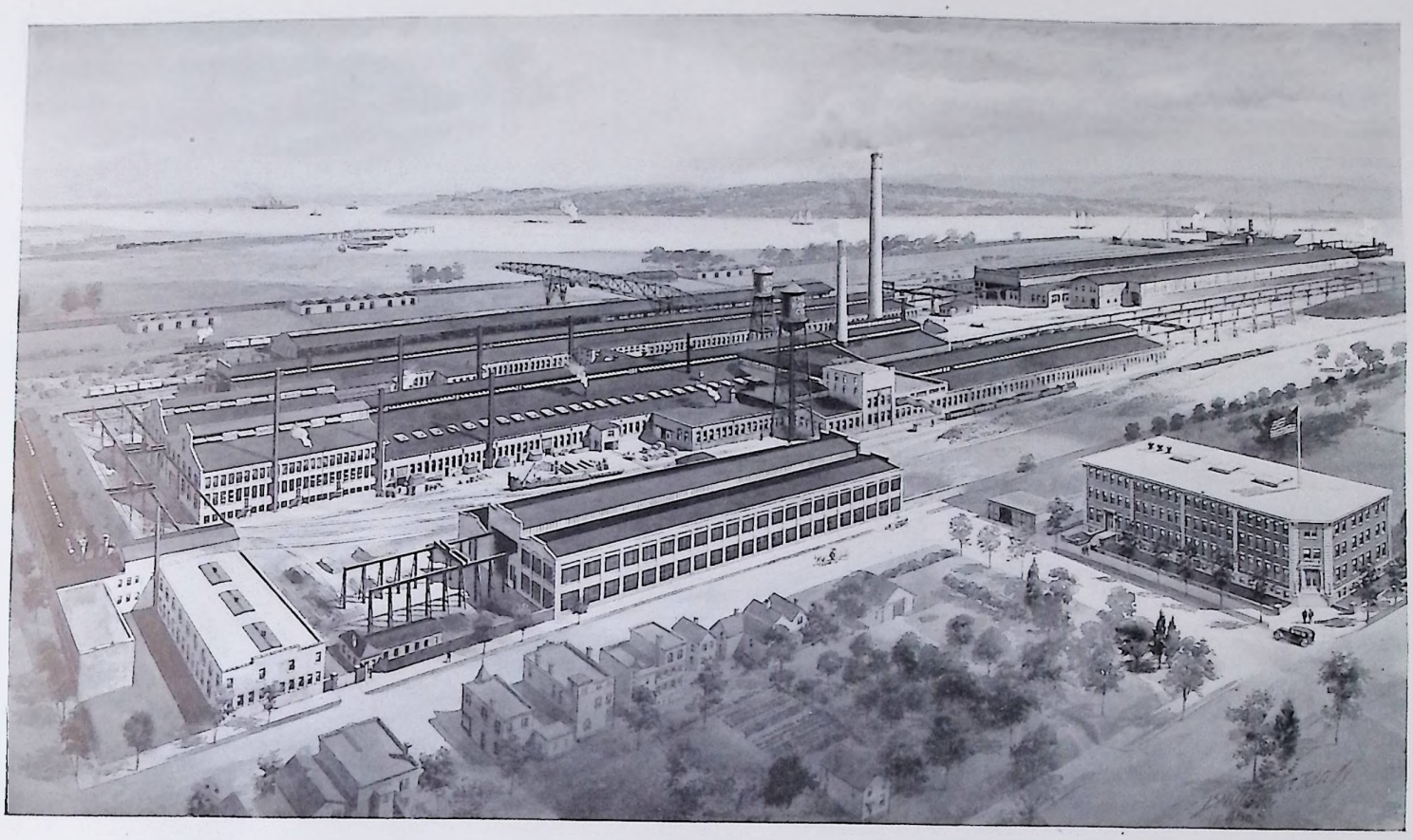
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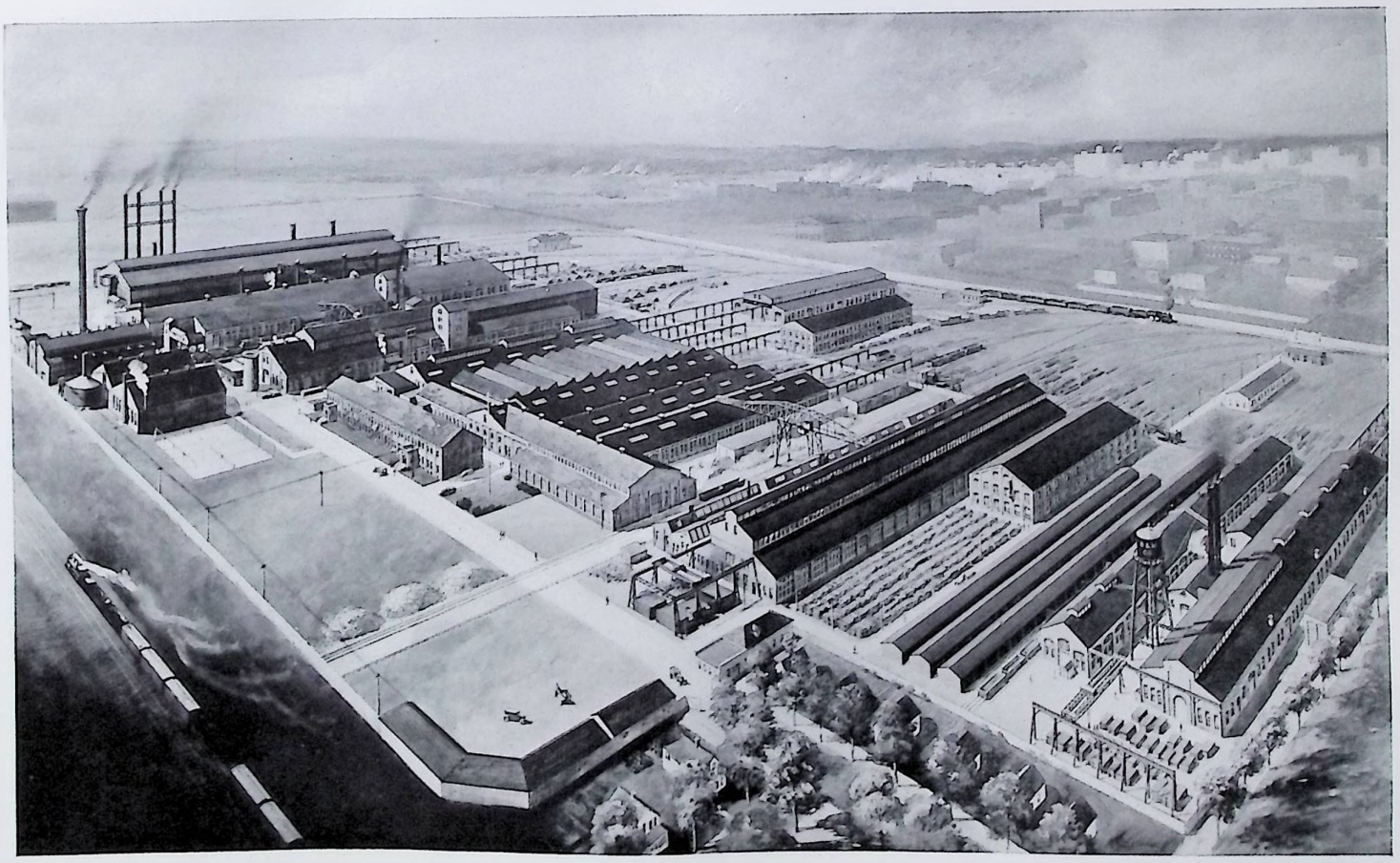
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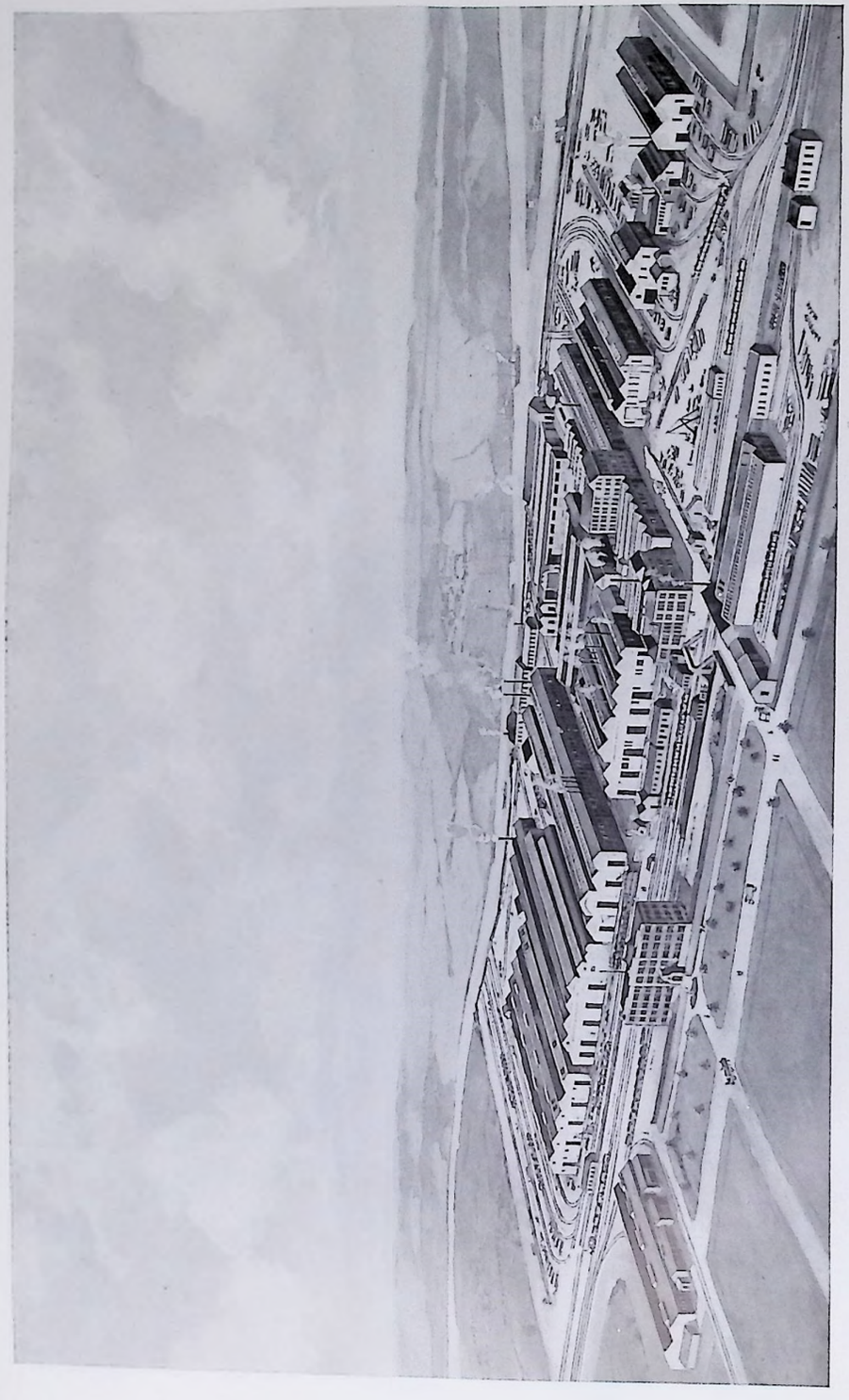
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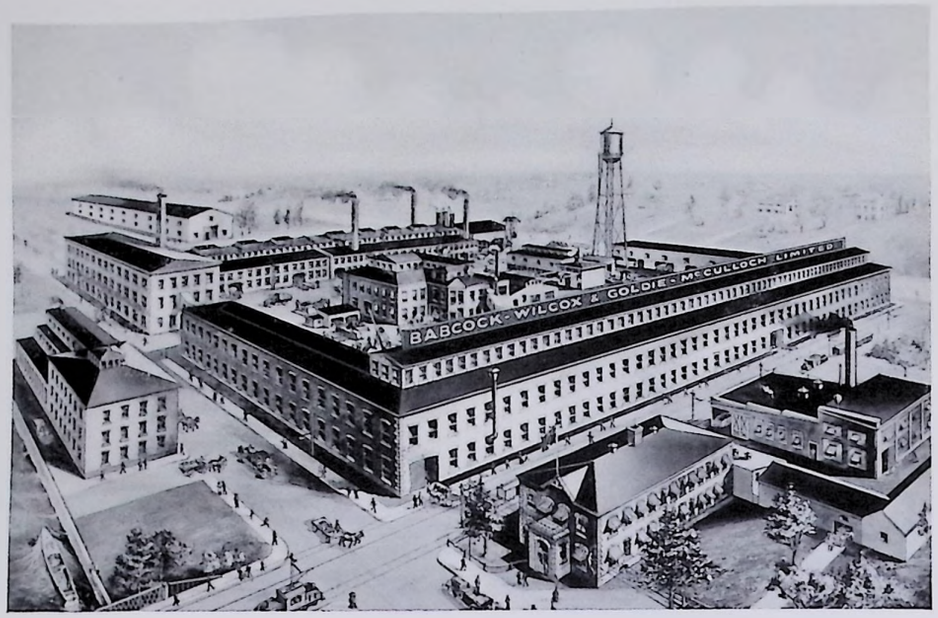
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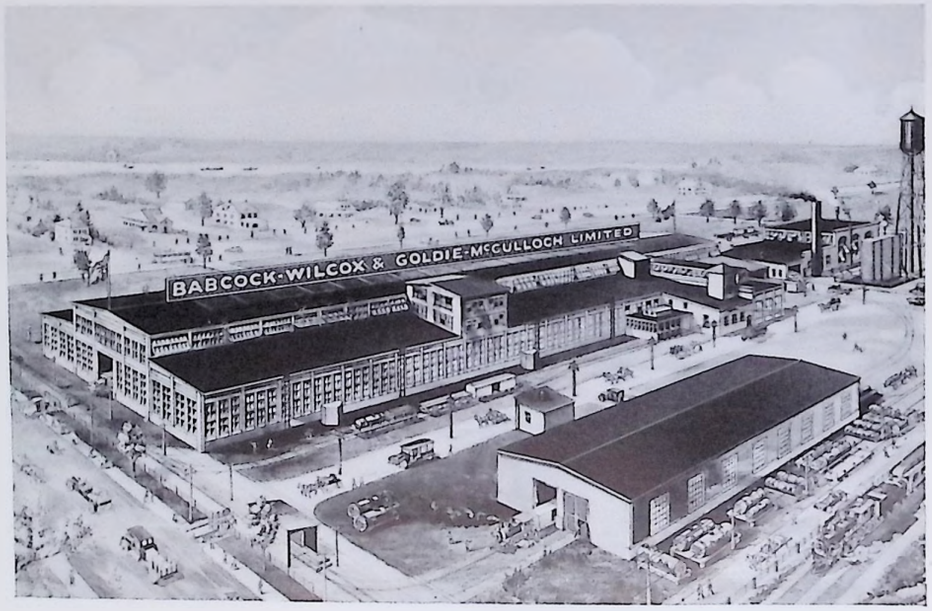
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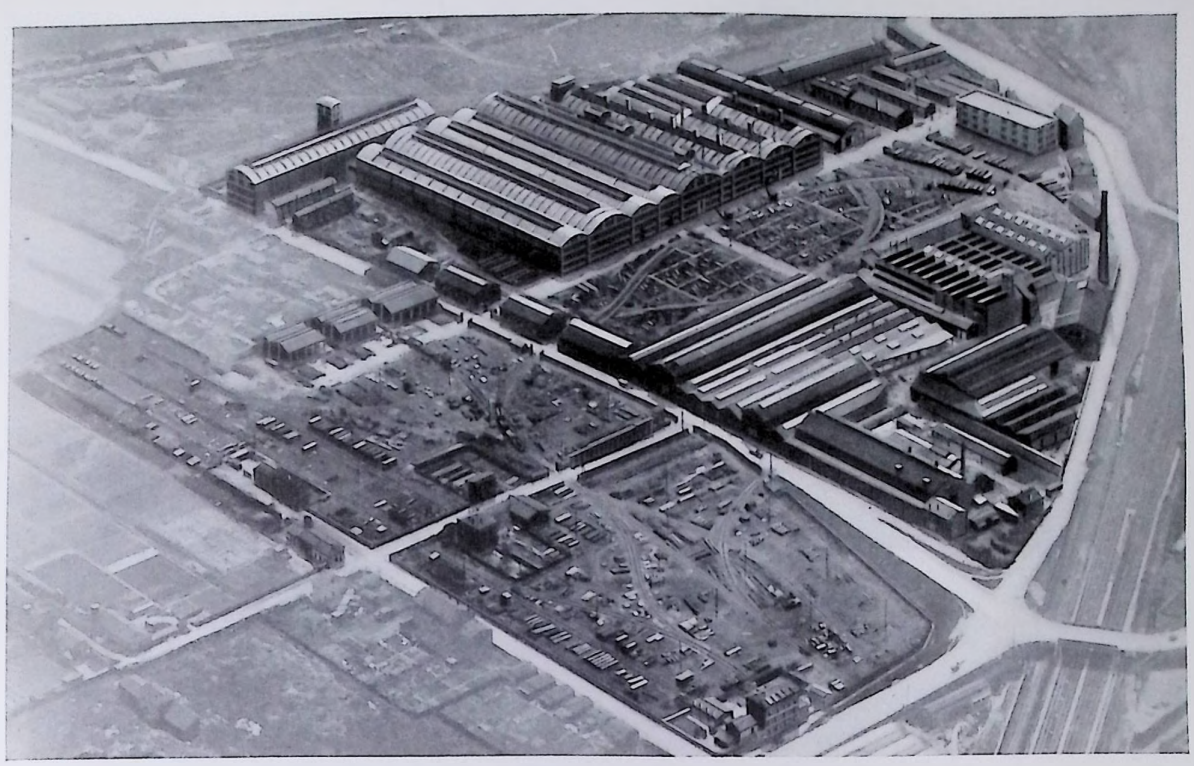
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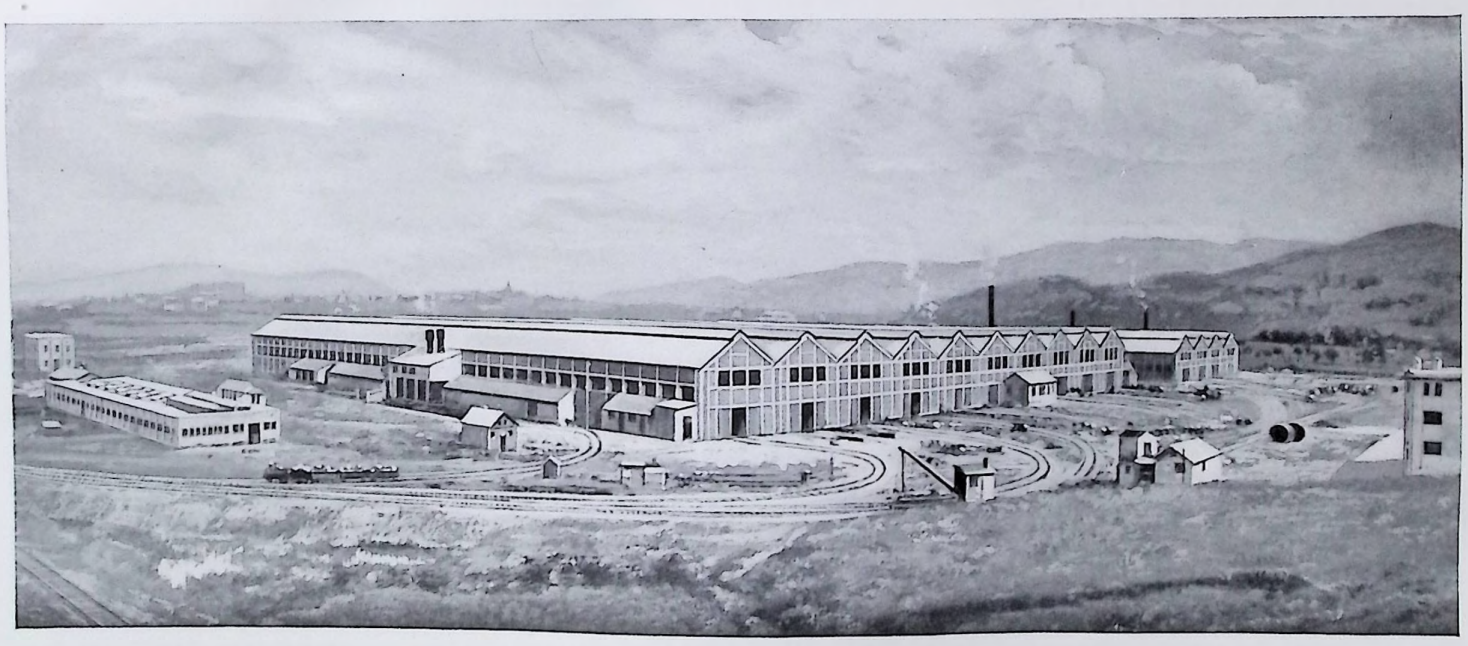
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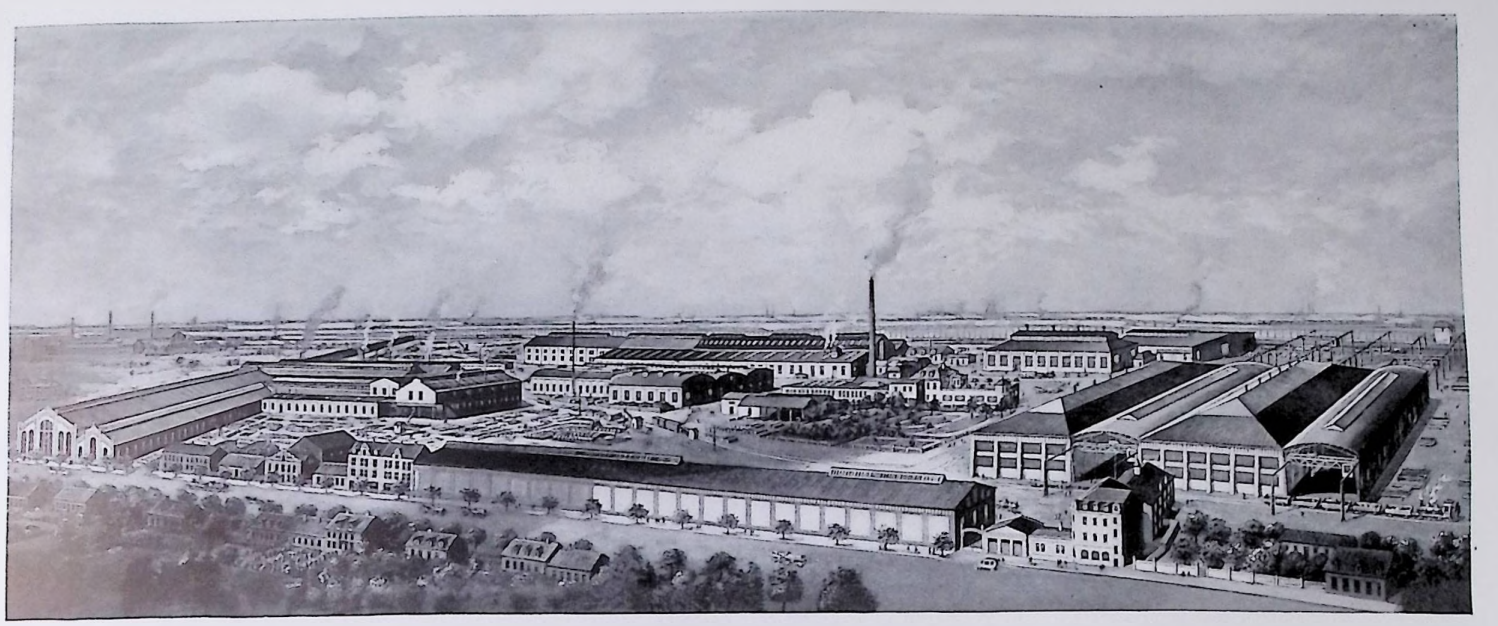
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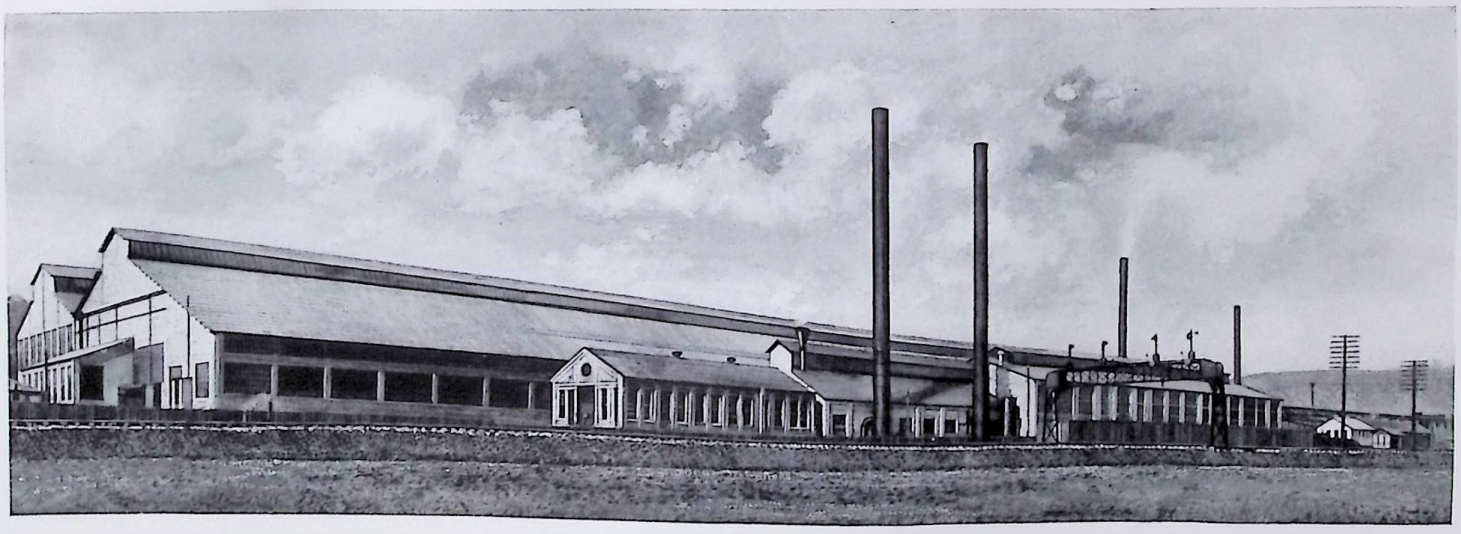
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The works of this company were established in 1901. In addition to making all the tubes used by The Babcock & Wilcox Company in boilers, superheaters, economizers and air heaters, the mills supply seamless steel tubes and pipe up to 5.5 inches outside diameter for general uses.

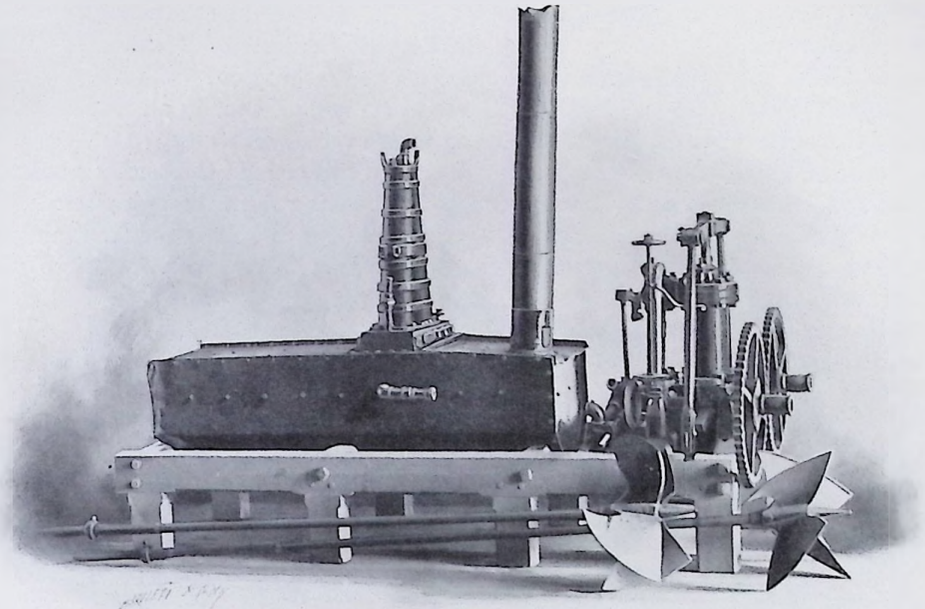


U. S. Airplane Carrier "Saratoga," 180,000 Shaft Horse-power, Babcock & Wilcox Express Type Boilers, Superheaters and Oil Burners

A BRIEF HISTORY OF THE MARINE WATER-TUBE BOILER

THE first boiler made of a combination of small tubes was a marine boiler invented by Col. John Stevens. This boiler was installed in a steamboat 68 feet long and 14 feet wide, operated in 1804 on the Hudson River.

The machinery of this vessel consisted of a single vertical cylinder whose piston rod, moving a cross head up and down, drove two cranks by means of connecting rods. From the cranks a pair of shafts led aft and were fitted with twin screws.

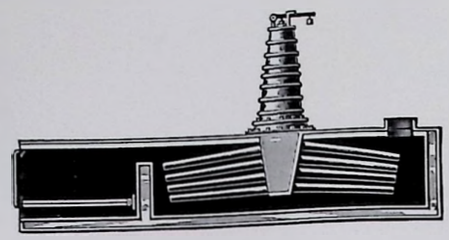


Machinery of Stevens Boat, 1804

The boiler supplying steam to this engine contained 100 tubes, 2 inches diameter and 18 inches long. One end of each tube was fastened to a central water leg, the other end being closed.

It is of interest to note that, as early as this, the advantage of small parts was appreciated. In the patent specification of this boiler, Colonel Stevens says: "The principle of this invention consists of forming a boiler by means of a system or combination of small vessels, instead of using, as in the common mode, one large one; the relative strength of the materials of which these vessels are composed increasing in proportion to the diminution of capacity."

In 1805, John Cox Stevens, the eldest son of Colonel Stevens, appreciating that a boiler made up of dead-end tubes did not have the elements of continued success, patented another type of water-tube boiler. This boiler consisted of 20 vertical tubes,

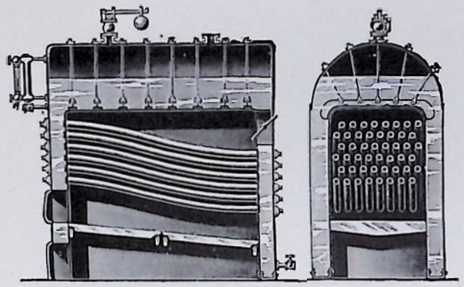


STEVENS, 1804

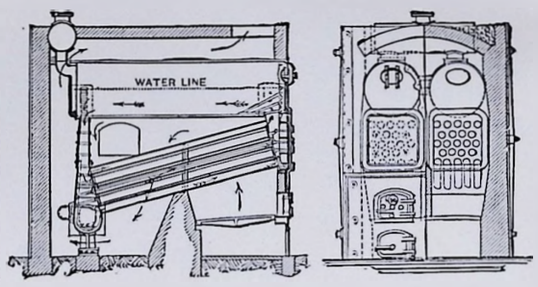
1¼ inches internal diameter and 40½ inches long, arranged in a circle, the outside diameter of which was approximately 12 inches, connecting a water chamber at the bottom to a steam chamber at the top. The steam and water chambers were annular spaces of small cross-section and contained approximately 33 cubic inches, the caps of these chambers being held on by bolts. The steam outlet pipe was 1 inch in diameter, the feed water entering through a similar opening at the bottom and the circulation was forced by the feed pump. One of these boilers was for a long time at the Stevens Institute of Technology at Hoboken and is now in the Smithsonian Institution at Washington.

From the time of the Stevens until 1849 or 1850, practically all marine boilers were of the box and, later, return tubular or Scotch type.

In 1856 Stephen Wilcox improved the design of the John Cox Stevens boiler by giving to the bank of tubes an inclination and connecting them through water legs at the front and rear to an overhead steam-drum, the sides as well as the front and rear headers being of the water-leg construction. This did away with the forced circulation, but contained the objection of flat-stayed surfaces. The side water legs in the Babcock & Wilcox design were replaced in 1868 by brick settings, though this design still had stayed surfaces opposite the tube ends.



WILCOX, 1856



BABCOCK & WILCOX, 1868

While neither the 1856 or 1868 designs were ever used in marine work, they included many principles of design later incorporated in the Babcock & Wilcox marine boiler.

The earliest patent granted to Babcock & Wilcox on a boiler which they believed might be used for marine purposes covered one of the box coil type. In this boiler the water was made to traverse the furnace several times before being delivered to the drum above. The tendency here, as in all similar boilers, was to form steam in the middle of the coil and blow the water from each end, leaving the tubes practically dry until the steam found an outlet and the water returned. This boiler, besides having a defective circulation, had a decidedly geyser-like action and produced wet steam.

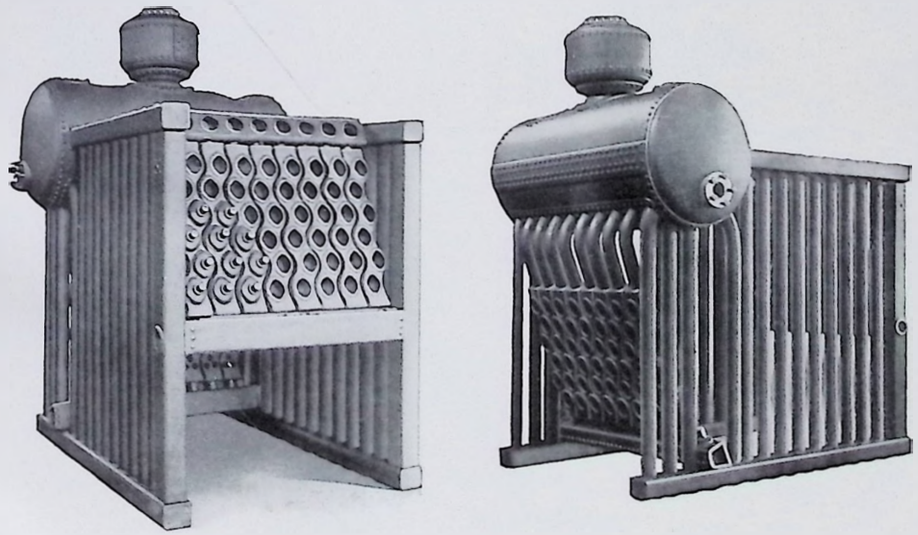
The first marine boiler actually built by Babcock & Wilcox was constructed in 1875 for the "Monroe," owned by the Quartermaster's Department, U.S.A. This

boiler, rated at 40 horse-power, was a combination fire- and water-tube boiler. The upper portion of the boiler was a fire-tube boiler with a combustion chamber at the back, as in Scotch marine boilers. Water legs were carried down at the front and back, being connected by a bank of water tubes inclined downward to the rear of the furnace. Three boilers of this design were built, the largest having two 48-inch drums and containing 1420 square feet of heating surface.

In order to eliminate flat stayed surfaces in boiler construction, the water legs which necessitated such construction were replaced in 1873 by the now familiar sinuous header construction, this being the first real move toward sectionalization. This sectional construction was first used in stationary work, though two boilers utilizing it were installed in the Clyde Line steamer "Benefactor." The headers in this design were at first of cast steel and later of cast iron.

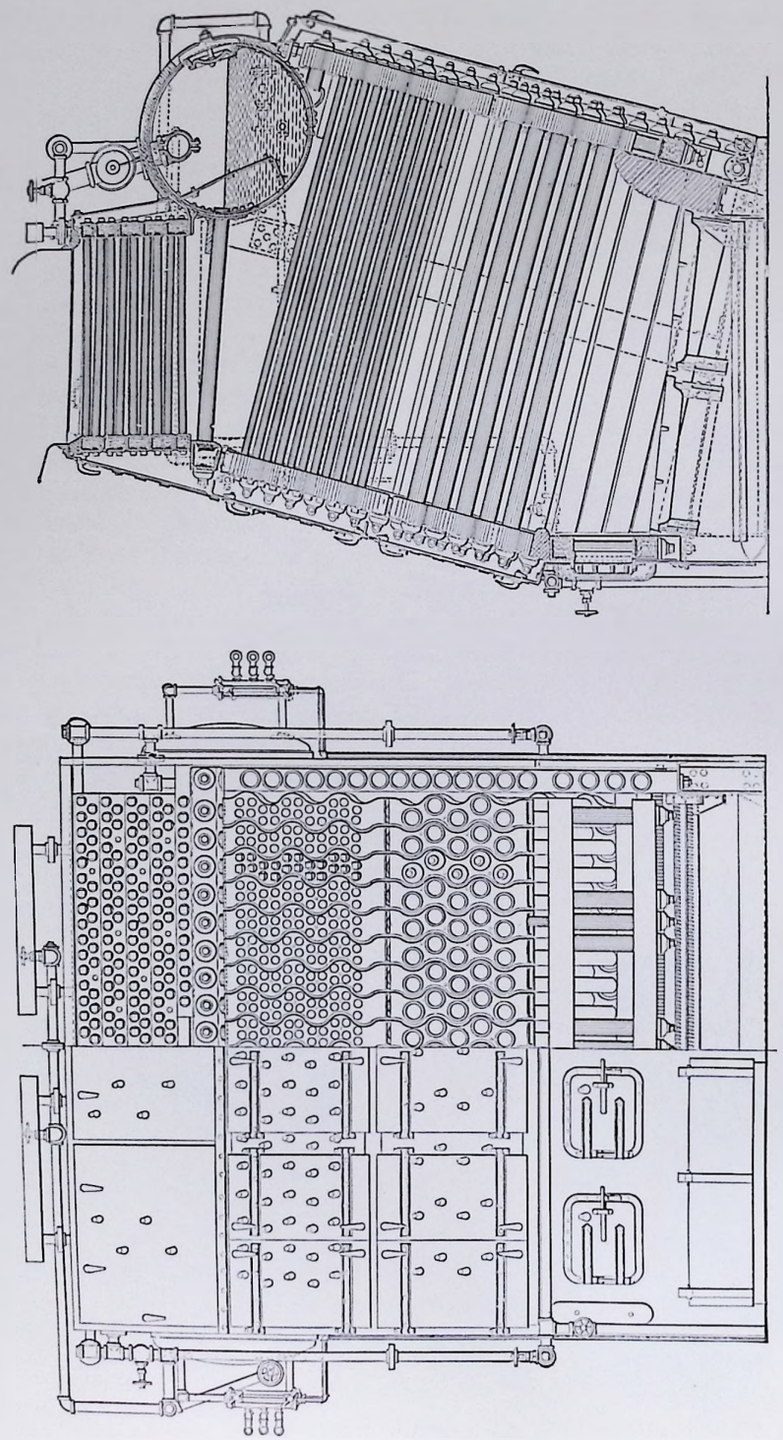
The cross-drum type of Babcock & Wilcox boiler was originally designed for low headroom conditions in stationary practice. This design, however, in the United States, was first used in marine work, the cast-iron headers of the stationary design being replaced by wrought-steel headers, the tubes shortened to meet marine space conditions, and vertical side tubes with light fire-tile casing being substituted for the brick setting, to reduce the weight. The steam drum extended across the rear of the boiler over the downtake circulators and there was a mud drum attached to all sections at the rear. The general construction is clearly shown in the illustration.

The first boiler of this design was built in 1889 for the steam yacht "Reverie." It contained 800 square feet of heating surface, 28 square feet of grate surface and supplied steam at 225 pounds pressure to a 250 horse-power quadruple expansion engine having cylinders 8, 11, 16½ and 26 inches in diameter by 12-inch stroke. This design received the early and favorable attention of Thomas Wilson & Sons, shipowners and operators of Hull, England, and, in 1891, a boiler containing 2263 square feet of heating surface and 53 square feet of grate surface was furnished to that concern for the S.S. "Nero." This installation was so successful that many other



"REVERIE" BOILER, 1889

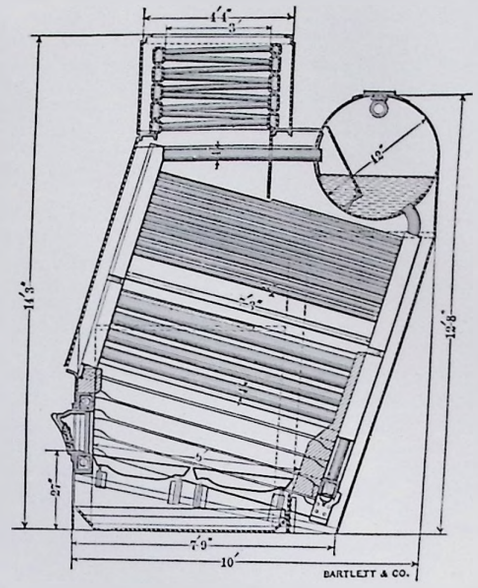
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Design of Babcock & Wilcox Water-tube Boilers for Lake Freight Steamer "Zenith City," 1895—Patented

installations were made for the same company, and a member of that firm, who was in Parliament, became one of the staunchest supporters of the British Admiralty when it determined to use Babcock & Wilcox boilers in the British Navy.

In 1895 the "Reverie" design was modified to increase accessibility. The vertical side tubes were replaced by forged steel boxes at the sides of the furnace, with tubes above the boxes forming the outer sections of the unit, both boxes and tubes being inclined at the same angle as the tubes of the boiler. One or more of the tubes at the bottom of the sections were ordinarily of larger diameter than those in the body of the boiler. Horizontal baffles were used, though in a few instances these were vertical. In a number of installations economizers were placed in the boiler uptake leading to the stack.



Babcock & Wilcox, 1895—Patented

The first firm foothold for Babcock & Wilcox boilers in the merchant marine was gained by boilers of this type. Realizing the full value of apparatus which would reduce the weight, space occupied and cost of operation, without reduction in power, the Zenith Transit Company, now the Pittsburgh Steamship Company, purchased, in 1895, the first two boilers of this type for the "Zenith City."

In 1896 the United States Navy Department ordered boilers of the "Reverie" type, as modified in 1895, for the "Marietta," "Annapolis" and "Chicago." While the boilers in these ships gave more satisfactory service than other boilers previously used by the Navy and saved approximately 30 per cent in weight over the earlier types, they were not entirely satisfactory from the standpoint of efficient combustion. The furnace, lower at the back than at the front, did not give a form of furnace that would result in good combustion conditions when the boilers were driven, and the grates, sloping downward to the rear to obtain furnace volume, made firing and ash handling difficult. Further, because of the shallow ash-pit at the rear, there was a tendency toward overheated grate bars. The steam drums and mud drums were at the rear of the boiler, making the drum fittings not readily accessible and awkwardly located.

While the use of Babcock & Wilcox boilers in the United States Navy was assured by the results obtained on these three vessels, the next change in design, while seemingly simple, in reality revolutionized to a great degree marine water-tube boiler practice. This consisted in firing the boiler under the downtake, instead of the uptake, headers, making what had previously been the rear of the boiler the front. This resulted in a furnace expanding in the direction in which combustion took place and led to much better combustion conditions. It brought the drum to the front, where the fittings were accessible and the water gauge where it could be seen without the use of long connections; made it possible to readily clean and replace the tubes, and allowed the grates to be properly located. This design was patented in 1897 by William D. Hoxie.

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While the modification was, as stated, seemingly simple, it was actually basic in invention, as indicated by the decision of the courts in upholding the patent in a suit resulting from what was perhaps an inevitable infringement following the success of this boiler.

This decision, handed down by Judge W. M. Lanning, says in part:

> The Hoxie patent seems almost to have revolutionized the art of constructing boilers for marine uses. Since 1897, Hoxie boilers have been installed in vessels of the United States Navy aggregating 497,946 horse-power and in vessels of the Navy of Great Britain aggregating 291,850 horse-power.* In the vessels of all countries the Hoxie boilers ordered or installed now aggregate nearly 1,000,000 horse-power. The Hoxie boiler combines compactness with safety, efficiency, durability and great convenience in cleaning and repairing. It was introduced in the Navy of the United States by Admiral Melville, while he was engineer-in-chief of the Navy, after examination of many other types of boilers offered to the Navy Department, and was selected by him because he deemed it "the best boiler of the class to be installed in the ships of the Navy of the United States." He declares that it has given "perfect satisfaction so far as the work of man can go," and that Hoxie was the first to solve the efforts of a hundred years to construct a light, safe and economical marine boiler. Previous to 1897 the type of boiler principally used in the larger vessels of the United States Navy was the fire-tube boiler. Between 1891 and 1897 a controversy had been carried on in engineering periodicals concerning the relative merits for marine uses of fire-tube boilers and water-tube boilers. This controversy became known as "the Battle of the Boilers." These facts show how strenuous were the efforts before the date of the Hoxie patent to invent an improved type of marine boiler. In that battle Hoxie was the victor. The simplicity of the invention may cause surprise that the battle was so long a one. But the fact that there was such a battle is conclusive proof that the combination described in the Hoxie patent was not obvious.

This decision is of particular interest in that it gives a wholly impartial and disinterested estimate of the value of The Babcock & Wilcox Company's major engineering contribution to our Navy's development in particular, and to marine engineering in general.

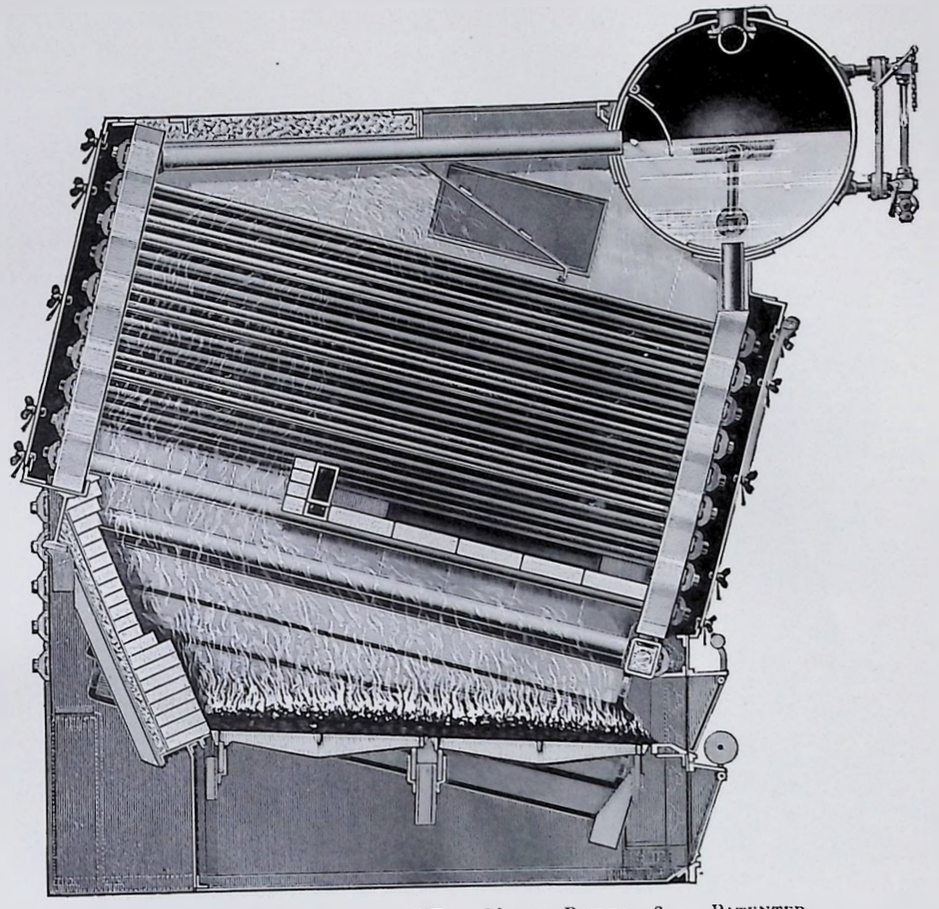
In 1899, this design was further improved by lengthening the tubes, thus increasing the furnace volume and, for a given heating surface, reducing the boiler height, and by making standard a three-pass baffle arrangement which led to better heat absorption. This baffle consisted of fire tile on the lower row of tubes extending toward the rear approximately 60 per cent of their length, thus forming a furnace roof, and two vertical baffles, giving the gases three passes over the heating surface. The side wall-cooling tubes above the furnace were abandoned in favor of lining the boiler casing above this point with fire tile, though the side water boxes in the furnace were retained. This permitted the placing of dusting doors in the side-walls and made it much easier to keep the boiler tubes clean.

The first of the boilers of this type was installed in 1899, in the United States cruiser "Alert," and this design has come to be known as the "Alert" design.

While there have been improvements made in details of design such as would naturally come in the progress of the boiler designer's art, as in any other, the "Alert" type Babcock & Wilcox boiler is still the standard for marine service. That such is the case is indicated by the numerous attempts made to imitate it, and, to a much greater extent, by its continued use in the United States Navy, navies of other countries, and the merchant marine. There are today in service throughout the world, Babcock & Wilcox marine boilers of this type containing in excess of 16,100,000 square feet of heating surface.

* Judge Lanning quoted these figures from testimony given by Mr. Hoxie on January 17, 1906, a date which shows how rapidly the Hoxie boiler was taken up by marine engineers throughout the world.

In the earlier days of the use of water-tube boilers for merchant service, the boiler room was, perhaps, not so carefully supervised as in the Navy. Certainly weight and space occupied were not considered of so great importance, and, in general, the power requirements for a given tonnage were less in the merchant service than for the Navy. For these reasons, when the Babcock & Wilcox marine boiler was first introduced into



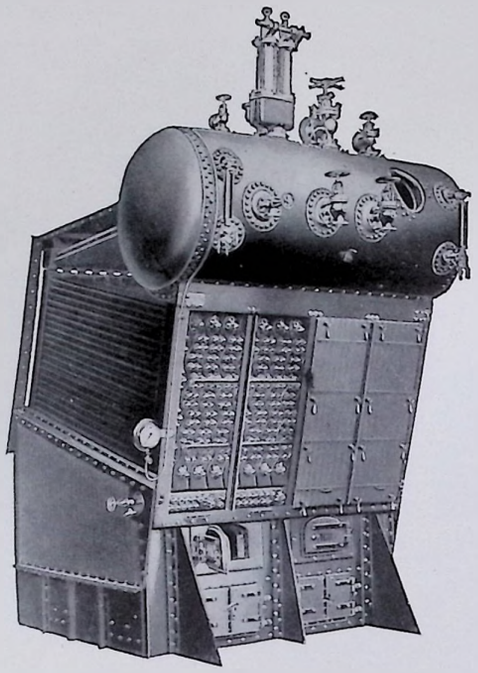
Babcock & Wilcox "Alert" Type Marine Boiler, 1899—Patented

the merchant marine, while the main features of design of the "Alert" type boiler were followed, the clusters of 2-inch tubes were replaced by single 4-inch tubes, these being more readily kept clean internally and giving a particularly rugged construction that would stand up under hard usage.

The first installation of the "Alert" type units using 4-inch tubes was made in 1900 on the Great Lakes, in the steamer "Asuncion." That these 4-inch "Alert" type boilers have given eminently satisfactory results in service is shown by the fact that there have been boilers of this design installed containing over 5,100,000 square feet of heating surface.

As the naval policies of the world countries expanded, new types of vessels were developed, particularly types calling for increasing speeds. Of such types the present-

day destroyer is typical. In such vessels the power plants must of necessity be of very large capacity as compared to the tonnage of the ships they are to serve. This makes the factors of weight and space occupied of the very greatest importance and necessitates an unusually light and compact design of boiler, in which steam can be raised very quickly and with the ability to operate satisfactorily at high rates of evaporation.



Babcock & Wilcox, 1897
Patented

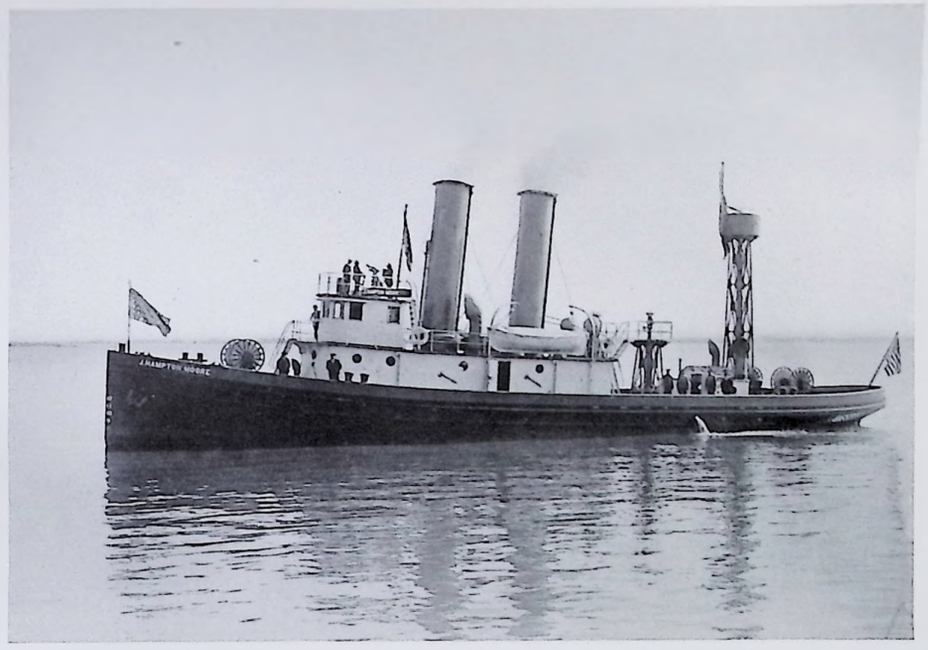
A number of types of boilers for this class of service had been developed before The Babcock & Wilcox Company decided to enter this branch of the marine field, and it was deemed better to adopt one of these types that had proved satisfactory through experience rather than to develop a new type. In the late nineties, therefore, a careful investigation of the relative merits of the existing successful types was undertaken, and eventually it found that one of these had, in addition to unexcelled operating characteristics, a special advantage, possessed by no other, in that any tube could be removed and replaced without disturbing any other tube. An earlier investigation of these boilers had led Babcock & Wilcox, Limited, London, to the same conclusion and it had acquired the rights to build the White-Forster boiler in its territory. The Babcock & Wilcox Company, New York, acquired the rights to build this boiler in the United States and after modifying certain details of design to meet conditions in this country, furnished the first Babcock & Wilcox Express type boilers to the United States Navy, in 1909, for the destroyer "Mayrant."

The original design of this boiler used lower drums or "water pockets" of oval cross-section. The Babcock & Wilcox Company improved the design in 1914 by making these drums of circular cross-section. This involved other changes, to retain the special feature of ready removal of tubes.

While the Express type of marine boiler has a limited field—*i.e.*, a field limited in the number of classes of service where it best meets operating conditions—the extent of this field may be appreciated from the fact that in 1926 the total heating surface of Babcock & Wilcox Express type boilers that had been installed was in excess of 5,300,000 square feet.

REQUIREMENTS OF STEAM BOILERS

SINCE the first appearance in our publication "Steam" of the following "Requirements of a Perfect Steam Boiler," the list has been copied many times, either word for word or clothed in different language, and applied to some specific type of boiler design or construction. In most cases, although full compliance with one or more of the requirements was structurally impossible, the reader was left to infer that the boiler under consideration possessed all of the desirable features. It is noteworthy that this list of requirements as drawn up by George H. Babcock and Stephen Wilcox, in 1875, while originally prepared as applicable to stationary boilers, represents the best practice of today, both in that and in the marine boiler field. Moreover,



City of Philadelphia Fireboat "J. Hampton Moore," Equipped with Babcock & Wilcox Boilers

coupled with their boiler itself, which is used in the navies of the world, in the merchant marine, and in the largest and most important stationary steam generating plants, the list forms a fitting monument to the foresight and genius of the inventors.

REQUIREMENTS OF A PERFECT STEAM BOILER

1st. Proper workmanship and simple construction, using materials which experience has shown to be the best, thus avoiding the necessity of early repairs.

2nd. A mud drum to receive all impurities deposited from the water, and so placed as to be removed from the action of the fire.

3rd. A steam and water capacity sufficient to prevent any fluctuation in steam pressure or water level.

4th. A water surface for the disengagement of the steam from the water, of sufficient extent to prevent foaming.

5th. A constant and thorough circulation of water throughout the boiler, so as to maintain all parts at the same temperature.

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6th. 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 passages between the different sections to equalize the water-line and pressure in all.

7th. A great excess of strength over any legitimate strain, the boiler being so constructed as to be free from strains due to unequal expansion, and, if possible, to avoid joints exposed to the direct action of the fire.

8th. 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.

9th. The heating surface as nearly as possible at right angles to the currents of heated gases, so as to break up the currents and extract the entire available heat from the gases.

10th. All parts readily accessible for cleaning and repairs. This is a point of the greatest importance as regards safety and economy.

11th. Proportioned for the work to be done, and capable of working to its full rated capacity with the highest economy.

12th. Equipped with the very best gauges, safety valves and other fixtures.



Bulk Oil Carrier "J. A. Moffett, Jr.," Diesel Propelled, Equipped with Babcock & Wilcox Boilers for Auxiliary Purposes
Owners, Standard Oil Company of New Jersey

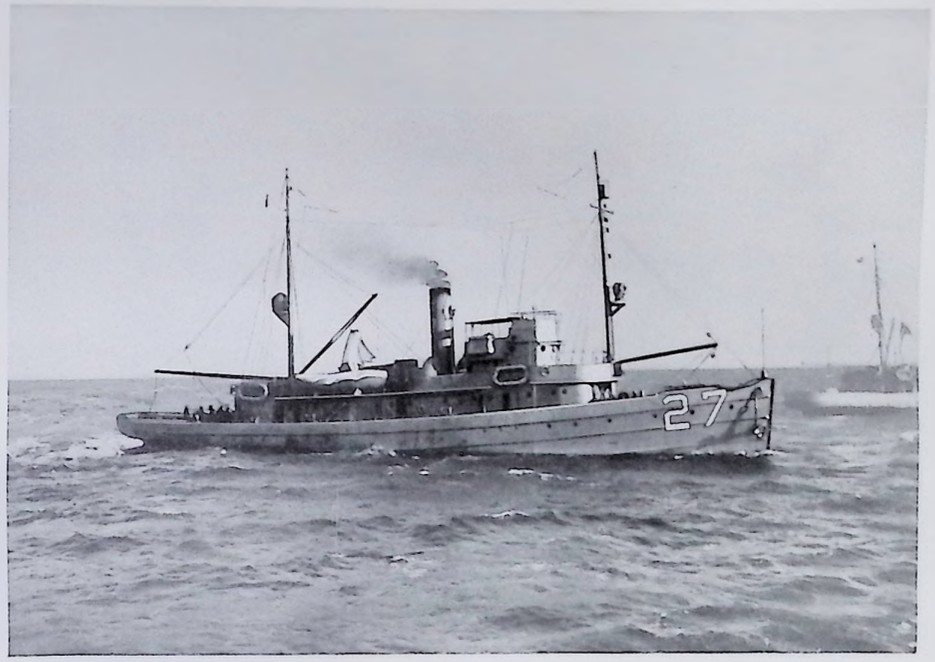
The late Admiral George W. Melville, engineer-in-chief of the United States Navy from 1887 to 1903, and recognized as one of the foremost leaders of the engineering profession, published in the *Engineering Magazine,* in January, 1912, an article entitled "The Development of the Marine Boiler in the Last Quarter Century." In this article, in outlining the characteristics that the thoroughly satisfactory water-tube boiler should possess, he includes, in somewhat modified form, the requirements as given by Babcock & Wilcox in 1875, and adds certain features as applying more specifically to marine practice. These features are:

1. Reasonable lightness, with scantlings sufficient to promise reasonable longevity.
2. Ability to raise steam quickly.
3. Economy of space.
4. Interchangeability of parts, and, as far as possible, the use of regular commercial sizes, so that repair material could be procured anywhere.

The description of the details of design and manufacture of Babcock & Wilcox marine boilers in the following pages indicates how fully these boilers meet those of the requirements outlined above that refer to design and structural features.

The manner in which the requirements other than those of design and construction are fulfilled are clearly demonstrated in the chapter comparing the Babcock & Wilcox boiler with other types and in the actual performance figures given later.

In the light of the performance of the Babcock & Wilcox boiler under the exacting conditions of present-day marine power-plant practice, the foregoing list of requirements reveals the insight of the inventors of the boiler into the fundamental principles of steam generator design and construction.



U. S. Minesweeper No. 27, Equipped with Babcock & Wilcox Boilers and Oil Burners

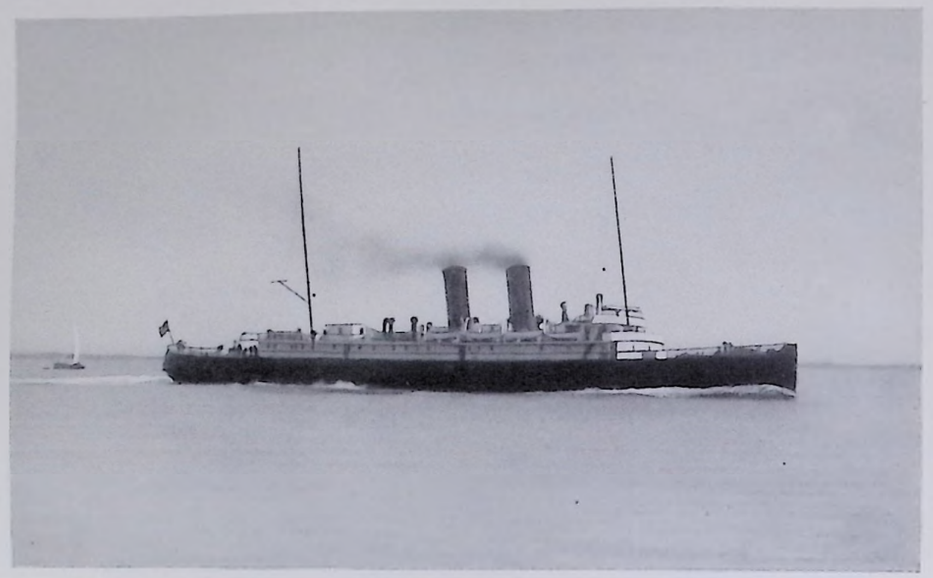
Since the Babcock & Wilcox marine boiler became thoroughly established as a durable and efficient steam generator, many types of water-tube marine boilers have appeared on the market. Most of them, failing to meet enough of the requirements of a perfect boiler, have fallen by the wayside, while a few, failing to meet all of the requirements, have only a limited field of usefulness. None has been superior, and in the majority of cases, the most ardent admirers of other boilers have been satisfied in looking up to the Babcock & Wilcox marine boiler as the standard, in claiming that the newer boilers are "just as good."

Records of recent performances, under the most severe conditions of service at sea and on land, show that the Babcock & Wilcox boiler can be operated continuously and regularly at higher rates of evaporation, with better efficiency, and with lower upkeep cost than any other boiler. It is especially adapted for marine service where it is necessary to use a boiler in which steam can be raised quickly and the boiler placed on

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the line, either from a cold state or from banked fires, in the shortest possible time, and with which the capacity, with good feed water, is limited only by the quantity of fuel that can be burned in the furnace.

The circulation through the separate sections, and the action of the headers in forcing a maximum and continuous circulation in the lower tubes, permit the operation of Babcock & Wilcox boilers without priming, with a higher degree of concentration of salts in the water than is possible in any other type of boiler.



Steamer "Stad Antwerpen," 14,000 Shaft Horse-power, Babcock & Wilcox Boilers
Owners, Belgian State Railways

The number and importance of the installations of Babcock & Wilcox marine boilers made during the last thirty years is a most gratifying demonstration of the merits of the apparatus, especially in view of their satisfactory operation under conditions which are perhaps more exacting than those of any other service.

Efficient durability, which is the test of all things, demonstrates for boilers, as for other apparatus, the survival of the fittest. When judged on this basis, the Babcock & Wilcox marine boiler stands pre-eminent in its ability to cover the whole field of marine steam generation with the highest commercial efficiency. Year after year, the Babcock & Wilcox boiler has become more firmly established as the standard of excellence in the boiler-making art.

THE BABCOCK & WILCOX MARINE BOILER

THE following description of the Babcock & Wilcox marine boiler will clearly indicate the manner in which it fulfills the requirements of the perfect steam boiler as previously given.

The Babcock & Wilcox marine boiler is built in two types, the details of construction being the same, the difference being in the use of 2-inch and 4-inch tubes.

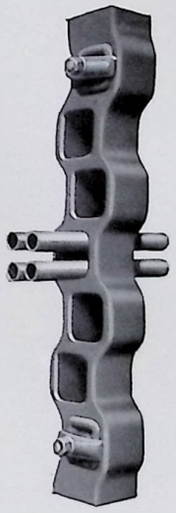
Sections—All Babcock & Wilcox boilers are made up of sections of seamless steel tubes, inclined at an angle with the horizontal and expanded at the ends into sinuous headers. In the case of the 2-inch tube boilers, the tubes are arranged in groups of four, the groups being staggered with respect to each other, while in the 4-inch tube boilers the individual tubes are so staggered. The amount of heating surface of the individual sections is made greater or less by a variation of the number of tubes in height and in the length of tubes. The heating surface of the boiler is varied by a change in the heating surface of individual sections and by the number of sections entering into the complete unit.



Forged Steel Header for Boiler Having 4-Inch Diameter Tubes

Each section is made up of a downtake header supplying water to the tubes, the tubes, and an uptake header, discharging water and steam from the tubes.

Headers—The headers entering into the construction of the sections are of wrought steel and of the inclined type, and are made in sinuous forms so that the tubes, either in groups of four 2-inch or individual 4-inch, are staggered with respect to each other. The width of the header, the thickness of material used in its construction, and its sinuous form, result in proper strength without the necessity of the use of staybolts.



Forged Steel Header for Boiler Having 2-Inch Diameter Tubes

Handholes and Fittings—Opposite the tube ends in the headers there is placed a handhole of sufficient size to permit the cleaning, removal and renewal of any tube.

In the case of the 2-inch tube boiler, the opening is approximately square in shape, one opening with its fitting taking care of a group of four tubes. In the 4-inch tube boiler the opening and its fitting are elliptical in shape, one opening for each tube.

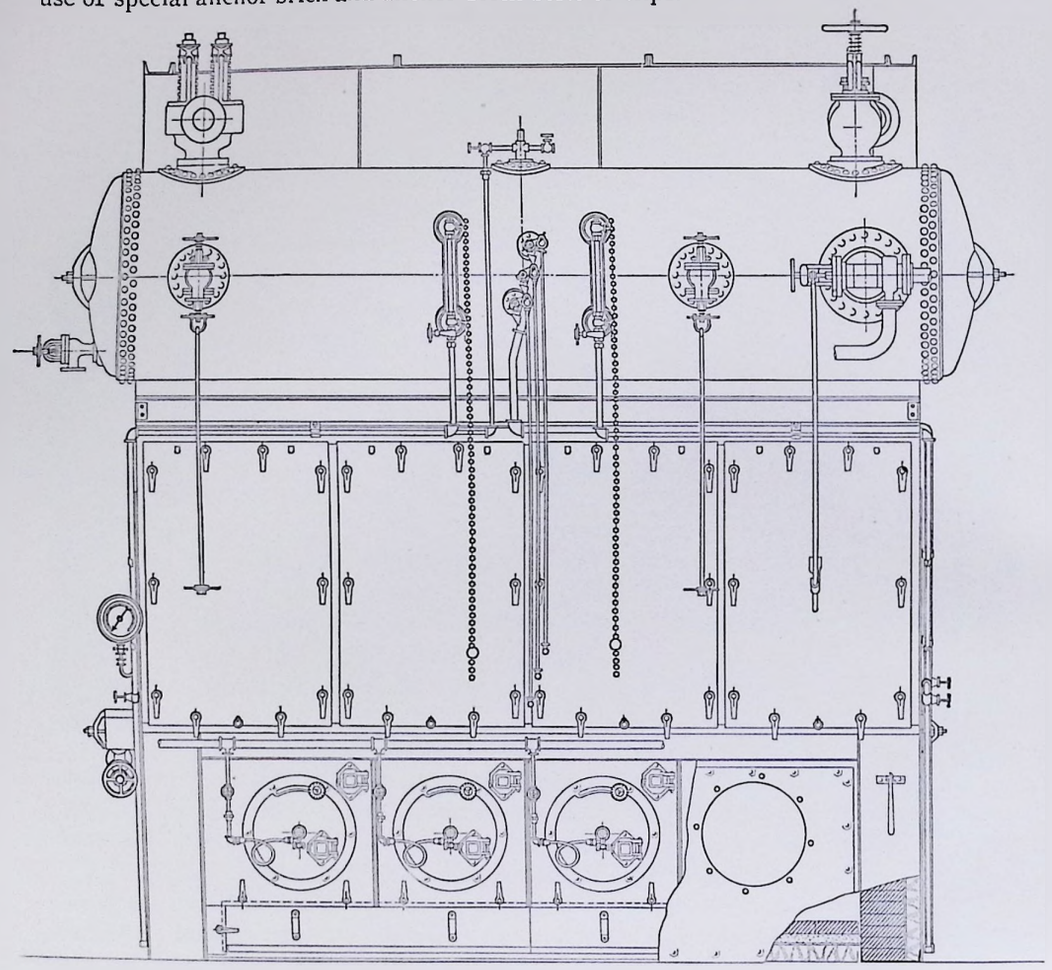
The handhole openings are machine-faced and milled back from the edge a sufficient distance to form a seat. The openings are closed by inside-fitting, forged-steel plates, shouldered to center in the opening, their flanged seats being milled to a true plane and the fittings being milled to fit the opening. These plates are held in position by studs and forged-steel binders and nuts. The joints between plates and header seats are made with thin gaskets.

Side-wall Water Boxes—In boilers to be fired with coal, the outside sections are continued down to the level of the grate, the tubes at the furnace sides being replaced with forged-steel boxes of rectangular section. These boxes are located one above the other at the same angle of inclination as the tubes and take the place of side-

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wall furnace brickwork. They insure cool side casing, prevent the adherence of clinkers and are of sufficient thickness to withstand the wear and tear of fire tools.

When the boilers are to be fired with oil the side-wall water boxes are not furnished and the sides, bottom, back and front walls are lined with fire-brick, giving ideal conditions for efficient combustion. This fire-brick lining is held in place by the use of special anchor brick and anchor-brick bolts or clips.

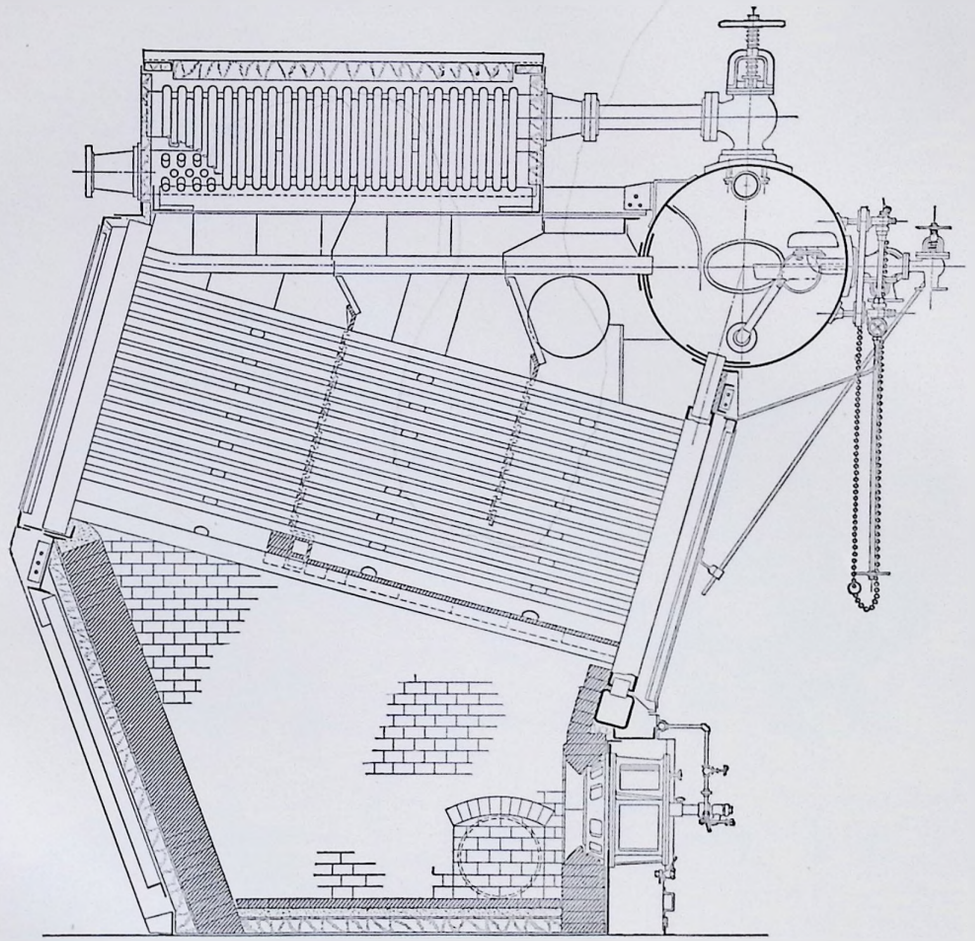


FRONT VIEW OF A BABCOCK & WILCOX MARINE BOILER, PARTLY IN SECTION, SHOWING ARRANGEMENT OF VALVES AND FITTINGS

Coal-burning boilers having brick-lined furnaces are occasionally furnished, this arrangement allowing for quick and satisfactory conversion to oil burning, should such conversion be considered advisable at any time. The brick-lined oil-burning furnace, again, can readily be converted to coal burning, by the addition of the usual fire fronts and grates, and the removal of the brick flooring in the furnace.

MUD DRUM—The lower ends of all downtake headers are connected by nipples to a transverse mud drum, which is a forged-steel box 7¼ inches square. The mud drum is machined for blow-off connections, and has handholes for cleaning, with the necessary handhole fittings.

DRUM—The steam drum of ample capacity is located at the front of the boiler over the downtake headers extending transversely across the boiler. The drum is connected to the downtake headers by vertical circulating tubes and to the tops of the uptake headers by horizontal circulating tubes, all circulators being 4-inch tubes. By the use of such connections, each section is provided with an entirely independent inlet and outlet for water and steam.



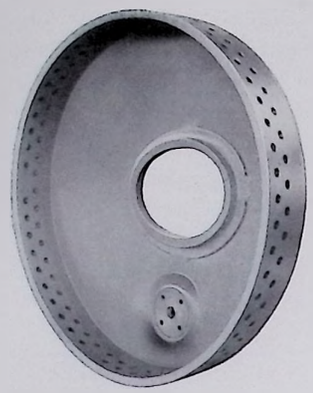
LONGITUDINAL SECTION OF A BABCOCK & WILCOX MARINE BOILER, SHOWING ARRANGEMENT OF VALVES AND FITTINGS

All drums have their longitudinal seams formed with inner and outer butt straps and have no circumferential seams other than head seams.

All shell plates and inner and outer butt straps are bent to a true circle to the extreme edge of the plates. All edges are planed. Such construction results in tight drums with a minimum of calking.

All rivet holes are drilled from the solid plate. The holes used for tack bolts are drilled in the flat plate, while all other rivet holes are drilled through the various pieces

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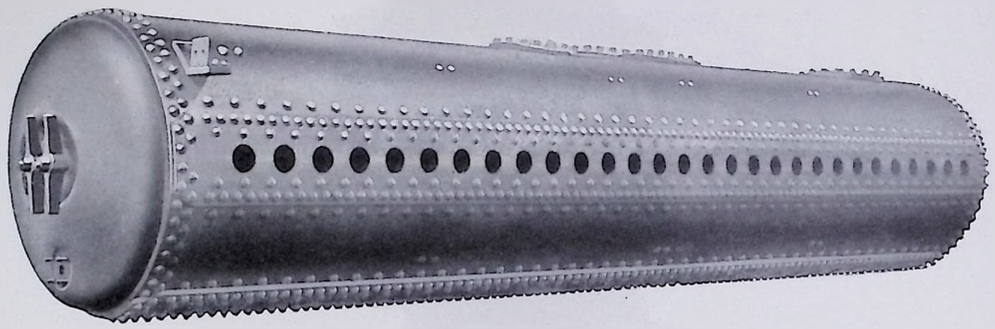
Forged Steel Drumhead
Inside View
Manhole Fitting Removed

entering into the joint after the drums are assembled. After the rivet holes are drilled, the drum is taken apart, the burrs are removed from the holes, the pieces reassembled and the rivets driven.

All rivets are driven under hydraulic pressure and held until black.

Drumheads are of wrought steel, forged at a single heat. During the forging process an elliptical manhole opening is cut in the plate, with the edges of the opening flanged inward to give the necessary strength to the opening, and affording, with a stiffening ring of sufficient thickness, a proper seat for the gasket between the drumhead and manhole fitting.

The flange of the drumhead is given a cylindrical



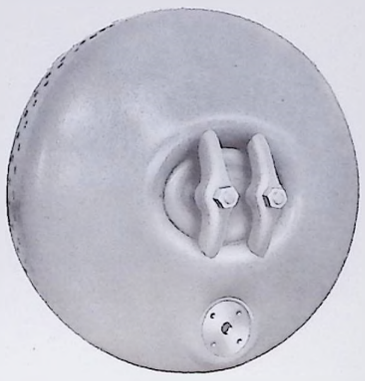
Steam Drum for Marine Boiler 29 Sections Wide. Working Pressure 275 Pounds

form of the exact size to fit within the drum shell, into which it is forced under a heavy hydraulic pressure. The head is then riveted to the shell with one or more rows of rivets, depending upon the pressure for which the drum is built.

The inner edges of the flanges of the drumhead are machined for calking, and the inner edge of the flange at the manhole opening is machined to form the gasket seat.

The manhole fitting is a pressed-steel forging, milled to fit the manhole opening, with its flange milled to form a seat for the manhole gasket.

Steam drums are fitted with internal feed connections, baffle plates, standard dry pipes, and surface blow off.

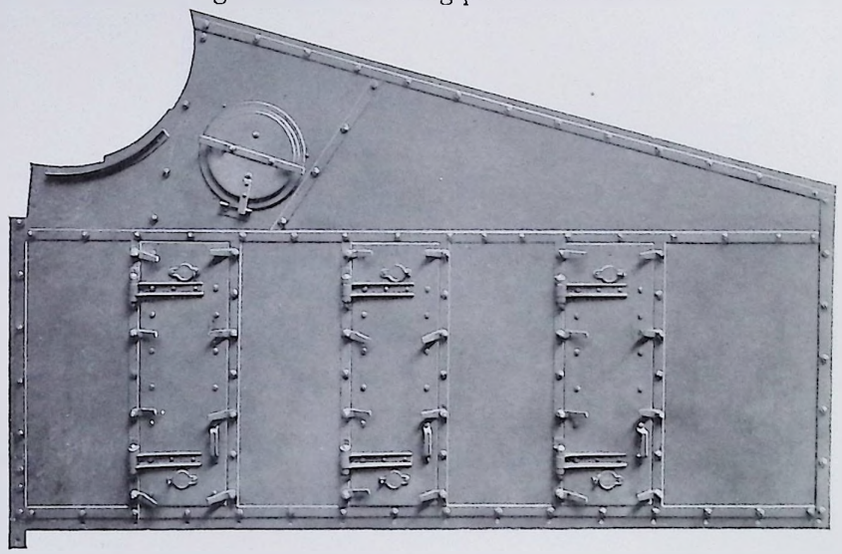


Forged Steel Drumhead
Outside View
Showing Manhole Fitting in Place

Drums are equipped with riveted pad for safety-valve connection, and riveted pads for steam outlets, water column and feed connections.

Supports—The pressure parts of a tile-side, or brick furnace, boiler are supported from beneath on foundations made of structural steel work. At the downtake end of the boiler the mud drum rests on two vertical "I" beams, one at each corner, just inside the steel casing. One flange of these "I" beams is riveted to the side casing and their bottoms secured to a foundation channel at the base of the side walls. At the top of these "I" beam supports there are provided cast-steel saddles, with yokes and bolts, to hold the mud drum. At the uptake end of the boiler, the headers are supported by a transverse "I" beam under the headers, between the side-casing plates, these plates at this place being reinforced by heavy angles.

Casing—Surrounding the pressure parts, and firmly secured to the foundation is a structural steel framing to which the casing plates are fastened.



Upper Side Casing of Boiler, Showing Arrangement of Side Swinging Dusting Doors and Removable Upper Cleaning Door

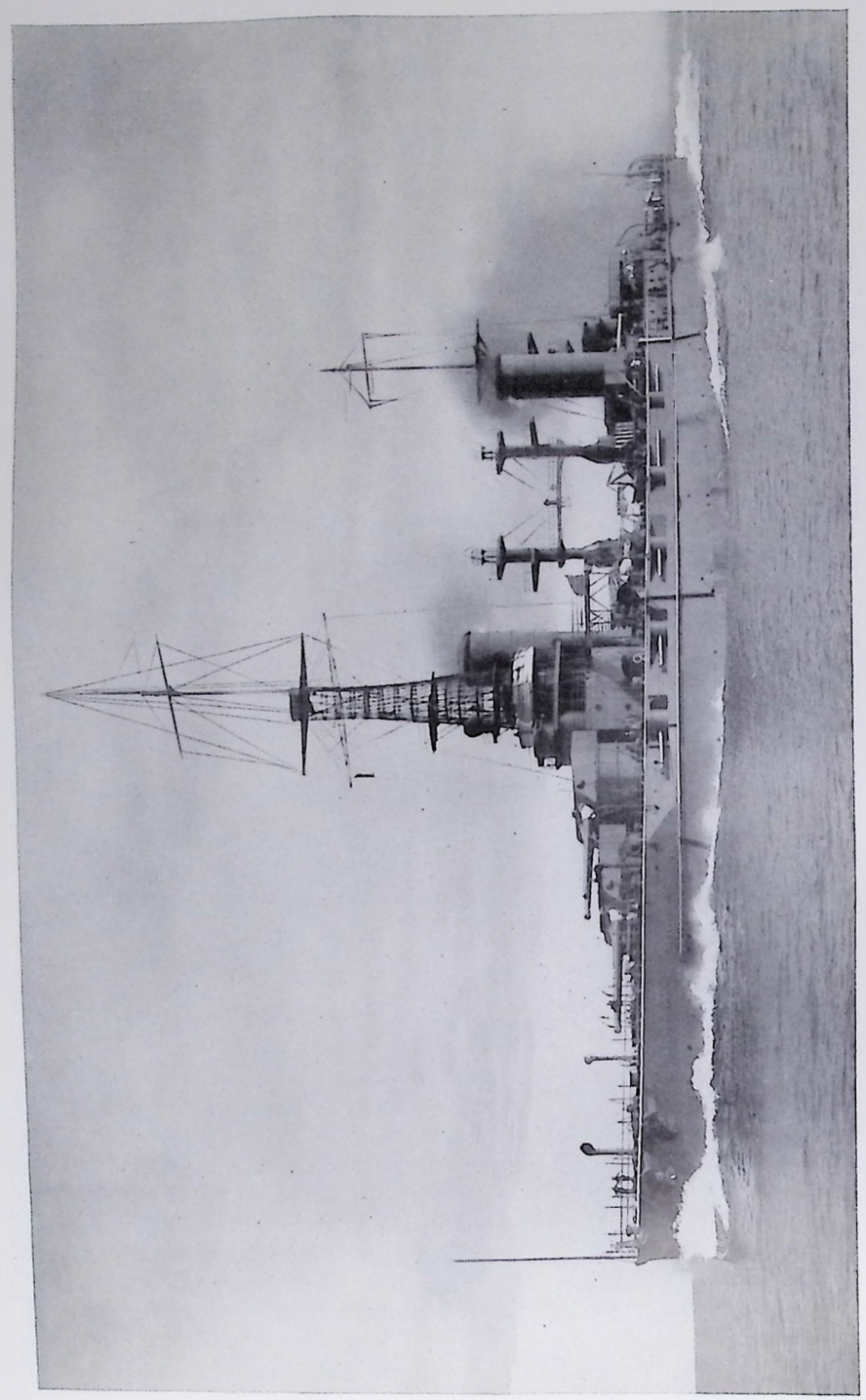
The spaces between the side tubes and casing plates are filled with light fire tiles of highly refractory material, backed with two inches of non-conducting material. The steel-casing plates, clamped to the structural framework, hold this non-conducting material in place. The casing plates are equipped with cleaning doors as described under that heading.

Fronts—Construction below the tube-door sills at the firing end of the boiler is varied to suit the furnace.

Where coal is to be used, a wrought-steel firefront is supplied having inswinging fire and ash-pit doors which automatically close in case of the rupture of a tube, thus protecting the firemen.

In the case of oil firing, the front is of wrought steel with suitable openings for oil burners. Where forced draft is used with oil burners these fronts are usually made with inner and outer plates to provide an air duct to the burners. With a closed fireroom installation, only one front plate is necessary.

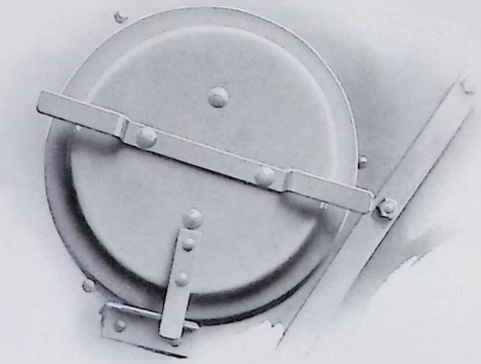
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Argentine Battleships "Rivadavia" and "Moreno," 39,500 Shaft Horse-power, Babcock & Wilcox Boilers

Tube Access and Cleaning Doors—Suspended from the framing at the front and rear of the boiler are steel tube doors which, when open, give full access to the headers and handhole fittings.

A full set of dusting and cleaning doors is provided, making all portions of the boiler accessible for cleaning and inspection. In the 4-inch tube boilers dusting panels, which are made up with a number of small doors with self-closing shutters, are supplied. Through the openings of these doors all portions of the heating surface may be cleaned with a steam or an air-dusting hand lance. In the 2-inch tube boilers this dusting panel is embodied in a swinging-door, as illustrated.



Wrought Steel Upper Cleaning Door

Fixtures—For hand-fired boilers, cast-iron dead plates and supports are supplied, together with suitable cast-iron door jambs and lintels. A full set of grate bars and grate bar bearers is furnished.

Baffle material, consisting of the "T" tile which forms the roof of the furnace, the cast-iron baffle blocks forming the end of this roof, and the cast-iron horizontal baffles plates forming the vertical baffles, is supplied. In the case of the 4-inch tube boiler, baffle tile for protecting the flame plates in the first pass of the gases are also furnished.

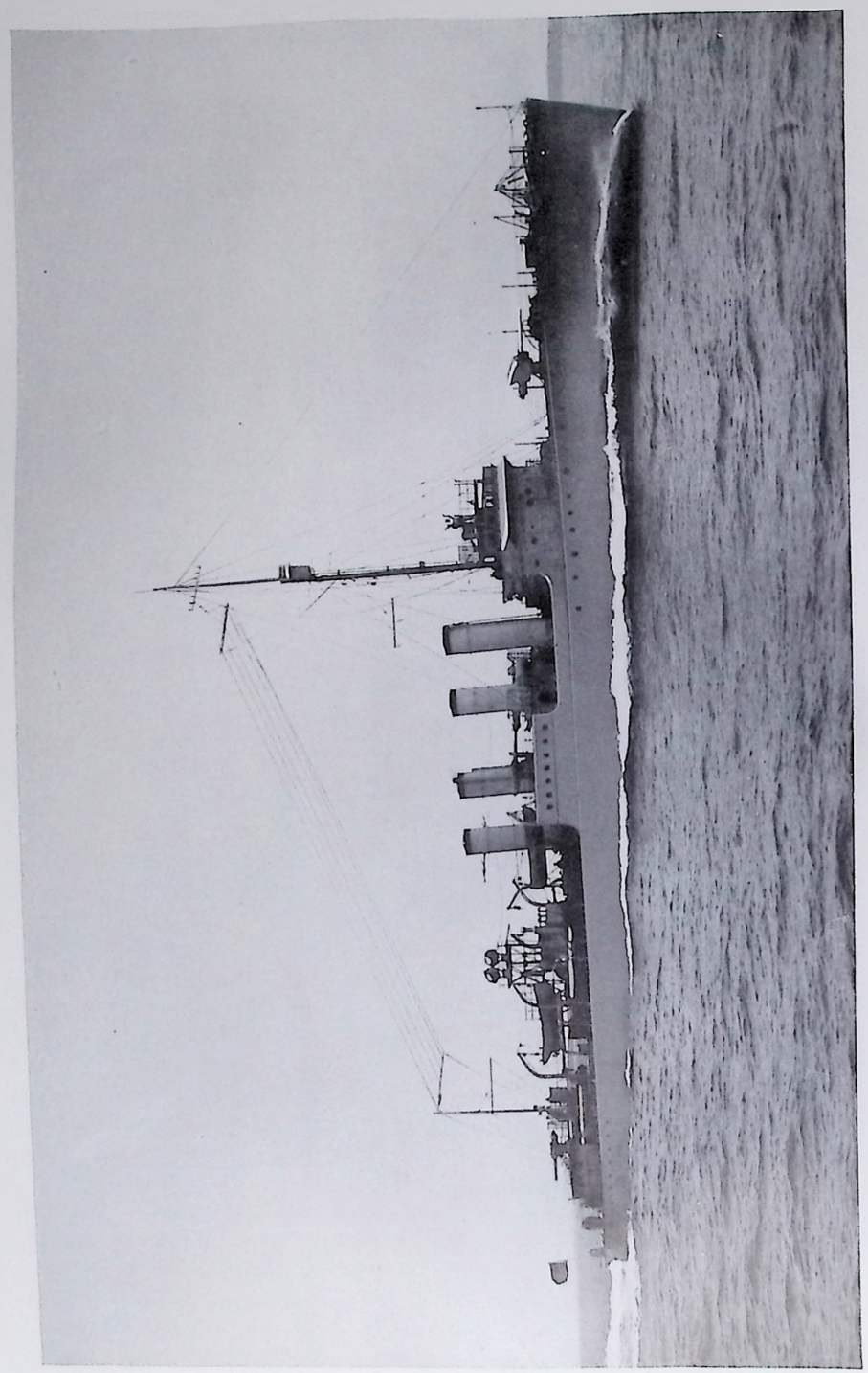
Fittings—Each boiler is equipped with safety valve, main and auxiliary feed stop and feed check valves, surface blow and bottom blow valves, water gauge glass fittings, gauge cocks and a fitting for the attachment of air valve, cleaning hose valve and steam gauge, which are also furnished. All valves and fittings are substantially built from designs of The Babcock & Wilcox Company and are attached to the drum by forged-steel nozzles or pads.

A liberal quantity of spare parts and tools is also provided, this latter including, in the case of coal-fired installations, a set of fire tools. Where vessels are intended for salt water service the company also furnishes a Babcock & Wilcox Water Testing Outfit. This equipment has recently been enlarged to include an alkalinity test, in addition to the chlorine test.

With the horizontal fire tile baffle carried on the lowermost tube from the front of the boiler to a point approximately 60 per cent of the depth of the furnace, a furnace form is obtained expanding in the direction in which combustion is taking place, a feature tending to give the most efficient combustion with any class of fuel. From the end of the fire-tile roof, where there are installed cast-iron baffle blocks, the products of combustion are led from the furnace over the heating surface by two vertical baffles, giving the gases three passes. The gases are ordinarily discharged through a breeching at the top of the third pass, immediately in rear of the steam drum.

The gas passage areas are ample and so proportioned that, with the length of gas travel resulting from the three passes, a maximum heat absorption is obtained from the gases without an undue frictional resistance and corresponding high draft loss.

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U. S. Destroyer, "James K. Paulding" Class, 29,340 Shaft Horse-power. Babcock & Wilcox Express Type Boilers

Feed water is introduced into the drum through nozzles on the front of the drum sheet and through internal mushroom feed nozzles. The water passes downward from the drum through the downtake tube nipples to the front headers, upward through the tubes of the sections to the rear headers, and, from the top of such uptake headers, through the horizontal circulating tubes which deliver it to the drum. Separation of steam, formed in the circulation through the tubes, and water takes place to an extent in the horizontal circulating tubes and is completed in the drum. The steam is stored in the steam space above the water line in the drum from which it is drawn through a so-called "dry pipe." The dry pipe as ordinarily supplied with Babcock & Wilcox marine boilers is misnamed in that it fulfills none of the functions ordinarily attributed



Steamer "New Hampshire," 2500 Indicated Horse-power. Babcock & Wilcox Boilers
Owners, The New England Steamship Company

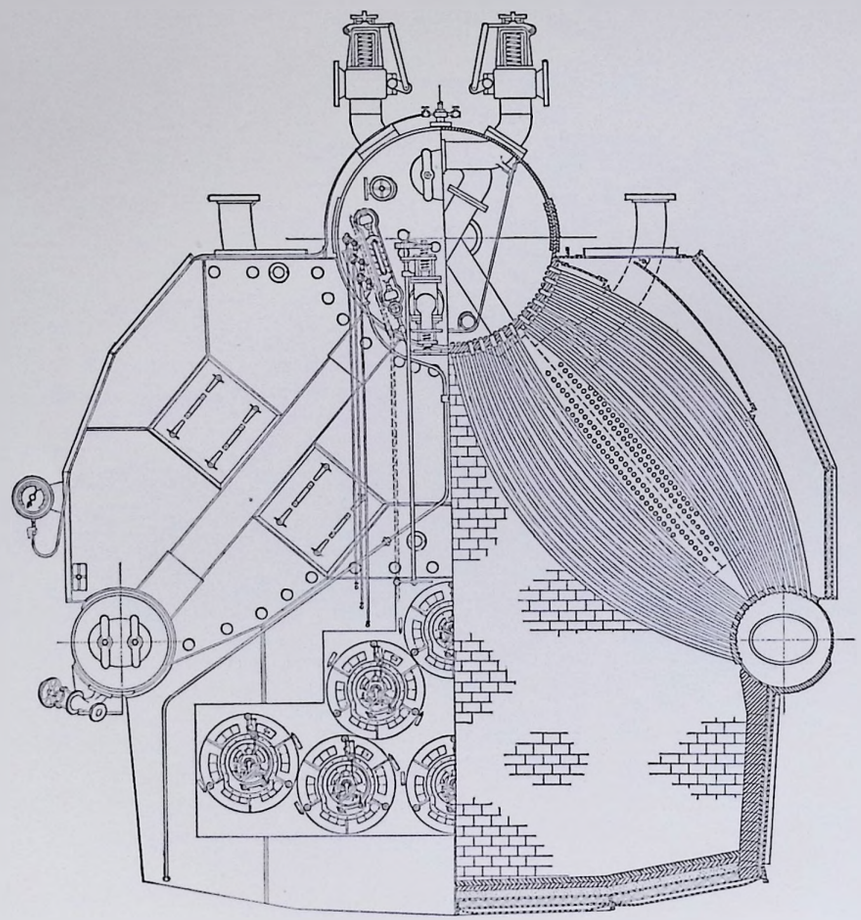
to such a device, particularly that of restricting the flow of steam from the boiler with a view to avoid priming. In the Babcock & Wilcox marine boiler, since the total area of the holes through the dry pipe is greater than that of the steam outlet, there is no restriction, and the pipe is in reality a collecting pipe, taking steam evenly from a large proportion of the length of the steam space of the drum.

All Babcock & Wilcox marine boilers are built in accordance with the General Rules and Regulations prescribed by the Board of Supervising Inspectors, United States Steamboat Inspection Service, or under General Specifications for Machinery, United States Navy. Boilers for merchant vessels are also built to meet the requirements of Lloyd's Register of Shipping, the American Bureau of Shipping and other classification societies, as required. The materials entering into the construction of Babcock & Wilcox boilers are the best obtainable for the special purpose for which they are used and are subjected to rigid inspection and tests. The boilers are manufactured by means of the most modern shop equipment and appliances in the hands of an old and well-tried organization of skilled mechanics under the supervision of experienced engineers.

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THE BABCOCK & WILCOX EXPRESS TYPE MARINE BOILER

AS stated in the first chapter, with the demands for increased speed in ships of the various navies throughout the world, it became necessary to offer boilers of light and compact design, capable of being operated at high rates of evaporation. The Babcock & Wilcox Company obtained the rights to manufacture and sell the White-Forster boiler, modified the design somewhat to meet the conditions of service in



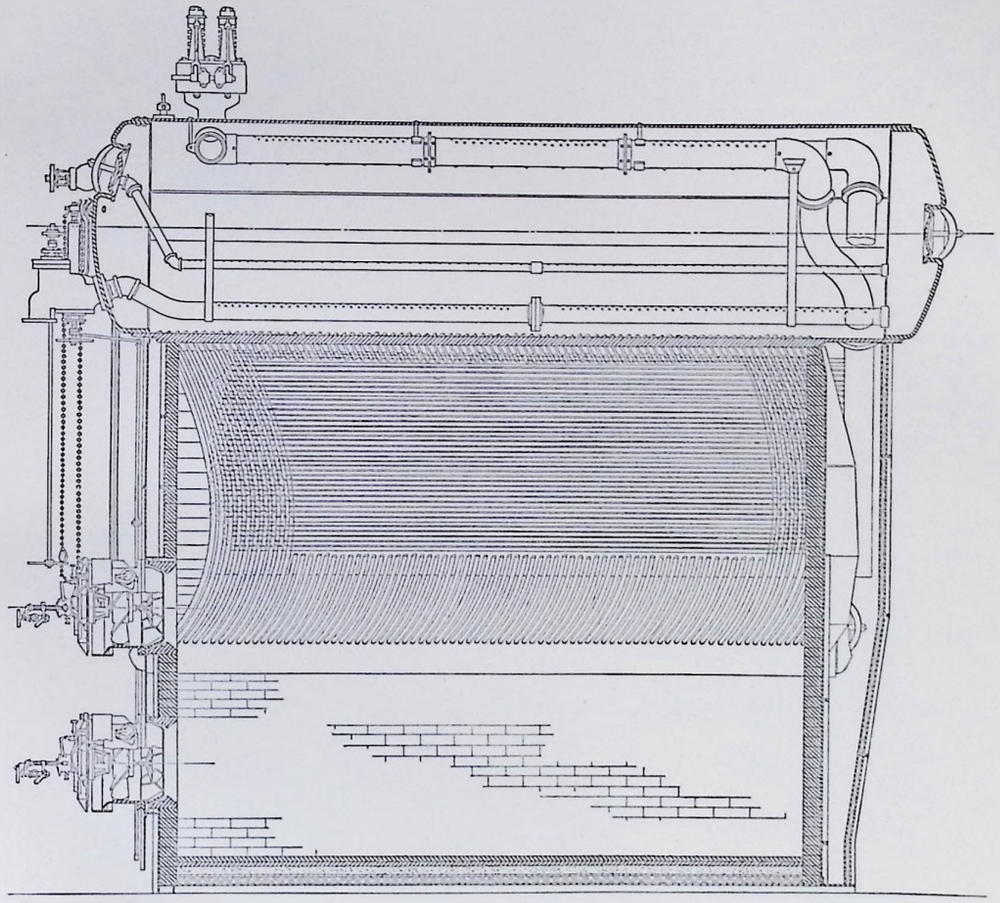
FRONT VIEW OF A BABCOCK & WILCOX EXPRESS TYPE BOILER, PARTLY IN SECTION, SHOWING ARRANGEMENT OF VALVES AND FITTINGS

this country, and, in addition, made certain specific improvements in design and construction. The modified boiler is now known as the Babcock & Wilcox Express type marine boiler.

This boiler, now common in destroyers, scout cruisers and similar classes of naval vessels, is known as the three-drum type. The Babcock & Wilcox Express type boiler differs from others of this same general design in that any tube can be removed and re-

placed without disturbing any other tube. This is accomplished by giving the tubes a slight uniform curvature in the plane parallel to the drums, with a slight enlargement of tube diameter at the steam-drum end, thus allowing any tube to be drawn into the steam drum and removed through the manhole opening, or other openings in the drum head. The curvature is slight and interferes in no way with interior cleaning. This curvature also allows for expansion and prevents distortion when operating at high rates.

The illustrations indicate this curvature and show that the boiler pressure parts are made up of a large steam-and-water drum connected by two banks of tubes diverging from this drum to two smaller water drums.



LONGITUDINAL SECTION OF A BABCOCK & WILCOX EXPRESS TYPE BOILER, SHOWING ARRANGEMENT OF VALVES AND FITTINGS

The steam drum is made up of two sheets, the tube sheet being thicker than the plate sheet to give sufficient strength of ligaments between tube holes. This tube sheet is planed to the same thickness as the upper sheet at the edges where the butt-strap joints connect the two sheets.

The smaller lower drums are made of a single sheet, that portion into which the tubes are expanded being thicker than the remainder of the drum shell, this latter

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U. S. Battleship "Colorado," 37,400 Shaft Horse-power, Babcock & Wilcox Boilers, Superheaters and Oil Burners

portion being planed down in the same manner as the edges of the steam-drum tube sheet.

The drumheads are equipped front and rear with manhcles, the method of manufacture of the heads and the manhole covers being as described for Babcock & Wilcox sectional marine boilers.

The front head of the steam drum is equipped with two special water-gauge glasses, water-gauge cocks, surface-blow valve and duplicate combination stop-and-check feed valves.

The upper shell of the steam drum is equipped with a wrought-steel pad for the reception of the main steam stop valve, where no superheaters are installed. Where superheaters are installed, connections are made direct from the internal dry pipes to the inlet superheater headers through the drum shell.

The steam drum is also equipped with wrought-steel pads carrying the safety valves.

The water drums are equipped with bottom-blow valves.

All valves and fittings are of special design and made in accordance with United States Navy specifications.

Internally the steam drum is equipped with duplicate submerged feed pipes, steam dry pipes and surface-blow connection.

In the manufacture of the pressure parts of the Babcock & Wilcox Express type marine boiler, the same methods are employed as described for the Babcock & Wilcox sectional marine boiler. While the requirements of the three-drum type of boiler are such that minimum weight is necessary, this lightness is obtained in the Babcock & Wilcox Express type boiler without sacrifice of strength or safety, all parts being designed to meet the requirements of the pressures they are to carry. The small size of the tubes used and the small amount of water carried minimize the danger of explosion.

The boiler is supported from beneath by special saddles on which the lower drums rest, these saddles in turn being supported on structural members extending to the ship frames.

The space between and below the lower drums forms a furnace of ample capacity, allowing sufficient fuel to be burned to give the high rates of evaporation required of this class of boiler. This furnace is lined on all sides with fire-brick held in place by anchor bolts and backed by insulating material.

With the exception of the furnace, no tile or brick are used in the Express type boiler. The casings consist of inner and outer steel plates with non-conducting block held firmly between. For convenience in handling, the casings are paneled, the panels being bolted together in such manner that no air or gas leakage is possible. The inner faces of the panels in the path of the hot gases are protected by asbestos millboard secured to the plate.

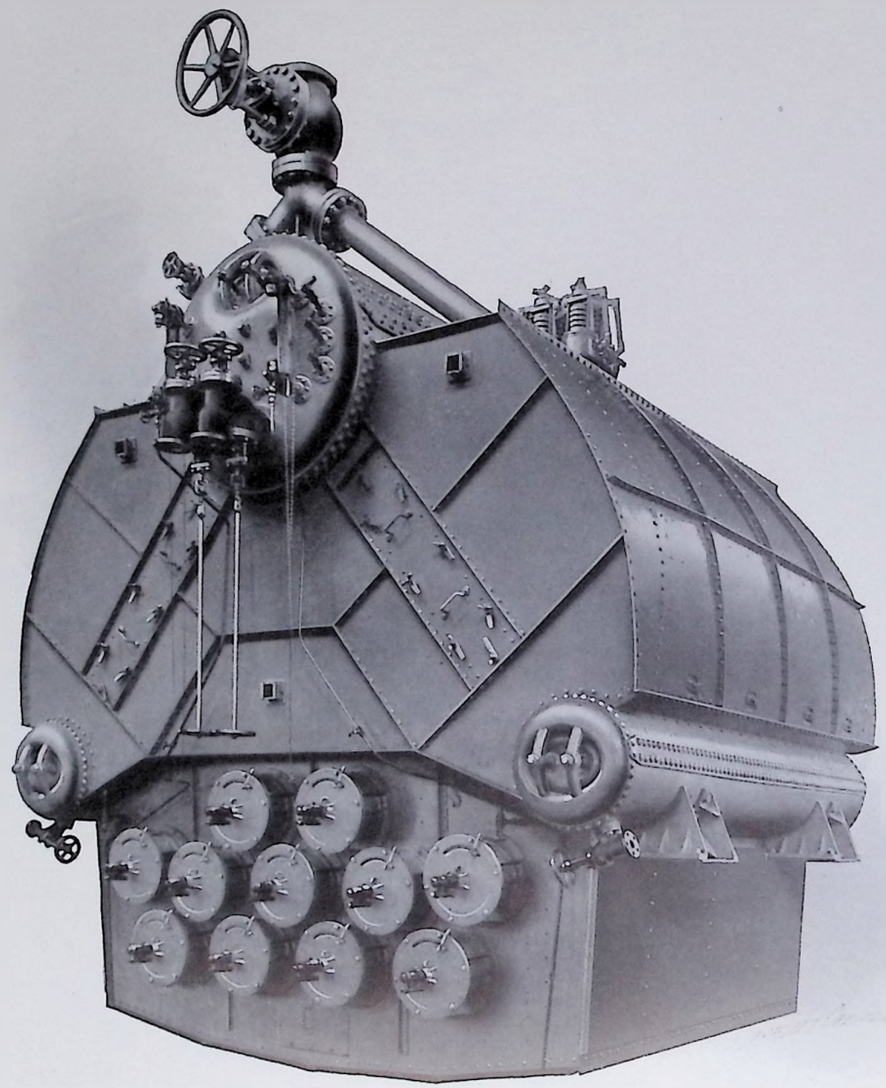
An ample number of sight holes in the casings allows for the proper observation of the interior of the furnace and uptakes.

The gases from the furnace make a single pass through the two banks of tubes to the uptakes at the sides of the steam drum. Proper distribution of the gases over the full length of the tubes in the two banks is assured by the use of two directing baffles at the top of each bank, shown in the illustrations. While the gases make but a single pass over the heating surface, the gas passage areas and the number of gas

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contacts are such that the efficiencies and exit gas temperatures are comparable with those obtained in the standard Babcock & Wilcox marine boilers. This is clearly shown by the tests of Babcock & Wilcox Express type boilers given elsewhere in this book.

The water circulation is direct and rapid. While the amount of water contained in the boiler in service is small—approximately two pounds per square foot of boiler heating surface—with ordinary care no difficulty is experienced in maintaining a proper water level.



Babcock & Wilcox Express Type Boiler. Heating Surface, 7565 Square Feet; Superheating Surface, 753 Square Feet; Working Pressure, 280 Pounds. Installed at the Fuel Oil Testing Plant, U. S. Navy Yard, Philadelphia, Pa.

ADVANTAGES OF THE BABCOCK & WILCOX MARINE BOILER

THE advantages of the Babcock & Wilcox marine boiler may perhaps be most clearly set forth by a consideration, first, of water-tube boilers as a class as compared with Scotch marine boilers, and, second, of the Babcock & Wilcox marine boiler specifically as compared with other designs of water-tube marine boilers.

WATER-TUBE VERSUS FIRE-TUBE BOILERS

Safety—The most important requirement of a steam boiler is that it shall be safe from disastrous explosion; that is, only the part damaged shall be affected.

A consideration of the thickness of material necessary for cylinders of various diameters under a steam pressure of 200 pounds and assuming an allowable stress of 12,000 pounds per square inch will indicate how the advent of high pressures was one of the primary factors in forcing the adoption of water-tube boilers. For example, under these conditions, cylinders, without joints, of 4, 48, 108 and 144 inches diameter, would require plate thicknesses of 0.033, 0.400, 0.900 and 1.200 inches, respectively. The rapidity with which plate thickness increases with diameter is apparent, and in practice, due to the fact that riveted joints must be used, the thickness, except in the case of the first, must be increased some 40 per cent.

In water-tube marine boilers, the drum diameter rarely exceeds 42 inches, and the thickness of plate required is never excessive. The thinner metal can be rolled to a more uniform quality, the seams admit of better proportioning, and the joints can be more easily and perfectly fitted than where thicker plates are necessary. All of these points contribute toward making the drums of water-tube marine boilers better able to withstand the stress which they are called upon to endure.

The essential constructive difference between the water-tube and Scotch marine boiler lies in the fact that the drum of the former is of relatively small diameter as against the large diameters necessary in the latter.

The factor of safety of boiler parts which come in contact with the most intense heat in water-tube marine boilers can be made much higher than would be practicable in a Scotch boiler.

Stresses due to unequal expansion have been a fruitful source of trouble in Scotch marine boilers. Unequal strains in such boilers result from difference in temperature of the various parts. This difference in temperature results from the lack of a positive, well-defined circulation. While such a circulation does not accompany all water-tube marine boiler designs, the circulation in water-tube boilers is in general much more definite than in Scotch boilers.

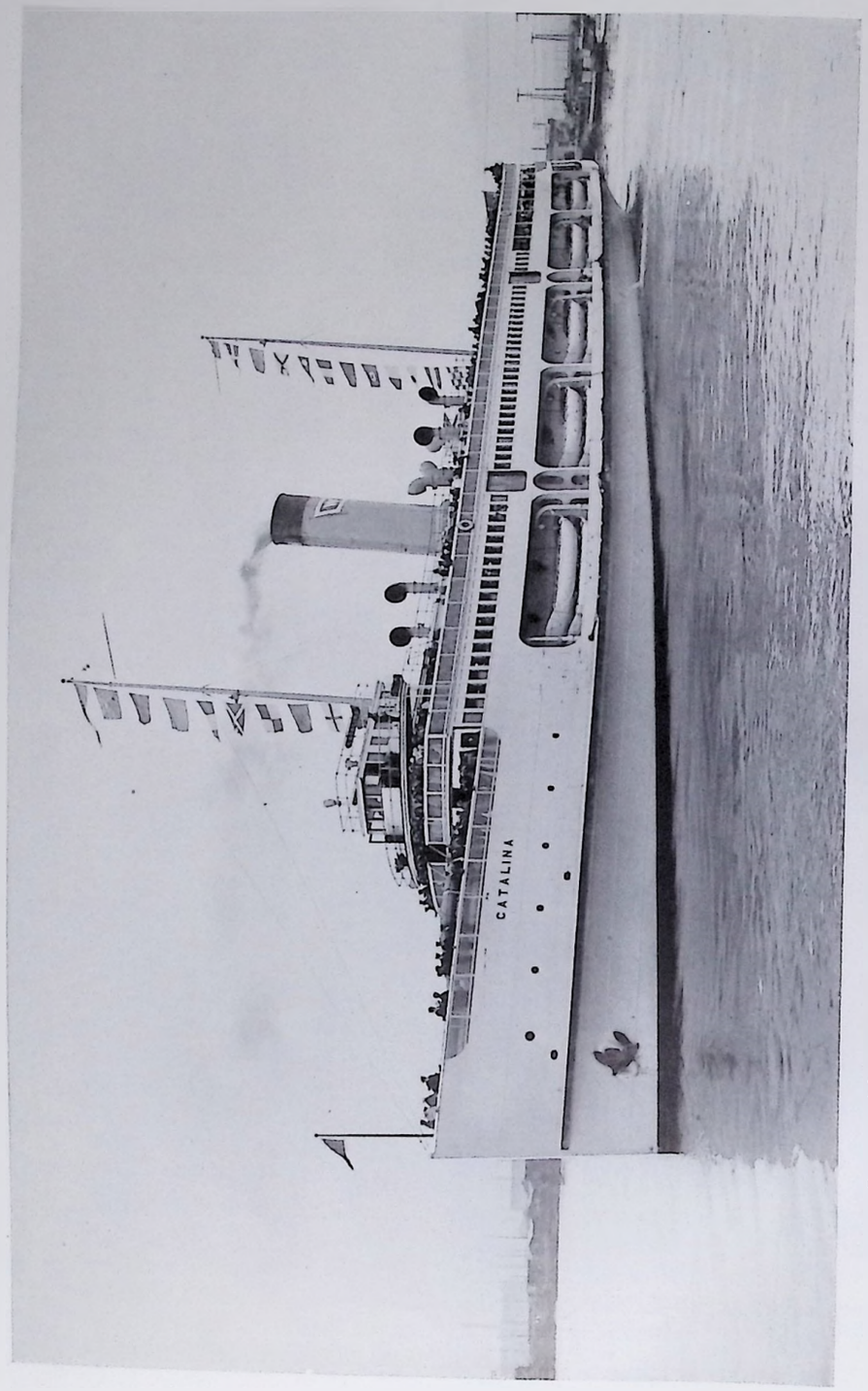
A positive and efficient circulation assures that all portions of the pressure parts will be at approximately the same temperature, and in this way strains resulting from unequal temperatures are obviated.

Economy—The requirement probably next in importance to safety in a steam boiler is economy in the use of fuel. To fulfil such a requirement, the three items of:

Proper grate or furnace for the class of fuel to be used;

A combustion chamber in which combustion can be completed before the heating surfaces are encountered;

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Steamer "Catalina," 4470 Indicated Horse-power, Babcock & Wilcox Boilers and Oil Burners
Owners, Wilmington Transportation Company

The heating surface of such character and arrangement that the maximum amount of available heat may be extracted;

must be co-ordinated.

Scotch marine boilers, from the nature of their design, do not permit the variety of combinations of heating surface and combustion space possible in practically any water-tube boiler.

Capacity—Due to the generally more efficient circulation found in water-tube than in Scotch boilers, rates of evaporation are possible with water-tube boilers that cannot be approached where Scotch boilers are employed.

Quick Steaming—Another important result of the better circulation ordinarily found in water-tube boilers is the ability to raise steam rapidly in starting and to meet sudden demands that may be thrown on them.

In a properly designed water-tube boiler steam may be raised from a cold boiler to 200 pounds pressure in ten minutes. Tests of Babcock & Wilcox boilers by officers of the Navy have proved this repeatedly. It is good practice, however, to retard the fires when setting away a cold boiler, allowing a matter of about one hour for getting a full head of steam, as such practice prolongs the life of the furnace brickwork.

For the sake of comparison with this figure it may be stated that with the large double-ended Scotch boilers such as are used in trans-Atlantic service, fires are usually started at least 15 hours before the time set for getting under way, artificial circulation being resorted to, to ensure an even temperature throughout the boiler. This length of time is necessary for such boilers, also, in order to eliminate as far as possible excessive strains resulting from the sudden application of heat.

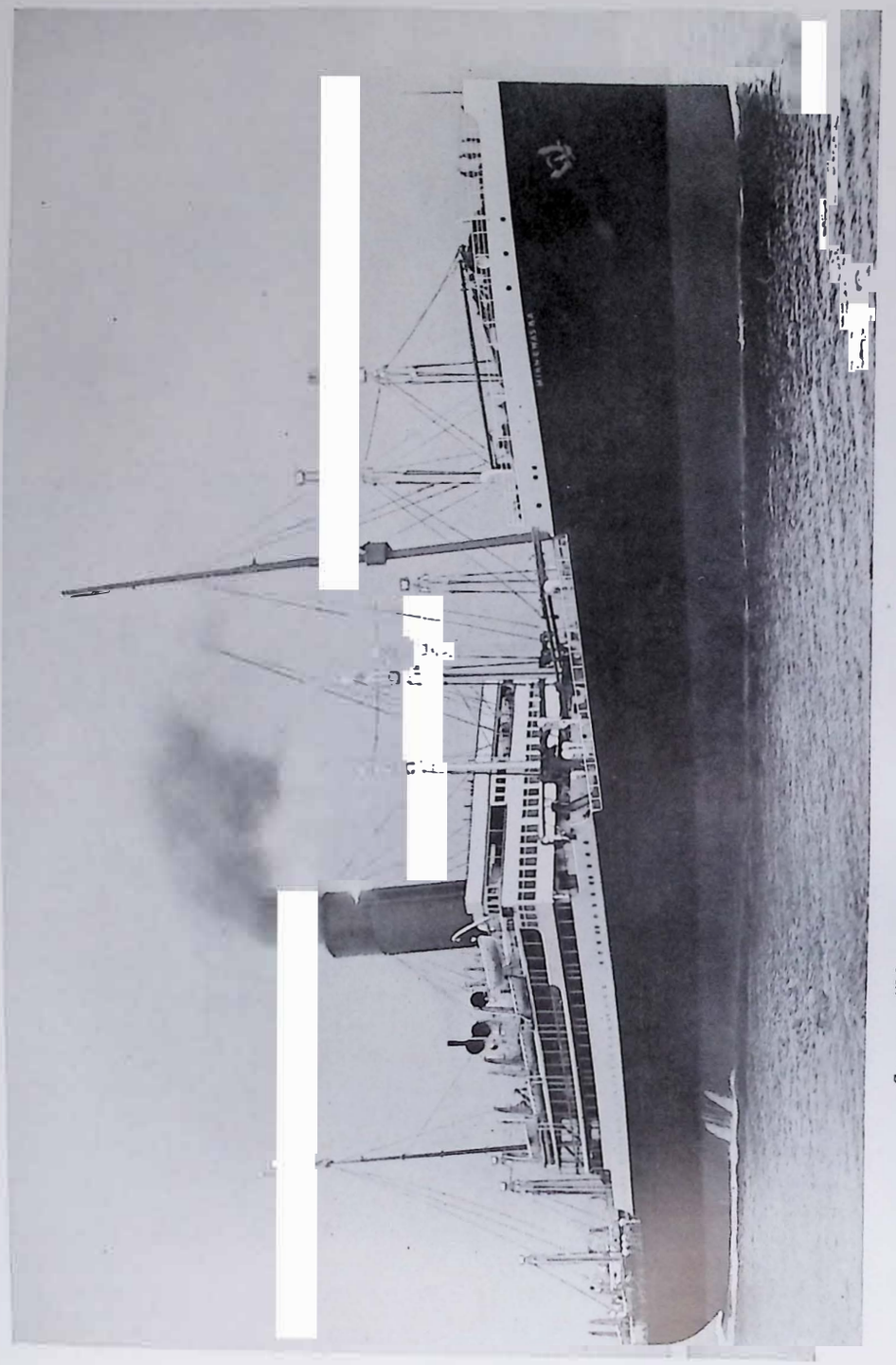
Weight—Water-tube boilers in general are much lighter than Scotch marine boilers. Babcock & Wilcox marine boilers are from 25 to 50 per cent lighter than Scotch boilers of the same heating surface, the difference depending upon the pressure for which the boilers are to be built.

Space Occupied—The space occupied by the amount of heating surface necessary for a given amount of steam generated is less for water-tube boilers. The difference in space so occupied varies with the size of the vessel and the amount of steam to be generated, and the range is large—from 10 to 30 per cent in favor of the water-tube marine boiler. This saving in space, together with the reduction in weight referred to above, is of particular importance when there are limitations set as to length and draught of a ship. Under such conditions, it is frequently possible to obtain the desired speed only through the use of water-tube boilers.

Cleaning—When the great difference is realized, both in economy and capacity, of a clean boiler and one in which the heating surfaces have been allowed to become fouled, it will be appreciated that the ability to keep the heating surface clean externally and internally is a factor of the highest importance.

Such results can be accomplished only by the use of a design in boiler construction that gives complete accessibility to all parts. In Scotch marine boilers the tubes are nested together with but little space between them and, as a consequence, nearly the entire tube surface is inaccessible. When scale forms on such tubes it is impossible to completely remove it from the inside of the boiler. If it is removed by a turbine hammer, there is no way of knowing how thorough a job has been done. With the formation of such scale there is danger of overheating, and frequent tube renewals are necessary.

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Steamer "Minnewaska," 18,000 Shaft Horse-power, Babcock & Wilcox Boilers
Owners, International Mercantile Marine Company (Atlantic Transport Line)

In Scotch marine boilers extensive tube renewals at intervals of six or seven years are usually required, while replacements are often necessary in less than one year.

Water-tube boilers in general are, from the nature of their design, more readily accessible for cleaning than are Scotch boilers.

Inspection—The objections given above, in the consideration of the inability to clean Scotch boilers properly, hold as well for the inspection of such boilers.

Repairs—The lack of accessibility in Scotch boilers further leads to difficulties where repairs are required.

In Scotch boilers, tube renewals may be a serious undertaking. Any accumulation of hard deposit on the exterior of the tube surfaces may so enlarge the tube as to make it impossible to draw it through the tube sheet, and in such cases it is necessary to cut out such tubes as will allow access to the one which has failed and remove them through the manhole.

BABCOCK & WILCOX MARINE BOILERS AS COMPARED WITH OTHER WATER-TUBE DESIGNS

It must be borne in mind that, because a boiler is of the water-tube design, it does not necessarily follow that it is a good or a safe boiler.

Safety—The main feature to which the Babcock & Wilcox marine boiler owes its safety is the construction made possible by the use of headers, by which the water in each vertical row of tubes is separated from that in the adjacent rows. This construction results in the very efficient circulation produced through the breaking up of the steam and water in the front headers. The use of a number of sections, thus composed of headers and tubes, has a distinct advantage over the use of a common chamber at the outlet ends of the tubes. In the former case, the circulation of water on one vertical row of tubes cannot interfere with that in the other rows, while in the latter construction there will be downward as well as upward currents. Such downward currents tend to neutralize any good effect there might be through the diminution of the density of the water column by the steam.

Further, the circulation results directly from the design of the boiler and requires no assistance from hydrokineters, internal automatic circulators and the like. All such mechanical devices in the interior of a boiler serve only to complicate the design and should not be required in a properly designed boiler.

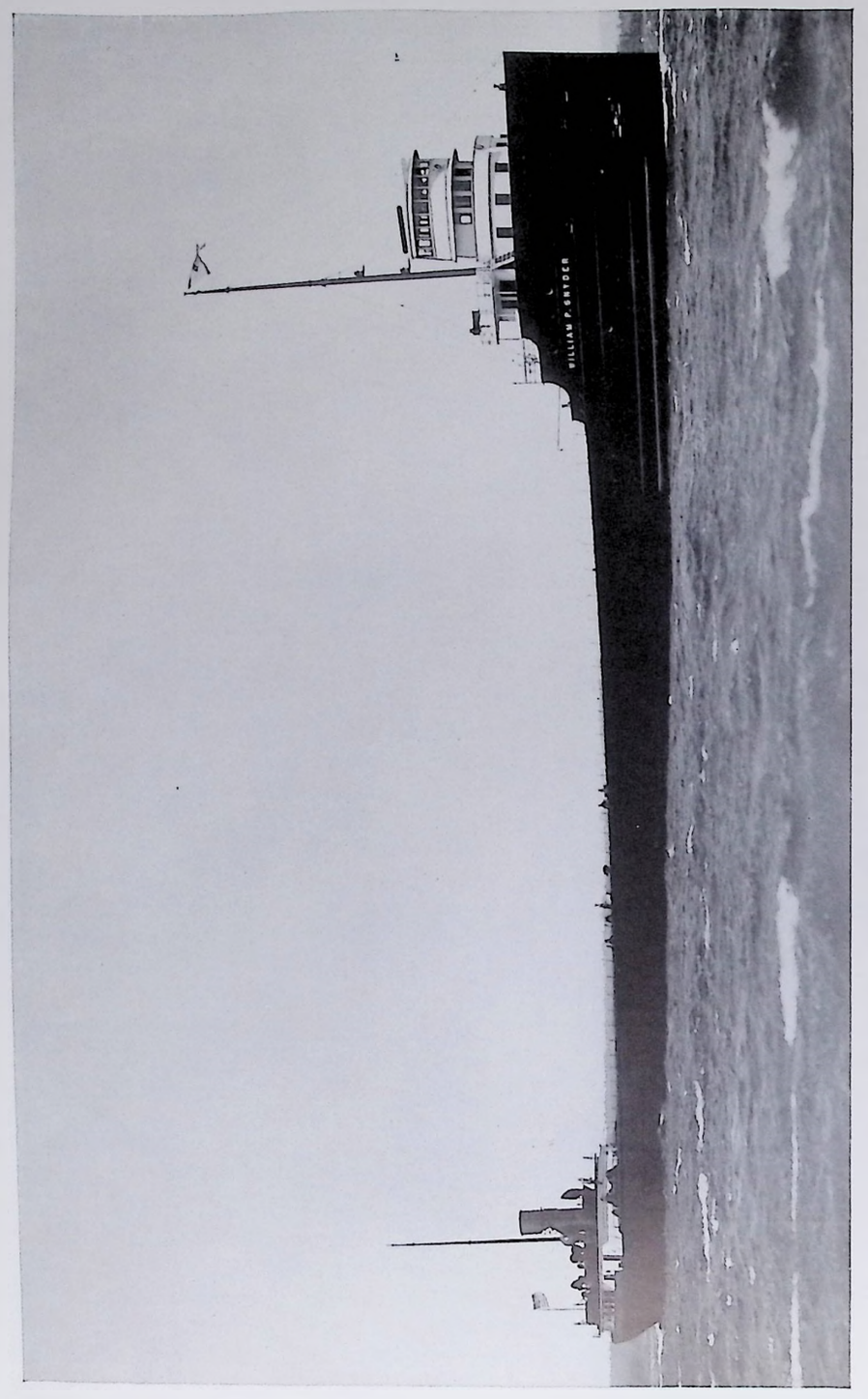
This positive and efficient circulation assures that all portions of the pressure parts of the Babcock & Wilcox boiler will be at approximately the same temperature, and in this way strains resulting from unequal temperatures are obviated.

Where the water throughout the boiler is at the temperature of the steam contained, a condition to be secured only by proper circulation, danger from internal pitting is minimized, or at least limited only to effects of the water fed the boiler. Where the temperature of the water in any portion of the boiler is lower than the temperature of the steam corresponding to the pressure carried, whether the fact that such lower temperatures exist as a result of lack of circulation or because of intentional design, internal pitting or corrosion will almost invariably result.

Staybolts are admittedly an undesirable element of construction in any boiler. They are wholly objectionable and the only reason for the presence of staybolts in a boiler is to enable a cheaper form of construction to be used than if they were eliminated.

In boilers utilizing in their design flat-stayed surfaces, or staybolt construction,

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Steamer "William P. Snyder," 1760 Indicated Horse-power, Babcock & Wilcox Boilers
Owners, Shenango Steamship Company, Cleveland, O.

under pressure, corrosion and wear and tear in service tend to weaken some single part subject to continual strain, the result being an increased strain on other parts greatly in excess of that for which an allowance can be made by any reasonable factor of safety. Where the construction is such that the weakening of a single part will produce a marked decrease in the safety and reliability of the whole, it follows of necessity that there will be a corresponding decrease in the working pressure which may be safely carried.

In water-leg boilers, the use of such flat-stayed surfaces under pressure presents difficulties that are practically insurmountable. Such surfaces, exposed to the heat of the fire, are subject to unequal expansion, distortion, leakage and corrosion, or, in general, to many of the objections that have already been advanced against the fire-tube boiler in the consideration of water-tube boilers as a class in comparison with fire-tube boilers.

Aside from the difficulties that may arise in actual service due to the failure of staybolts, or, in general, due to the use of flat-stayed surfaces, constructional features are encountered in the actual manufacture of such boilers that make it difficult if not impossible to produce a first-class mechanical job. It is practically impossible in the building of such a boiler to so design and place the staybolts that all will be under equal strain. Such unequal strains, resulting from constructional difficulties, will be greatly multiplied when such a boiler is placed in service. Much of the riveting in boilers of this design must of necessity be handwork, which is never the equal of machine riveting. The use of water-leg construction ordinarily requires the flanging of large plates, which is difficult, and because of the number of heats necessary and the continual working of the material, may lead to the weakening of such plates.

Where stayed surfaces or water legs are features in the design of a water-tube marine boiler, the factor of safety of such parts must be most carefully considered. In such parts, too, is the determination of this factor most difficult, and because of the "rule of thumb" determination frequently necessary, the factor of safety becomes in reality a factor of ignorance. As opposed to such indeterminate factors of safety, when the factor of safety of the drum has been determined in the Babcock & Wilcox marine boiler, and such a factor can be determined *accurately,* the factors for all the other portions of the pressure parts are greatly in excess of that of the drum. All Babcock & Wilcox marine boilers are built with a factor of safety in accordance with the marine regulations under which they are built, and inasmuch as the factor of safety of the tubes and the headers is greatly in excess of this figure, it applies specifically to the drum. This factor represents a greater degree of safety than a considerably higher factor applied to a boiler in which any riveted portion is acted upon directly by the fire, or the same factor applied to a boiler utilizing flat-stayed surface construction, where the accurate determination of the limiting factor of safety is difficult, if not impossible.

In view of the above, the absence of stayed surfaces in the Babcock & Wilcox marine boiler is obviously a distinguishing advantage where safety is a factor. It is of interest to note that in the evolution of the Babcock & Wilcox boiler, both marine and stationary, staybolt construction was used in several designs, found unsatisfactory and unsafe, and discarded.

Economy—While it has been stated that, in general, water-tube marine boilers are, from their design, more efficient than Scotch marine boilers, this does not mean

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STEAMER "ALASKA," 5600 INDICATED HORSE-POWER, BABCOCK & WILCOX BOILERS
ALASKA STEAMSHIP CO., SEATTLE, WASH.

that because a boiler is of the water-tube type it will necessarily show the maximum efficiency. For such maximum efficiency, a proper co-ordination of fuel, furnace, and heating surface is essential. An extended experience with different fuels for different operating conditions in furnace design and in the proper arrangement of boiler heating surface with reference to the furnace has made The Babcock & Wilcox Company leaders in the field of economy, and the great number of Babcock & Wilcox marine boilers now in satisfactory operation constitute an unimpeachable testimonial to their ability to meet all conditions of service.

In some designs of water-tube marine boilers, the baffle arrangement is such as to render certain portions of the heating surface ineffective. This is not only the result of by-passing a portion of the heating surface—*i.e.* having such portion out of the direct path of the products of combustion, but also because there is a tendency for soot and dirt to collect in such dead spaces and form an insulating layer on the tubes.

In Babcock & Wilcox marine boilers, the vertical baffles give a cross passage of the gases over the tubes, a direction of gas flow which has been proven to result in a maximum heat transfer rate, and place all of the heating surface in the direct path of the gases in their passage through the boiler.

Tests of Babcock & Wilcox marine boilers given elsewhere in this book will clearly indicate the high efficiencies being obtained over a wide range of conditions.

CAPACITY—It has been shown that the construction which utilizes headers and tubes to form sections in water-tube marine boilers leads to the maximum definite and rapid circulation. Because of the use of such construction, Babcock & Wilcox marine boilers have been operated at higher rates of evaporation per square foot of heating surface than any other marine water-tube boiler in actual service.

When burning coal, hand fired, Babcock & Wilcox marine boilers have been operated at an evaporation of 14.76 pounds from and at 212 degrees per square foot of heating surface per hour. Using oil fuel, Babcock & Wilcox marine boilers have been operated at an evaporation of 15.83 pounds of water from and at 212 degrees per square foot of heating surface per hour. While the Express type of boiler does not employ a sectional header construction, the design of the Babcock & Wilcox Express type marine boiler is such as to assure the desired rapid and definite circulation, and with such a boiler, burning oil fuel with Babcock & Wilcox mechanical burners, an evaporation of 22.73 pounds from and at 212 degrees per square foot of heating surface per hour has been obtained.

QUICK STEAMING—The advantages of water-tube boilers as a class over Scotch boilers in ability to raise steam quickly has been indicated.

Due to the positive and rapid circulation resulting from the design of the Babcock & Wilcox marine boiler, steam may be raised more rapidly than in practically any other design of water-tube marine boiler. In the Babcock & Wilcox marine boiler, the water is divided into many small streams, which circulate without undue frictional resistance in thin envelopes passing through the hottest part of the furnace, the steam being carried rapidly to the disengaging surface. The pressure parts that are exposed to the heat of the furnace are everywhere in contact with water internally and, as a result, there is no danger of overheating on starting up quickly, nor can leaks occur from unequal expansion, such as might be the case if an attempt is made to raise steam quickly in a boiler using stay-bolted water legs or which is of rigid construction otherwise.

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With a 2-inch tube Babcock & Wilcox marine boiler, burning coal, in a test where forced draft was available, steam was raised from an initial temperature of boiler and its contained water of 72 degrees to a pressure of 200 pounds, in 12½ minutes after lighting fires.

In a test of a 2-inch tube Babcock & Wilcox marine boiler burning oil with Babcock & Wilcox mechanical oil burners, and with an air pressure of 2½ inches in the fireroom, steam was raised from an initial temperature of boiler and its contained water of 105 degrees, to a pressure of 190 pounds, when the boiler was cut into the line, in 8 minutes and 45 seconds after the first burner was lighted.

In both of these tests, regardless of the short length of time between lighting fires and reaching full boiler pressure, there was no sign of strain or leakage resulting from any unequal temperature of pressure parts.

Accessibility—That water-tube marine boilers are, as a class, more accessible for inspection and cleaning than are Scotch boilers has been indicated. All water-tube boilers, however, are not equally accessible. In certain designs, due to the arrangement of baffling used, it is practically impossible to remove all deposits of soot and dirt. Frequently, in order to cheapen the product, cleaning and access doors are not supplied in sufficient number as part of the boiler equipment.

It is almost impossible to thoroughly clean water legs internally, and in such places also there is a tendency toward unsuspected corrosion under deposits that cannot be removed.

In Babcock & Wilcox marine boilers, every portion of the interior of the heating surfaces can be reached and kept clean, while any soot deposited on the exterior surfaces may be blown off while the boiler is under pressure or removed with hand tools when the boiler is cold.

Repairs—In Babcock & Wilcox marine boilers, the possession of great strength, the elimination of stresses due to uneven temperatures and of the resulting danger of leaks and corrosion, the protection of the drums from the intense heat of the furnace, and the decreased liability of any scale-forming matter to lodge on the hottest tube surfaces, all tend to minimize the necessity for repairs. The tubes of the Babcock & Wilcox marine boiler are practically the only parts which may need renewal and, with proper operation, these only at infrequent intervals. A small stock of tubes of the proper diameter and length for the boilers in question is all that need be carried to make renewals and such renewals may be made cheaply and quickly by the ship's force, thus eliminating the considerable expense of shore gang repairs.

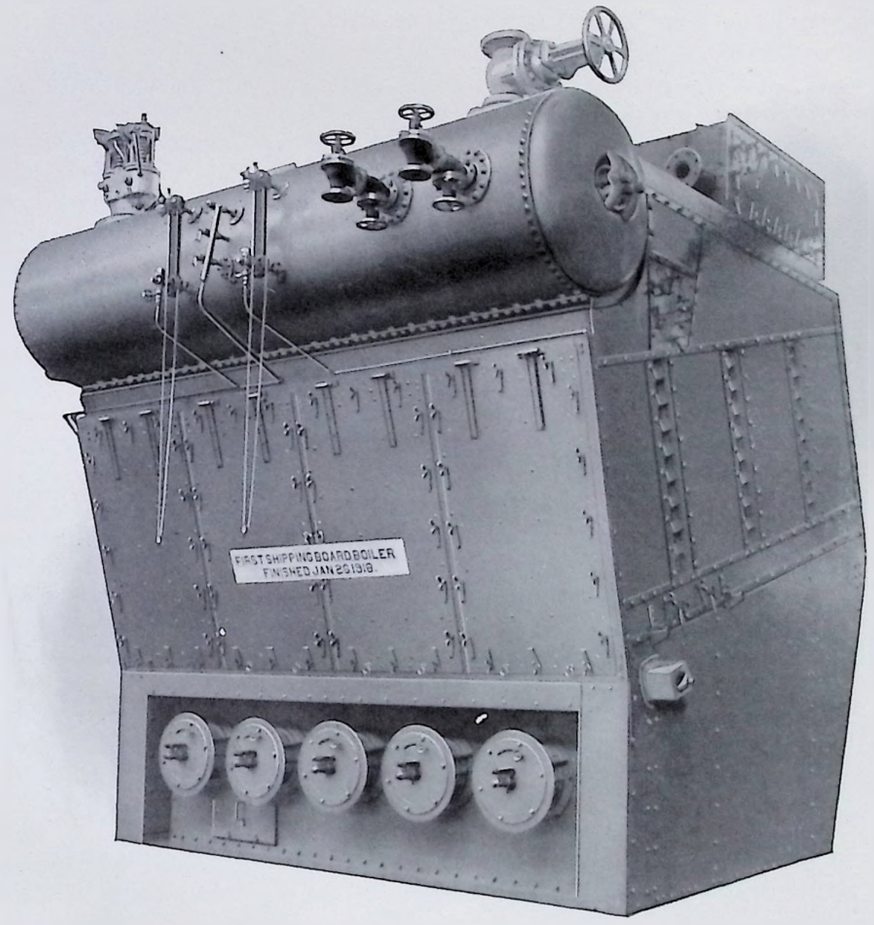
Repairs in water-leg boilers are difficult at best, and frequently unsatisfactory when completed. When staybolt replacements are necessary, in order to get at the inner sheet of the water leg, a number of tubes must sometimes be cut out. Not infrequently a replacement of an entire water leg is necessary and this is difficult and requires a lengthy shutdown. With the Babcock & Wilcox marine boilers, on the other hand, even if it is necessary to replace a section, this may be done in a few hours after the boiler is cool.

In the case of certain staybolt failures, the working pressure of a repaired boiler utilizing such construction will frequently be lowered by the inspectors when the boiler is again placed in service. The sectional nature of the Babcock & Wilcox marine boiler enables it to maintain its original working pressure over long periods of time, almost regardless of the nature of any repair that may be required. There is no case on

record of the pressure for which a Babcock & Wilcox marine boiler was originally built having been reduced by the Steamboat Inspection Service.

Durability—Babcock & Wilcox marine boilers that have been operated over twenty-five years are still in every-day service, with entirely satisfactory results and under the same steam pressure as that for which they were originally designed.

Probably the very best evidence of the value of the Babcock & Wilcox boiler, both marine and stationary, as a steam generator and of the reliability of the apparatus, is seen in the sales of the company. Since the company was formed, there have been sold throughout the world over 100,000,000 square feet of heating surface in this type of boiler alone.



Babcock & Wilcox Water-tube Marine Boiler—Shipping Board Type—Equipped with Babcock & Wilcox Superheater and Oil Burners. Heating Surface 2902 Square Feet; Superheating Surface 290 Square Feet; Working Pressure 200 Pounds

A feature that cannot be overlooked in the consideration of the advantages of the Babcock & Wilcox marine boiler is the fact that as part of the organization back of the boiler there is a body of engineers of recognized ability, ready at all times to assist its customers in every way.

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HEAT AND ITS MEASUREMENT

THE usual conception of heat is that it is a form of energy produced by the vibratory motion of the minute particles or molecules of a body. All bodies are assumed to be composed of these molecules, which are held together by mutual cohesion and yet are in a state of continual vibration. The hotter the body or the more heat added to it, the more vigorous is the vibration of the molecules.

The effect of heat on a body may be to change its temperature, its volume, or its state—that is, from solid to liquid or from liquid to gaseous. Where ice is melted to water and evaporated into steam, the various changes are admirably described in the lecture by Mr. Babcock on "The Theory of Steam Making" given in the thirty-sixth edition of our treatise on "Steam."

Maxwell defines the temperature of a body as "its thermal state with reference to its power of communicating heat to other bodies."

From this definition as indicating temperature as purely relative, an arbitrary scale of some sort must be assumed for temperature measurement. The ideal scale is the thermodynamic scale which Kelvin defines as a scale such that "the absolute values of two temperatures are to one another in proportion of the heat taken in to the heat rejected in a reversible thermodynamic engine working with a source and refrigerator at the higher and lower temperatures, respectively." The arbitrary unit degree is defined by giving the temperature interval between the freezing and the boiling points of pure water under standard conditions a value of 100 degrees Centigrade. No method of temperature measurement uses this principle, but scales defined by other means are in agreement with the thermodynamic scale or vary from it by small known amounts.

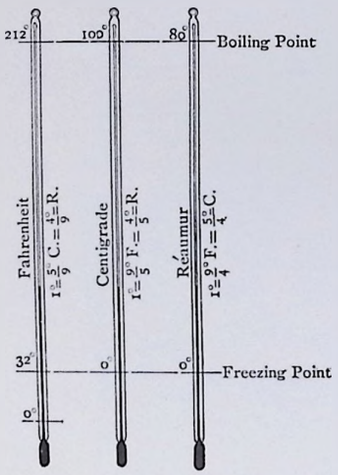


FIG. 1

The high temperature scale is defined by nitrogen expansion in the constant-volume gas thermometer. Such thermometers are not used commercially but serve as fundamental standard instruments. A series of fixed points defined by the boiling or melting points of chemical elements or compounds is determined by the gas thermometer and corrected to agree with the true thermodynamic scale. These points determine the scale as defined by other forms of pyrometers. The standard fixed points used in thermometry are given in Table 1, on page 39.

The temperature interval between the freezing and the boiling points of pure water under standard conditions is the unit of temperature used in all thermometric scales. The division of this interval into smaller units or degrees, however, differs in different scales. There are in use today three thermometric scales, known as the Fahrenheit, the Centigrade or Celsius, and the Réaumur. As shown in Fig. 1, in the Fahrenheit scale, the interval between the two fixed points is divided into 180 parts; the boiling point is marked 212 and the freezing point 32, and zero is a temperature which, at the time this thermometer was invented, was incorrectly imagined to be the lowest temperature

obtainable. In the Centigrade and Réaumur scales, the fixed interval is divided into 100 and 80 parts or degrees, respectively. In each of these scales, the freezing point is marked zero, and the boiling point in the Centigrade scale 100 and in the Réaumur scale 80.

Table 2 and appended formulæ are useful in converting from one scale to another.

In measuring any temperature it is essential for the measuring instrument to be in such relation to the body whose temperature is desired that the two are in

TABLE 1

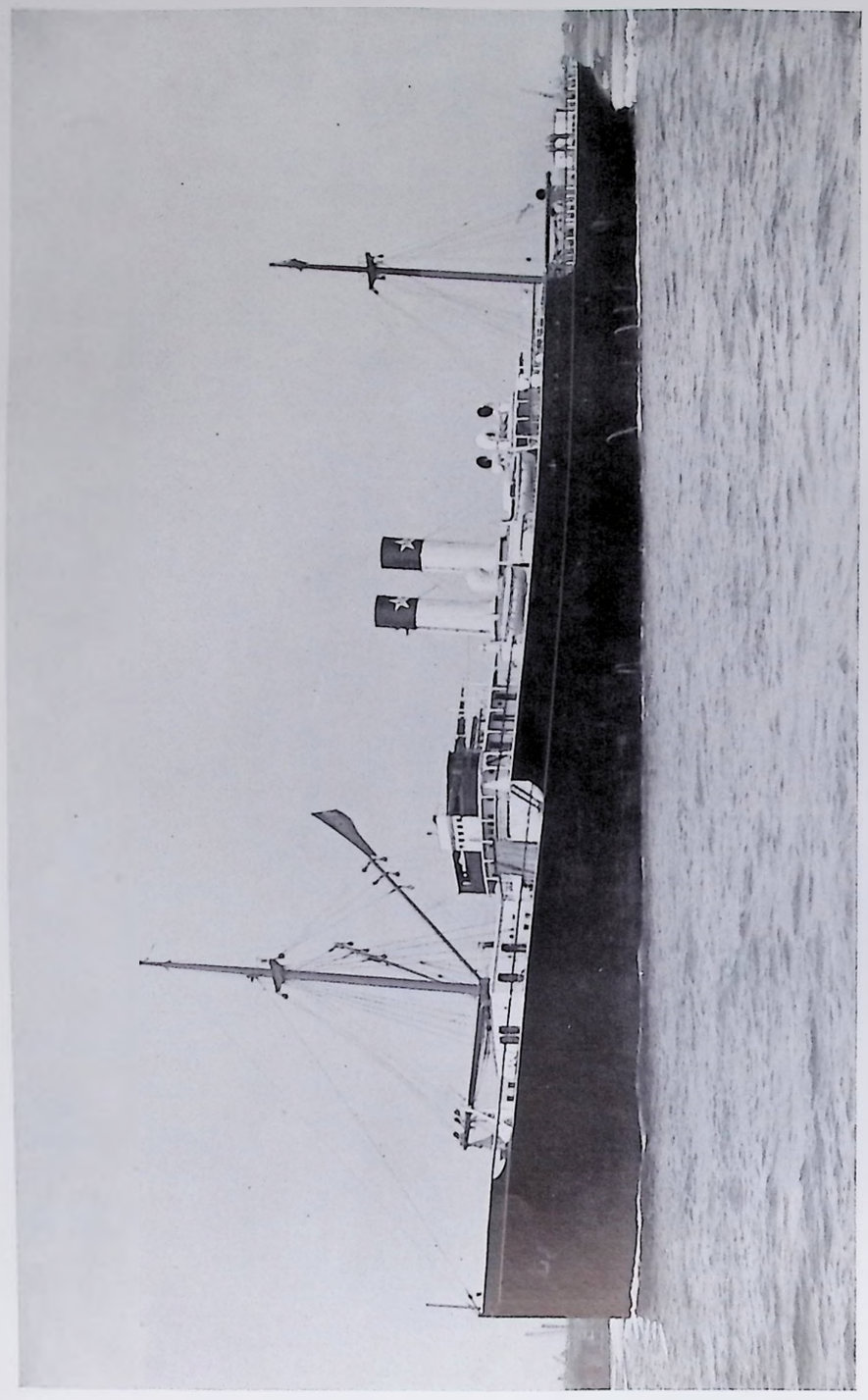
STANDARD FIXED POINTS USED IN THERMOMETRY

Substance	Transformation	Temperature	
		Degrees Centigrade	Degrees Fahrenheit
Oxygen	Boiling Point	−182.98	−297.36
Carbon Bisulphide	Melting Point	−112.0	−169.6
Carbon Dioxide	Boiling Point	−78.5	−109.3
Mercury	Melting Point	−38.88	−37.98
Water	Melting Point	0.00	32.00
Water	Boiling Point	100.00	212.00
Naphthalene	Boiling Point	217.95	424.31
Tin	Melting Point	231.9	449.4
Benzophenone	Boiling Point	305.9	582.6
Cadmium	Melting Point	320.9	609.6
Zinc	Melting Point	419.4	786.9
Sulphur	Boiling Point	444.55	832.19
Antimony	Melting Point	630.0	1166.0
Aluminum	Melting Point	658.7	1217.7
Silver	Melting Point	960.2	1760.4
Gold	Melting Point	1062.6	1944.7
Copper	Melting Point	1082.8	1981.0
Lithium Metasilicate	Melting Point	1201.	2293.8
Diopside	Melting Point	1391.5	2536.7
Nickel	Melting Point	1452.6	2646.7
Palladium	Melting Point	1549.5	2821.1
Platinum	Melting Point	1755.	3191.

N. B.—The boiling points are at standard pressure.

complete thermal equilibrium, *i. e.*, that they are at the "same" temperature. Because of variation in temperatures to be measured and of bodies whose temperatures are desired, and because of limitations in the different measuring devices, thermometry is necessarily subdivided into two divisions: The first involves the insertion of the measuring device into the body whose temperature is desired and obtaining results through heat conduction, and the second, the use of a measuring instrument at a distance, involving radiation. Radiation methods are dependent upon the fact that all bodies are radiating energy, that the amount radiated varies with the temperature, and that the energy may be measured.

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Steamer "El Coston," 7100 Shaft Horse-power, Babcock & Wilcox Boilers, Superheaters and Oil Burners
Owners, Southern Pacific Company, New York, N. Y.

Low and moderate temperatures are ordinarily measured by thermometers. High temperatures are measured by some form of pyrometer, discussed hereafter under the heading, "High Temperature Measurements." Mercury is the fluid most commonly used in the construction of thermometers. In the United States, the bulbs of high-grade thermometers are usually made of either Jena 59^{III} borosilicate thermometer glass or Jena 16^{III} glass, the stems being made of ordinary glass. The Jena 16^{III} is not suitable for use at temperatures much above 850 degrees Fahrenheit and the harder Jena 59^{III} should be used in thermometers for temperatures higher than this.

TABLE 2

COMPARISON OF THERMOMETER SCALES

	Fahrenheit	Centigrade	Réaumur		Fahrenheit	Centigrade	Réaumur
Absolute Zero . .	−459.6	−273.1	−218.5		50	10	8
	0	−17.78	−14.22		75	23.89	19.11
	10	−12.22	−9.78		100	37.78	30.22
	20	−6.67	−5.33		200	93.33	74.67
	30	−1.11	−0.89	Boiling Point . .	212	100	80
Freezing Point .	32	0	0		250	121.11	96.89
Maximum Density					300	148.89	119.11
of Water . . .	39.1	3.94	3.15		350	176.67	141.33

$F=\frac{9}{5}C+32°=\frac{9}{4}R+32°$ $\quad C=\frac{5}{9}(F-32°)=\frac{5}{4}R$ $\quad R=\frac{4}{9}(F-32°)=\frac{4}{5}C$

Below the boiling point, the hydrogen-gas thermometer is the almost universal standard with which mercurial thermometers may be compared, while above this point the nitrogen-gas thermometer is used. In both of these standards, the change in temperature is measured by the change in pressure of a constant volume of the gas.

In graduating a mercurial thermometer for the Fahrenheit scale, ordinarily a degree is represented as $\frac{1}{180}$ part of the volume of the stem between the readings at the melting point of ice and the boiling point of water. For temperatures above the latter, the scale is extended in degrees of the same volume. For very accurate work, however, the thermometer may be graduated to read true gas-scale temperatures by comparing it with the gas thermometer and marking the temperatures at 25 or 50 degree intervals. Each degree is, then, $\frac{1}{25}$ or $\frac{1}{50}$ of the volume of the stem in each interval.

Every thermometer, especially if intended for use above the boiling point, should be suitably annealed before it is used. If this is not done, the true melting point and also the "fundamental interval," that is, the interval between the melting and the boiling points, may change considerably. After continued use at the higher temperatures, also, the melting point will change, so that the thermometer must be calibrated occasionally to insure accurate readings.

As a general rule, thermometers are graduated to read correctly for total immersion, that is, with bulb and stem of the thermometer at the same temperature, and they should be used in this way when compared with a standard thermometer. If the stem emerges into space either hotter or colder than that in which the bulb is placed,

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a " stem correction " must be applied to the observed temperature in addition to any correction that may be found in the comparison with the standard. For instance, for a particular thermometer, comparison with the standard with both fully immersed made necessary the following corrections:

Temperature	*Correction*	*Temperature*	*Correction*
40°F.	0.0	300°F.	+2.5
100	0.0	400	—0.5
200	0.0	500	—2.5

When the sign of the correction is positive (+) it must be added to the observed reading, and when the sign is a negative (—) the correction must be subtracted.

The formula for the stem correction is as follows:

$$\text{Stem correction} = 0.000085 \times n\,(T - t) \qquad (1)$$

in which T is the observed temperature, t is the mean temperature of the emergent column, n is the number of degrees of mercury column emergent, and 0.000085 is the difference between the coefficient of expansion of the mercury and that in the glass in the stem.

Suppose the observed temperature is 400 degrees and the thermometer is immersed to the 200-degree mark, so that 200 degrees of the mercury column project into the air. The mean temperature of the emergent column may be found by tying another thermometer on the stem with the bulb at the middle of the emergent mercury column, as in Fig. 2. Suppose this mean temperature is 85 degrees, then

$$\text{Stem correction} = 0.000085 \times 200 \times (400 - 85) = 5.3 \text{ degrees}$$

As the stem is at a lower temperature than the bulb, the thermometer will evidently read too low, so that this correction must be added to the observed reading to find the reading corresponding to total immersion. The corrected reading will therefore be 405.3 degrees. If this thermometer is to be corrected in accordance with the calibrated corrections given above, we note that a further correction of 0.5 must be applied to the observed reading at this temperature, so that the correct temperature is 405.3—0.5=404.8 degrees or 405 degrees.

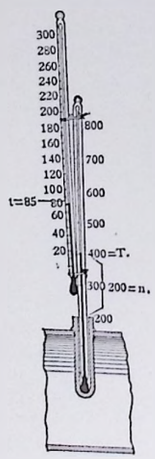


FIG. 2

Fig. 2 shows how a stem correction can be obtained for the case just described.

Fig. 3 affords an opportunity for comparing the scale of a thermometer correct for total immersion with one which will read correctly when submerged to the 300-degree mark, the stem being exposed at a mean temperature of 100 degrees Fahrenheit, a temperature often prevailing when thermometers are used for measuring temperatures in steam mains.

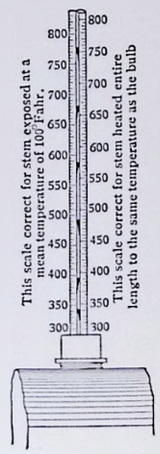


FIG. 3

ABSOLUTE ZERO—Experiments show that at 32 degrees Fahrenheit a perfect gas expands $\frac{1}{491.6}$ part of its volume if its pressure remains constant and its temperature is increased one degree. Thus if gas at 32 degrees Fahrenheit occupies 100 cubic feet and its temperature is increased one degree,

its volume will be increased to $100+\frac{100}{491.6}=100.203$ cubic feet, while for a temperature increase of two degrees the volume will be increased to $100+\frac{100\times 2}{491.6}=100.407$ cubic feet. If this rate of expansion per one degree held good at all temperatures, and experiment shows that it does above the freezing point, if the pressure remains constant, the volume would double if raised to a temperature of 32+491.6=523.6 degrees Fahrenheit, while under a diminution of temperature it would decrease in volume and finally disappear at a temperature of 491.6—32=459.6 degrees below zero Fahrenheit. While undoubtedly some change in the law would take place before the lower temperature could be reached, there is no reason why the law may not be used over the range of temperature within which it is known to hold good. It is evident from this explanation that under a constant pressure, the volume of a gas will vary directly as the number of degrees between its temperature and —459.6 degrees Fahrenheit. The value, —459.6, is uncertain by approximately two units in the last place, and for all ordinary engineering purposes, such as reductions of gas volumes, the value —460 is more than sufficiently accurate.

To simplify the application of this law, a new thermometric scale is constructed as follows: The point corresponding to —460 degrees Fahrenheit is taken as the zero point on the new scale, and the degrees are identical in magnitude with those of the Fahrenheit scale. Temperatures referred to this scale are called absolute temperatures and the point —460 degrees Fahrenheit is called absolute zero. It is obvious that the relation between the absolute temperature ($T°$) and the temperature of the Fahrenheit scale ($t°$) is

$$T° = t° + 460. \qquad (2)$$

Thus if one pound of gas is at 60 degrees Fahrenheit and another pound at 140 degrees Fahrenheit, the pressure at both temperatures being the same, the respective volumes will be in the ratio of 520 to 600.

BRITISH THERMAL UNIT—The quantitative measure of heat is the British thermal unit, ordinarily expressed as B.t.u. While this has until recently been ordinarily defined as the amount of heat necessary to raise the temperature of one pound of water at a definite temperature, one degree Fahrenheit, the value now commonly accepted is $\frac{1}{180}$th of the amount of heat required to raise one pound of pure water from the ice point (32 degrees Fahrenheit) to the steam point (212 degrees Fahrenheit), both at standard pressure. In the metric system, the quantitative measure of heat is the calorie, and is $\frac{1}{100}$th of the amount of heat required to raise the temperature of one kilogram of pure water from 0 to 100 degrees Centigrade. The relations of the two units are as follows:

Unit	*Water*	*Temperature Rise*
1 B.t.u.	1 Pound	1 Degree Fahrenheit
1 Calorie	1 Kilogram	1 Degree Centigrade

1 kilogram = 2.2046 pounds; 1 degree Centigrade = $\frac{9}{5}$ degrees Fahrenheit.

Hence 1 calorie = $(2.2046 \times \frac{9}{5}) = 3.968$ B.t.u.

The heat values are ordinarily given in B.t.u. per pound, and in calories per kilogram, in which case the B.t.u. per pound are approximately equivalent to $\frac{9}{5}$ the calories per kilogram.

As stated in the first law of thermodynamics, heat and mechanical energy are mutually convertible. The relation of heat energy to work as determined by Joule's

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original experiments indicated one B.t.u. as being equivalent to 772 foot-pounds. Joule's later experiments and those of Rowland and others indicated that this value was low. Some of the values more recently determined are: Mollier (1906), 778.23; Goodenough (1914), 777.64; Callendar (1915), 777.8; Marks and Davis (1916), 777.52. The value ordinarily accepted is 778 foot-pounds, the greatest variation from this value in any of those given being insignificant. Inasmuch as this value is uncertain in the last place, the use of figures beyond the decimal point is useless.

One horse-power is equivalent to 550 foot-pounds per second, or 1,980,000 foot-pounds per hour, while one kilowatt-hour is equivalent to 1.3410 horse-power-hours. On this basis, with the mechanical equivalent of one B.t.u. taken as 778 foot-pounds, we have

$$1 \text{ horse-power-hour} = \frac{1{,}980{,}000}{778} = 2545 \text{ B.t.u.}$$

$$1 \text{ kilowatt-hour} = \frac{1{,}980{,}000}{778 \times .7457} = 3413 \text{ B.t.u.}$$

Specific Heat—The specific heat of a substance is the quantity of heat expressed in thermal units required to raise unit weight of the substance through one degree of temperature. The specific heat of all substances varies with temperature, and since all substances vary in volume or pressure with temperature changes, it is necessary to distinguish between specific heats at constant volume and at constant pressure, expressed ordinarily as C_v and C_p, respectively.

Because of the low coefficients of expansion of solids and liquids, there is but little difference in the specific heat at constant volume and that at constant pressure. With gases, on the other hand, the difference is appreciable. When heat is added to a gaseous substance its volume may be kept constant, in which case no external work is done, or the gas may be allowed to expand during the addition of heat, the pressure being kept constant. The specific heat at constant volume, therefore, is always less than that at constant pressure by an amount equal to the heat required to do the work of expansion against external pressure.

In both specific heat at constant volume and at constant pressure, it is necessary to distinguish between *instantaneous* and *mean* specific heat. The instantaneous specific heat of a substance is the amount of heat that must be added to unit weight of such substance at some definite temperature to increase its temperature one degree under given conditions of pressure or volume. The mean specific heat of a substance, over a given temperature range, is the value by which such range must be multiplied to determine the quantity of heat required to raise unit weight of the substance through the range under the conditions of pressure or volume which exist.

In practically all boiler and combustion work, mean specific heats are used.

If a pound of pure water is considered a standard, the mean specific heat over a temperature range of 32 to 212 degrees Fahrenheit, from the definition of a B.t.u., is unity. The pound is the unit of weight considered in expressing the specific heat of all substances.

The specific heat of gases is of the greatest importance in combustion work, and is treated more fully in the chapter on "Combustion."

The specific heats of various solids and liquids are given in Table 3.

TABLE 3

SPECIFIC HEAT OF VARIOUS LIQUIDS AND SOLIDS

SOLIDS					
	Temperature* Degrees Fahrenheit	Specific Heat		Temperature* Degrees Fahrenheit	Specific Heat
Copper	59–460	.0951	Glass (normal ther. 16^{III})	66–212	.1988
Gold	32–212	.0316	Lead	59	.0299
Wrought Iron	59–212	.1152	Platinum	68–212	.0319
Cast Iron	68–212	.1189	Silver	32–212	.0559
Steel (soft)	68–208	.1175	Tin	–105–64	.0518
Steel (hard)	68–208	.1165	Ice	. . . .	.5040
Zinc	32–212	.0931	Sulphur (newly fused)	. . . .	.2025
Brass (yellow)	32	.0883			

LIQUIDS					
	Temperature* Degrees Fahrenheit	Specific Heat		Temperature* Degrees Fahrenheit	Specific Heat
Water†	32–212	1.0000	Sulphur (melted)	246–297	.2350
Alcohol	32	.5475	Tin (melted)	482	.0578
	176	.7694	Sea Water (sp. gr. 1.0043)	64	.980
Mercury	32	.03346	Sea Water (sp. gr. 1.0463)	64	.903
Benzol	50	.340	Oil of Turpentine	32	.411
	122	.423	Petroleum	70–136	.511
Glycerine	59–102	.576	Sulphuric Acid	68–133	.3363
Lead (melted)	500	.0356			

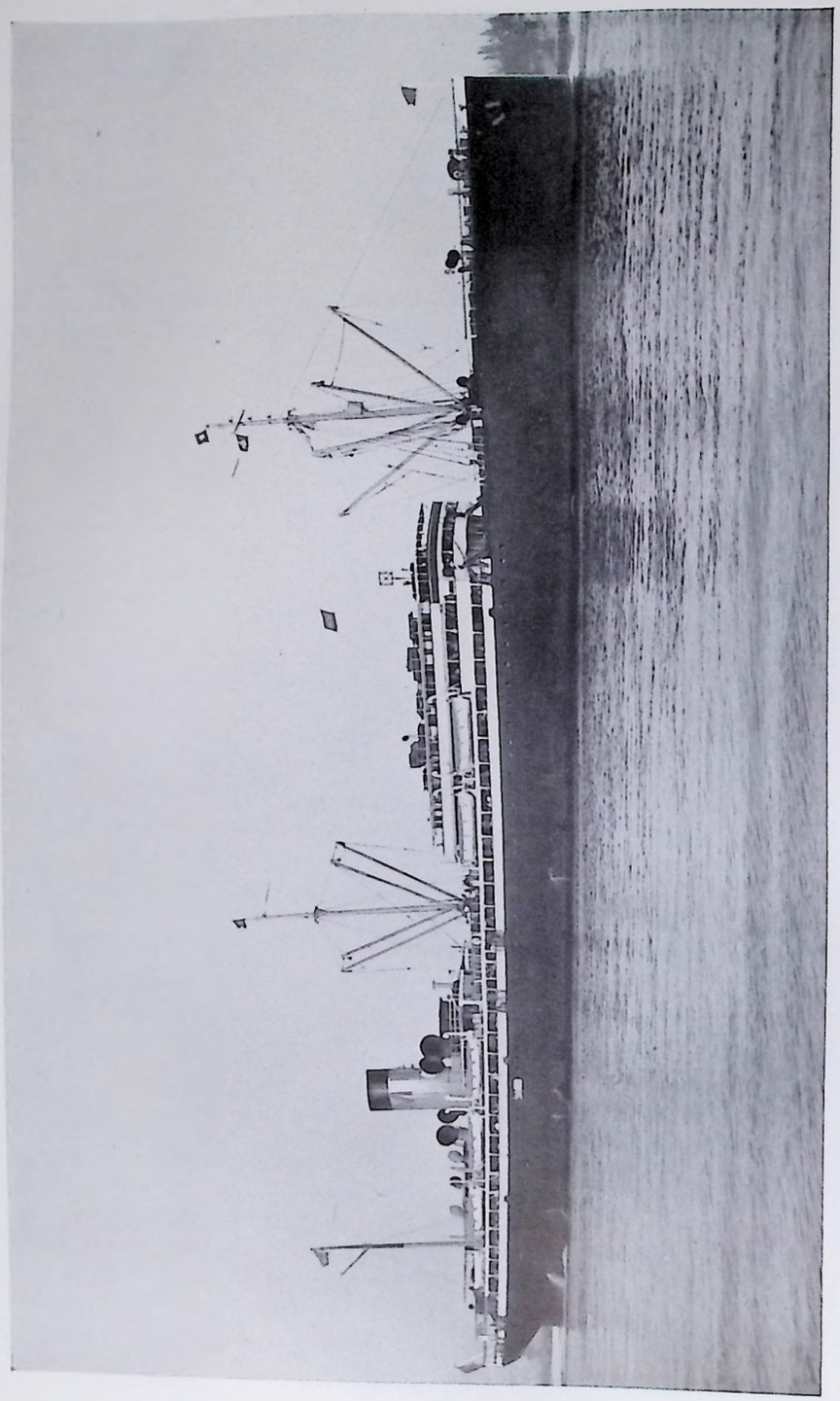
* When one temperature alone is given the "true" specific heat is given; otherwise the value is the "mean" specific heat for the range of temperature given.

† For variation, see Table 10.

Sensible Heat—The heat utilized in raising the temperature of a body is termed "sensible heat." In the case of water, the sensible heat required to raise it from 32 degrees Fahrenheit to the boiling point corresponding to the pressure at which steam is formed is termed the "heat of the liquid."

Latent Heat—Latent heat is the heat which apparently disappears in producing some change in the condition of a body without increasing its temperature. If heat be added to ice at freezing temperature, the ice will melt but its temperature will not be raised. The heat so utilized in changing the condition of the ice is the latent heat, and in this particular case is known as the "latent heat of fusion." If heat be added to water at 212 degrees under atmospheric pressure, the water will not become hotter but will be evaporated into steam, the temperature of which will also be 212 degrees. The heat so utilized is called the "latent heat of evaporation" and is the heat which apparently disappears in causing the substance to pass from a liquid to a gaseous state.

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STEAMER "MAUI," 10,000 SHAFT HORSE-POWER, BABCOCK & WILCOX BOILERS, SUPERHEATERS AND OIL BURNERS
OWNERS, MATSON NAVIGATION COMPANY, SAN FRANCISCO, CAL.

Latent heat is not lost, but reappears whenever the substances pass through a reverse cycle, from a gaseous to a liquid or from a liquid to a solid state. It may, therefore, be defined, as stated, as the heat which apparently disappears, or is lost to thermometric measurement, when the molecular constitution of a body is being changed. Latent heat is expended in overcoming the molecular cohesion of the particles of the substance and in overcoming the resistance of external pressure to a change of volume of the heated body. Latent heat of evaporation, therefore, may be said to consist of internal and external heat, the former being utilized in overcoming the molecular resistance of the water in changing to steam, while the latter is expended in overcoming any resistance to the increase of its volume during formation. In evaporating a pound of water at 212 degrees to steam at 212 degrees, 897.6 B.t.u. are expended as internal latent heat and 72.8 B.t.u. as external latent heat. For a more detailed description of the changes brought about in water by sensible and latent heat, the reader is again referred to the lecture by Mr. Babcock on "The Theory of Steam Making" in our treatise "Steam."

EBULLITION—The temperature of ebullition of any liquid, or its boiling point, may be defined as the temperature which exists where the addition of heat to the liquid no longer increases its temperature, the heat added being absorbed or utilized in converting the liquid into vapor. This temperature is dependent upon the pressure under which the liquid is evaporated, being higher as the pressure is greater.

TOTAL HEAT* OF EVAPORATION—The quantity of heat required to raise a unit of any liquid from the freezing point to any given temperature, and to entirely evaporate it at that temperature, is the total heat of evaporation of the liquid for that temperature. It is the sum of the heat of the liquid and the latent heat of evaporation.

To recapitulate, the heat added to a body is divided as follows:

Total heat = Heat to change the temperature + heat to overcome the molecular cohesion + heat to overcome the external pressure resisting an increase of volume of the body.

Where water is converted into steam, this total heat is divided as follows:

Total heat = Heat to change the temperature of the water + heat to separate the molecules of the water + heat to overcome resistance to increase in volume of the steam,
= Heat of the liquid + internal latent heat + external latent heat,
= Heat of the liquid + total latent heat of steam,
= Total heat of evaporation.

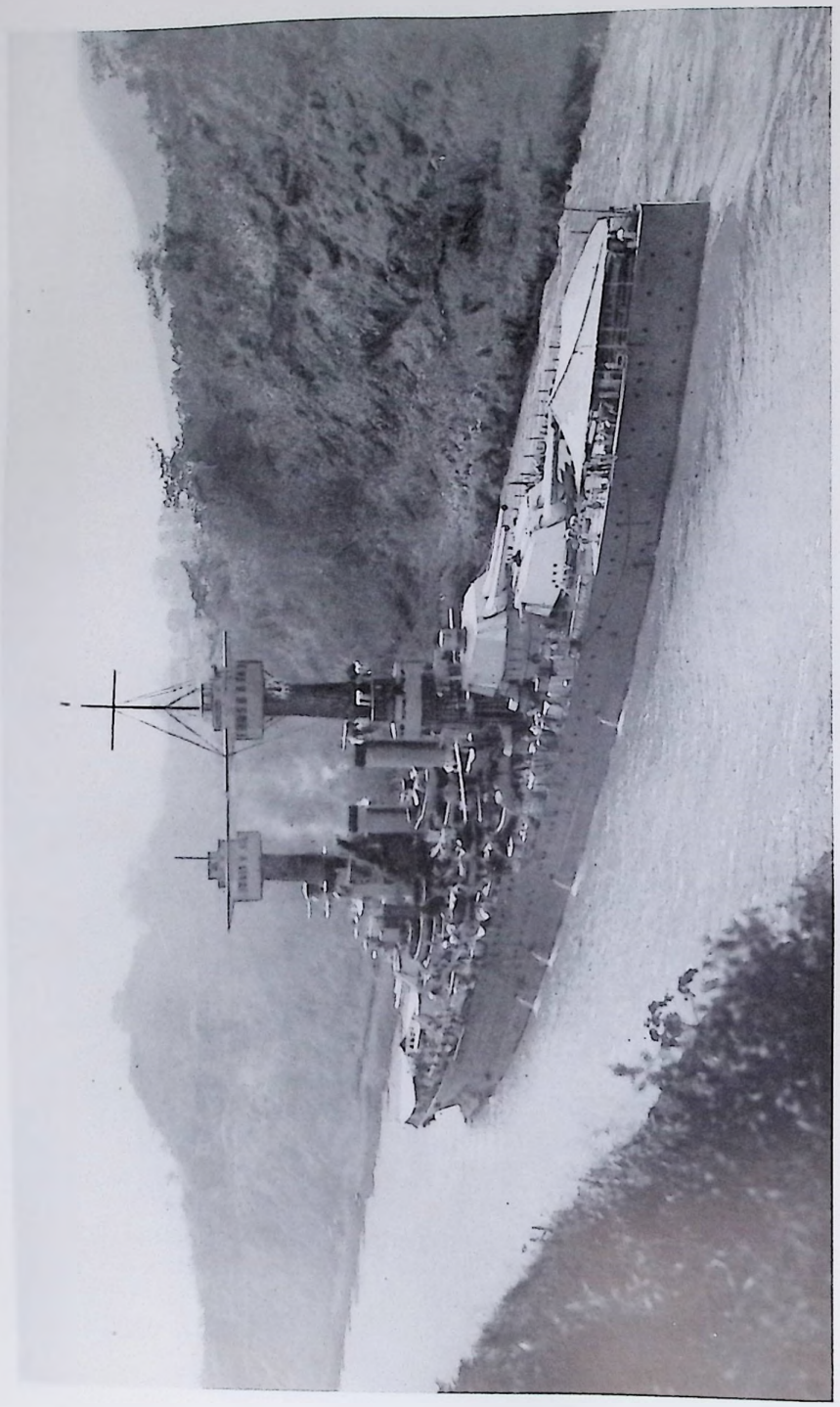
The steam tables given on pages 79 to 81 give the heat of the liquid and the total latent heat through a wide range of temperatures.

GASES—When heat is added to gases there is no internal work done; hence the total heat is that required to change the temperature plus that required to do the external work. If the gas is not allowed to expand but is preserved at constant volume, the entire heat added is that required to change the temperature only.

LINEAR EXPANSION OF SUBSTANCES BY HEAT—To find the increase in the length of a bar of any material due to an increase of temperature, multiply the number of

* The Committee on Definitions and Values of the A. S. M. E. Power Test Codes Committee has recommended that the term "Total Heat" be replaced by the term "Heat Content."

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U. S. Battleship "Maryland," 36,167 Shaft Horse-power, Babcock & Wilcox Boilers, Superheaters and Oil Burners Passing Through Culebra Cut, Panama Canal

degrees of increase in temperature by the coefficient of expansion for one degree and by the length of the bar. Where the coefficient of expansion is given for 100 degrees, as in Table 4, the result should be divided by 100. The expansion of metals per one degree rise of temperature increases slightly as high temperatures are reached, but for all practical purposes it may be assumed to be constant for a given metal.

TABLE 4

LINEAL EXPANSION OF SOLIDS AT ORDINARY TEMPERATURES

(Tabular values represent increase per foot per 100 degrees increase in temperature, Fahrenheit or Centigrade)

Substance	Temperature Conditions* Degrees Fahrenheit	Coefficient per 100 Degrees Fahrenheit	Coefficient per 100 Degrees Centigrade
Brass (cast)	32 to 212	.001042	.001875
Brass (wire)	32 to 212	.001072	.001930
Copper	32 to 212	.000926	.001666
Glass (English flint)	32 to 212	.000451	.000812
Glass (French flint)	32 to 212	.000484	.000872
Gold	32 to 212	.000816	.001470
Granite (average)	32 to 212	.000482	.000868
Iron (cast)	104	.000589	.001061
Iron (soft forged)	0 to 212	.000634	.001141
Iron (wire)	32 to 212	.000800	.001440
Lead	32 to 212	.001505	.002709
Mercury	32 to 212	.01014†	.018252†
Platinum	104	.000499	.000899
Limestone	32 to 212	.000139	.000251
Silver	104	.001067	.001921
Steel (Bessemer rolled, hard)	0 to 212	.00056	.00101
Steel (Bessemer rolled, soft)	0 to 212	.00063	.00117
Steel (cast, French)	104	.000734	.001322
Steel (cast annealed, English)	104	.000608	.001095

*Where range of temperature is given, coefficient is mean over range.
†Coefficient of cubical expansion.

High Temperature Measurements*—The temperatures to be dealt with in boiler practice range from those of ordinary air to the temperature of burning fuel. The gases of combustion, originally at the temperature of the furnace, cool as they pass over the boiler-heating surface, to a temperature approximating that due to the pressure within the boiler, resulting in a wide range of temperatures through which definite measurements are frequently required.

Of the different methods devised for ascertaining these temperatures, the most important are as follows:

Mercurial Thermometer or Pyrometers—While mercurial thermometers are ordinarily given a scale up to 1000 degrees, their accuracy is questionable above 750 or 800 degrees Fahrenheit.

Expansion Pyrometers—These may be divided into two classes, bimetallic and pressure thermometers. The former, because of instability—*i.e.* a change with varia-

* For a full discussion of high temperature measurements and the instruments for such measurements, see the Bureau of Standards Technologic Paper No. 170, "Pyrometric Practice."

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tion of the zero point—is not reliable unless frequently adjusted and checked, and is not extensively used.

Pressure thermometers may be subdivided into vapor pressure, liquid-filled, and gas-filled thermometers. These, provided chemically pure materials, and, in the case of the liquid and gas-filled thermometers, materials which have a critical temperature below the maximum temperature to be measured, are reliable up to 1500 degrees Fahrenheit.

Platinum Resistance Pyrometers—Such pyrometers are used for temperatures up to 1200 or 1400 degrees Fahrenheit. While remarkably accurate within the limits of their range, because of these limits and because of the precautions necessary in securing accuracy, platinum resistance pyrometers find a greater field of usefulness in laboratory, rather than commercial, work, such as boiler rooms.

Thermoelectric Pyrometers—Thermoelectric pyrometry is based upon the principle that if, in a closed circuit of two metals, the two junctions are at different temperatures, an electric current will flow through the circuit.

The metals in general use for thermoelectric couples vary with the temperature range to be covered and the degree of accuracy desired in temperature measurement. Since, however, any two dissimilar metals might be employed, it is necessary in the selection of such metals, to consider:

1st. The ability to resist corrosion and oxidation.

2nd. The development of a relatively high electromotive force.

3rd. That the temperature-electromotive force relation be such that the latter increases constantly with temperature rise over the range for which the couple is to be employed.

Thermocouples are usually divided into base metal and rare metal couples, the principle in the two being the same, the distinction being in the difference in initial cost, durability and accuracy.

Base metal couples may be used for temperatures up to 2000 degrees Fahrenheit, and rare metal couples for temperatures up to 2900 degrees, though in the latter the accuracy at the higher temperatures is questionable.

Radiation Pyrometers—This class of pyrometer measures the intensity of all wave lengths emitted by a glowing body, the light and the heat rays combined, this combined radiated energy ordinarily being focused in some manner on the hot junction of a small thermocouple. They may be used for measuring temperatures up to 3600 degrees Fahrenheit.

Due to the difference in the power of emission of energy, and light of different substances, errors in true temperatures as measured by radiation pyrometers may be as great as 400 degrees. Since the error is practically constant it is not of particular importance when radiation pyrometers are used in connection with heat control in industrial processes.

Optical Pyrometers—This class of pyrometer measures the intensity of a narrow spectral band of radiation emitted from a glowing body. They may be used for measuring temperatures up to 12,600 degrees Fahrenheit* though for ordinary boiler work their range is from 1000 to 3600 degrees.

* Le Chateliers investigations.

Radiation and optical pyrometers offer the only pyrometric means of measuring temperatures above 2725 degrees Fahrenheit. While the human element enters into the temperatures as determined by either type, they are rather remarkably accurate. Optical pyrometers, when designed specially for precision, are accurate to within 3.6 degrees Fahrenheit, and in commercial form are accurate within 10 degrees.

TABLE 5

APPROXIMATE MELTING POINTS OF METALS

Metal	Temperature Degrees Fahrenheit	Metal	Temperature Degrees Fahrenheit
Wrought Iron	2737	Lead	621
Pig Iron (gray)	2190–2327	Bismuth	520
Cast Iron (white)	2075	Tin	449
Steel	2460–2550	Platinum	3191
Steel (cast)	2500	Gold	1945
Copper	1981	Silver	1760
Zinc	786	Aluminum	1218
Antimony	1166		

Melting Point of Metals—Approximate temperatures, especially of gases, may be determined by introducing a series of chemically pure metals of which the melting points are known. By using such a series of metals in progressive rotation, a point will be reached where the temperature to be measured may be determined as being between the melting point of two of the metals in the series, one of them having melted, the other having resisted the temperature in question. Such a method lacks accuracy, but it suffices for determinations where approximate results are satisfactory.

The melting points of various metals that may be used for determinations of this nature are given in Table 5.

Determination of Temperature from Character of Emitted Light—The temperature evolved in combustion may be approximated from the appearance of the fuel mass or the flame. These approximate temperatures are indicated by the figures of Table 6. Such figures are of necessity but the roughest approximations, but in connection with flame length, are of some value where apparatus for more accurate determinations is not available.

TABLE 6

TEMPERATURE AND APPEARANCE OF FLAME*

Appearance of Flame	Temperature Degrees Fahrenheit
Dark Red	975
Dull Red	1290
Dull Cherry Red	1470
Full Cherry Red	1650
Clear Cherry Red	1830
Deep Orange	2010
White	2370
Bright White	2550
Dazzling White	2730

* C. S. M. Pouillet.

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PROPERTIES OF WATER

PURE water is a chemical compound of one volume of oxygen and two volumes of hydrogen, its chemical symbol being H_2O.

The density of water depends upon its temperature. Its density at four temperatures, much used in physical calculations, is given in Table 7.

TABLE 7

DENSITY OF WATER AT TEMPERATURES USED IN PHYSICAL CALCULATIONS

Temperature Degrees Fahrenheit	Density per Cubic Foot Pounds	Density per Cubic Inch Pounds
At 32 degrees or freezing point at sea level . .	62.418	0.03612
At 39.2 degrees or point of maximum density . .	62.427	0.03613
At 62 degrees or standard temperature	62.355	0.03608
At 212 degrees or boiling point at sea level . .	59.846	0.03469

The density of water has been very carefully determined and the results of the more trustworthy determinations are in very good agreement. For temperatures up to 212 degrees Fahrenheit, the values are known to an accuracy of one or two parts in 100,000. The values in Table 7 are quoted from the Bureau of Standards Circular No. 19, "Standard Density and Volumetric Tables."

The United States standard gallon is a volume of 231 cubic inches. A gallon (231 cubic inches) of water at 62 degrees Fahrenheit weighs 8.3357 pounds.

In the above a pound of water means the quantity of water which has a mass of one pound; in other words, the quantity which has a weight, corrected for the effect of the buoyancy of the air, of one pound. If water is weighed in air in the ordinary way, against brass or iron weights, the weight will be smaller than the mass by about one-tenth of one per cent, a difference which is negligible except where results are required to more than three significant figures. Table 14 gives the density of water at different temperatures, as stated by Marks and Davis.

TABLE 8

VOLUME AND WEIGHT OF DISTILLED WATER AT VARIOUS TEMPERATURES*

Temperature Degrees Fahrenheit	Relative Volume Water at 39.2 Degrees=1	Weight per Cubic Foot Pounds	Temperature Degrees Fahrenheit	Relative Volume Water at 39.2 Degrees=1	Weight per Cubic Foot Pounds	Temperature Degrees Fahrenheit	Relative Volume Water at 39.2 Degrees=1	Weight per Cubic Foot Pounds	Temperature Degrees Fahrenheit	Relative Volume Water at 39.2 Degrees=1	Weight per Cubic Foot Pounds
32	1.000176	62.42	160	1.02337	61.00	290	1.0830	57.65	430	1.197	52.2
39.2	1.000000	62.43	170	1.02682	60.80	300	1.0890	57.33	440	1.208	51.7
40	1.000004	62.43	180	1.03047	60.58	310	1.0953	57.00	450	1.220	51.2
50	1.00027	62.42	190	1.03431	60.36	320	1.1019	56.66	460	1.232	50.7
60	1.00096	62.37	200	1.03835	60.12	330	1.1088	56.30	470	1.244	50.2
70	1.00201	62.30	210	1.04256	59.88	340	1.1160	55.94	480	1.256	49.7
80	1.00338	62.22	212	1.04343	59.83	350	1.1235	55.57	490	1.269	49.2
90	1.00504	62.11	220	1.0469	59.63	360	1.1313	55.18	500	1.283	48.7
100	1.00698	62.00	230	1.0515	59.37	370	1.1396	54.78	510	1.297	48.1
110	1.00915	61.86	240	1.0562	59.11	380	1.1483	54.36	520	1.312	47.6
120	1.01157	61.71	250	1.0611	58.83	390	1.1573	53.94	530	1.329	47.0
130	1.01420	61.55	260	1.0662	58.55	400	1.167	53.5	540	1.35	46.3
140	1.01705	61.38	270	1.0715	58.26	410	1.177	53.0	550	1.37	45.6
150	1.02011	61.20	280	1.0771	57.96	420	1.187	52.6	560	1.39	44.9

*Marks and Davis

The British Imperial gallon is defined as the volume of 10 Imperial pounds of distilled water weighed in air against brass weights, with the water and the air at a temperature of 62 degrees Fahrenheit, the barometer being at 30 inches. This volume equals 277.41 cubic inches.

Water is but slightly compressible and for all practical purposes may be considered non-compressible. The coefficient of compressibility ranges from 0.000051 per atmosphere at 35 degrees Fahrenheit to 0.000044 at 128 degrees, decreasing as the temperature increases.

Table 8 gives the weight in vacuo and the relative volume of a cubic foot of distilled water at various temperatures.

The density of water at the standard temperature being taken as 62.355 pounds per cubic foot, the pressure exerted by the column of water of any stated height, and conversely the height of any column required to produce a stated pressure, may be computed as follows:

The pressure in pounds per square foot = 62.355 × height of column in feet.
The pressure in pounds per square inch = 0.433 × height of column in feet.
Height of column in feet = pressure in pounds per square foot ÷ 62.355.
Height of column in feet = pressure in pounds per square inch ÷ 0.433.
Height of column in inches = pressure in pounds per square inch × 27.71.
Height of column in inches = pressure in ounces per square inch × 1.73.

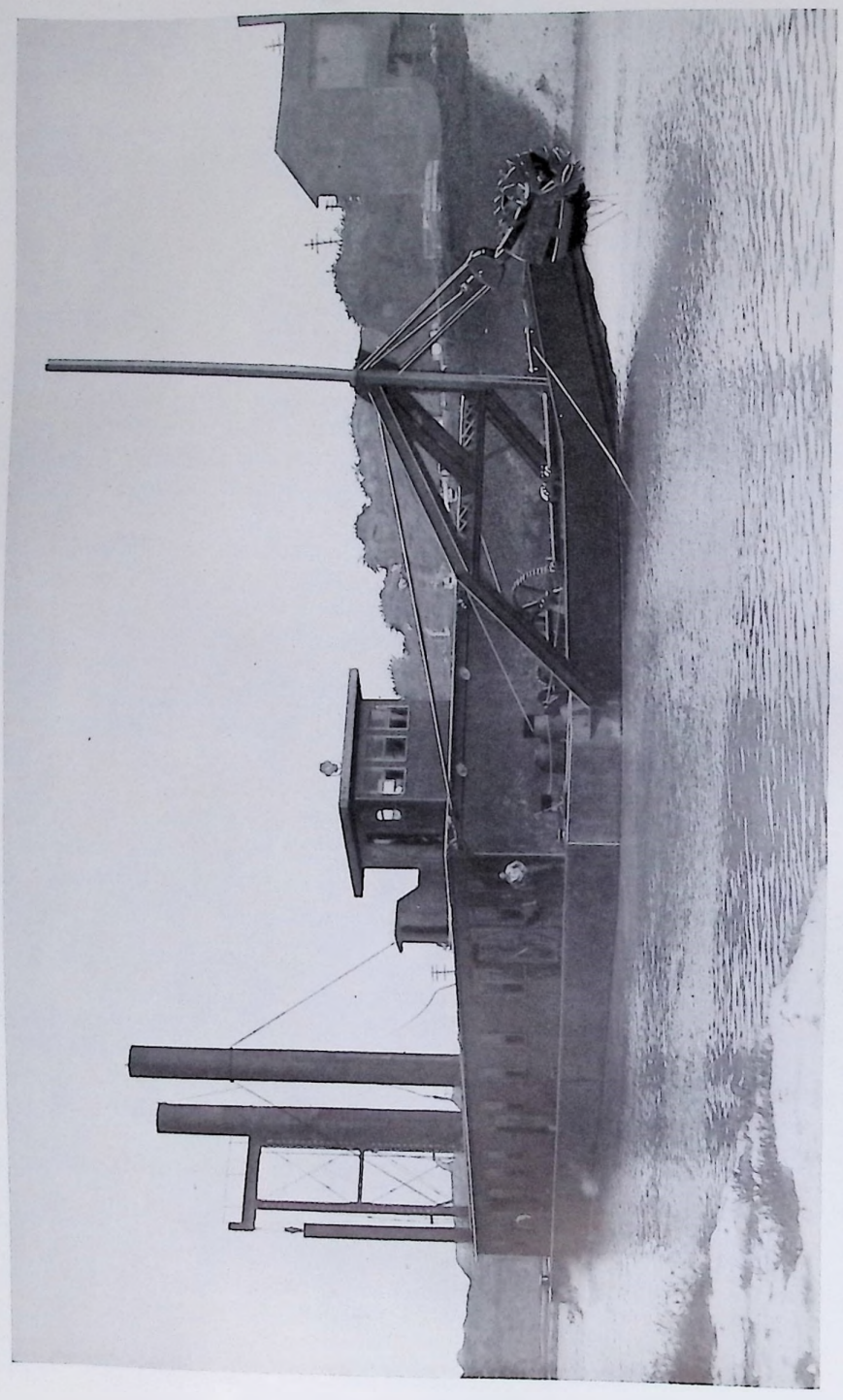
By a change in the densities given above, the pressure exerted and height of column may be computed for temperatures other than 62 degrees. Table 14 gives the densities at temperatures from 32 to 340 degrees Fahrenheit.

Where an accuracy of more than about one-tenth of one per cent is required, the pressures in pounds per square foot computed by the relations just stated must be corrected for the displaced air column, the variation of g and the variation of density with temperature. The error due to neglecting the difference between the local and standard values of g may be greater than the error due to neglecting the difference between the standard temperature and the actual temperature of the water column. Similarly the error due to neglecting the effect of the displaced air in calculating the pressure in pounds per square foot due to a water column, by the above relations, may amount to as much as the change due to a variation of 10 to 15 degrees in the temperature of the water. The data generally adopted by American physicists and engineers are given in Circular 19 of the U. S. Bureau of Standards, "Standard Density and Volumetric Tables."

A pressure of one pound per square inch is exerted by a column of water 2.3093 feet or 27.71 inches high at 62 degrees Fahrenheit.

Water in its natural state is never found absolutely pure. It has a solvent action of wider range than any other liquid possesses, and this action is affected by temperature. The solubility of sodium chloride increases somewhat with rising temperature, the solubility of magnesium sulphate increases much more rapidly, while the solubility of sodium sulphate increases rapidly with the temperature up to about 90 degrees Fahrenheit, and then decreases as the temperature rises above that point.

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Dredge "Lyons," Working on New York State Barge Canal, 2000 Indicated Horse-power, Babcock & Wilcox Boilers
Owners, Arundel Corporation

Sea water contains about 3.5 per cent, by weight, of dissolved solids comprising sodium chloride, 2.72 per cent; magnesium chloride, 0.38 per cent; magnesium sulphate, 0.17 per cent; calcium sulphate, 0.13 per cent; potassium sulphate, 0.08 per cent; calcium carbonate, 0.01 per cent; magnesium bromide, 0.01 per cent (Encyclopædia Britannica, eleventh edition, XIX, 977).

TABLE 9

EFFECT OF ALTITUDE IN REDUCING THE BOILING POINT OF WATER

Altitude Feet	Barometer at 32 Degrees Fahrenheit Inches	Boiling Point Degrees Fahrenheit	Atmospheric Pressure Pounds per Square Inch	Altitude Feet	Barometer at 32 Degrees Fahrenheit Inches	Boiling Point Degrees Fahrenheit	Atmospheric Pressure Pounds per Square Inch
0	29.92	212	14.70	8,000	22.38	198	10.98
500	29.43	211	14.46	8,500	21.96	197	10.78
1,000	28.93	210	14.21	9,000	21.55	196	10.58
1,500	28.44	209	13.97	9,500	21.15	195	10.38
2,000	27.94	209	13.72	10,000	20.75	194	10.19
2,500	27.44	208	13.48	10,500	20.36	193	10.00
3,000	26.94	207	13.23	11,000	19.98	192	9.81
3,500	26.44	206	12.99	11,500	19.60	191	9.63
4,000	25.95	205	12.75	12,000	19.24	191	9.45
4,500	25.47	204	12.52	12,500	18.88	190	9.27
5,000	25.01	203	12.29	13,000	18.52	189	9.09
5,500	24.55	202	12.06	13,500	18.17	188	8.92
6,000	24.09	201	11.83	14,000	17.82	187	8.75
6,500	23.65	200	11.61	14,500	17.47	186	8.58
7,000	23.22	199	11.39	15,000	17.12	185	8.41
7,500	22.80	199	11.18	15,500	16.77	184	8.24

The boiling point of water decreases as the altitude above the sea level increases. Table 9 gives the reduction in barometric pressure with the altitude as reported by Gregg in the Monthly Weather Review, January, 1918, with the corresponding boiling points of water and atmospheric pressures in pounds per square inch, computed from data in the Smithsonian Physical Tables, 1921. Tables of this nature are only approximately correct and need revision as research establishes more closely the relation between altitude and barometric pressure.

The specific heat of a substance is defined as the quantity of heat, per degree, required to raise the temperature of unit mass of the substance under specified conditions. As measured under ordinary conditions, at constant (atmospheric) pressure the specific heat of water is found to be larger than that of all but a few other substances. The specific heat of water depends upon the temperature, and numerous careful measurements have been made to determine the manner of this variation, but the results are not in sufficiently good agreement to lead to conclusions that may be considered as final. The values that are used for the Marks and Davis Steam Tables and for the Callendar Steam Tables are given in Table 10. The values used by Callendar are based on his own measurements and those of Barnes (Philosophical Transactions, Series A, 1913, volume 212, page 30).

The British thermal unit is defined as 1/180th of the quantity of heat required to raise the temperature of one pound of water from 32 to 212 degrees Fahrenheit, and, according to this definition, the average specific heat of water between 32 and

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212 degrees Fahrenheit is unity. It will be noted in Table 10 that the specific heat is arbitrarily chosen as equal to unity at 55 degrees Fahrenheit and 20 degrees Centigrade, respectively, in the two parts of the table.

TABLE 10

SPECIFIC HEAT OF WATER AT VARIOUS TEMPERATURES

MARKS AND DAVIS From Values of Barnes and Dieterici				CALLENDAR From Values of Barnes and Callendar					
				Temperature			Temperature		
Temperature Degrees Fahrenheit	Specific Heat	Temperature Degrees Fahrenheit	Specific Heat	Degrees Centigrade	Degrees Fahrenheit	Specific Heat	Degrees Centigrade	Degrees Fahrenheit	Specific Heat
30	1.0098	130	0.9979	0	32	1.0106	55	131	0.9994
40	1.0045	140	0.9986	5	41	1.0065	60	140	0.9999
50	1.0012	150	0.9994	10	50	1.0031	65	149	1.0005
55	1.0000	160	1.0002	15	59	1.0009	70	158	1.0012
60	0.9990	170	1.0010	20	68	1.0000	75	167	1.0020
70	0.9977	180	1.0019	25	77	0.9992	80	176	1.0029
80	0.9970	190	1.0029	30	86	0.9988	85	185	1.0038
90	0.9967	200	1.0039	35	95	0.9986	90	194	1.0048
100	0.9967	210	1.0052	40	104	0.9986	95	203	1.0059
110	0.9970	220	1.007	45	113	0.9987	100	212	1.0070
120	0.9974	230	1.009	50	122	0.9990	. .	. .	. .

In consequence of this variation in specific heat, the variation in the heat of the liquid of the water at different temperatures is not a constant. Table 14* gives the heat of the liquid in a pound of water at temperatures ranging from 32 to 340 degrees Fahrenheit.

The specific heat of ice at 32 degrees Fahrenheit is 0.506. The latent heat of fusion is 143.5 B.t.u. per pound (Bulletin, Bureau of Standards, 1915, volume 12, page 49). The specific heat of water vapor, the third form in which water may exist, depends very much on the manner in which the vapor is heated, whether at constant pressure, constant volume, or otherwise. The specific heat of saturated vapor requires special mention. To change saturated vapor at one temperature to saturated vapor at a higher temperature, the pressure must be increased. If the compression were effected adiabatically, that is, without adding or subtracting heat, the vapor would become superheated, consequently, during compression, some heat must be abstracted in order to keep the vapor in a saturated condition. In this process, the rise in temperature was accompanied by the removal of heat from the vapor, and the specific heat of saturated vapor is therefore negative. The specific heat of superheated steam is discussed later in this book.

* See page 77.

BOILER-FEED WATER

IN view of the increasing size of marine boilers and particularly the higher rates of evaporation being demanded, the importance of the use of proper water for boiler-feed purposes cannot be overestimated.

In marine practice, the chief danger arising from the use of poor feed is corrosion. Corrosion, or the chemical action leading to the actual destruction of the boiler metal, is due to the solvent or oxidizing properties of feed water.

The basic causes of corrosion are today better understood than formerly and it is now possible to so treat water as to render it comparatively harmless from the corrosive standpoint.

In stationary boiler practice feed water is ordinarily treated outside of the boiler. In marine practice, however, because of the weight of the treating plant, the space occupied, and more particularly because the contamination is almost entirely the result of leakage of sea water into the boiler system, an outside treating plant is impracticable.

The main causes of corrosion and pitting in marine boilers may be classified as due to—

1. Dissolved oxygen in the feed.
2. Organic or fatty acids from greases and oils.
3. Presence of sea water.

The oxygen component of air is soluble in water. When pure water is exposed to air containing 20.9 per cent of oxygen at a barometric pressure of 760 millimeters of mercury, such solubility is:

At 32 degrees Fahrenheit—10.23 cubic centimeters per liter.
At 70 degrees Fahrenheit— 6.29 cubic centimeters per liter.
At 86 degrees Fahrenheit— 5.33 cubic centimeters per liter.

There is not a complete agreement among chemists and engineers as to the exact mechanism of corrosion through the agency of dissolved oxygen, but it remains a fact that such oxygen dissolved from air by water is probably the greatest accelerator of corrosion.

The solubility of oxygen in water increases with an increase in pressure, but decreases with an increase in temperature. These factors governing the relative degree of solubility, suggest the following methods of preventing corrosion from this source:

1. Heating the water in the boiler under atmospheric pressure. This may be done either by means of circulators or by boiling the water for a short time by means of a direct fire before closing the boiler.
2. Keeping the water in the feed supply tank at the maximum practicable temperature with a minimum amount of surface exposed to the atmosphere.
3. Keeping the feed pump glands and suction joints tight, to avoid drawing in air, and seeing that all feed pump suctions are kept entirely covered at all times.

Deaerating apparatus such as is used in stationary practice is not practicable in marine work because of weight and space occupied.

Organic acids find their way into the feed-water system when lubricants compounded with animal or vegetable oils are used. Such oils will decompose, the glycerine component being dissolved by the water and the fatty acids left free to attack the metal of the boiler. Such acids will cause severe pitting wherever the sludge can find a lodging place.

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Ferryboat "San Leandro," 1500 Shaft Horse-power, Babcock & Wilcox Boilers, Superheaters and Oil Burners
Owners, Key System Transit Co., Oakland, Calif.

When corrosion occurs from this source it can be stopped by neutralizing the water through the use of carbonate of soda. Obviously, however, the logical method of preventing corrosion from fatty acids is the use of oils having a pure mineral base and not compounded. It should be borne in mind that it is much more important to protect the boiler from possible acid attack than to obtain perfect internal lubrication.

The effects of the presence of organic or fatty acids on the action of the boiler as regards priming will be discussed later.

The entrance of sea water into the boiler is the greatest probable source of corrosion in marine practice. Such water may be introduced into the boiler in four possible ways—condenser leakage, interconnecting-valve leakage, steam heaters of salt water, and leakage or priming of evaporators.

Sea water is composed in part of the constituents shown in the following representative analysis:

Calcium carbonate . . .	10 grains per United States gallon
Calcium sulphate	114 grains per United States gallon
Magnesium sulphate . . .	135 grains per United States gallon
Magnesium chloride . . .	244 grains per United States gallon
Sodium chloride	1706 grains per United States gallon

The magnesium chloride is the most harmful of these constituents, since, under the influence of temperature and pressure within the boiler, it dissociates into hydrochloric acid and magnesium hydrate. The acid in attacking the metal of the boiler forms iron chloride which, in turn, further hydrolizes. This cycle of reactions is repeated, so that a small amount of magnesium chloride is capable of doing serious damage. Under certain conditions magnesium sulphate will also decompose in a similar manner into magnesium hydrate and sulphuric acid, but this reaction does not take place to as great an extent as the decomposition of magnesium chloride.

Obviously the best method of avoiding corrosion due to the presence of magnesium chloride would be to prevent the leakage of any sea water into the feed system from any source. Unfortunately, conditions on shipboard are such that this is practically impossible, and the method used for prevention of magnesium chloride corrosion is ordinarily one of chemical neutralization.

Magnesium chloride can be neutralized by soda ash. In such a system, the sodium chloride, or common salt, content of the water is not to be taken as a measure of the quantity of soda ash required for the neutralization of the magnesium chloride present.

Special methods to be used to prevent or minimize corrosion from all sources combined may be classified under two heads: 1st, Lime addition to the water in the boiler and, 2nd, Maintenance of alkaline water in the boiler by periodic additions of soda ash.

Lime additions to the feed water, which should be made only when the boiler is new or retubed, or has been turbined down to the bare metal, are made in part to bring about the conversion of magnesium chloride into magnesium hydrate and calcium chloride, neither of which is corrosive, but primarily to form a light scale on the interior of the boiler surfaces, thus preventing the corrosive agents from coming in contact with the metal surfaces.

When starting with new or retubed boilers, after a thorough cleaning,* ten pounds of lime, dissolved in cold water and run in through the manhole, should be put into the boiler for every 1000 indicated horse-power, and from four to six pounds per day for

* See page 233, "The Care and Operation of Marine Boilers."

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every 1000 indicated horse-power should be passed through the hot well as milk of lime for approximately six days. Milk of lime is a mixture of one pound of unslaked lime or 1½ pounds of hydrated lime with a gallon of water.

The fact has been definitely established that by maintaining an alkaline condition of the water in the boiler, the tendency toward corrosion is greatly lessened. It is not necessary here to go into the theoretical considerations supporting this statement of fact.

The limit of the amount of alkalinity has been purposely fixed at 15 grains a gallon. This means an excess of 15 grains a gallon over and above the amount necessary to combine with the magnesium chloride and scale-forming constituents of the feed.

While in practice an alkalinity of 15 grains may be difficult to maintain, it is the degree that should be striven for. However, the minimum alkalinity should not be allowed to go below 12 grains, and the maximum above 20 grains. While water is improved from the corrosion prevention standpoint through the agency of alkalinity, too great an excess, on the other hand, might bring about dangerous operating conditions through embrittling of the boiler steel. Therefore, the importance of establishing and maintaining a limited degree of alkalinity is to be emphasized.

It might be well to emphasize the distinction between the two uses of soda ash, namely, the one use of maintaining alkaline water in the boiler and the other use of neutralizing the magnesium chloride content of the water.

In the first case, the soda ash is intended to remain uncombined and, if no substances are present with which it can react, no further additions would be necessary. In the second case, where salt-water leakage occurs, additions have to be repeated as long as magnesium chloride is finding its way into the system.

The amount of soda to be added is dependent upon the analysis of the water within the boiler. Soda should never be added by guess, but should be carefully weighed and added in the amount indicated by a reasonably accurate test of the water within the boiler. Further, the careful handling of soda cannot be too strongly stressed, since, as stated, an excess of soda can, within a short period, result in dangerous operating conditions resulting from caustic embrittlement.

As said, the soda should be added only as a result of a test of the water within the boilers. It is recommended that samples be taken each watch and an accurate record of the tests kept. The testing apparatus should be simple and the test reasonably accurate. An apparatus meeting these requirements is the Babcock & Wilcox Water Testing Outfit, furnished as standard equipment with Babcock & Wilcox marine boilers. The results obtained with this apparatus naturally depend upon the care used by the engineer, but when the tests are carefully made the results are accurate.

B. & W. Water-Testing Kit—The testing kit consists of the following reagents* and apparatus, securely packed in a substantial case:

1—Bottle Silver Nitrate Solution, 4.7328 gm. p. l.
1—Bottle Sulphuric Acid Solution, 1/50 normal.
1—Bottle Chromate of Potassium, 5 per cent Solution of Pure Neutral Potassium Chromate.
1—Bottle Methyl Orange Solution.
3—Test Tubes, graduated for chlorine and alkalinity.
2—Glass Beakers.
2—Droppers.

* In replacing any of the reagents, they should be ordered exactly as described above.

Alkalinity Test—The graduated test tube should be filled to the zero reading on the alkalinity scale, with the water to be tested, the sample first being allowed to cool. Two drops of methyl orange indicator should then be added, which addition will impart to the sample a straw color. The sulphuric acid should then be slowly added, drop by drop, shaking the sample frequently. Before the test is completed a faint reddish color will appear, which will disappear as the sample is shaken. When this faint red color remains after shaking, no more sulphuric acid solution should be added. The reading on the alkalinity scale of the test tube at the level of the liquid gives the alkalinity in grains per gallon.



Babcock & Wilcox Water Testing Kit

Since the change from straw color to a faint red is, in some lights, difficult to distinguish, it is advisable, at least until this color change has become entirely familiar, that the engineer fill a second test tube with the water to be tested, add two drops of methyl orange to impart the straw color, and compare the color of the sample under test with that of the second water sample, to assure that it has actually the faint red color.

In testing for alkalinity the greatest care must be used in adding the acid, and to stop such addition at the very first indication of a permanent faint red throughout the sample. A few additional drops may result in a highly erroneous reading.

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Tank Steamer "Dirigo," 3100 Indicated Horse-power, Babcock & Wilcox Boilers, Superheaters and Oil Burners
Owners, The Texas Co., New York, N. Y.

If the test for alkalinity shows less than 15 grains per gallon, sufficient soda ash should be added to bring the alkalinity to this point. It is necessary to know the amount of water within the boiler at steaming level before attempting to add soda ash, and the addition should be made by exact weight.

Since practically all raw waters contain soda ash consuming constituents, it is not possible, in the case of a freshly filled boiler, to state definitely how much soda ash should be added to obtain 15 grains alkalinity without an analysis of the raw water. The best method of procedure when starting boilers freshly filled with raw water, is to add 2¼ pounds of soda ash for every thousand gallons of contained water. The boiler should then be allowed to steam for a short period, a test for alkalinity then be made, and if such test shows less than 15 grains alkalinity, additional soda ash should be added at the rate of 2½ ounces for each one thousand gallons of water for each grain that the alkalinity is below 15. If a second test of the water still shows less than 15 grains alkalinity, a similar amount of soda ash should be added, namely, 2½ ounces for each one thousand gallons of water for each grain less than 15.

The alkalinity will be reduced through the use of make-up feed and by salt leaks into the system and alkalinity tests should be made frequently, preferably, as stated, at least once on every watch.

Chlorine Test—The graduated test tube should be filled to the zero mark on the chlorine scale, CL, with the water to be tested. One drop of the potassium chromate should then be added and, next, the silver nitrate solution, a few drops at a time. While the latter solution is being added the sample should be shaken constantly. Before the test is completed, a red color will appear in the sample, but the first red indications will disappear as the sample is shaken. When the sample becomes a faint red, and *remains red* upon being shaken, no more silver nitrate should be added. The reading of the scale on the graduated test tube at the level of the liquid will indicate the amount of chlorine present in grains per gallon.

If it is found that the chlorine content of the water to be tested exceeds 250 grains per gallon (the upper limit of the test tube scale), the water to be tested should be diluted by three times its volume of fresh water, in which case the reading on the scale, after the test is completed, multiplied by four, will give the amount of chlorine in the original sample.

The amount of chlorine in the water is a general indication of the extent of salt leak into the system. An excess of soda, however, will give a false chlorine reading, that is, the test will indicate a greater amount of chlorine than is actually present. If just sufficient soda is being used to maintain the alkalinity at approximately 15 grains, the difference between the true chlorine content and the test reading may be disregarded.

From the analysis of sea water previously given it is evident that there are two sources of chlorine—sodium chloride and magnesium chloride. The former, or common salt, is not injurious to the boiler from the standpoint of corrosion, since its reaction is neither acid nor alkaline. Common salt in sufficient quantities, however, will cause priming, and at very high densities will deposit in crystalline form. The danger of the magnesium chloride from the standpoint of corrosion has already been indicated.

If soda has been used in sufficient quantity to neutralize the magnesium chloride content, the chlorine reading indicates sodium chloride alone. If a test of water to

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which no soda has been added indicates 600 grains of chlorine, approximately 85 per cent would be due to sodium chloride and 15 per cent due to magnesium chloride. If sufficient soda is added to neutralize the magnesium chloride and a second test made, the chlorine reading would still be 600 grains. In other words, the addition of soda does not change the total chlorine reading, but does change the magnesium chloride to sodium chloride, and the reading of the second test indicates the entire chlorine content due to common salt.

The chlorine readings are not to be used as the basis for the addition of soda. If the soda treatment is properly followed to maintain an alkalinity of 15 grains, the chlorine reading represents salt alone and may, until indicating a high content, be disregarded in so far as feed treatment is concerned.

It is not possible to make any definite statement as to allowable degrees of chlorine concentration before difficulties will be encountered from priming. Experience with different designs of boilers under individual sets of operating conditions will best indicate this.

In addition to frequent and careful tests of the boiler water, tests of the condensate and water in the feed and filter tank should be made. For the purpose of testing the condensate, the elongated neck of the test tube is graduated in fifths of a grain, the procedure followed in making this test being the same as laid down for the test of boiler water.

Scale—Scale is the formation on the boiler-heating surface which results from the deposition of impurities in the feed water in the form of a more or less hard adherent crust. Scale-forming matter at sea is largely magnesium carbonate and calcium sulphate resulting from sea-water leakage, though some scale-forming matter may be present in the fresh feed taken from shore lines.

When scale-forming matter finds its way into the system, the addition of the proper amount of soda ash to maintain the proper degree of alkalinity will tend to combine a portion of the soda ash with such substances and deposit them in the form of loose muddy sediment which will not bake hard on the surfaces and which can be washed or blown out when the boiler is not in use.

Foaming—Reference has been made to the possibility of corrosion resulting from the use of vegetable lubricating oils. In addition to such a result, the use of these oils may lead to foaming, particularly in the case of water over-treated with soda ash. This foaming is the result of a combination of the fatty acids with the alkalies to form soluble soaps, and, obviously, soap dissolved in boiler water would lead to a condition of severe foaming.

The prevention of foaming, as in the case of corrosion from this source, is the use of mineral oils only.

Priming—Priming, or the passing off of water with steam in slugs, may result, as stated, from a concentration of sodium chloride in solution. Experience will indicate the degree of concentration that may be carried before priming will occur and, when such point is approached, the water in the boiler should be freshened.

In discussing boiler operation from the feed-water standpoint it is necessary to consider the accessories to the boiler as well as the boiler itself. This phase will be discussed in the chapter on "The Care and Operation of Marine Boilers."

BOILER FEED WATER HEATERS, PUMPS AND FEED WATER REGULATORS

BEFORE water fed to a boiler can be converted into steam, it must be heated to the temperature corresponding to the pressure within the boiler. The temperature due to 200 pounds gauge pressure is approximately 388 degrees Fahrenheit, and if water is fed to the boiler at 60 degrees, each pound of water must have added to it approximately 333 B.t.u. to bring its temperature to the point at which it can be converted into steam. As it requires 1171.1 B.t.u. to raise the temperature of one pound of water from 60 to 388, and to convert it into steam at 200 pounds gauge pressure, the 333 B.t.u. required simply to raise the temperature of the water to that corresponding to the pressure will be approximately 28 per cent of the total. If, therefore, this increase in the temperature of the water can be secured before it is introduced into the boiler, by the utilization of heat other than that resulting from the combustion of fuel under the boiler, there will be a saving in fuel burned under the boiler of $333 \div 1171.1 = 28$ per cent. Such a saving, if accomplished by the use of a heating medium that would otherwise be wasted, and, provided that the cost of operating and maintaining the apparatus used to bring about the saving is less than the value of the heat saved, is a direct net saving.

The saving in fuel due to the heating of the feed water by means of heat that would otherwise be wasted may be computed from the formula:

$$\text{Fuel saving, per cent} = \frac{100\,(h - h_1)}{H - h_1}$$

where h=heat of the liquid at the temperature of the feed water after heating, h_1=heat of the liquid at the temperature of the feed water before heating, and H=total heat above 32 degrees per pound of steam at the boiler pressure. Values of H may be found from Table 15. It will be noted that the heat in the water (in B.t.u.) above 32 degrees does not agree with the temperature difference. The variation, however, is small enough to be neglected in computing the net saving. Table 11 has been computed from this formula to show the fuel saving resulting from various increases in feed temperatures at a boiler pressure of 200 pounds gauge. The saving at other pressures will vary somewhat, being less for a given temperature rise as the pressure increases and greater as the pressure decreases.

Besides the saving in fuel effected by the use of feed water heaters, other advantages are secured. The time required for the conversion of the water into steam is diminished, and the steam capacity of the boiler correspondingly increased. Further, the feeding of cold water into a boiler has a tendency toward setting up temperature strains, which are diminished as the temperature of the feed approaches that due to the steam pressure. An important additional advantage of feed water heating in certain classes of apparatus is the precipitation in the heater of a large portion of the scale-forming ingredients before they enter the boiler, with a consequent saving in cleaning and losses through decreased efficiency and capacity.

Feed Water Heaters—In general, feed water heaters may be included in three main classes: closed heaters, open heaters and economizers. The first two classes

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may use either live or exhaust steam as the heating medium, while the third utilizes the heat in the spent gases leaving the boiler.

Open feed water heaters are not ordinarily used in American marine practice, though they are used to some extent in European marine work. While the use of water economizers is old in marine practice, for a long period their use was practically abandoned. More recently, however, certain marine engineers have advocated their reintroduction. It would seem, however, that if marine follows stationary practice in its tendency toward higher and higher interstage heating of feed water, their use will not again become common. This is due to the fact that with the high temperature feed obtained with interstage bleeder heating, an economizer has not the temperature range through which to function that it would have with ordinary feed temperatures. This is particularly true at moderate pressures, since the water leaving an economizer should have a temperature about 50 degrees below the temperature, due to the pressure carried in the boiler. If an increase in efficiency is desired equivalent to that which a water economizer would give, the solution would appear to be the use of an air heater.*

Feed water heaters of the closed type are made with coils or straight tubes. In closed heaters, steam and water do not come into direct contact with each other, either the steam or water passing through the tubes. Because of this lack of actual contact, the temperature of the water can never be brought up to the temperature of the steam, ordinarily being from 10 to 15 degrees lower. With impure feed water there is a formation on the heater surfaces of a scale that is difficult to remove. These two factors may lead to a lower efficiency than would be obtained with open heaters, but the closed heater is advantageous where there is oil in the steam, in that it cannot mingle with the feed water, and, in general, the closed heater leads to a simplification of the boiler room.

Where closed heaters of the straight-tube type are used, care must be taken in their design to minimize the possibility of leakage, particularly where the service is intermittent.

In a closed type of heater, any dissolved gases contained in the feed water entering the heater ordinarily pass through the heater to the boiler. Corrosion of the piping or boiler may result unless care is taken to remove such gases from the feed leaving the heater.

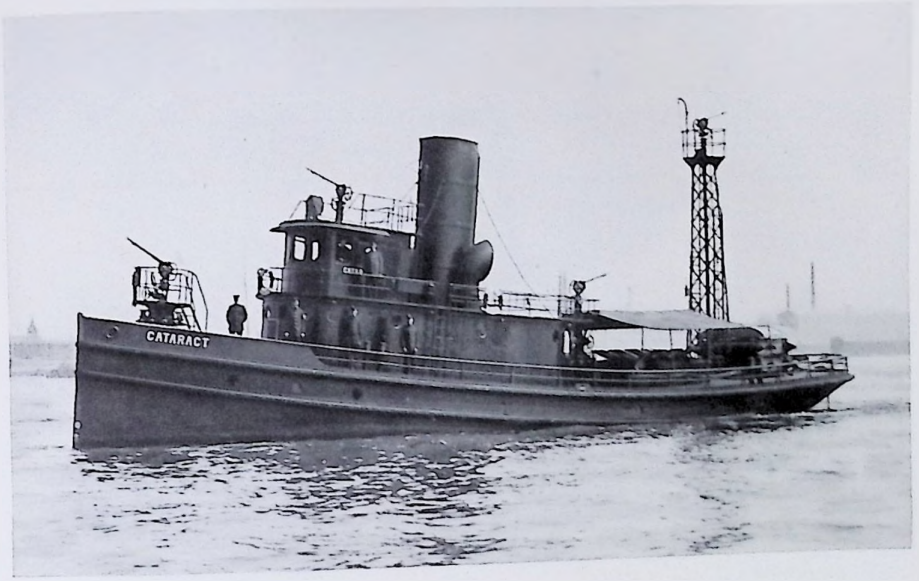
Boiler Feed Pumps—The pumps used for boiler feeding purposes are of two general classes: direct-acting steam pumps and centrifugal pumps.

While direct-acting pumps may be used with either simple, compound or triple-expansion steam ends, the first is ordinarily used. The steam consumption of the latter two designs will be appreciably less than that of the first, but this may be more than offset by cheaper first and maintenance costs, and in cases where all of the exhaust may be used for heating the feed, there will be but little difference in plant efficiency. The simple, direct-acting pump may have one or two water cylinders, being known as simplex or duplex. The former, because of fewer wearing parts, has a lower maintenance cost, but delivers the feed with a certain irregularity that may lead to strains in the feed piping. This irregularity of discharge is largely overcome with duplex pumps, but it may exist to some extent under certain conditions of service. Plungers in direct-acting pumps are preferable to pistons because of greater ease in adjusting packing to minimize slippage.

* See page 71, Air Heaters.

Direct-acting steam pumps are ordinarily used with the smaller boiler installations. Their steam consumption is high, and for compound steam ends will be approximately 100 pounds per indicated horse-power when operated under ordinary service conditions, while, with simple steam ends, this figure may be as high as 200 pounds. The power used by feed pumps, however, even with such high steam consumption, represents but a small percentage of the total steam generated.

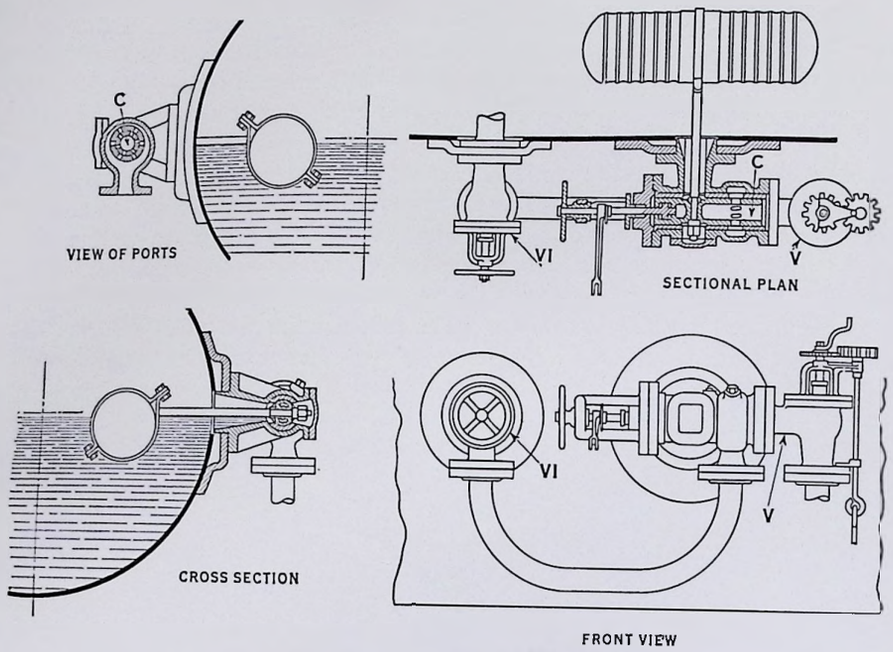
Centrifugal pumps have not been developed in the smaller sizes to an extent where their advantages are sufficient to warrant their use in place of direct-acting pumps. In such pumps the limit of pressure obtainable in a single stage is not in excess of 100 pounds, and to meet the requirements of high pressure it would be necessary to go to additional stages which would make the cost of the centrifugal pump with its drive such that it could not compete with the direct-drive pumps for the smaller plants.

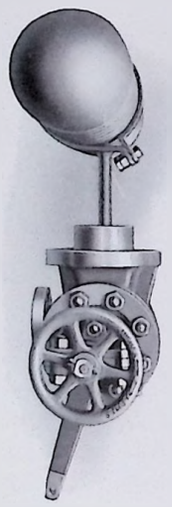


City of Baltimore Fireboat "Cataract," Equipped with Babcock & Wilcox Boilers and Oil Burners

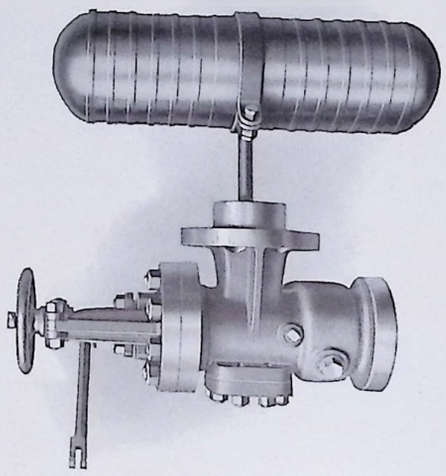
There is perhaps a tendency, particularly in connection with the larger installations, toward the use of centrifugal feed pumps. They are built in from two to five stages, depending upon the pressure at which feed is to be delivered. This delivery pressure is ordinarily based upon from 60 to 100 pounds for each stage. Centrifugal pumps are continuous in their action and set up no pulsating strains in the feed piping. Because of relatively small clearances, the slippage is less than in the direct-acting pump, particularly in the case of the larger centrifugals. In the event of a sudden shutting of the feed valves, this design of pump simply churns the water within it, without delivering any water, and no accident can occur. They are relatively small and require but little attention, the latter factor representing a saving in labor. While a centrifugal pump costs appreciably more than a direct-acting pump, the lower maintenance cost probably more than offsets the difference in original expenditure.

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Babcock & Wilcox Feed Water Regulator—Side View



Babcock & Wilcox Feed Water Regulator—Top View

Centrifugal feed pumps are either turbine or motor-driven, though until very recently the former method of drive was used almost entirely. This was largely because of its reliability and the ability to regulate the speed when so driven; this avoids the change in pressure characteristics of the centrifugal pump, with varying discharge rates at constant speed, which exists when driven by alternating current motors. With a motor drive, because of lower speeds, the pump efficiency is somewhat less than with a turbine drive, and to offset the lower efficiency it is ordinarily necessary to supply an extra stage in the pump to give the same pressure.

Feed Water Regulators—While in the case of boilers having a free circulation and with ample water storage capacity, water levels may be kept steady through hand regulation with but little attention, feed water regulators are frequently used and, particularly in the case of large installations, are to be recommended.

A design of feed water regulator with which the Babcock & Wilcox Company has obtained satisfactory results is shown on page 68. The advantages of this particular type are simplicity, certainty of action, ease of adjustment, small space occupied, minimum friction of working parts, and freedom from possibility of derangement.

The regulator consists of a hollow cylinder, which is rotated on its axis by means of a float and lever placed inside the boiler drum. The cylinder has ports cut diagonally along its walls, and works in a liner having corresponding ports. The liner is fitted in a casting or box which is attached to the boiler shell, and has one branch for a feed-check valve, and another branch for a pipe leading to the feed-stop valve on the boiler drum.

The cylinder is moved lengthwise along the liner by means of a suitable adjusting screw and spindle, the spindle being connected to the cylinder by ball bearings, so that friction is reduced to a minimum.

Attached to the spindle, between the adjusting wheel and the gland, is a lever and rod, the latter extending to a position in the stokehold where it can be easily worked by hand from the firing level. The object of this rod and lever is to enable the water tender to ascertain that the float is moving freely when getting up steam, or to keep the float in the open or shut position, should circumstances at any time so demand.

The regulator operates as follows:—

Feed water enters through the check valve V into the hollow cylinder C, and if the water is low, passes through the ports of both cylinder and liner to the valve box and through the feed stop valve, V_1, into the boiler.

The position of the cylindrical valve is regulated by the float. When the water is low the ports are open, and as the level rises, carrying the float with it, the cylindrical valve is moved around and the ports gradually close. The regulator maintains a remarkably constant water level regardless of load.

The water level may be maintained at the height desired by moving the cylinder lengthwise along the liner by turning the hand-wheel of the spindle attached to the cylinder. As the ports in both cylinder and liner are placed diagonally, the travel of the cylinder insures an alteration in the point of cut-off relative to the position of the float.

In the improbable event of failure of the float, the regulator opens automatically, and the feed may be controlled by hand.

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Steamer "Isle of Thanet," 8250 Shaft Horse-power Babcock & Wilcox Boilers and Oil Burners
Owners, Southern Railway Company, England

AIR HEATERS

AIR HEATERS, or as they are called abroad, air economizers, utilize the heat in the gases coming from the boiler for heating the air for combustion. From the standpoint of efficiency, it would appear to be as logical to reduce the exit gas temperature to any desired minimum temperature through the use of this class of apparatus as by an economizer. Any quantity of sensible heat introduced into the furnace with the air for combustion represents heat available for absorption by the boiler to as great an extent as does heat evolved in the actual combustion of fuel in the furnace. Further, it is of interest to note that the increase in efficiency, due to the use of air heaters, may be greater than that represented by the actual gas temperature drop through the air heater. This is due to the fact that the use of preheated air for combustion results in a higher furnace temperature, and such increase means better combustion conditions and increased boiler efficiency. The effect of furnace temperature on efficiency is too often overlooked. A certain percentage of the heat absorbed by all boilers is absorbed through direct radiation. When it is realized that the amount of heat so received varies as the fourth power of the absolute temperature of the radiating medium, it is evident that even a small increase in furnace temperature will result in a large increase in radiation absorption, and such increase, again, leads to increased boiler efficiency. It is not possible to state definitely the increase in furnace temperature due to a given increase in the temperature of air used for combustion. This is due to the uncertainty of any reported furnace temperature, the accurate measurement of which is ordinarily decidedly open to question, but from available data, such increase would appear to be from 10 to 30 per cent of the increase in the temperature of the air used for combustion.

Air heaters have been in common use in both marine and stationary practice in Europe for a number of years, but it is only within the last few years that they have been used to any extent in stationary practice in this country, and even more recently that any serious consideration has been given them in marine water-tube boiler work here. Increasing fuel costs, however, and the demands for higher efficiencies, will undoubtedly lead to a greater use of this class of apparatus in marine practice.

While more surface is required to produce a given gas temperature drop through an air heater than through an economizer, the weight of the air heater would be less than that of the economizer, because the former is built to operate at atmospheric pressures and the latter must be heavy enough to withstand boiler pressures. It might be argued by those favoring economizers as against air heaters that the necessity of hot-air ducts would lead to a certain amount of complication. This argument, however, would be offset by the necessity with economizers for the use of deaerating apparatus. The Babcock & Wilcox Company believes that this is the case where steel-tube economizers are installed. The space occupied by air heaters is little greater, if any, than that required by economizers.

To repeat a statement made in the chapter on feed water heating, it would appear that the better overall efficiency would be obtained by the use of interstage bleeder heating of feed water and the use of air preheaters. Because of the very high heat transfer rates obtained with bleeder heaters, such heaters are comparatively small.

Air heaters in general may be classed under two types, plate and tubular. There

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are numerous designs of both types, the general construction of which is indicated by the names given, which differ mainly in details of construction.

When tube heaters are used having the tubes placed vertically, it is usual to pass the gases through the tubes and the air around the outside of the tubes. Where horizontal tube air heaters are used, it is customary to pass the air through the tubes and the gases around the outside of the tubes. In the case of the plate heaters the air and gases ordinarily pass through adjacent channels formed by plates.

One design of plate heater is a rotative type, made up of nested plates forming a cylindrical heater, with a multiplicity of passages through the plates. Approximately half of the metal of the plates is in the path of the gases at one time and somewhat less than half in the path of air. The cylinder in rotating acts as a regenerator.

Tests of some air heaters have shown a considerable amount of leakage, and to assure tightness, plate heaters are ordinarily welded. Leakage in welded heaters is difficult to locate and even more difficult to correct. Tubular heaters, because of expanded joints, are readily made tight. The leakage in rotative regenerative heaters is probably greater than in other types, though the manufacturers claim that this has been reduced to a degree that is not objectionable. Where such leakage occurs from the forced draft fan into the flue gases through the rotor, due to air entrainment and to pressure leakage, the measured exit gas temperature may be appreciably less than the actual temperature from the rotor, and thus lead to an erroneous belief in low exit gas temperatures.

Plate heaters, under continuous operation, are probably more difficult to keep clean than are tubular heaters. The rotative heater, except with "gummy" fuels, is probably more readily kept clean than either of the other types, but this advantage may be more than offset by air leakage.

In air heaters as at present designed, the heat transfer rates are comparatively low. Such transfer rates, as in the case of all heat interchanger apparatus, are dependent primarily upon the mass flow of absorbing and heat-yielding media—that is, the ratio of gas or air weight to the gas or air flow area—and to a lesser extent, upon the form of the flow channels.

As the weight of gas and air per unit of flow area increases, the transfer rates obtained become greater. The factors limiting the gas and air velocities through any air heater are allowable draft loss through both the gas and air sections, and the arrangement of surface in a manner that permits proper cleaning.

With the ratios of gas and air weights to areas of flow channels now commonly offered, the transfer rates will vary from 1.5 to something over 3 B.t.u. per square foot of air heater surface per degree difference in mean temperature between gas and air per hour.

The maximum temperature that may be used with safety in preheated air, before furnace difficulties are encountered, has not been definitely established but is known, in the case of both oil burning and coal burning, to be in the neighborhood of 600 degrees Fahrenheit. There are a large number of installations in stationary practice using air temperatures of 350 degrees Fahrenheit in which no difficulties have been encountered from combustion of the coal or oil, operation or upkeep of the stokers or oil burners, or excessive brickwork maintenance. In view of the fact that space and weight limitations will always act to restrict the size of air heaters commercially desirable in the marine field, it is not likely that air temperatures in excess of 350 degrees Fahrenheit will ever become common in marine practice.

STEAM

WHEN a given weight of a perfect gas is compressed or expanded at a constant temperature, the product of the pressure and volume is a constant. Vapors, which are liquids in aeriform condition, on the other hand, can exist only at a definite pressure corresponding to each temperature if in the saturated state, that is, the pressure is a function of the temperature only. Steam is water vapor, and at a pressure of, say, 150 pounds absolute per square inch, saturated steam can exist only at a temperature of 358 degrees Fahrenheit. Hence, if the pressure of saturated steam be fixed, its temperature is also fixed, and *vice versa*.

Saturated steam is water vapor in the condition in which it is generated from water with which it is in contact. Or it is steam which is at the maximum pressure and density possible at its temperature. If any change be made in the temperature or pressure of steam, there will be a corresponding change in its condition. If the pressure be increased or the temperature decreased, a portion of the steam will be condensed. If the temperature be increased or the pressure decreased, a portion of the water with which the steam is in contact will be evaporated into steam. Steam will remain saturated just so long as it is of the same pressure and temperature as the water with which it can remain in contact without a gain or loss of heat. Moreover, saturated steam cannot have its temperature lowered without a lowering of its pressure, any loss of heat being made up by the latent heat of such portion as will be condensed. Nor can the temperature of saturated steam be increased except when accompanied by a corresponding increase in pressure, any added heat being expended in the evaporation into steam of a portion of the water with which it is in contact.

Dry saturated steam contains no water. In some cases, saturated steam is accompanied by water which is carried along with it, either in the form of a spray or is blown along the surface of the piping, and the steam is then said to be wet. The percentage weight of the steam in a mixture of steam and water is called the quality of the steam. Thus, if in a mixture of 100 pounds of steam and water there is three-quarters of a pound of water, the quality of the steam will be 99.25.

Heat may be added to steam not in contact with water, such an addition of heat resulting in an increase of temperature and pressure if the volume be kept constant, or an increase in temperature and volume if the pressure remain constant. Steam whose temperature thus exceeds that of saturated steam at a corresponding pressure is said to be superheated and its properties approximate those of a perfect gas.

As pointed out in the chapter on heat, the heat necessary to raise one pound of water from 32 degrees Fahrenheit to the point of ebullition is called the *heat of the liquid*. The heat absorbed during ebullition consists of that necessary to dissociate the molecules, or the *inner latent heat,* and that necessary to overcome the resistance to the increase in volume, or the *outer latent heat*. These two make up the *latent heat of evaporation* and the sum of this latent heat of evaporation and the heat of the liquid make the *total heat* of the steam. These values for various pressures are given in the steam tables, pages 79 to 85.

The specific volume of saturated steam at any pressure is the volume in cubic feet of one pound of steam at that pressure.

The density of saturated steam, that is, its weight per cubic foot, is obviously the reciprocal of the specific volume. This density varies as the $\frac{16}{17}$ power over the

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ordinary range of pressures used in steam-boiler work and may be found by the formula, $D = .003027\ p^{.941}$, which is correct within 0.15 per cent up to 250 pounds pressure.

The relative volume of steam is the ratio of the volume of a given weight to the volume of the same weight of water at 39.2 degrees Fahrenheit and is equal to the specific volume times 62.427.

As vapors are liquids in their gaseous form and the boiling point is the point of change in this condition, it is clear that this point is dependent upon the pressure under which the liquid exists. This fact is of great practical importance in steam condenser work and in many operations involving boiling in an open vessel, since in the latter case its altitude will have considerable influence. The relation between altitude and boiling point of water is shown in Table 9.

The conditions of feed temperature and steam pressure in boiler tests, fuel performances and the like will be found to vary widely in different trials. In order to secure a means for comparison of different trials, it is necessary to reduce all results to some common basis. The method which has been adopted for the reduction to a comparable basis is to transform the evaporation under actual conditions of steam pressure and feed temperature which exist in the trial to an equivalent evaporation under a set of standard conditions. These standard conditions presuppose a feed-water temperature of 212 degrees Fahrenheit and a steam pressure equal to the normal atmospheric pressure at sea level, 14.7 pounds absolute. Under such conditions, steam would be generated *at* a temperature of 212 degrees, the temperature corresponding to atmospheric pressure at sea level, *from* water at 212 degrees. The weight of water which *would* be evaporated under the assumed standard conditions by exactly the amount of heat absorbed by the boiler under actual conditions existing in the trial, is, therefore, called the equivalent evaporation "from and at 212 degrees."

The factor for reducing the weight of water actually converted into steam from the temperature of the feed, at the steam pressure existing in the trial, to the equivalent evaporation under standard conditions is called the *factor of evaporation*. This factor is the ratio of the heat added to each pound of steam in heating the water from the temperature of the feed in the trial to the temperature corresponding to the pressure existing in the trial to the total heat added to evaporate one pound of water from and at 212 degrees. The heat added during the trial is obviously the difference between the total heat of evaporation of the steam at the pressure existing in the trial and the heat of the liquid in the water at the temperature at which it was fed in the trial. To illustrate by an example:

In a boiler trial the temperature of the feed water is 60 degrees Fahrenheit and the pressure under which steam is delivered is 160.3 pounds gauge pressure or 175 pounds absolute pressure. The total heat of one pound of steam at 175 pounds pressure is 1195.9 B.t.u. measured above the standard temperature of 32 degrees Fahrenheit. But the water fed to the boiler contained 28.08 B.t.u. as the heat of the liquid measured above 32 degrees Fahrenheit. Therefore, to each pound of steam there has been added 1167.82 B.t.u. To evaporate one pound of water under standard conditions would, on the other hand, have required but 970.4 B.t.u., which, as described, is the latent heat of evaporation at 212 degrees Fahrenheit. Expressed differently, the total heat of one pound of steam at the pressure corresponding to a

temperature of 212 degrees is 1150.4 B.t.u. One pound of water at 212 degrees contains 180 B.t.u. of sensible heat above 32 degrees Fahrenheit. Hence, under standard conditions, 1150.4—180=970.4 B.t.u. is added in the changing of one pound of water into steam at atmospheric pressure and a temperature of 212 degrees. This is in effect the definition of the latent heat of evaporation.

Hence, if conditions of the trial had been standard, only 970.4 B.t.u. would be required and the ratio of 1167.82 to 970.4 B.t.u. is the ratio determining the factor of evaporation. The factor in the assumed case is $1167.82 \div 970.4 = 1.2034$ and if the same amount of heat had been absorbed under standard conditions as was absorbed in the trial condition, 1.2034 times the amount of steam would have been generated. Expressed as a formula for use with any set of conditions, the factor is,

$$F = \frac{H - h}{970.4} \qquad (4)$$

Where H=the total heat of steam above 32 degrees Fahrenheit from steam tables,

h=sensible heat of feed water above 32 degrees Fahrenheit from Table 20.

In the form above, the factor may be determined with either saturated or superheated steam, provided that in the latter case values of H are available for varying degrees of superheat and pressures.

In the case of saturated steam containing moisture, the factor may be computed from the formula

$$F = \frac{H - \frac{L \times m}{100} - h}{970.4} \qquad (5)$$

Where L = latent heat of steam at observed pressure, and

m = percentage of moisture in steam.

A table of the factors of evaporation for saturated steam boiler trials is published in the thirty-sixth edition of our treatise "Steam." This covers a large range of conditions and except for the most refined work intermediate values between those given may be determined by interpolation.

Steam gauges indicate the pressure above the atmosphere. As has been pointed out, the atmospheric pressure changes according to the altitude and the variation in the barometer. Hence, calculations involving the properties of steam are based on *absolute* pressures, which are equal to the gauge pressure plus the atmospheric pressure in pounds to the square inch. This latter is generally assumed to be 14.7 pounds per square inch at sea level, but for other levels it must be determined from the barometric reading at that place.

Vacuum gauges indicate the difference, expressed in inches of mercury, between atmospheric pressure and the pressure within the vessel to which the gauge is attached. For approximate purposes, 2.04 inches height of mercury may be considered equal to a pressure of one pound per square inch at the ordinary temperatures at which mercury gauges are used. Hence for any reading of the vacuum gauge in inches, G, the absolute pressure for any barometer reading in inches, B, will be $(B - G) \div 2.04$. If the barometer is 30 inches measured at ordinary temperatures and not corrected to 32 degrees Fahrenheit and the vacuum gauge 24 inches, the absolute pressure will be $(30 - 24) \div 2.04 = 2.9$ pounds per square inch.

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The temperature, pressure and other properties of steam for varying amounts of vacuum and the pressure above vacuum corresponding to each inch of reading of the vacuum gauge are given in Table 12.

TABLE 12

PROPERTIES OF SATURATED STEAM FOR VARYING AMOUNTS OF VACUUM

CALCULATED FROM MARKS AND DAVIS TABLES

Vacuum Height Inches	Absolute Pressure Pounds	Temperature Degrees Fahrenheit	Heat of the Liquid Above 32 Degrees B.t.u.	Latent Heat Above 32 Degrees B.t.u.	Total Heat Above 32 Degrees B.t.u.	Density or Weight per Cubic Foot Pounds
29.5	.207	54.1	22.18	1061.0	1083.2	0.000678
29	.452	76.6	44.64	1048.7	1093.3	0.001415
28.5	.698	90.1	58.09	1041.1	1099.2	0.002137
28	.944	99.9	67.87	1035.6	1103.5	0.002843
27	1.44	112.5	80.4	1028.6	1109.0	0.00421
26	1.93	124.5	92.3	1022.0	1114.3	0.00577
25	2.42	132.6	100.5	1017.3	1117.8	0.00689
24	2.91	140.1	108.0	1013.1	1121.1	0.00821
22	3.89	151.7	119.6	1006.4	1126.0	0.01078
20	4.87	161.1	128.9	1001.0	1129.9	0.01331
18	5.86	168.9	136.8	996.4	1133.2	0.01581
16	6.84	175.8	143.6	992.4	1136.0	0.01827
14	7.82	181.8	149.7	988.8	1138.5	0.02070
12	8.80	187.2	155.1	985.6	1140.7	0.02312
10	9.79	192.2	160.1	982.6	1142.7	0.02554
5	12.24	202.9	170.8	976.0	1146.8	0.03148

From the steam tables, the condensed Table 13 of the properties of steam at different pressures may be constructed. From such a table there may be drawn the following conclusions:

TABLE 13

VARIATION IN PROPERTIES OF SATURATED STEAM WITH PRESSURE

Pressure Pounds Absolute	Temperature Degrees Fahrenheit	Heat of Liquid B.t.u.	Latent Heat B.t.u.	Total Heat B.t.u.
14.7	212.0	180.0	970.4	1150.4
20.0	228.0	196.1	960.0	1156.2
100.0	327.8	298.3	888.0	1186.3
300.0	417.5	392.7	811.3	1204.1

As the pressure and temperature increase, the latent heat decreases. This decrease, however, is less rapid than the corresponding increase in the heat of the liquid and hence the total heat increases with an increase in the pressure and temperature. The percentage increase in the total heat is small, being 0.5, 3.1 and 4.7 per cent for 20, 100 and 300 pounds absolute pressure, respectively, above the total heat in one pound of steam at 14.7 pounds absolute. The temperatures, on the other hand, increase at the rates of 7.5, 54.6 and 96.9 per cent. The efficiency of a perfect steam engine is proportional to the expression $(t—t_1) \div t$, in which t and t_1 are the absolute temperatures of the saturated steam at admission and exhaust, respectively. While actual engines only approximate the ideal engine in efficiency, yet they follow the same general law. Since the exhaust temperature cannot be lowered beyond present practice, it follows that the only available method of increasing the efficiency is by an increase in the temperature of the steam at admission. How

this may be accomplished by an increase of pressure is clearly shown, for the increase of fuel necessary to increase the pressure is negligible, as shown by the total heat, while the increase in economy, due to the higher pressure, will result directly from the rapid increase of the corresponding temperature.

TABLE 14

HEAT UNITS PER POUND AND WEIGHT PER CUBIC FOOT OF WATER BETWEEN 32 DEGREES FAHRENHEIT AND 340 DEGREES FAHRENHEIT

Temperature Degrees Fahrenheit	Heat Units per Pound	Weight per Cubic Foot	Temperature Degrees Fahrenheit	Heat Units per Pound	Weight per Cubic Foot	Temperature Degrees Fahrenheit	Heat Units per Pound	Weight per Cubic Foot	Temperature Degrees Fahrenheit	Heat Units per Pound	Weight per Cubic Foot	Temperature Degrees Fahrenheit	Heat Units per Pound	Weight per Cubic Foot	Temperature Degrees Fahrenheit	Heat Units per Pound	Weight per Cubic Foot
32	0.00	62.42	70	38.06	62.30	108	75.95	61.90	146	113.86	61.27	184	151.89	60.49	222	190.1	59.58
33	1.01	62.42	71	39.06	62.30	109	76.94	61.88	147	114.86	61.25	185	152.89	60.47	223	191.1	59.55
34	2.01	62.42	72	40.05	62.29	110	77.94	61.86	148	115.86	61.24	186	153.89	60.45	224	192.1	59.53
35	3.02	62.43	73	41.05	62.28	111	78.94	61.85	149	116.86	61.22	187	154.90	60.42	225	193.1	59.50
36	4.03	62.43	74	42.05	62.27	112	79.93	61.83	150	117.86	61.20	188	155.90	60.40	226	194.1	59.48
37	5.04	62.43	75	43.05	62.26	113	80.93	61.82	151	118.86	61.18	189	156.90	60.38	227	195.2	59.45
38	6.04	62.43	76	44.04	62.26	114	81.93	61.80	152	119.86	61.16	190	157.91	60.36	228	196.2	59.42
39	7.05	62.43	77	45.04	62.25	115	82.92	61.79	153	120.86	61.14	191	158.91	60.33	229	197.2	59.40
40	8.05	62.43	78	46.04	62.24	116	83.92	61.77	154	121.86	61.12	192	159.91	60.31	230	198.2	59.37
41	9.05	62.43	79	47.04	62.23	117	84.92	61.75	155	122.86	61.10	193	160.91	60.29	231	199.2	59.34
42	10.06	62.43	80	48.03	62.22	118	85.92	61.74	156	123.86	61.08	194	161.92	60.27	232	200.2	59.32
43	11.06	62.43	81	49.03	62.21	119	86.91	61.72	157	124.86	61.06	195	162.92	60.24	233	201.2	59.29
44	12.06	62.43	82	50.03	62.20	120	87.91	61.71	158	125.86	61.04	196	163.92	60.22	234	202.2	59.27
45	13.07	62.42	83	51.02	62.19	121	88.91	61.69	159	126.86	61.02	197	164.93	60.19	235	203.2	59.24
46	14.07	62.42	84	52.02	62.18	122	89.91	61.68	160	127.86	61.00	198	165.93	60.17	236	204.2	59.21
47	15.07	62.42	85	53.02	62.17	123	90.90	61.66	161	128 86	60.98	199	166.94	60.15	237	205.3	59.19
48	16.07	62.42	86	54.01	62.16	124	91.90	61.65	162	129.86	60.96	200	167.94	60.12	238	206.3	59.16
49	17.08	62.42	87	55.01	62.15	125	92.90	61.63	163	130.86	60.94	201	168.94	60.10	239	207.3	59.14
50	18.08	62.42	88	56.01	62.14	126	93.90	61.61	164	131.86	60.92	202	169.95	60.07	240	208.3	59.11
51	19.08	62.41	89	57.00	62.13	127	94.89	61.59	165	132.86	60.90	203	170.95	60.05	241	209.3	59.08
52	20.08	62.41	90	58.00	62.12	128	95.89	61.58	166	133.86	60.88	204	171.96	60.02	242	210.3	59.05
53	21.08	62.41	91	59.00	62.11	129	96.89	61.56	167	134.86	60.86	205	172.96	60.00	243	211.4	59.03
54	22.08	62.40	92	60.00	62.09	130	97.89	61.55	168	135.86	60.84	206	173.97	59.98	244	212.4	59.00
55	23.08	62.40	93	60.99	62.08	131	98.89	61.53	169	136.86	60.82	207	174.97	59.95	245	213.4	58.97
56	24.08	62.39	94	61.99	62.07	132	99.88	61.52	170	137.87	60.80	208	175.98	59.93	246	214.4	58.94
57	25.08	62.39	95	62.99	62.06	133	100.88	61.50	171	138.87	60.78	209	176.98	59.90	247	215.4	58.91
58	26.08	62.38	96	63.98	62.05	134	101.88	61.49	172	139.87	60.76	210	177.99	59.88	248	216.4	58.89
59	27.08	62.37	97	64.98	62.04	135	102.88	61.47	173	140.87	60.73	211	178.99	59.85	249	217.4	58.86
60	28.08	62.37	98	65.98	62.03	136	103.88	61.45	174	141.87	60.71	212	180.00	59.83	250	218.5	58.83
61	29.08	62.36	99	66.97	62.02	137	104.87	61.43	175	142.87	60.69	213	181.0	59.80	260	228.6	58.55
62	30.08	62.36	100	67.97	62.00	138	105.87	61.41	176	143.87	60.67	214	182.0	59.78	270	238.8	58.26
63	31.07	62.35	101	68.97	61.99	139	106.87	61.40	177	144.88	60.65	215	183.0	59.75	280	249.0	57.96
64	32.07	62.35	102	69.96	61.98	140	107.87	61.38	178	145.88	60.62	216	184.0	59.73	290	259.3	57.65
65	33.07	62.34	103	70.96	61.97	141	108.87	61.36	179	146.88	60.60	217	185.0	59.70	300	269.6	57.33
66	34.07	62.33	104	71.96	61.95	142	109.87	61.34	180	147.88	60.58	218	186.1	59.68	310	279.9	57.00
67	35.07	62.33	105	72.95	61.94	143	110.87	61.33	181	148.88	60.56	219	187.1	59.65	320	290.2	56.66
68	36.07	62.32	106	73.95	61.93	144	111.87	61.31	182	149.89	60.53	220	188.1	59.63	330	300.6	56.30
69	37.06	62.31	107	74.95	61.91	145	112.86	61.29	183	150.89	60.51	221	189.1	59.60	340	311.0	55.94

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The gain due to superheat cannot be predicted from the formula for the efficiency of a perfect steam engine. This formula is not applicable in cases where superheat is present, since only a relatively small amount of the heat in the steam is imparted at the maximum or superheated temperature.

The advantage of the use of high-pressure steam may be also indicated by considering the question from the aspect of volume. With an increase of pressure comes a decrease in volume, thus one pound of saturated steam at 100 pounds absolute pressure occupies 4.43 cubic feet, while at 200 pounds pressure it occupies 2.29 cubic feet. If then, in separate cylinders of the same dimensions, one pound of steam at 100 pounds absolute pressure and one pound at 200 pounds absolute pressure enter and are allowed to expand to the full volume of each cylinder, the high-pressure steam, having more room and a greater range for expansion than the low-pressure steam, will thus do more work. This increase in the amount of work, as was the increase in temperature is large relative to the additional fuel required as indicated by the total heat. In general it may be stated that the fuel required to impart a given amount of heat to a boiler is practically independent of the steam pressure, since the temperature of the fire is so high as compared with the steam temperature that a variation in the steam temperature does not produce an appreciable effect.

The formulæ for the algebraic expression of the relation between saturated steam pressures, temperatures and steam volumes have been up to the present time empirical. These relations have, however, been determined by experiment and, from the experimental data, tables have been computed which render unnecessary the use of empirical formulæ. Such formulæ may be found in any standard work of thermodynamics. The following tables cover all practical cases.

Table 14 gives the heat units contained in water above 32 degrees Fahrenheit at different temperatures.

Table 15 gives the properties of saturated steam for various pressures.

Table 16 gives the properties of superheated steam at various pressures and temperatures.

These tables are based on those computed by Lionel S. Marks and Harvey N. Davis.

TABLE 15

PROPERTIES OF SATURATED STEAM

REPRODUCED BY PERMISSION FROM MARKS AND DAVIS "STEAM TABLES AND DIAGRAMS"

(Copyright, 1909, by Longmans, Green & Co.)

Pressure Pounds Absolute	Temperature Degrees Fahrenheit	Specific Volume Cubic Feet per Pound	Heat of the Liquid, B.t.u.	Latent Heat of Evaporation B.t.u.	Total Heat of Steam, B.t.u.	Pressure Pounds Absolute
1	101.83	333.0	69.8	1034.6	1104.4	1
2	126.15	173.5	94.0	1021.0	1115.0	2
3	141.52	118.5	109.4	1012.3	1121.6	3
4	153.01	90.5	120.9	1005.7	1126.5	4
5	162.28	73.33	130.1	1000.3	1130.5	5
6	170.06	61.89	137.9	995.8	1133.7	6
7	176.85	53.56	144.7	991.8	1136.5	7
8	182.86	47.27	150.8	988.2	1139.0	8
9	188.27	42.36	156.2	985.0	1141.1	9
10	193.22	38.38	161.1	982.0	1143.1	10
11	197.75	35.10	165.7	979.2	1144.9	11
12	201.96	32.36	169.9	976.6	1146.5	12
13	205.87	30.03	173.8	974.2	1148.0	13
14	209.55	28.02	177.5	971.9	1149.4	14
15	213.0	26.27	181.0	969.7	1150.7	15
16	216.3	24.79	184.4	967.6	1152.0	16
17	219.4	23.38	187.5	965.6	1153.1	17
18	222.4	22.16	190.5	963.7	1154.2	18
19	225.2	21.07	193.4	961.8	1155.2	19
20	228.0	20.08	196.1	960.0	1156.2	20
22	233.1	18.37	201.3	956.7	1158.0	22
24	237.8	16.93	206.1	953.5	1159.6	24
26	242.2	15.72	210.6	950.6	1161.2	26
28	246.4	14.67	214.8	947.8	1162.6	28
30	250.3	13.74	218.8	945.1	1163.9	30
32	254.1	12.93	222.6	942.5	1165.1	32
34	257.6	12.22	226.2	940.1	1166.3	34
36	261.0	11.58	229.6	937.7	1167.3	36
38	264.2	11.01	232.9	935.5	1168.4	38
40	267.3	10.49	236.1	933.3	1169.4	40
42	270.2	10.02	239.1	931.2	1170.3	42
44	273.1	9.59	242.0	929.2	1171.2	44
46	275.8	9.20	244.8	927.2	1172.0	46
48	278.5	8.84	247.5	925.3	1172.8	48
50	281.0	8.51	250.1	923.5	1173.6	50
52	283.5	8.20	252.6	921.7	1174.3	52
54	285.9	7.91	255.1	919.9	1175.0	54
56	288.2	7.65	257.5	918.2	1175.7	56
58	290.5	7.40	259.8	916.5	1176.4	58
60	292.7	7.17	262.1	914.9	1177.0	60
62	294.9	6.95	264.3	913.3	1177.6	62
64	297.0	6.75	266.4	911.8	1178.2	64
66	299.0	6.56	268.5	910.2	1178.8	66
68	301.0	6.38	270.6	908.7	1179.3	68

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PROPERTIES OF SATURATED STEAM—CONTINUED

Pressure Pounds Absolute	Temperature Degrees Fahrenheit	Specific Volume Cubic Feet per Pound	Heat of the Liquid, B.t.u.	Latent Heat of Evaporation B.t.u.	Total Heat of Steam, B.t.u.	Pressure Pounds Absolute
70	302.9	6.20	272.6	907.2	1179.8	70
72	304.8	6.04	274.5	905.8	1180.4	72
74	306.7	5.89	276.5	904.4	1180.9	74
76	308.5	5.74	278.3	903.0	1181.4	76
78	310.3	5.60	280.2	901.7	1181.8	78
80	312.0	5.47	282.0	900.3	1182.3	80
82	313.8	5.34	283.8	899.0	1182.8	82
84	315.4	5.22	285.5	897.7	1183.2	84
86	317.1	5.10	287.2	896.4	1183.6	86
88	318.7	5.00	288.9	895.2	1184.0	88
90	320.3	4.89	290.5	893.9	1184.4	90
92	321.8	4.79	292.1	892.7	1184.8	92
94	323.4	4.69	293.7	891.5	1185.2	94
96	324.9	4.60	295.3	890.3	1185.6	96
98	326.4	4.51	296.8	889.2	1186.0	98
100	327.8	4.429	298.3	888.0	1186.3	100
105	331.4	4.230	302.0	885.2	1187.2	105
110	334.8	4.047	305.5	882.5	1188.0	110
115	338.1	3.880	309.0	879.8	1188.8	115
120	341.3	3.726	312.3	877.2	1189.6	120
125	344.4	3.583	315.5	874.7	1190.3	125
130	347.4	3.452	318.6	872.3	1191.0	130
135	350.3	3.331	321.7	869.9	1191.6	135
140	353.1	3.219	324.6	867.6	1192.2	140
145	355.8	3.112	327.4	865.4	1192.8	145
150	358.5	3.012	330.2	863.2	1193.4	150
155	361.0	2.920	332.9	861.0	1194.0	155
160	363.6	2.834	335.6	858.8	1194.5	160
165	366.0	2.753	338.2	856.8	1195.0	165
170	368.5	2.675	340.7	854.7	1195.4	170
175	370.8	2.602	343.2	852.7	1195.9	175
180	373.1	2.533	345.6	850.8	1196.4	180
185	375.4	2.468	348.0	848.8	1196 8	185
190	377.6	2.406	350.4	846.9	1197.3	190
195	379.8	2.346	352.7	845.0	1197.7	195
200	381.9	2.290	354.9	843.2	1198.1	200
205	384.0	2.237	357.1	841.4	1198.5	205
210	386.0	2.187	359.2	839.6	1198.8	210
215	388.0	2.138	361.4	837.9	1199.2	215
220	389.9	2.091	363.4	836.2	1199.6	220
225	391.9	2.046	365.5	834.4	1199.9	225
230	393.8	2.004	367.5	832.8	1200.2	230
235	395.6	1.964	369.4	831.1	1200.6	235
240	397.4	1.924	371.4	829.5	1200.9	240
245	399.3	1.887	373.3	827.9	1201.2	245
250	401.1	1.850	375.2	826.3	1201.5	250

PROPERTIES OF SATURATED STEAM—CONTINUED

Pressure Pounds Absolute	Temperature Degrees Fahrenheit	Specific Volume Cubic Feet per Pound	Heat of the Liquid, B.t.u.	Latent heat of Evaporation B.t.u.	Total Heat of Steam, B.t.u.	Pressure Pounds Absolute
260	404.5	1.782	378.9	823.1	1202.1	260
270	407.9	1.718	382.5	820.1	1202.6	270
280	411.2	1.658	386.0	817.1	1203.1	280
290	414.4	1.602	389.4	814.2	1203.6	290
300	417.5	1.551	392.7	811.3	1204.1	300
310	420.5	1.502	395.9	808.5	1204.5	310
320	423.4	1.456	399.1	805.8	1204.9	320
330	426.3	1.413	402.2	803.1	1205.3	330
340	429.1	1.372	405.3	800.4	1205.7	340
350	431.9	1.334	408.2	797.8	1206.1	350
360	434.6	1.298	411.2	795.3	1206.4	360
370	437.2	1.264	414.0	792.8	1206.8	370
380	439.8	1.231	416.8	790.3	1207.1	380
390	442.3	1.200	419.5	787.9	1207.4	390
400	444.8	1.17	422.	786.	1208.	400
410	447.2	1.14	425.	783.	1208.	410
420	449.6	1.11	427.	780.	1208.	420
430	452.0	1.09	430.	778.	1208.	430
440	454.3	1.06	433.	776.	1208.	440
450	456.5	1.04	435.	774.	1209.	450

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TABLE 16

PROPERTIES OF SUPERHEATED STEAM

REPRODUCED BY PERMISSION FROM MARKS AND DAVIS "STEAM TABLES AND DIAGRAMS"

(Copyright, 1909, by Longmans, Green & Co.)

Pressure Pounds Absolute		Saturated Steam	Degrees of Superheat						Pressure Pounds Absolute	
			50	100	150	200	250	300		
5	t	162.3	212.3	262.3	312.3	362.3	412.3	462.3	t	5
	v	73.3	79.7	85.7	91.8	97.8	103.8	109.8	v	
	h	1130.5	1153.5	1176.4	1199.5	1222.5	1245.6	1268.7	h	
10	t	193.2	243.2	293.2	343.2	393.2	443.2	493.2	t	10
	v	38.4	41.5	44.6	47.7	50.7	53.7	56.7	v	
	h	1143.1	1166.3	1189.5	1212.7	1236.0	1259.3	1282.5	h	
15	t	213.0	263.0	313.0	363.0	413.0	463.0	513.0	t	15
	v	26.27	28.40	30.46	32.50	34.53	36.56	38.58	v	
	h	1150.7	1174.2	1197.6	1221.0	1244.4	1267.7	1291.1	h	
20	t	228.0	278.0	328.0	378.0	428.0	478.0	528.0	t	20
	v	20.08	21.69	23.25	24.80	26.33	27.85	29.37	v	
	h	1156.2	1179.9	1203.5	1227.1	1250.6	1274.1	1297.6	h	
25	t	240.1	290.1	340.1	390.1	440.1	490.1	540.1	t	25
	v	16.30	17.60	18.86	20.10	21.32	22.55	23.77	v	
	h	1160.4	1184.4	1208.2	1231.9	1255.6	1279.2	1302.8	h	
30	t	250.4	300.4	350.4	400.4	450.4	500.4	550.4	t	30
	v	13.74	14.83	15.89	16.93	17.97	18.99	20.00	v	
	h	1163.9	1188.1	1212.1	1236.0	1259.7	1283.4	1307.1	h	
35	t	259.3	309.3	359.3	409.3	459.3	509.3	559.3	t	35
	v	11.89	12.85	13.75	14.65	15.54	16.42	17.30	v	
	h	1166.8	1191.3	1215.4	1239.4	1263.3	1287.1	1310.8	h	
40	t	267.3	317.3	367.3	417.3	467.3	517.3	567.3	t	40
	v	10.49	11.33	12.13	12.93	13.70	14.48	15.25	v	
	h	1169.4	1194.0	1218.4	1242.4	1266.4	1290.3	1314.1	h	
45	t	274.5	324.5	374.5	424.5	474.5	524.5	574.5	t	45
	v	9.39	10.14	10.86	11.57	12.27	12.96	13.65	v	
	h	1171.6	1196.6	1221.0	1245.2	1269.3	1293.2	1317.0	h	
50	t	281.0	331.0	381.0	431.0	481.0	531.0	581.0	t	50
	v	8.51	9.19	9.84	10.48	11.11	11.74	12.36	v	
	h	1173.6	1198.8	1223.4	1247.7	1271.8	1295.8	1319.7	h	
55	t	287.1	337.1	387.1	437.1	487.1	537.1	587.1	t	55
	v	7.78	8.40	9.00	9.59	10.16	10.73	11.30	v	
	h	1175.4	1200.8	1225.6	1250.0	1274.2	1298.1	1322.0	h	
60	t	292.7	342.7	392.7	442.7	492.7	542.7	592.7	t	60
	v	7.17	7.75	8.30	8.84	9.36	9.89	10.41	v	
	h	1177.0	1202.6	1227.6	1252.1	1276.4	1300.4	1324.3	h	
65	t	298.0	348.0	398.0	448.0	498.0	548.0	598.0	t	65
	v	6.65	7.20	7.70	8.20	8.69	9.17	9.65	v	
	h	1178.5	1204.4	1229.5	1254.0	1278.4	1302.4	1326.4	h	
70	t	302.9	352.9	402.9	452.9	502.9	552.9	602.9	t	70
	v	6.20	6.71	7.18	7.65	8.11	8.56	9.01	v	
	h	1179.8	1205.9	1231.2	1255.8	1280.2	1304.3	1328.3	h	
75	t	307.6	357.6	407.6	457.6	507.6	557.6	607.6	t	75
	v	5.81	6.28	6.73	7.17	7.60	8.02	8.44	v	
	h	1181.1	1207.5	1232.8	1257.5	1282.0	1306.1	1330.1	h	
80	t	312.0	362.0	412.0	462.0	512.0	562.0	612.0	t	80
	v	5.47	5.92	6.34	6.75	7.17	7.56	7.95	v	
	h	1182.3	1208.8	1234.3	1259.0	1283.6	1307.8	1331.9	h	
85	t	316.3	366.3	416.3	466.3	516.3	566.3	616.3	t	85
	v	5.16	5.59	5.99	6.38	6.76	7.14	7.51	v	
	h	1183.4	1210.2	1235.8	1260.6	1285.2	1309.4	1333.5	h	

t = Temperature, degrees Fahrenheit.
v = Specific volume, in cubic feet, per pound.
h = Total heat from water at 32 degrees, B.t.u.

Pressure Pounds Absolute		Saturated Steam	Degrees of Superheat						Pressure Pounds Absolute	
			50	100	150	200	250	300		
90	t	320.3	370.3	420.3	470.3	520.3	570.3	620.3	t	90
	v	4.89	5.29	5.67	6.04	6.40	6.76	7.11	v	
	h	1184.4	1211.4	1237.2	1262.0	1286.6	1310.8	1334.9	h	
95	t	324.1	374.1	424.1	474.1	524.1	574.1	624.1	t	95
	v	4.65	5.03	5.39	5.74	6.09	6.43	6.76	v	
	h	1185.4	1212.6	1238.4	1263.4	1288.1	1312.3	1336.4	h	
100	t	327.8	377.8	427.8	477.8	527.8	577.8	627.8	t	100
	v	4.43	4.79	5.14	5.47	5.80	6.12	6.44	v	
	h	1186.3	1213.8	1239.7	1264.7	1289.4	1313.6	1337.8	h	
105	t	331.4	381.4	431.4	481.4	531.4	581.4	631.4	t	105
	v	4.23	4.58	4.91	5.23	5.54	5.85	6.15	v	
	h	1187.2	1214.9	1240.8	1265.9	1290.6	1314.9	1339.1	h	
110	t	334.8	384.8	434.8	484.8	534.8	584.8	634.8	t	110
	v	4.05	4.38	4.70	5.01	5.31	5.61	5.90	v	
	h	1188.0	1215.9	1242.0	1267.1	1291.9	1316.2	1340.4	h	
115	t	338.1	388.1	438.1	488.1	538.1	588.1	638.1	t	115
	v	3.88	4.20	4.51	4.81	5.09	5.38	5.66	v	
	h	1188.8	1216.9	1243.1	1268.2	1293.0	1317.3	1341.5	h	
120	t	341.3	391.3	441.3	491.3	541.3	591.3	641.3	t	120
	v	3.73	4.04	4.33	4.62	4.89	5.17	5.44	v	
	h	1189.6	1217.9	1244.1	1269.3	1294.1	1318.4	1342.7	h	
125	t	344.4	394.4	444.4	494.4	544.4	594.4	644.4	t	125
	v	3.58	3.88	4.17	4.45	4.71	4.97	5.23	v	
	h	1190.3	1218.8	1245.1	1270.4	1295.2	1319.5	1343.8	h	
130	t	347.4	397.4	447.4	497.4	547.4	597.4	647.4	t	130
	v	3.45	3.74	4.02	4.28	4.54	4.80	5.05	v	
	h	1191.0	1219.7	1246.1	1271.4	1296.2	1320.6	1344.9	h	
135	t	350.3	400.3	450.3	500.3	550.3	600.3	650.3	t	135
	v	3.33	3.61	3.88	4.14	4.38	4.63	4.87	v	
	h	1191.6	1220.6	1247.0	1272.3	1297.2	1321.6	1345.9	h	
140	t	353.1	403.1	453.1	503.1	553.1	603.1	653.1	t	140
	v	3.22	3.49	3.75	4.00	4.24	4.48	4.71	v	
	h	1192.2	1221.4	1248.0	1273.3	1298.2	1322.6	1346.9	h	
145	t	355.8	405.8	455.8	505.8	555.8	605.8	655.8	t	145
	v	3.12	3.38	3.63	3.87	4.10	4.33	4.56	v	
	h	1192.8	1222.2	1248.8	1274.2	1299.1	1323.6	1347.9	h	
150	t	358.5	408.5	458.5	508.5	558.5	608.5	658.5	t	150
	v	3.01	3.27	3.51	3.75	3.97	4.19	4.41	v	
	h	1193.4	1223.0	1249.6	1275.1	1300.0	1324.5	1348.8	h	
155	t	361.0	411.0	461.0	511.0	561.0	611.0	661.0	t	155
	v	2.92	3.17	3.41	3.63	3.85	4.06	4.28	v	
	h	1194.0	1223.6	1250.5	1276.0	1300.8	1325.3	1349.7	h	
160	t	363.6	413.6	463.6	513.6	563.6	613.6	663.6	t	160
	v	2.83	3.07	3.30	3.53	3.74	3.95	4.15	v	
	h	1194.5	1224.5	1251.3	1276.8	1301.7	1326.2	1350.6	h	
165	t	366.0	416.0	466.0	516.0	566.0	616.0	666.0	t	165
	v	2.75	2.99	3.21	3.43	3.64	3.84	4.04	v	
	h	1195.0	1225.2	1252.0	1277.6	1302.5	1327.1	1351.5	h	
170	t	368.5	418.5	468.5	518.5	568.5	618.5	668.5	t	170
	v	2.68	2.91	3.12	3.34	3.54	3.73	3.92	v	
	h	1195.4	1225.9	1252.8	1278.4	1303.3	1327.9	1352.3	h	

t = Temperature, degrees Fahrenheit.
v = Specific volume, in cubic feet, per pound.
h = Total heat from water at 32 degrees, B.t.u.

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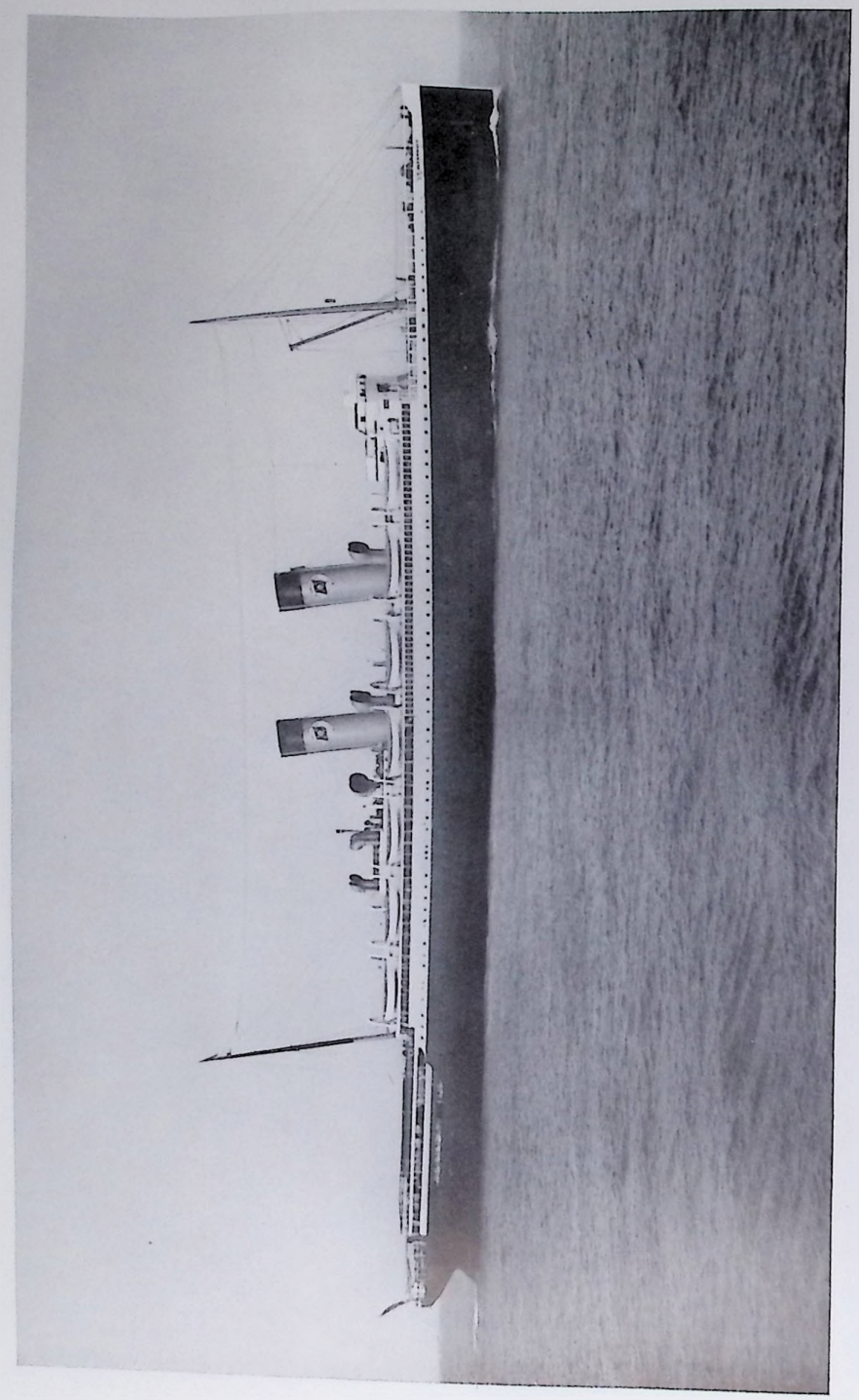
Pressure Pounds Absolute		Saturated Steam	Degrees of Superheat						Pressure Pounds Absolute	
			50	100	150	200	250	300		
175	t	370.8	420.8	470.8	520.8	570.8	620.8	670.8	t	
	v	2.60	2.83	3.04	3.24	3.44	3.63	3.82	v	175
	h	1195.9	1226.6	1253.6	1279.1	1304.1	1328.7	1353.2	h	
180	t	373.1	423.1	473.1	523.1	573.1	623.1	673.1	t	
	v	2.53	2.75	2.96	3.16	3.35	3.54	3.72	v	180
	h	1196.4	1227.2	1254.3	1279.9	1304.8	1329.5	1353.9	h	
185	t	375.4	425.4	475.4	525.4	575.4	625.4	675.4	t	
	v	2.47	2.68	2.89	3.08	3.27	3.45	3.63	v	185
	h	1196.8	1227.9	1255.0	1280.6	1305.6	1330.2	1354.7	h	
190	t	377.6	427.6	477.6	527.6	577.6	627.6	677.6	t	
	v	2.41	2.62	2.81	3.00	3.19	3.37	3.55	v	190
	h	1197.3	1228.6	1255.7	1281.3	1306.3	1330.9	1355.5	h	
195	t	379.8	429.8	479.8	529.8	579.8	629.8	679.8	t	
	v	2.35	2.55	2.75	2.93	3.11	3.29	3.46	v	195
	h	1197.7	1229.2	1256.4	1282.0	1307.0	1331.6	1356.2	h	
200	t	381.9	431.9	481.9	531.9	581.9	631.9	681.9	t	
	v	2.29	2.49	2.68	2.86	3.04	3.21	3.38	v	200
	h	1198.1	1229.8	1257.1	1282.6	1307.7	1332.4	1357.0	h	
205	t	384.0	434.0	484.0	534.0	584.0	634.0	684.0	t	
	v	2.24	2.44	2.62	2.80	2.97	3.14	3.30	v	205
	h	1198.5	1230.4	1257.7	1283.3	1308.3	1333.0	1357.7	h	
210	t	386.0	436.0	486.0	536.0	586.0	636.0	686.0	t	
	v	2.19	2.38	2.56	2.74	2.91	3.07	3.23	v	210
	h	1198.8	1231.0	1258.4	1284.0	1309.0	1333.7	1358.4	h	
215	t	388.0	438.0	488.0	538.0	588.0	638.0	688.0	t	
	v	2.14	2.33	2.51	2.68	2.84	3.00	3.16	v	215
	h	1199.2	1231.6	1259.0	1284.6	1309.7	1334.4	1359.1	h	
220	t	389.9	439.9	489.9	539.9	589.9	639.9	689.9	t	
	v	2.09	2.28	2.45	2.62	2.78	2.94	3.10	v	220
	h	1199.6	1232.2	1259.6	1285.2	1310.3	1335.1	1359.8	h	
225	t	391.9	441.9	491.9	541.9	591.9	641.9	691.9	t	
	v	2.05	2.23	2.40	2.57	2.72	2.88	3.03	v	225
	h	1199.9	1232.7	1260.2	1285.9	1310.9	1335.7	1360.3	h	
230	t	393.8	443.8	493.8	543.8	593.8	643.8	693.8	t	
	v	2.00	2.18	2.35	2.51	2.67	2.82	2.97	v	230
	h	1200.2	1233.2	1260.7	1286.5	1311.6	1336.3	1361.0	h	
235	t	395.6	445.6	495.6	545.6	595.6	645.6	695.6	t	
	v	1.96	2.14	2.30	2.46	2.62	2.77	2.91	v	235
	h	1200.6	1233.8	1261.4	1287.1	1312.2	1337.0	1361.7	h	
240	t	397.4	447.4	497.4	547.4	597.4	647.4	697.4	t	
	v	1.92	2.09	2.26	2.42	2.57	2.71	2.85	v	240
	h	1200.9	1234.3	1261.9	1287.6	1312.8	1337.6	1362.3	h	
245	t	399.3	449.3	499.3	549.3	599.3	649.3	699.3	t	
	v	1.89	2.05	2.22	2.37	2.52	2.66	2.80	v	245
	h	1201.2	1234.8	1262.5	1288.2	1313.3	1338.2	1362.9	h	
250	t	401.0	451.0	501.0	551.0	601.0	651.0	701.0	t	
	v	1.85	2.02	2.17	2.33	2.47	2.61	2.75	v	250
	h	1201.5	1235.4	1263.0	1288.8	1313.9	1338.8	1363.5	h	
255	t	402.8	452.8	502.8	552.8	602.8	652.8	702.8	t	
	v	1.81	1.98	2.14	2.28	2.43	2.56	2.70	v	255
	h	1201.8	1235.9	1263.6	1289.3	1314.5	1339.3	1364.1	h	

t = Temperature, degrees Fahrenheit.
v = Specific volume, in cubic feet, per pound.
h = Total heat from water at 32 degrees, B.t.u.

Pressure Pounds Absolute		Saturated Steam	Degrees of Superheat						Pressure Pounds Absolute	
			50	100	150	200	250	300		
260	t	404.5	454.5	504.5	554.5	604.5	654.5	704.5	t	
	v	1.78	1.94	2.10	2.24	2.39	2.52	2.65	v	260
	h	1202.1	1236.4	1264.1	1289.9	1315.1	1340.0	1364.7	h	
270	t	407.9	457.9	507.9	557.9	607.9	657.9	707.9	t	
	v	1.72	1.87	2.02	2.16	2.30	2.43	2.56	v	270
	h	1202.6	1237.4	1265.2	1290.9	1316.2	1341.1	1365.9	h	
280	t	411.2	461.2	511.2	561.2	611.2	661.2	711.2	t	
	v	1.66	1.81	1.95	2.09	2.22	2.35	2.48	v	280
	h	1203.1	1238.4	1266.2	1291.9	1317.2	1342.2	1367.0	h	
290	t	414.4	464.4	514.4	564.4	614.4	664.4	714.4	t	
	v	1.60	1.75	1.89	2.02	2.15	2.28	2.40	v	290
	h	1203.6	1239.3	1267.1	1293.0	1318.3	1343.2	1368.0	h	
300	t	417.5	467.5	517.5	567.5	617.5	667.5	717.5	t	
	v	1.55	1.69	1.83	1.96	2.09	2.21	2.33	v	300
	h	1204.1	1240.3	1268.2	1294.0	1319.3	1344.3	1369.2	h	
310	t	420.5	470.5	520.5	570.5	620.5	670.5	720.5	t	
	v	1.50	1.64	1.78	1.90	2.02	2.14	2.26	v	310
	h	1204.5	1241.1	1269.1	1294.9	1320.3	1345.3	1370.2	h	
320	t	423.4	473.4	523.4	573.4	623.4	673.4	723.4	t	
	v	1.46	1.59	1.72	1.85	1.96	2.08	2.19	v	320
	h	1204.9	1242.0	1270.0	1295.9	1321.3	1346.3	1371.3	h	
330	t	426.3	476.3	526.3	576.3	626.3	676.3	726.3	t	
	v	1.41	1.54	1.67	1.80	1.91	2.02	2.13	v	330
	h	1205.3	1242.9	1270.9	1296.8	1322.2	1347.3	1272.3	h	
340	t	429.1	479.1	529.1	579.1	629.1	679.1	729.1	t	
	v	1.37	1.50	1.63	1.75	1.86	1.97	2.08	v	340
	h	1205.7	1243.8	1271.8	1297.8	1323.2	1348.3	1373.3	h	
350	t	431.9	481.9	531.9	581.9	631.9	681.9	731.9	t	
	v	1.33	1.46	1.58	1.70	1.81	1.92	2.02	v	350
	h	1206.1	1244.6	1272.7	1298.7	1324.1	1349.3	1374.3	h	
360	t	434.6	484.6	534.6	584.6	634.6	684.6	734.6	t	
	v	1.30	1.42	1.54	1.65	1.76	1.87	1.97	v	360
	h	1206.4	1245.4	1273.6	1299.6	1325.0	1350.2	1375.3	h	
370	t	437.2	487.2	537.2	587.2	637.2	687.2	737.2	t	
	v	1.26	1.38	1.50	1.61	1.72	1.82	1.92	v	370
	h	1206.8	1246.2	1274.4	1300.4	1325.9	1351.1	1376.2	h	
380	t	439.8	489.8	539.8	589.8	639.8	689.8	739.8	t	
	v	1.23	1.35	1.47	1.58	1.68	1.78	1.88	v	380
	h	1207.1	1247.0	1275.3	1301.8	1326.8	1352.0	1377.1	h	
390	t	442.3	492.3	542.3	592.3	642.3	692.3	742.3	t	
	v	1.20	1.32	1.43	1.54	1.64	1.74	1.83	v	390
	h	1207.4	1247.8	1276.1	1302.3	1327.7	1353.0	1378.1	h	
400	t	444.8	494.8	544.8	594.8	644.8	694.8	744.8	t	
	v	1.17	1.28	1.40	1.50	1.60	1.70	1.79	v	400
	h	1207.7	1248.6	1276.9	1303.0	1328.6	1353.9	1379.1	h	
450	t	456.5	506.5	556.5	606.5	656.5	706.5	756.5	t	
	v	1.04	1.14	1.25	1.35	1.44	1.53	1.61	v	450
		1204.	1252.	1281.	1307.	1333.	1358.	1383.	h	

t = Temperature, degrees Fahrenheit.
v = Specific volume, in cubic feet, per pound
h = Total heat from water at 32 degrees, B.t.u.

Marine Steam (1928)
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Steamer "H. F. Alexander," 26,000 Shaft Horse-power, Babcock & Wilcox Boilers
Owners, Pacific Steamship Company (Admiral Line), Seattle, Wash.

MOISTURE IN STEAM

THE presence of moisture in steam causes a loss, not only in the practical waste of the heat utilized to raise this moisture from the temperature of the feed water to the temperature of the steam, but also through the increased initial condensation in an engine cylinder and through friction and other actions in a steam turbine. The presence of such moisture also interferes with proper cylinder lubrication, causes a knocking in the engine and a water hammer in the steam pipes. In steam turbines it will cause erosion of the blades.

The percentage by weight of steam in a mixture of steam and water is called the *quality of the steam.*

The apparatus used to determine the moisture content of steam is called a calorimeter, though since it may not measure the heat in the steam, the name is not descriptive of the function of the apparatus. The first form used was the "barrel calorimeter," but the liability of error was so great that its use was abandoned. Modern calorimeters are in general of either the throttling or separator type.

Throttling Calorimeter—Fig. 6 shows a typical form of throttling calorimeter. Steam is drawn from a vertical main through the sampling nipple, passes around the first thermometer cup, then through a one-eighth-inch orifice in a disk between two flanges, and lastly, around the second thermometer cup and to the atmosphere. Thermometers are inserted in the wells, which should be filled with mercury or heavy cylinder oil.

Fig. 6. Throttling Calorimeter and Sampling Nozzle

The instrument and all pipes and fittings leading to it should be thoroughly insulated to diminish radiation losses. Care must be taken to prevent the orifice from becoming choked with dirt and to see that no leaks occur. The exhaust pipe should be short to prevent back pressure below the disk.

When steam passes through an orifice from a higher to a lower pressure, as is the case with the throttling calorimeter, no external work has to be done in overcoming a resistance. Hence, if there is no loss from radiation, the quantity of heat in the steam will be exactly the same after passing the orifice as before passing. If the higher steam pressure is 160 pounds gauge and the lower pressure that of the atmosphere, the total heat in a pound of dry steam at the former pressure is 1195.9 B.t.u. and at the latter pressure 1150.4 B.t.u., a difference of 45.4 B.t.u. As this heat will still exist in the steam at the lower pressure, since there is no external work done, its effect must be to superheat the steam. Assuming the specific heat of superheated steam to be 0.47, each pound passing through will be superheated $45.4 \div 0.47 = 96.6$ degrees. If, however, the steam had contained one per cent of moisture, it would have contained less heat units per pound than if it were dry. Since the latent heat of

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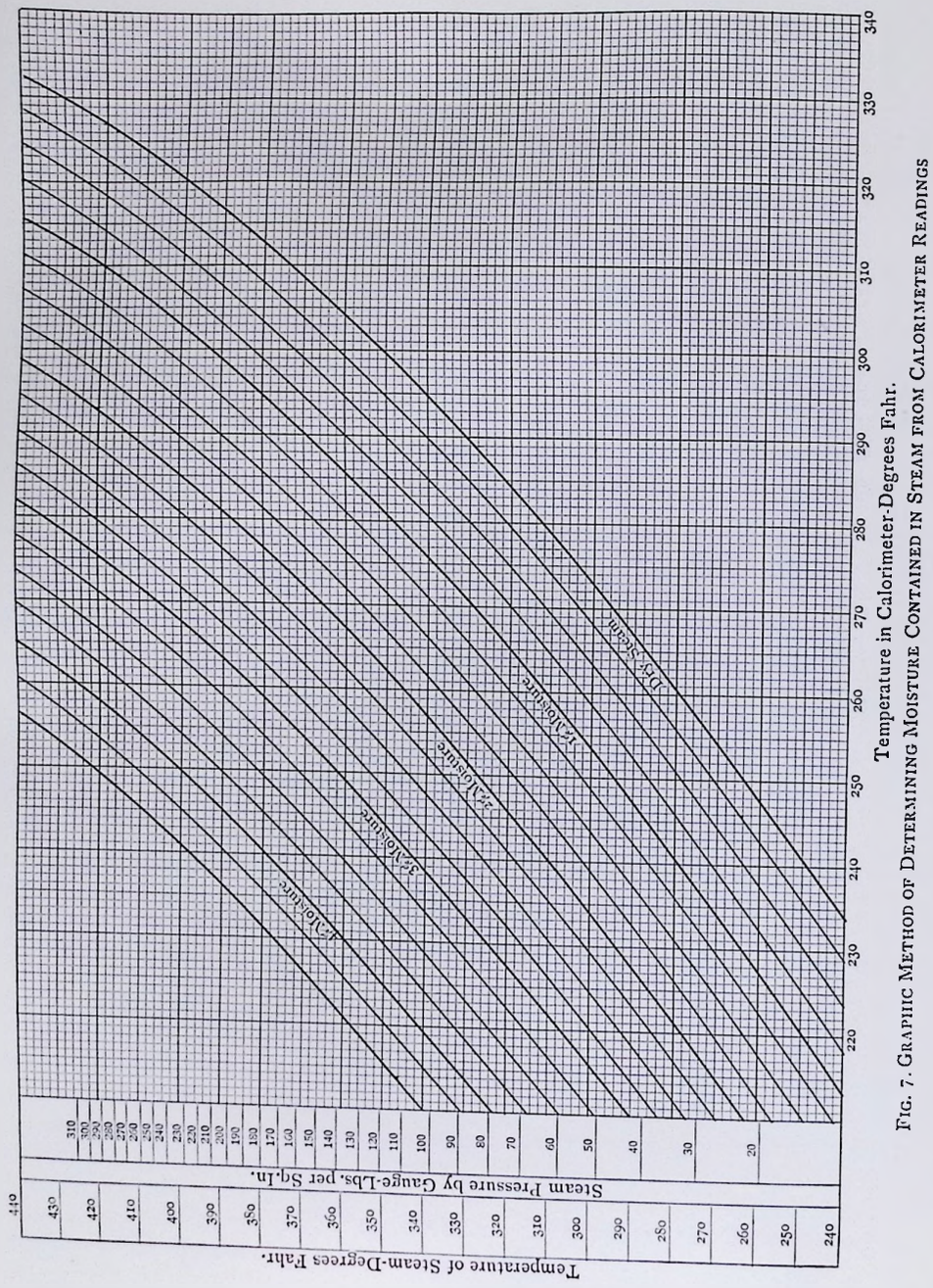


FIG. 7. GRAPHIC METHOD OF DETERMINING MOISTURE CONTAINED IN STEAM FROM CALORIMETER READINGS

steam at 160 pounds gauge pressure is 852.8 B.t.u., it follows that the one per cent of moisture would have required 8.5 B.t.u. to evaporate it, leaving only 45.4—8.5 = 36.9 B.t.u. available for superheating; hence, the superheat would be 36.9 ÷ 0.47 = 78.5 degrees, as against 96.6 degrees for dry steam. In a similar manner, the degree of superheat for other percentages of moisture may be determined. The action of the throttling calorimeter is based upon the foregoing facts, as shown below:

Let H = total heat of one pound of steam at boiler pressure,
L = latent heat of steam at boiler pressure,
h = total heat of steam at the reduced pressure after passing the orifice,
t_1 = temperature of saturated steam at the reduced pressure,
t_2 = temperature of steam after expanding through the orifice in the disk,
0.47 = the specific heat of saturated steam at atmospheric pressure,
x = proportion by weight of moisture in steam.

The difference in B.t.u. in a pound of steam at the boiler pressure and after passing the orifice is the heat available for evaporating the moisture content and superheating the steam. Therefore,

$$H - h = xL + 0.47\ (t_2 - t_1)$$

$$\text{or } x = \frac{H - h - 0.47\ (t_2 - t_1)}{L} \qquad (6)$$

Almost invariably the lower pressure is taken as that of the atmosphere. Under such conditions, $h = 1150.4$ and $t_1 = 212$ degrees. The formula thus becomes:

$$x = \frac{H - 1150.4 - 0.47\ (t_2 - 212)}{L} \qquad (7)$$

For practical work it is more convenient to dispense with the upper thermometer in the calorimeter and to measure the pressure in the steam main by an accurate steam pressure gauge.

A chart may be used for determining the value of x for approximate work without the necessity for computation. Such a chart is shown in Fig. 7 and its use is as follows: Assume a gauge pressure of 180 pounds and a thermometer reading of 295 degrees. The intersection of the vertical line from the scale of temperatures as shown by the calorimeter thermometer and the horizontal line from the scale of gauge pressures will indicate directly the percentage of moisture in the steam as read from the diagonal scale. In the present instance, this percentage is 1.0.

SOURCES OF ERROR IN THE APPARATUS—A slight error may arise from the value, 0.47, used as the specific heat of superheated steam at atmospheric pressure. This value, however, is very nearly correct and any error resulting from its use will be negligible.

There is ordinarily a larger source of error due to the fact that the stem of the thermometer is not heated to its full length, to an initial error in the thermometer and to radiation losses.

With an ordinary thermometer immersed in the well to the 100 degrees mark, the error when registering 300 degrees would be about 3 degrees and the true temperature be 303 degrees.*

The steam is evidently losing heat through radiation from the moment it enters the sampling nipple. The heat available for evaporating moisture and superheating

* See Stem Correction, page 42.

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Steamer "Joseph H. Frantz," 3000 Indicated Horse-power, Babcock & Wilcox Boilers and Superheaters
Owners, Columbia Steamship Company, Cleveland, Ohio

steam after it has passed through the orifice into the lower pressure will be diminished by just the amount lost through radiation, and the value of t_2, as shown by the calorimeter thermometer, will, therefore, be lower than if there were no such loss. The method of correcting for the thermometer and radiation error recommended by the Power Test Committee of the American Society of Mechanical Engineers is by referring the readings as found on the boiler trial to a "normal" reading of the thermometer. This normal reading is the reading of the lower calorimeter thermometer for dry saturated steam, and should be determined by attaching the instrument to a horizontal steam pipe in such a way that the sampling nozzle projects upward to near the top of the pipe, there being no perforations in the nozzle and the steam taken only through its open upper end. The test should be made with the steam in a quiescent state and with the steam pressure maintained as nearly as possible at the pressure observed in the main trial, the calorimeter thermometer to be the same as was used on the trial or one exactly similar.

With a normal reading thus obtained for a pressure approximately the same as existed in the trial, the true percentage of moisture in the steam, that is, with the proper correction made for radiation, may be calculated as follows:

Let T denote the normal reading for the conditions existing in the trial. The effect of radiation from the instrument as pointed out will be to lower the temperature of the steam at the lower pressure. Let x_1 represent the proportion of water in the steam which will lower its temperature an amount equal to the loss by radiation. Then,

$$x_1 = \frac{H - h - 0.47\ (T - t_1)}{L}$$

This amount of moisture, x_1, was not in the steam originally but is the result of condensation in the instrument through radiation. Hence, the true amount of moisture in the steam represented by X is the difference between the amount as determined in the trial and that resulting from condensation, or,

$$\begin{aligned} X &= x - x_1 \\ &= \frac{H - h - 0.47\ (t_2 - t_1)}{L} - \frac{H - h - 0.47\ (T - t_1)}{L} \\ &= \frac{0.47\ (T - t_2)}{L} \qquad (8) \end{aligned}$$

As T and t_2 are taken with the same thermometer under the same set of conditions, any error in the reading of the thermometers will be approximately the same for the temperatures T and t_2 and the above method therefore corrects for both the radiation and thermometer errors. The theoretical readings for dry steam, where there are no losses due to radiation, are obtainable from formula (7) by letting $x = 0$ and solving for t_2. The difference between the theoretical reading and the normal reading for no moisture will be the thermometer and radiation correction to be applied in order that the correct reading of t_2 may be obtained.

For any calorimeter within the range of its ordinary use, such a thermometer and radiation correction taken from one normal reading is approximately correct for any conditions with the same or a duplicate thermometer.

The percentage of moisture in the steam, corrected for thermometer error and radiation and the correction to be applied to the particular calorimeter used, would be

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determined as follows: Assume a gauge pressure in the trial to be 180 pounds and the thermometer reading to be 295 degrees. A normal reading, taken in the manner described, gives a value of $T=303$ degrees; then, the percentage of moisture corrected for thermometer error and radiation is,

$$x=\frac{0.47\ (303-295)}{845.0}$$

$$=0.45 \text{ per cent.}$$

The theoretical reading for dry steam will be,

$$0=\frac{1197.7-1150.4-0.47\ (t_2-212)}{845.0}$$

$t_2=313$ degrees.

The thermometer and radiation correction to be applied to the instrument used, therefore, over the ordinary range of pressure is

Correction = 313 — 303 = 10 degrees.

The chart may be used in the determination of the correct reading of moisture percentage and the permanent radiation correction for the instrument used, without computation, as follows: Assume the same trial pressure, feed temperature and normal reading as above. If the normal reading is found to be 303 degrees, the correction for thermometer and radiation will be the theoretical reading for dry steam as found from the chart, less this normal reading, or 10 degrees correction. The correct temperature for the trial in question is, therefore, 305 degrees. The moisture corresponding to this temperature and 180 pounds gauge pressure will be found from the chart to be 0.45 per cent.

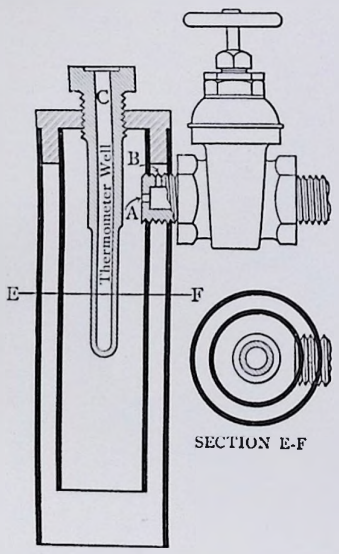


Fig. 8. Compact Throttling Calorimeter

There are many forms of throttling calorimeters, all of which work upon the same principle. The simplest one is probably that shown in Fig. 6. An extremely convenient and compact design is shown in Fig. 8. This calorimeter consists of two concentric metal cylinders screwed to a cap containing a thermometer well. The steam pressure is measured by a gauge placed in the supply pipe or other convenient location. Steam passes through the orifice *A* and expands to atmospheric pressure, its temperature at this pressure being measured by a thermometer placed in the cup *C*. To prevent radiation losses, as far as possible, the annular space between the two cylinders is used as a jacket, steam being supplied to this space through the hole *B*.

The limits of moisture within which the throttling calorimeter will work are, at sea level, from 2.88 per cent at 50 pounds gauge pressure and 7.17 per cent moisture at 250 pounds pressure.

Sampling Nozzle—The principal source of error in steam calorimeter determinations is the failure to obtain an average sample of the steam delivered by the boiler,

and it is extremely doubtful whether such a sample is ever obtained. The two governing features in obtaining such a sample are the type of sampling nozzle used and its location.

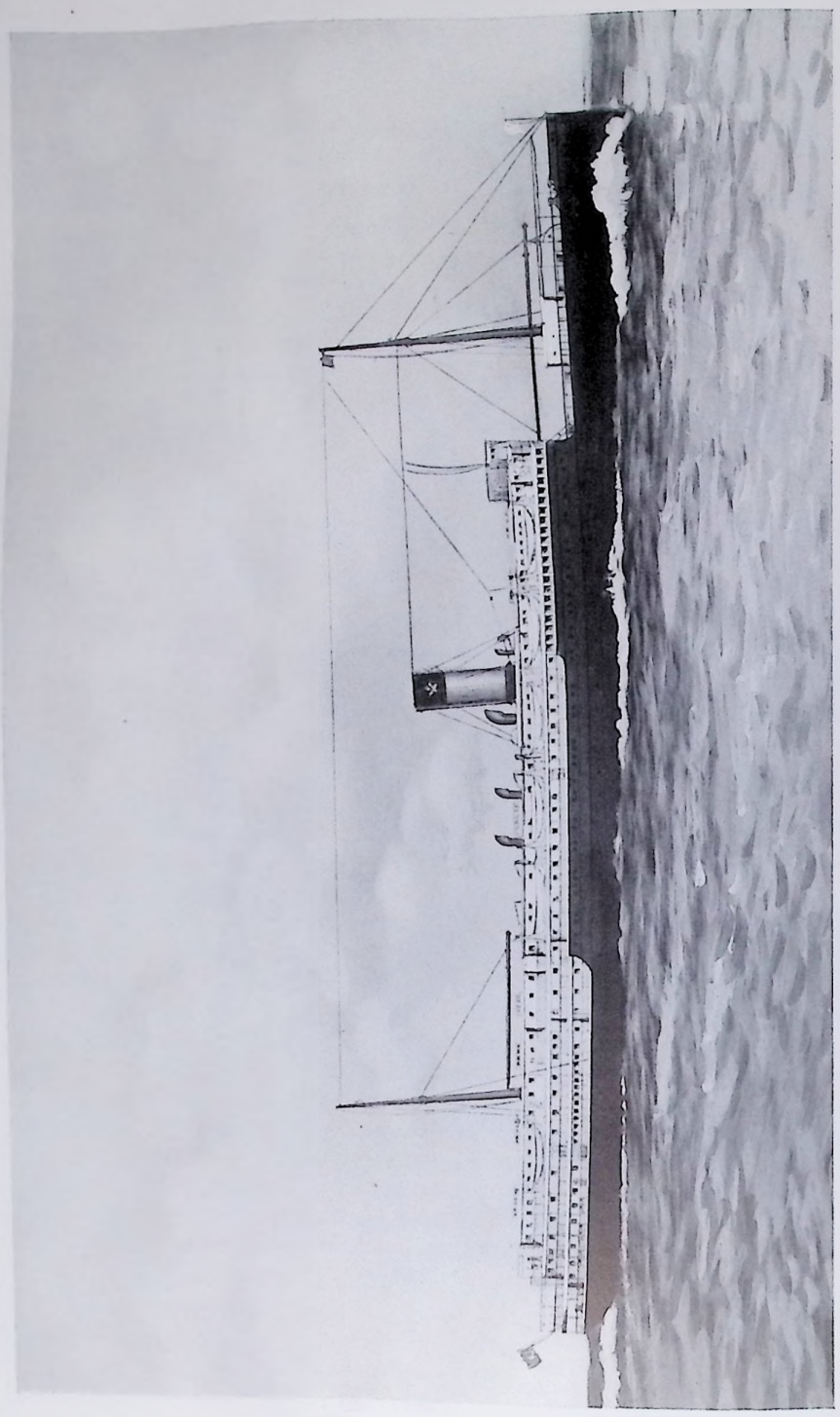
The American Society of Mechanical Engineers recommends a sampling nozzle made of one-half-inch iron pipe closed at the inner end and the interior portion perforated with not less than twenty one-eighth-inch holes equally distributed from end to end and preferably drilled in irregular or spiral rows, with the first hole not less than one-half inch from the wall of the pipe. Many engineers object to the use of a perforated sampling nozzle because it ordinarily indicates a higher percentage of moisture than is actually present in the steam. This is due to the fact that if the perforations come close to the inner surface of the pipe, the moisture, which in many instances clings to this surface, will flow into the calorimeter and cause a large error. Where a perforated nozzle is used, in general it may be said that the perforations should be at least one inch from the inner pipe surface.

A sampling nozzle, open at the inner end and unperforated, undoubtedly gives as accurate a measure as can be obtained of the moisture in the steam passing that end. It would appear that a satisfactory method of obtaining an average sample of the steam would result from the use of an open end unperforated nozzle passing through a stuffing box which would allow the end to be placed at any point across the diameter of the steam pipe.

Location of Sampling Nozzle—The calorimeter should be located as near as possible to the point from which the steam is taken and the sampling nozzle should be placed in a section of the main pipe near the boiler and where there is no chance of moisture pocketing in the pipe. The American Society of Mechanical Engineers recommends that a sampling nozzle of which a description has been given, should be located in a vertical main, rising from the boiler with its closed end extending nearly across the pipe. Where non-return valves are used, or where there are horizontal connections leading from the boiler to a vertical outlet, water may collect at the lower end of the uptake pipe and be blown upward in a spray which will not be carried away by the steam owing to a lack of velocity. A sample taken from the lower part of this pipe will show a greater amount of moisture than a true sample. With gooseneck connections a small amount of water may collect on the bottom of the pipe near the upper end where the inclination is such that the tendency to flow backward is ordinarily counterbalanced by the flow of steam forward over its surface; but when the velocity momentarily decreases the water flows back to the lower end of the gooseneck and increases the moisture at that point, making it an undesirable location for sampling. In any case, it should be borne in mind that with low velocities the tendency is for drops of entrained water to settle to the bottom of the pipe, and to be temporarily broken up into spray whenever an abrupt bend or other disturbance is met.

Before taking any calorimeter reading, steam should be allowed to flow through the instrument freely until it is thoroughly heated. The method of using a throttling calorimeter is evident from the description of the instrument given and the principle upon which it works.

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Steamer "Dixie," 7100 Shaft Horse-power, Babcock & Wilcox Boilers, Superheaters and Oil Burners
Owners, the Southern Pacific Company, New York, N. Y.

SUPERHEATED STEAM

WHILE, as has been seen, the use of superheated steam is old, but little was known until recently of its properties.

In the preparation and computation of a table giving the properties of superheated steam the important factor is its specific heat. Without this value it is impossible to determine the total heat of superheated steam, that is, the amount of heat that must be applied to bring saturated steam to any given superheated condition.

Regnault, in 1862, determined the specific heat of superheated steam, basing his value, which he gives as 0.48, upon four series of experiments, all of which covered approximately the same temperature range and all at atmospheric pressure. While Regnault's experiments in no way proved that the value of specific heat as given by him was independent of either pressure or temperature, this value was accepted for some forty years and applied to higher pressures and temperatures as well as to those within the range of his experiments.

With the revival of the use of superheated steam in the nineties, the assumption that the specific heat was constant regardless of pressure and temperature was found to be incorrect, and a number of investigators turned their attention to the subject. Grindley and Greissmann's experiments, in 1900, appeared to indicate that the specific heat increased with the temperature and was independent of the pressure. Other investigators, however, proved this untrue. Among those giving their services in this investigation were Lorenz, Linde, Holborn and Henning, Callendar, Carpenter, Thomas, and Knoblauch and Jakob. The experiments of the last two investigators were laborious and comprehensive and they made special efforts to eliminate the presence of moisture in the steam in observations near the saturation point, an error which unavoidably crept into other investigations.

Table 16* gives, in condensed form, the properties of superheated steam as calculated by Lionel S. Marks and Harvey N. Davis.

Because of the continually increasing use of steam at pressures and temperatures above the range covered by the various sets of steam tables available (*i.e.* because over the higher range the values have been extrapolated), and because of the discrepancies in values in the different tables of well-known authorities, The American Society of Mechanical Engineers, in 1921, became sponsors of a program of research by prominent engineers and physicists to determine definitely the properties of steam over a range of pressure and temperatures such as would cover all conditions that might be encountered. This work is being carried on jointly by the United States Bureau of Standards, Massachusetts Institute of Technology, and the School of Engineering, Harvard University. The results of these and earlier investigations are to be used as the basis in preparing new steam tables which, it is hoped, will be adopted as standard.

Specific Heat—It is frequently necessary to know the mean specific heat of superheated steam at a given pressure and temperature. Fig. 9 shows graphically values of this mean specific heat for various temperatures and pressures and has been computed from values of total heat as given in Marks and Davis' steam tables. The calculation involved, expressed as a formula, is:

* See pages 82 to 85.

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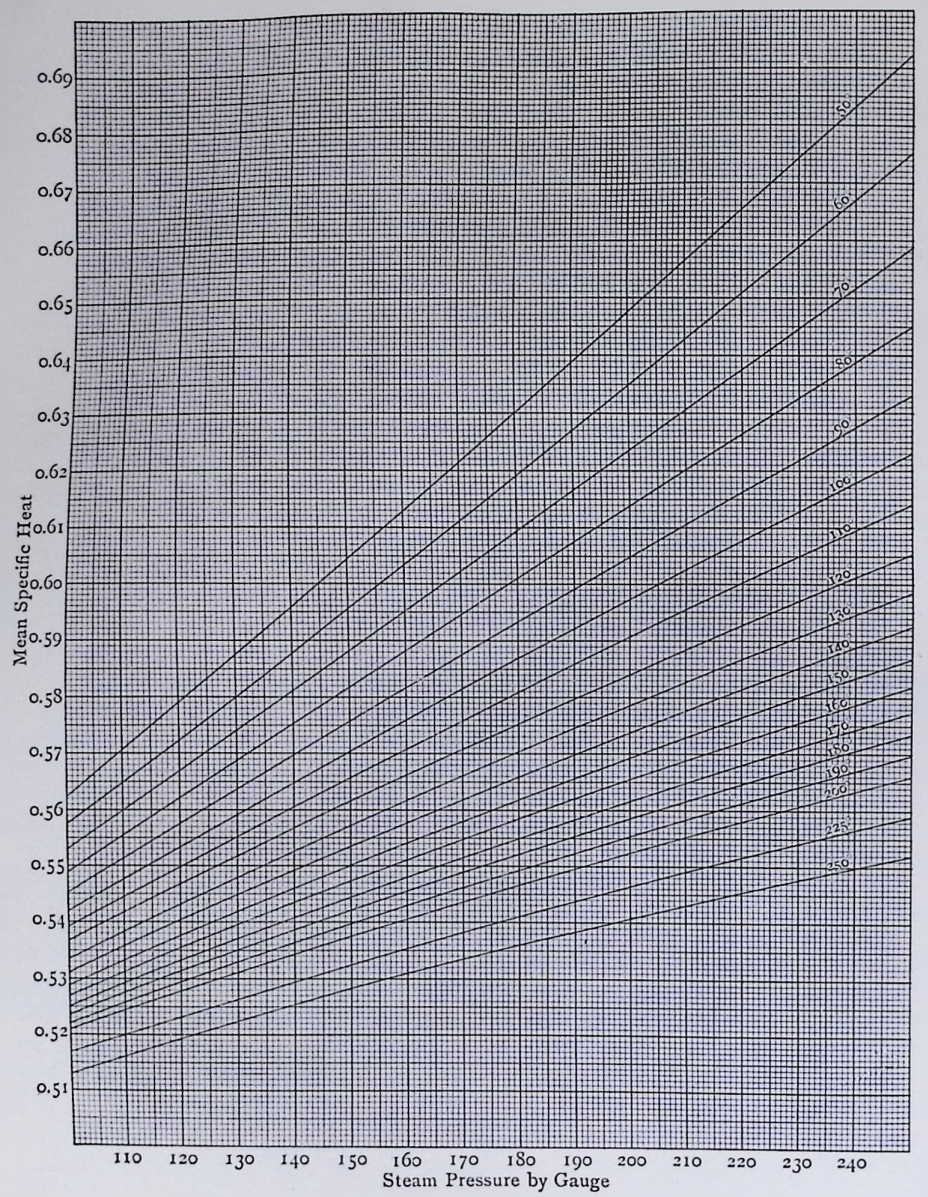


FIG. 9. MEAN SPECIFIC HEAT OF SUPERHEATED STEAM
CURVES REPRESENT DEGREES OF SUPERHEAT
Calculated from Marks and Davis

$$Sp.\,Ht. = \frac{H - h}{T_{sup} - T_{sat}} \qquad (9)$$

Where $Sp.\,Ht.$ = mean specific heat at a given pressure and temperature,
H = total heat of superheated steam at a given temperature and pressure,
h = total heat of saturated steam at the corresponding pressure,
T_{sup} and T_{sat} = temperatures of superheated and saturated steam.

FACTOR OF EVAPORATION WITH SUPERHEATED STEAM—When superheat is present in steam during a boiler trial and superheated steam tables are available, the method of calculating the factor of evaporation is the same as in the case of saturated steam, namely:

$$Factor = \frac{H - h}{L} \qquad (4)$$

Where H = total heat in superheated steam,
h = sensible heat in the feed water above 32 degrees Fahrenheit,
L = 970.4, the latent heat of evaporation at atmospheric pressure.

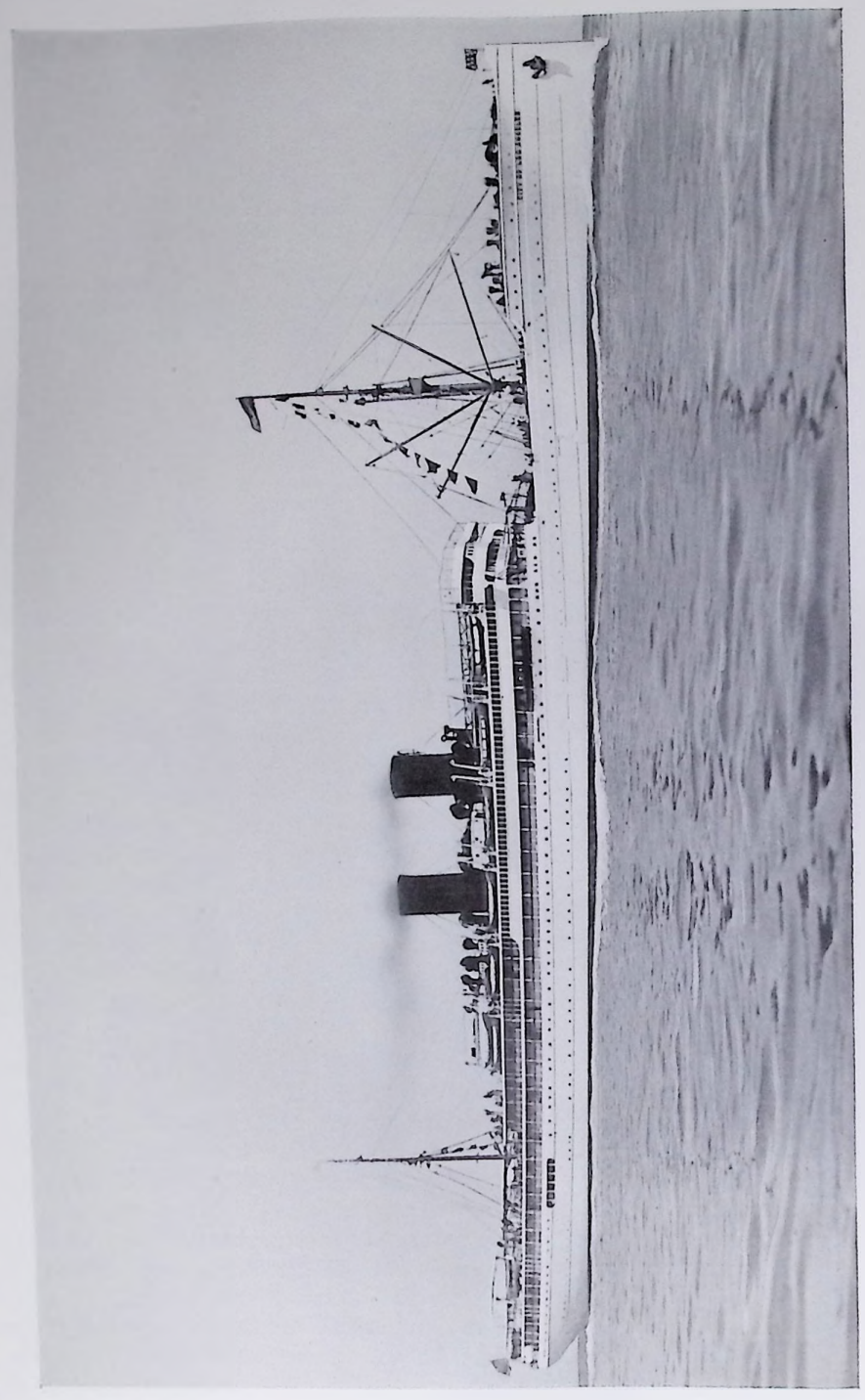
ADVANTAGES OF SUPERHEATED STEAM—That advantages are to be secured through the use of superheated steam is perhaps most conclusively shown by the fact that in stationary practice the largest and most economical power plants throughout the world are equipped with superheaters. Such advantages have also been appreciated by marine engineers, though it is only within recent years that the higher degrees of superheat common in stationary work have been considered for marine practice. Regardless of these facts, in numerous instances the conviction has to an extent held that the use of superheated steam would involve operating difficulties which, taken in connection with the added first cost, would more than offset any fuel saving. There are, of course, conditions under which the installation of superheaters would in no way be advisable and when a superheater installation is contemplated, it must be considered in all of its phases. The actual saving possible by the use of superheated steam must be balanced against such factors as the initial cost and upkeep of the superheater, the efficiency of the design of superheater to be installed, the nature of the service in question, the design of the prime movers, a consideration of pipe and fittings, and the like.

No general statement of the saving possible may be made since this may vary widely with a number of the factors above enumerated.

The logical method of approaching the subject would appear to be, first, a consideration of the saving in fuel possible through the use of superheated steam regardless of other factors; and, second, a consideration of the bearing of such factors on the advisability of a superheater installation.

In considering the saving possible by the use of superheated steam, it is too often assumed that there is only a saving in the prime movers, a saving which is at least partially offset by an increase in the fuel consumption of the boilers generating steam. This misconception is due to the fact that the fuel consumption of the boiler is only considered in connection with a definite weight of steam. It is true that where such a definite weight is to be superheated, an added amount of fuel must be

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Steamer "City of Los Angeles," 12,000 Shaft Horse-power, Babcock & Wilcox Boilers, Superheaters and Oil Burners
Owners, Los Angeles Steamship Co., Wilmington, Calif.

burned. With a properly designed superheater, where the combined efficiency of the boiler and superheater will be at least as high as of a boiler alone, the approximate increase in coal consumption for producing a given weight of steam will be as follows:

Superheat Degrees	*Added Fuel Per Cent*	*Superheat Degrees*	*Added Fuel Per Cent*
25	1.59	100	5.69
50	3.07	150	8.19
75	4.38	200	10.58

These figures represent the added fuel necessary for superheating a definite weight of steam to the number of degrees given. A heat unit basis is standard in the consideration of boiler evaporation and from this standpoint, again providing the efficiency of the boiler and superheater is as high as of the boiler alone, there is no additional fuel required to generate steam containing a definite number of heat units whether such units be due to superheat or saturation. That is, if 6 per cent more fuel is required to generate and superheat to 100 degrees a definite weight of steam, over what would be required to produce the same weight of saturated steam, that steam, when superheated, will contain 6 per cent more heat units above the feedwater temperature than if saturated.

The statement that for an additional number of heat units generated in the furnace of a boiler equipped with a superheater, there will be an equivalent increase in the number of heat units appearing in the superheated steam over saturated, is based, as indicated, upon an equal efficiency for the boiler and superheater as for the boiler in which no superheater is installed. As a matter of fact, the efficiency of a boiler and superheater, where the latter is properly designed and located, will be slightly higher for the same set of furnace conditions than will be the efficiency of a boiler in which no superheater is installed. A superheater, properly placed within the boiler setting in such a way that the products of combustion for generating saturated steam are utilized as well for superheating that steam, will not in any way alter furnace conditions. With a given set of such furnace conditions, for a given amount of fuel burned, the fact that additional surface, whether as boiler heating or superheating surface, is placed in such a manner that the gases must sweep over it, will tend to lower the temperature of the exit gases. It is such a lowering of exit-gas temperature that is the ultimate indication of added efficiency. Though the amount of this added efficiency is difficult to determine by test, that there is an increase is unquestionable.

Where a properly designed superheater is installed in a boiler, the boiler heating surface, in the generation of a definite number of heat units, is relieved of a portion of the work which would be required were these heat units delivered in saturated steam. This results either in a reduction in the capacity at which it is necessary to operate the boiler itself, apart from the superheater, in the developing of a definite number of boiler horse-power, with a consequent saving in the apparatus due to a decreased load, or it enables the same number of horse-power to be developed from a smaller number of boilers, with the boiler heating surface doing the same amount of work as if no superheaters were installed. Such a superheater needs practically no attention, is not subject to a large upkeep cost or depreciation, and performs its function without in any way interfering with the operation of the boiler.

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Steamer "El Oceano," 6000 Shaft Horse-power, Babcock & Wilcox Boilers, Superheaters and Oil Burners
Owners, the Southern Pacific Company, New York, N. Y.

Following the course of the steam, the advantage of the use of superheated steam is next seen in the general absence of water in the pipes. While it is possible for a pipe through which superheated steam is flowing to carry water also, there is usually an entire absence of such water in the piping system, especially where the piping is well covered. The thermal conductivity of superheated steam, that is, its power to receive from or to give out heat to surrounding bodies, is much lower than that of saturated steam and its heat, therefore, will not be transmitted so rapidly to the walls of the pipe as will the heat from saturated steam. When a pipe is carrying saturated steam, assuming no loss in pressure, the amount of heat radiated usually represents an equivalent condensation. Where such a pipe carries superheated steam, again assuming no loss in pressure, the amount of radiation represents only a decrease in the amount of superheat, for condensation cannot take place until the temperature of the steam is lowered to that of the saturated steam at the existing pressure and the temperature of the walls is higher than the temperature of saturated steam where the pipes are well covered and the steam is superheated an ordinary amount. Obviously, therefore, where the degree of superheat is sufficiently high at the boiler, an amount of heat could be radiated far in excess of what is found in well-covered steam lines, and the steam as delivered to the prime movers and auxiliaries would still be dry or superheated.

The loss through drips resulting from such line condensation in the average plant using saturated steam is one which is ordinarily largely underestimated. Such a loss, which is frequently in excess of 5 per cent, can be greatly reduced, if not entirely eliminated, where superheated steam is used.

It is in the prime movers that the advantages of the use of superheated steam are most clearly seen.

In a reciprocating engine, steam is admitted into a space that has been cooled by the steam exhausted during the previous stroke. The heat necessary to warm the cylinder walls from the temperature to which they have been reduced by the exhaust can be supplied, in the absence of jackets, only by the entering steam, and even where jackets are used a large amount of heat must be supplied in this way. If this steam is saturated, such an adding of heat to the walls at the expense of the heat of the entering steam results in the condensation of a portion. This initial condensation is seldom less than from 20 to 30 per cent of the total weight of steam entering the cylinder. It is obvious that if the steam entering be superheated, it must be reduced to the temperature of saturated steam at a corresponding pressure before any condensation can take place. If the steam be superheated sufficiently to allow a reduction in temperature equivalent to the quantity of heat that must be imparted to the cylinder walls and still remain superheated, it is clear that initial condensation is avoided. In the case of a simple engine, where the range of temperature change is a maximum, the degree of superheat necessary to offset a cylinder condensation of, say, 20 per cent in the case of saturated steam, would be excessive, notwithstanding the lower conductivity of superheated steam. As cylinders are added, however, the range of temperature change between the entering steam and the cylinder walls is decreased and proportionately the degree of superheat necessary to prevent initial condensation.

With saturated steam, the heat utilized in warming the cylinder walls to the temperature of the entering steam is mainly lost, in so far as its ability to perform

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work in the cylinder is concerned. It is true that as expansion progresses a portion of the steam that has been so condensed will re-evaporate, though the greater portion of such re-evaporation takes place at the point of exhaust. The latent heat of the water given up in re-evaporation is utilized in changing the condition of the working fluid and is not available for work. Furthermore, a portion of the saturated steam condenses during adiabatic expansion, this condensation increasing as expansion progresses. In high-speed engines using saturated steam the condensation due to adiabatic expansion is just about offset by the re-evaporation of the initial condensation. Since superheated steam cannot condense until its temperature has been reduced to that of saturated steam at a corresponding pressure, not only is initial condensation prevented by its use but also such condensation as would occur during expansion. When superheated sufficiently, the steam delivered by the exhaust will be dry. The number of heat units lost in overcoming condensation effects will be the same whether superheated or saturated steam is the working fluid, but in the case of saturated steam the water of condensation has no power to do work, while superheated steam, even after having given up the equivalent number of heat units to correspond to the condensation in the case of saturated steam, will still have the power of expansion and the ability to do work.

With superheated steam, therefore, a larger proportion of the heat is utilized in the developing of power than with saturated steam, where a large amount is lost in changing the condition of the working fluid. This results in a lower heat consumption in an engine using superheated steam, that is, the expenditure of a less number of heat units in the developing of one indicated horse-power. The "heat consumption" furnishes the true basis for the comparison of efficiencies of different engines, just as a comparison of boiler results is based on an evaporation from and at 212 degrees.

The water consumption of an engine in pounds per indicated horse-power is in no sense a true indication of its efficiency. The initial pressures and corresponding temperatures in two different cases may vary widely and thus through the resulting difference in the temperature of the exhaust affect the temperature of condensed steam returned to the boiler.

The lower the heat consumption of an engine per indicated horse-power, the higher its economy. Since the use of superheated steam decreases this heat consumption, as has been shown, the number of heat units to be imparted in generating steam is reduced, this in turn leading to the lowering of the amount of fuel which must be burned.

No accurate statement can be made as to the saving possible through the use of superheated steam with reciprocating engines. In highly economical plants, where the water consumption per indicated horse-power is low, the gain would be less than would result from the use of superheated steam in less economical plants where the water consumption is higher. Broadly speaking, it may vary from 3 to 5 per cent for 100 degrees of superheat in large and economical plants using engines in which there is a high ratio of expansion, to from 10 to 25 per cent for 100 degrees of superheat where less economical steam engines are in service.

Experience has unquestionably shown that the use of superheated steam with turbines leads to an appreciable gain in economy.

If the saturated steam entering a turbine contains moisture, the effect of such moisture is an appreciable lowering of the turbine's economy. It is stated on good authority that one per cent of moisture contained in the steam will reduce the economy approximately 2 per cent.

The water rate of a large economical steam turbine with superheated steam is reduced about one per cent for every 12 degrees of superheat up to 200 degrees of superheat. To superheat one pound of steam 12 degrees requires about 7 B.t.u. and if 1050 B.t.u. are required at the boiler to evaporate one pound of water into saturated steam from the temperature of the feed water, the heat required for the superheated steam will be 1057 B.t.u. One per cent of saving, therefore, in the water consumption corresponds to a net saving of about one-third of one per cent in the coal consumption. On this basis 100 degrees of superheat with an economical steam turbine will result in somewhat over 3 per cent of saving in the coal for equal boiler efficiencies. As a boiler with a properly designed superheater placed within the setting is more economical for a given capacity than a boiler without a superheater, the minimum gain in the coal consumption will be, say, 4 or 5 per cent as compared to a plant with the same boilers without superheaters.

The above estimates are on the basis of a thoroughly dry saturated steam or steam just at the point of being superheated or containing a few degrees of superheat. If the saturated steam is moist, the saving due to superheat is more, and ordinarily the gain in economy due to superheated steam for equal boiler efficiencies, as compared with commercially dry steam, is, say, 5 per cent for each 100 degrees of superheat.

For any type of steam motor it may be broadly stated that the lower its economy, the greater the saving that will be effected through the use of superheat.

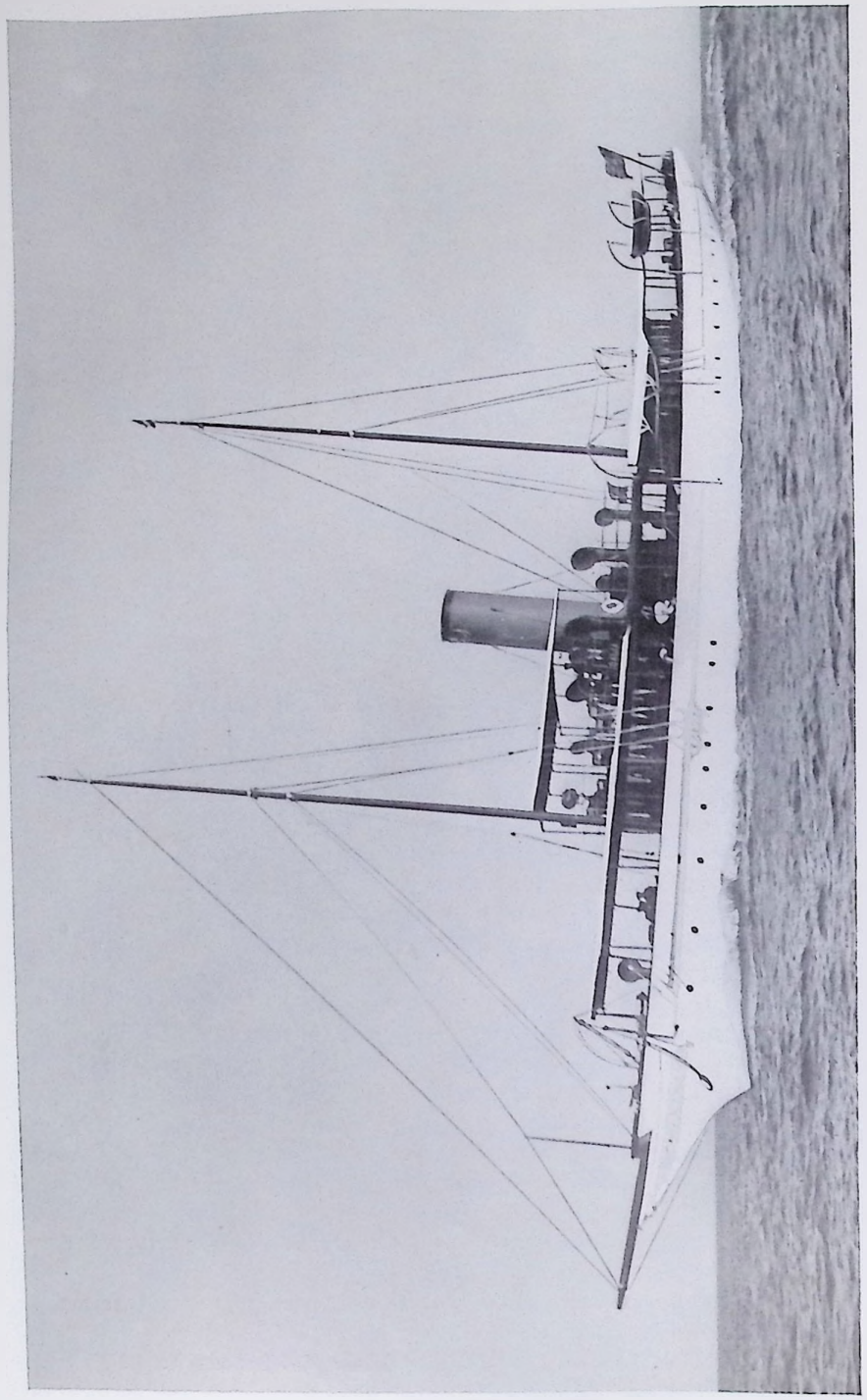
The figures that have been given refer to the possible saving through the use of superheated steam in the prime movers alone. Where the auxiliaries are of a design that can properly handle superheated steam there will be a larger percentage of gain than in the prime movers. This is due to the fact that it is the auxiliaries in a plant that ordinarily show the lowest economy and the lower the efficiency of the steam apparatus, the greater the saving possible.

An example from actual practice will perhaps best illustrate and emphasize the foregoing facts. In October, 1909, a series of comparable tests was conducted by The Babcock & Wilcox Company on the steam yacht "Idalia" to determine the steam consumption both with saturated and superheated steam of the main engine on that yacht, including as well the feed pump, circulating pump and air pump. These tests are more representative than are most tests of like character in that the saving in the steam consumption of the auxiliaries, which were much more wasteful than the main engine, formed an important factor. A résumé of these tests was published in the Journal of the Society of Naval Engineers, November, 1909.

The main engine of the "Idalia" is four cylinder, triple expansion, $11\frac{1}{2} \times 19 \times (2)\ 22\frac{11}{16} \times 18$ inches stroke. Steam is supplied by a Babcock & Wilcox marine boiler having 2500 square feet of boiler-heating surface, 340 square feet of superheating surface and 65 square feet of grate surface.

The auxiliaries consist of a feed pump $6 \times 4 \times 6$ inches, an independent air pump $6 \times 12 \times 8$ inches, and a centrifugal pump driven by a reciprocating engine

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STEAM YACHT "IDALIA," 1,000 INDICATED HORSE-POWER, BABCOCK & WILCOX BOILER, SUPERHEATER AND OIL BURNERS

$5\frac{7}{16} \times 5$ inches. Under ordinary operating conditions the superheat existing is about 100 degrees Fahrenheit.

Tests were made with various degrees of superheat, the amount being varied by by-passing the gases, and in the tests with the lower amounts of superheat by passing a portion of the steam from the boiler to the steam main without passing it through the superheater. Steam temperature readings were taken at the engine throttle. In the tests with saturated steam, the superheater was completely cut out of the system. Careful calorimeter measurements were taken, showing that the saturated steam delivered to the superheater was dry.

The weight of steam used was determined from the weight of the condensed steam discharged from the surface condenser, the water being pumped from the hot well into a tank mounted on platform scales. The same indicators, thermometers and gauges were used in all the tests, so that the results are directly comparable. The indicators used were of the outside spring type, so that there was no effect of the temperature of the steam. All tests were of sufficient duration to show a uniformity of results by hours. A summary of the results secured is given in Table 17, which shows the water rate per indicated horse-power and the heat consumption. The latter figures are computed on the basis of the heat imparted to the steam above the actual temperature of the feed water, and, as stated, these are the figures that are directly comparable.

The table shows that the saving in steam consumption with 105 degrees of superheat was 15.3 per cent and in heat consumption about 10 per cent. This may be safely stated to be a conservative representation of the saving that may be accomplished by the use of superheated steam in a plant as a whole, where superheated

TABLE 17

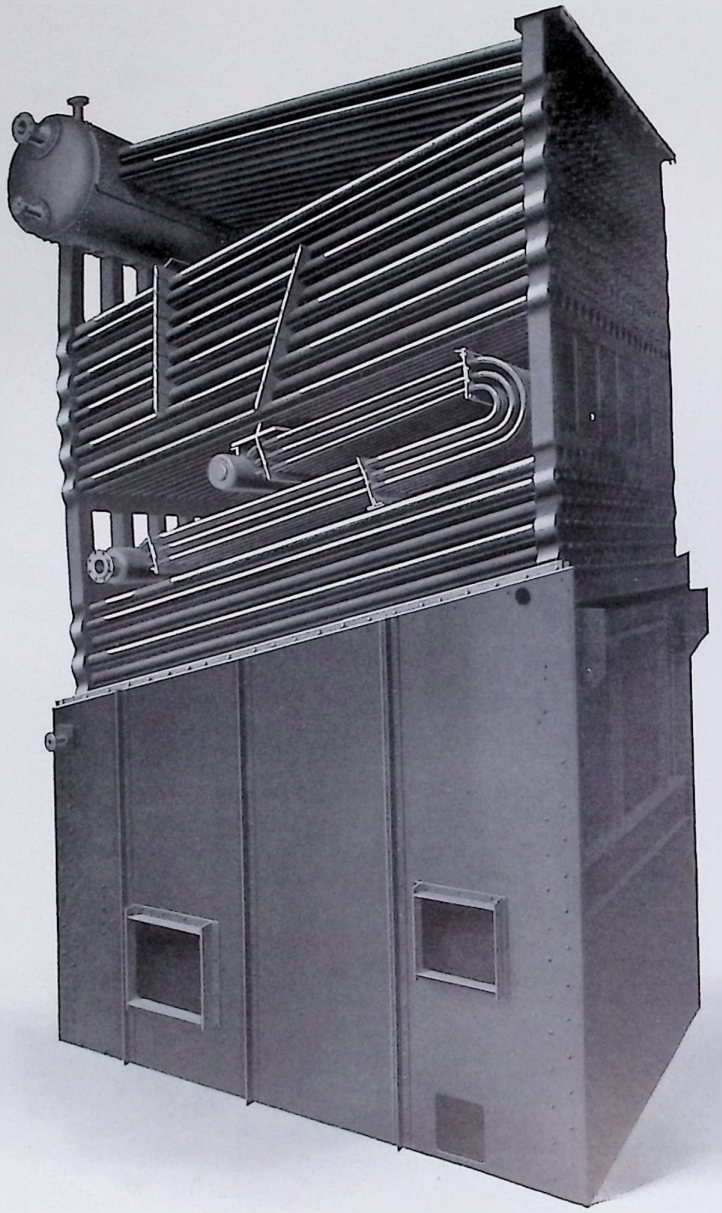
RESULTS OF "IDALIA" TESTS

Date	1909	Oct. 11	Oct. 14	Oct. 14	Oct. 12	Oct. 13
Degrees of Superheat, Fahrenheit		0	57	88	96	105
Pressures, pounds per square inch above Atmospheric Pressure	Throttle	190	196	201	198	203
	First Receiver	68.4	66.0	64.3	61.9	63.0
	Second Receiver	9.7	9.2	8.7	7.8	8.4
Vacuum, inches		25.5	25.9	25.9	25.4	25.2
Temperature, Degrees Fahrenheit	Feed	201	206	205	202	200
	Hot Well	116	109.5	115	111.5	111
Revolutions per minute	Air Pump	57	56	53	54	45
	Circulating Pump	196	198	196	198	197
	Main Engine	194.3	191.5	195.1	191.5	193.1
Indicated Horse-Power, Main Engine		512.3	495.2	521.1	498.3	502.2
Water per hour, total pounds		9397	8430	8234	7902	7790
Water per indicated Horse-Power, pounds		18.3	17.0	15.8	15.8	15.5
B.t.u. per minute per indicated Horse-Power		314	300	284	286	283
Percentage Saving of Steam		. .	7.1	13.7	13.7	15.3
Percentage Saving of Fuel (computed)		. .	4.4	9.5	8.9	9.9

steam is furnished not only to the main engine but also to the auxiliaries. The figures may be taken as conservative for the reason that in addition to the saving shown in

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the table, there would be, in the larger installations where the steam lines would be longer, a saving much greater than generally realized in the drips, where the loss with saturated steam is greatly in excess of that with superheated steam.



One of a Battery of Two Babcock & Wilcox Boilers, with Interdeck Superheaters and Air Heaters, Installed on the Steamer "Carl D. Bradley" of the Bradley Transportation Company, Rogers City, Mich. Furnaces Arranged for Mechanical Stokers. View Shows Boiler Partly Erected

THE PROPERTIES OF AIR AND GASES

ATMOSPHERIC air is a mechanical mixture and not a chemical compound. Its constituents are oxygen, nitrogen, and small amounts of carbon dioxide, water vapor, argon and other inert gases. In combustion work, the gases other than carbon dioxide are included in the nitrogen content, and while there is some slight difference in the proportion of oxygen and nitrogen as given by different authorities, the generally accepted values are:

By volume: oxygen 20.91 per cent, nitrogen 79.09 per cent.

By weight: oxygen 23.15 per cent, nitrogen 76.85 per cent.

All gases have a characteristic equation based upon that of a perfect gas, from which, with three values known, the fourth may be computed. This equation is:

$$PV = RT \qquad (10)$$

Where P = absolute pressure in pounds per square foot,
V = volume per pound in cubic feet,
T = absolute temperature, degrees Fahrenheit,
R = a constant, varying with the gas and derived from the relations existing between the volume, pressure and temperature of the gas in question.

Air is ordinarily considered the standard to which other gases are referred, and for air the value of R is 53.33. The values of R for gases other than air are given in Table 19.

The density of a gas other than air is the weight of unit volume of the gas divided by the weight of an equal volume of pure dry air. The weight per cubic

TABLE 18

VOLUME AND WEIGHT OF AIR

AT ATMOSPHERIC PRESSURE

Temperature Degrees Fahrenheit	Volume One Pound Cubic Foot	Weight per Cubic Foot Pounds	Temperature Degrees Fahrenheit	Volume One Pound Cubic Foot	Weight per Cubic Foot Pounds	Temperature Degrees Fahrenheit	Volume One Pound Cubic Foot	Weight per Cubic Foot Pounds
32	12.390	.080710	160	15.615	.064041	340	20.151	.049625
50	12.843	.077863	170	15.867	.063024	360	20.655	.048414
55	12.969	.077107	180	16.119	.062039	380	21.159	.047261
60	13.095	.076365	190	16.371	.061084	400	21.663	.046162
65	13.221	.075637	200	16.623	.060158	425	22.293	.044857
70	13.347	.074923	210	16.875	.059259	450	22.923	.043624
75	13.473	.074223	212	16.925	.059084	475	23.554	.042456
80	13.599	.073535	220	17.127	.058388	500	24.184	.041350
85	13.725	.072860	230	17.379	.057541	525	24.814	.040300
90	13.851	.072197	240	17.631	.056718	550	25.444	.039302
95	13.977	.071546	250	17.883	.055919	575	26.074	.038352
100	14.103	.070907	260	18.135	.055142	600	26.704	.037448
110	14.355	.069662	270	18.387	.054386	650	27.964	035760
120	14.607	.068460	280	18.639	.053651	700	29.224	.034219
130	14.859	.067299	290	18.891	.052935	750	30.484	.032804
140	15.111	.066177	300	19.143	.052238	800	31.744	.031502
150	15.363	.065092	320	19.647	.050898	850	33.004	.030299

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Steamer "Princess Margaret," 14,000 Shaft Horse-power, Babcock & Wilcox Boilers
Built for the Canadian Pacific Steamship Company; Owners, British Admiralty

foot of a gas is thus the weight per cubic foot of air times the density, air and gas being under the same conditions of pressure and temperature. The volume of a gas is the reciprocal of its weight. It is perhaps more convenient to compute weight and volumetric data of gases from their relative densities and a table of weights and volumes of air than from the characteristic equation.

Since the volume, and hence the weight, of a gas is a function of both temperature and pressure, it is necessary, in order that there may be a suitable basis for comparison, that all volumes be referred to some standard set of conditions. Such conditions, for gases now ordinarily accepted as standard, are a pressure of 14.6963 pounds per square inch (2116.27 pounds per square foot), and a temperature of 32 degrees Fahrenheit.

Table 18 gives the volume and weight of air at atmospheric pressure and varying temperatures.

Table 19 gives the relative densities of the gases other than air encountered in combustion work, together with the weights and volumes under standard conditions. With these values and those of Table 18, the volume and weight of these gases at temperatures other than 32 degrees may be readily computed.

TABLE 19

DENSITY, WEIGHT AND VOLUME OF GASES

AT ATMOSPHERIC PRESSURE AND 32 DEGREES FAHRENHEIT

Substance	Molecular Symbol	Relative Density		Weight per Cubic Foot Pounds	Cubic Feet per Pound Cubic Feet	Value of Constant R in PV=RT
		Air=1	Hydrogen=1*			
Air	....	1.0000	....	.08071	12.390	53.33
Oxygen	O_2	1.1053	16	.08921	11.209	48.24
Hydrogen	H_2	0.0696	1	.00562	177.936	765.8
Nitrogen	N_2	0.9673	14	.07807	12.809	55.13
Carbon Monoxide	CO	0.9672	14	.07806	12.811	55.15
Carbon Dioxide	CO_2	1.5291	22	.12341	8.103	34.88
Methane	CH_4	0.5576	8	.04500	22.222	95.64
Acetylene	C_2H_2	0.9200	13	.07425	13.468	57.97
Ethylene	C_2H_4	0.9674	14	.07808	12.807	55.12
Ethane	C_2H_6	1.0494	15	.08470	11.806	50.81
Sulphur Dioxide	SO_2	2.2639	32	.18272	5.473	23.56
Carbon † ‡	C_2	....	....	145	.0069	....
Sulphur	....	....	....	125	.0080	....

* Based on approximate molecular weights.

† Solid.

‡ If carbon can be conceived to exist as a gas under standard conditions its relative density would be 0.820, its weight per cubic foot .0668 pound, and its volume 14.97 cubic feet per pound.

The specific heat of air at constant pressure varies with its temperature. A number of determinations of this value have been made, certain of those accepted as most authentic being given in Table 20.

Because of the discrepancy in the values as given by different authorities, it is customary to use a constant value of 0.24 for the specific heat of air at constant pressure regardless of temperature. This value finds its most common use in combustion work in the computation of the amount of heat lost in the dry chimney

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gases* leaving a boiler or economizer. Since such loss is based on dry gases, the error arising from the arbitrarily assumed constant value of the specific heat of air, particularly at moderate temperatures, is probably within the limits of error of boiler testing as a whole. On the other hand, in problems involving the total weight of gas, including moisture, the variation in specific heat with the composition of the gas as well as with its temperature may be appreciable, and where accurate results are desired the specific heat must be computed for each fuel and set of combustion

TABLE 20

SPECIFIC HEAT OF AIR AT CONSTANT PRESSURE AND VARIOUS TEMPERATURES

Temperature Range		Specific Heat	Authority
Degrees Centigrade	Degrees Fahrenheit		
−30– 10	−22– 50	0.2377	Regnault
0–100	32– 212	0.2374	Regnault
0–200	32– 392	0.2375	Regnault
20–440	68– 824	0.2366	Holborn and Curtis
20–630	68–1166	0.2429	Holborn and Curtis
20–800	68–1472	0.2430	Holborn and Curtis
0–200	32– 392	0.2389	Wiedemann

conditions. Problems in which the variation in specific heat with the composition and temperature of the gas are of primary importance include those of theoretical furnace temperature, superheater, economizer and air heater performance.

Over a given temperature range, the specific heat of any gas is primarily affected by the moisture and hydrogen content of the fuel burned to produce the gas. These constituents appear in the products of combustion as superheated steam, the specific heat of which is high.

The general formula for the specific heat of any gas at constant pressure may be expressed:

$$C_p = A + Bt + Ct^2 + Dt^3 \qquad (11)$$

The mean specific heat over a temperature range t_1— t_2 is then

$$C_p = \int_{t_1}^{t_2} \frac{A + Bt + Ct^2 + Dt^3}{(t_2 - t_1)} dt \qquad (12)$$

or by integration

$$C_{p_{t_1-t_2}} = A + \frac{B}{2}(t_2+t_1) + \frac{C}{3}\left[(t_2+t_1)^2 - t_2 t_1\right] + \frac{D}{4}(t_2+t_1)\left[(t_2+t_1)^2 - 2 t_2 t_1\right] \qquad (13)$$

Where the lower limit of the temperature range, t_1, is zero, the mean specific heat between zero and t_2, from formula *(13)*, becomes

$$C_{p_{0-t_2}} = A + \frac{B}{2} t_2 + \frac{C}{3} t_2^2 + \frac{D}{4} t_2^3 \qquad (14)$$

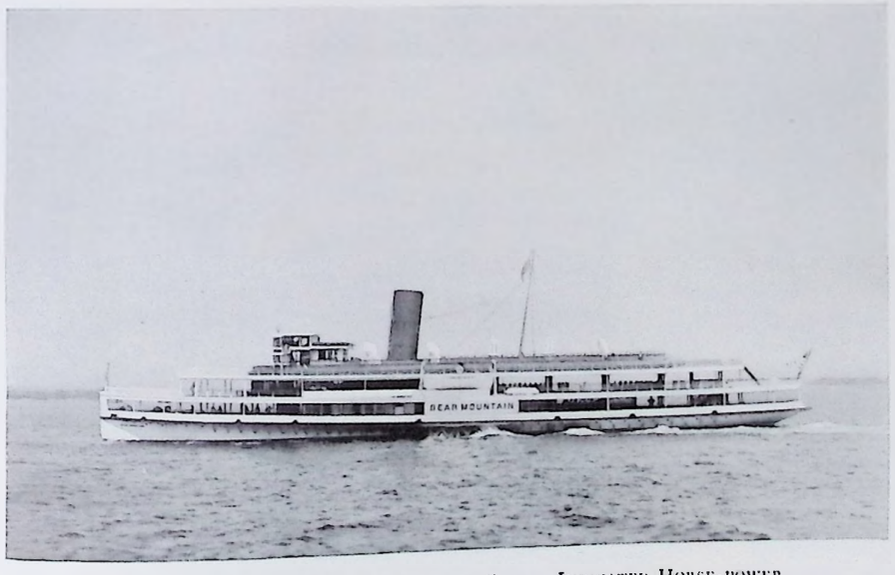
* See page 137

Distinction was made in the chapter on "Heat and Its Measurement" between specific heat at constant volume and specific heat at constant pressure. Except in the case of water vapor, the variation with pressure in the specific heat of the gases ordinarily encountered in combustion work is negligible. In the case of water vapor, where it is necessary to deal with any considerable range of pressures, this variation would be an appreciable factor, but in the commercial gases involved in combustion, the partial pressure exerted by water vapor, either in gases before combustion or in the exhaust gases, is rarely over one pound absolute. With such a limited pressure range and in view of the fact that the water vapor content of the gases is small, the effect of such variation in pressure on the specific heat of the gas as a whole may be neglected.

The range of pressure in the gases encountered in boiler work is so limited, varying from that at which the commercial gases are introduced into the furnace for combustion to the suction under which they are drawn over the boiler heating surfaces, that in the computation of combustion data the gases may be safely assumed to be at a constant pressure. The specific heat at constant pressure is the specific heat which should be used, and any results based on the assumption of a constant pressure of the gases as a whole, and in which the variation in the specific heat of the water vapor content with change of pressure is neglected, will be well within the limits of accuracy of practically all combustion data computation.

The values of the constants A, B and C of formula *(11)* for the instantaneous specific heat of the gases commonly encountered in combustion work are given in Table 21. The values of $B/2$, $C/3$ and $D/4$ for use in the integrated formula for determining the mean specific heat, may readily be computed from the values of this table.

The values of the constants for carbon dioxide below 2200 degrees are those of Holborn and Henning, while above this temperature a modification of their formula



EXCURSION STEAMER "BEAR MOUNTAIN," 2200 INDICATED HORSE-POWER
BABCOCK & WILCOX BOILERS AND OIL BURNERS
OWNERS, MCALLISTER STEAMBOAT COMPANY, NEW YORK, N. Y.

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is used. The values for carbon monoxide, nitrogen and hydrogen are also from their investigations, while those for oxygen are a modification of the values as determined by Langen and Pier to make them come into agreement with other determinations. The values for water vapor are based on Marks and Davis' steam tables. The specific heat of methane and ethylene has not been as definitely determined as that of the other gases, but the values of the constants in the formula for these gases as given in Table 21 are those ordinarily accepted as being authentic.

TABLE 21

SPECIFIC HEAT CONSTANTS

Gas	A	B	C	D
Oxygen	0.2154	.000019		
Nitrogen	0.2343	.000021		
Carbon Monoxide	0.2343	.000021		
Carbon Dioxide	0.1983	835×10^{-7}	-16.7×10^{-9}	
Carbon Dioxide*	0.1991	873×10^{-7}	$-23\,4 \times 10^{-9}$	0.22×10^{-11}
Water Vapor	0.4541	32×10^{-7}	2825×10^{-11}	
Methane	0.481	.00056		
Ethylene	0.335	.00042		

* For temperatures above 2200 degrees Fahrenheit.

The mean specific heat at constant pressure and at ordinary temperatures of the commoner gases is given in Table 22.

TABLE 22

MEAN SPECIFIC HEATS AT CONSTANT PRESSURE AND ORDINARY TEMPERATURES

	Molecular Symbol	Mean Specific Heat	
		0—60 Degrees Fahrenheit	0—600 Degrees Fahrenheit
Air		.2381	.2484
Oxygen	O_2	.2160	.2211
Hydrogen	H_2	3.3850	3.4750
Nitrogen	N_2	.2349	.2406
Carbon Monoxide	CO	.2349	.2406
Carbon Dioxide	CO_2	.2008	.2214
Water Vapor	H_2O	.4542	.4586
Methane*	CH_4	.498	.649
Ethylene†	C_2H_4	.348	.461
Sulphur Dioxide	SO_2	.1544	

* Methane—$Cp_{0-t} = 0.481 + 0.00028\ t$.

† Ethylene—$Cp_{0-t} = 0.335 + 0.00021\ t$.

The specific heat of a gaseous mixture is found by multiplying the percentages by weight of the constituent gases by the specific heat of the individual constituents and dividing the sum of the products by 100. To illustrate, assume oil so burned as to give a weight of products of combustion per pound of oil, and the corresponding percentages by weight, as follows:

	Weight of Products per Pound of Oil Burned Pounds	*Percentage by Weight of Products*
CO_2	3.088	18.239
O_2	0.357	2.108
N_2	12.343	72.902
H_2O	1.143	6.751

Assume that this gas is cooled in passing through an economizer or an air heater from 600 to 300 degrees Fahrenheit. The mean specific heat over this temperature range is computed in the following manner:

CO_2 $C_{300\text{-}600} = 0.1983 + 417.5 \times 10^{-7} (600 + 300)$
$\quad - 5.567 \times 10^{-9} [(600 + 300)^2 - (600 \times 300)]$

O_2 $C_{300\text{-}600} = 0.2154 + 0.0000095 (600 + 300)$

N_2 $C_{300\text{-}600} = 0.2343 + 0.0000105 (600 + 300)$

H_2O $C_{300\text{-}600} = 0.4541 + 16 \times 10^{-7} (600 + 300)$
$\quad + 942 \times 10^{-11} [(600 + 300)^2 - (600 \times 300)]$

or over the temperature range assumed the specific heats of the various constituents are:

$CO_2 = 0.2324$
$O_2 = 0.2240$
$N_2 = 0.2438$
$H_2O = 0.4614$

The specific heat of the gas as a whole is then:

CO_2	$18.239 \times 0.2324 =$	4.239
O_2	$2.108 \times 0.2240 =$	0.472
N_2	$72.902 \times 0.2438 =$	17.774
H_2O	$6.751 \times 0.4614 =$	3.114
		25.599

and the mean specific heat is:

$25.599 \div 100 = 0.2560$

In the fuel chosen for the example, oil was purposely assumed as being high in hydrogen content, to indicate that in certain classes of computations, even where the temperatures are low, the assumption of a specific heat of flue gases of 0.24 may lead to an error of at least 6.5 per cent.

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Steamer "Iroquois," 8500 Shaft Horse-power, Babcock & Wilcox Boilers, Superheaters and Oil Burners
Owners, Clyde Steamship Company (Clyde Line), New York, N. Y.

COMBUSTION

COMBUSTION may be defined as the phenomenon resulting from any reaction evolving heat. Oxygen is the sole supporter of combustion, and a combustible may therefore be defined as a substance capable of combining with oxygen to produce heat. The speed of combustion, which is primarily dependent upon the affinity of a combustible element for oxygen, covers a range from very slow, as in the case of rust formation, to instantaneous, as in the explosion of confined powder.

As applied to heat production for steam-generating purposes, combustion may be defined as the rapid combination of the combustible elements of a fuel with oxygen, while in this sense the term combustible implies the capacity of an element for rapid combination with oxygen for the evolving of heat.

The elements and compounds encountered in commercial combustion work, together with their molecular symbols and weights, are given in Table 23.

TABLE 23.

ELEMENTS AND COMPOUNDS ENCOUNTERED IN COMBUSTION

Substance	Molecular Symbol	Atomic Weight		Molecular Weight		Form
		Accurate	Approximate	Accurate	Approximate	
Carbon	C*	12.005	12	†	. . .	Solid
Hydrogen	H_2	1.008	1	2.015	2	Gas
Oxygen	O_2	16.00	16	32.00	32	Gas
Sulphur	S_2	32.07	32	64.14	64	Solid
Nitrogen‡	N_2	14.01	14	28.02	28	Gas
Carbon Monoxide	CO	. . .	. . .	28.01	28	Gas
Carbon Dioxide	CO_2	. . .	. . .	44.01	44	Gas
Methane	CH_4	. . .	. . .	16.03	16	Gas
Acetylene	C_2H_2	. . .	. . .	26.03	26	Gas
Ethylene	C_2H_4	. . .	. . .	28.03	28	Gas
Ethane	C_2H_6	. . .	. . .	30.05	30	Gas
Sulphur Dioxide	SO_2	. . .	. . .	64.07	64	Gas
Hydrogen Sulphide	H_2S	. . .	. . .	34.08	34	Gas
Water Vapor	H_2O	. . .	. . .	18.02	18	Vapor
Air	. . .	. . .	. . .	28.94	29	Gas

*Atomic symbol.

†The molecular weight of C has not been definitely determined. Carbon exists in a number of forms each of which probably has its own molecular weight. The latest investigations indicate that a molecule of carbon in any form consists of at least 12 atoms.

‡Atmospheric nitrogen as distinguished from chemically pure nitrogen, which has an atomic weight slightly less than 14.01.

The chemical reaction of combustion, as in the case of any chemical reaction, results from a rearrangement of the atoms of the constituent elements into a new combination of molecules. Such reactions occur in accordance with fixed and invariable weight relations, which are characteristic of the elements involved, and definite volumetric changes based upon the number of gaseous molecules reacting and produced. The chemical reactions and resulting combinations of combustion are given in Table 24.

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For the reactions to be as indicated in this table, it is necessary that the amount of oxygen supplied for combustion be the exact amount required and be wholly utilized. Such a condition necessitates a distinction between perfect and complete combustion. Combustion may be complete when an excess of oxygen over that actually required is supplied, such excess appearing as unused oxygen in the products of combustion. Combustion is only perfect when complete without the presence of oxygen in the products resulting from such combustion. An example of combustion that is neither complete nor perfect is that of carbon to carbon monoxide. Since carbon monoxide is capable of a further combination with oxygen to form carbon dioxide, combustion is obviously incomplete. Combustion that is complete but not perfect is discussed hereafter under the heading of excess air.

TABLE 24

CHEMICAL REACTIONS OF COMBUSTION

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

IGNITION TEMPERATURE—While, as stated, the speed of combustion is primarily dependent upon the affinity of the combustible for combination with oxygen, it is also dependent upon the conditions under which combustion takes place, and the chief of such conditions is that of temperature. Introducing a combustible into the presence of oxygen does not of necessity result in combustion.

TABLE 25

IGNITION TEMPERATURES

Combustible Substance	Molecular Symbol	Ignition Temperature Degrees Fahrenheit
Sulphur	S_2	470
Fixed Carbon—Bituminous Coal	. . .	766
Fixed Carbon—Semi-bituminous Coal	. . .	870
Fixed Carbon—Anthracite Coal	. . .	925
Acetylene	C_2H_2	900
Ethane	C_2H_6	1000
Ethylene	C_2H_4	1022
Hydrogen	H_2	1130
Methane	CH_4	1202
Carbon Monoxide	CO	1210

Every combustible substance has a temperature called its "ignition temperature," to which it must be brought before it will unite with oxygen in chemical combination, and below which such combination will not take place. This ignition temperature must exist with oxygen present or no combustion will occur. The ignition temperature of various fuels and of the combustible constituents of such fuels is given in Table 25.

The temperature of ignition of the gases of a coal varies and is higher than the ignition temperature of the fixed carbon constituent. The ignition temperature of coal is that of its fixed carbon content, the gaseous constituents being ordinarily distilled off, though not ignited, before this temperature is reached.

When combustion has started, the heat evolved in the oxidization of the combustible matter will, under proper conditions, maintain sufficiently high temperatures for a continuity of ignition.

HEAT OF COMBUSTION—Elements entering into a direct combination to form a compound either evolve or absorb a definite amount of heat, called the "heat of combination," which, from its definition, may be positive or negative. Since the term combustion, as applied to boiler practice, is limited to the rapid chemical combination of the combustible elements of a fuel with oxygen, with a resulting production of heat, the heat of combustion of a fuel is the heat evolved in the complete oxidization of the combustible elements through their union with oxygen. Thus, the heat of combustion of a fuel is the heat of combination of a specific set of elements and compounds, the combination of which with oxygen always results in the production of heat. It follows that the heat of combination of a compound which results from the union of a single combustible element with oxygen to produce heat is the same as the heat of combustion of that element.

In the commercial evolving of heat for steam generation, the "second law" of Berthelot is of importance. This law may be summarized as follows:

The heat energy evolved in any chemical change in the boiler furnace, where no mechanical work is done, *i. e.*, evolved through the union of combustible elements with oxygen, is dependent upon the final products of combustion and in no way upon any intermediate combination or combinations that may have occurred in reaching the final result.

The application of this law may be readily shown by example:

A coal fire from which all of the volatile constituents have been driven and which consists of incandescent coke may, for the present purpose, be considered as consisting entirely of carbon. If air is introduced under the fire, the oxygen immediately breaks its mechanical union with nitrogen and enters into chemical combination with carbon to form carbon dioxide ($C + 2O = CO_2$). Each unit of carbon has combined with the maximum amount of oxygen with which it can exist as a compound. The oxygen, on the other hand, is capable of uniting with additional carbon and as the unit of carbon dioxide passes upward through the fuel bed under the influence of draft it encounters other free carbon with which it unites to form carbon monoxide ($CO_2 + C = 2CO$), thus "satisfying" the affinity of oxygen for carbon. If no additional oxygen is encountered in the further passage through the fuel bed, these particular molecules, as representative of the products of combustion, will issue from the fuel bed as carbon monoxide. If no additional oxygen is encountered in the furnace, the total heat available for later absorption by the boiler is that due to the combustion of carbon to carbon monoxide, regardless of the fact that at one stage of the process the carbon had been completely oxidized and carbon dioxide had been produced. If, on the other hand, additional oxygen is encountered in the furnace, the temperature is above the ignition point of carbon monoxide, and this temperature is maintained a sufficient length of time for further combustion, *i. e.*, if the gases are not cooled below the ignition temperature by the boiler-heating surface before further combustion can be completed, the carbon of the carbon monoxide will unite with additional oxygen to form carbon dioxide ($2CO + 2O = 2CO_2$). The total heat evolved and available for absorption in such cases will be that due to the burning of carbon to carbon dioxide regardless of the two intermediate steps.

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STEAMER "VOLENDAM," 8000 SHAFT HORSE-POWER, BABCOCK & WILCOX BOILERS; OWNERS, HOLLAND-AMERICA LINE

The methods of determining the heat of combustion of solid and liquid fuels are discussed in a later chapter. The heat of combustion of elements and combustible compounds entering into gaseous fuels has been determined by so many authorities that definite values may be accepted as correct without determination. Such values are given in Table 26.

TABLE 26*

HEAT OF COMBUSTION

BY CALORIMETRIC DETERMINATION

Combustible	Molecular Symbol	Heat Value—B.t.u. per Pound		Per Cubic Foot‡
		Higher	Lower or Net†	Higher
Hydrogen . . .	H_2	62000	52920	348
Carbon (to CO) .	C	4380	. . .	. . .
Carbon (to CO_2).	C	14540	. . .	. . .
Carbon Monoxide	CO	4380	. . .	342
Carbon in CO§ .	C	10160	. . .	. . .
Methane . . .	CH_4	23850	21670	1073
Acetylene . . .	C_2H_2	21460	21020	1590
Ethylene . . .	C_2H_4	21450	20420	1675
Ethane	C_2H_6	22230	20500	1883
Sulphur (to SO_2) .	S_2	4050	. . .	. . .
Sulphur (to SO_3) .	S_2	5940	. . .	. . .

† There is a considerable discrepancy between lower heat values as given by different authorities, the variation being due to methods of computation and assumptions. (See text.) The values given are those of G. A. Goodenough.

‡ At 32 degrees Fahrenheit and atmospheric pressure.

§ Per pound of carbon in carbon monoxide, *i. e.*, 2.33 pounds of CO.

* Heating Value by Calorimetry, see discussion, page 161.

While this table gives two heat values, the higher and net, the former is that which should be used. The difference is discussed hereafter in connection with the determination of the heating values of fuels.*

AIR AND COMBUSTION—In the foregoing more or less abstract consideration of combustion, it has been assumed that there existed a supply of oxygen sufficient for combination with the combustible elements of a fuel and a temperature sufficient to bring about the chemical combinations of combustion. In practice, it is the physical introduction of oxygen to the combustible elements or compounds of a fuel, in such manner as to assure complete oxidization and at the same time to assure the utilization of all or the maximum proportion so supplied, that presents the most difficult problem in the burning of all fuels.

Oxygen, as stated, is the sole supporter of combustion. The source of supply of oxygen is the air. From the proportionate parts by weight of oxygen and nitrogen as given, namely, oxygen 23.15 per cent and nitrogen 76.85 per cent, in order to supply one pound of oxygen for combustion it is necessary to supply $1 \div 0.2315 = 4.320$ pounds of air, and such weight represents, for each pound of oxygen supplied, 3.320 pounds of nitrogen, which serve no useful function in combustion and is a source of direct loss in combustion work.

From the manner of chemical combinations occurring in the union of oxygen with combustible elements and compounds, as indicated in Table 24, and the atomic weights of the elements involved, the proportionate parts by weight of the elements entering into the resulting compounds and the weights of the products of combustion may be readily computed. With the amount of oxygen required for

* See page 158.

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Steamer "Everett," 1475 Indicated Horse-power, Babcock & Wilcox Boilers and Oil Burners
Owners, McCormick Steamship Company, San Francisco, Cal.

combustion thus determined, the amount of air required is directly indicated by the oxygen-nitrogen ratio of air.

The methods of such computations are clearly indicated by example, and since the relation of the products of combustion to the combustible elements of the fuel is the most important factor in the determination of the efficiency of combustion, it appears advisable to illustrate such computations fully.

Consider first carbon. From Table 24 it was seen that one atom of carbon united with two atoms of oxygen to form carbon dioxide, $C + 2O = CO_2$. From the atomic weights, $12 + (2 \times 16) = 44$, or in the burning of one pound of carbon to carbon dioxide, twelve parts by weight of carbon combine with thirty-two parts by weight of oxygen to form forty-four parts by weight of carbon dioxide. Hence, any weight of carbon dioxide must be composed of 27.27 per cent by weight of carbon and 72.73 per cent by weight of oxygen, or 1 pound $CO_2 = 0.2727$ pound $C + 0.7273$ pound O_2.

Since the ratio of carbon to oxygen in carbon dioxide is 1 to 2.667, it is obvious that in burning one pound of carbon to carbon dioxide, 2.667 pounds of oxygen will be required. If one pound of oxygen is contained in 4.32 pounds of air it will be necessary to supply for the complete combustion of one pound of carbon $2.667 \times 4.32 = 11.52$ pounds of air, and since each pound of oxygen is accompanied by 3.32 pounds of nitrogen, there will pass off with the carbon dioxide $2.667 \times 3.32 = 8.85$ pounds of nitrogen.

In the complete combustion of one pound of carbon, then, the resulting products of combustion will be

1 pound C + 2.667 pounds O_2 = 3.667 pounds CO_2
2.667 × 3.32 pounds N_2 = 8.885 pounds N_2

TABLE 27

COMBUSTION DATA

IN TERMS OF POUNDS PER POUND OF FUEL

	Molecular Symbol	Theoretically Required Pounds		Products of Combustion Pounds				
		O_2	Air	CO_2	H_2O	N_2	CO	SO_2
Carbon (to CO_2) . . .	C	2.667	11.52	3.667	. . .	8.85	. . .	. . .
Carbon (to CO) . . .	C	1.333	5.76	. . .	. . .	4.43	2.333	. . .
Carbon Monoxide . . .	CO	0.572	2.46	1.57	. . .	1.89	. . .	. . .
Sulphur	S	1.000	4.32	. . .	. . .	3.32	. . .	2.00
Hydrogen	H_2	8.000	34.56	. . .	9.00	26.56	. . .	. . .
Methane	CH_4	4.000	17.28	2.75	2.25	13.28	. . .	. . .
Acetylene	C_2H_2	3.077	13.29	3.39	0.69	10.21	. . .	. . .
Ethylene	C_2H_4	3.429	14.81	3.14	1.29	11.38	. . .	. . .
Ethane	C_2H_6	3.733	16.13	2.93	1.80	12.40	. . .	. . .
Hydrogen Sulphide . .	H_2S	1.412	6.10	. . .	0.53	4.69	. . .	1.88

Again, consider hydrogen. Table 24 indicates that two atoms of hydrogen will combine with one atom of oxygen to form water vapor, $2H + O = H_2O$. From the atomic weights, $(2 \times 1) + 16 = 18$, or in the burning of one pound of hydrogen to

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water vapor, one part by weight of hydrogen will combine with eight parts by weight of oxygen to form nine parts by weight of water vapor. Hence in one pound of water vapor we have

$$1 \text{ pound } H_2O = 0.111 \text{ pound } H_2 + 0.889 \text{ pound } O_2$$

Since the ratio of hydrogen to oxygen in water vapor is thus 1 to 8, it will require 8 pounds of oxygen for the complete combustion of one pound of hydrogen, which means, as for the combustion of carbon, $8 \times 4.32 = 26.56$ pounds N_2, and the products of combustion of one pound of hydrogen will be

$$1 \text{ pound } H_2 + 8 \text{ pounds } O_2 = 9 \text{ pounds } H_2O$$
$$8 \times 3.32 \text{ pounds } N_2 = 26.56 \text{ pounds } N_2$$

The methods of computation are simple, but, as stated, are considered at length because of their importance. Table 27 gives the results of such computations in terms of weight for all of the combustible elements and compounds encountered in commercial fuels.

TABLE 28

AIR AND OXYGEN REQUIRED FOR AND PRODUCTS OF COMBUSTION

	Weight per Pound Oil Pound	Required—Pounds		Products of Combustion—Pounds per Pound of Oil				
		O_2	Air	CO_2	O_2	N_2	H_2O	SO_2
C	0.8400	2.240	9.677	3.080	. . .	7.437	. . .	. . .
H_2	0.1270	1.016	4.389	. . .	. . .	3.373	1.143	. . .
O_2	0.0120	. . .	. . .	. . .	0.0120	. . .	. . .	. . .
S	0.0040	0.004	0.017	. . .	. . .	0.013	. . .	0.008
N_2	0.0170	. . .	. . .	. . .	. . .	0.017	. . .	. . .
		3.260	14.083	3.080	0.012	10.840	1.143	0.008
O_2 in Oil		0.012	0.052*	. . .	0.012	0.040*	. . .	. . .
		3.248	14.031	3.080	0.000	10.800	1.143	0.008
SO_2 as CO_2		. . .	. . .	0.008	. . .	. . .	. . .	0.008
		3.248	14.031	3.088	0.000	10.800	1.143	0.000

*Air and N_2 equivalents of O_2 in oil.

The methods of computing the products of perfect combustion may best be illustrated by example. If we assume an oil having an ultimate analysis as follows:

	Per Cent
Carbon	84.00
Hydrogen	12.70
Oxygen	1.20
Sulphur	0.40
Nitrogen	1.70
	100.00

the weight of oxygen and air theoretically required for combustion and the weight of products of combustion may be computed from Table 27 and are as given in Table 28. In this table it is assumed that the oxygen content of the fuel is utilized in

combustion, and that the sulphur content is burned to sulphur dioxide and appears in the analysis of the products of combustion as carbon dioxide. This method of computation of the products of perfect combustion of any fuel is of importance in connection with analyses of such products as are discussed in the next chapter.

Excess Air—While supplying just the proper amount of oxygen to assure complete combustion would appear from the chemical standpoint, as indicated by Table 27, to be a simple matter, it is the difficulty of supplying sufficient oxygen for complete combustion, and the maintaining such supply with its accompanying nitrogen at a point as nearly as possible at the theoretical amount required, that is the primary problem of combustion. In other words, it is the relation of the amount of air actually supplied to that theoretically required for combustion that is the measure of the efficiency of such combustion.

Nitrogen, which almost of necessity must be introduced into the boiler furnace with the oxygen required for the combustion of any fuel, is, as stated, an inert gas which performs no function in combustion. It passes through the furnace and boiler without change, except in temperature and volume, dilutes the air, absorbs heat and reduces the temperature of the products of combustion. For this reason and because of the high proportion of nitrogen to oxygen in air, it is the chief source of heat loss in combustion. Any oxygen supplied to the furnace in excess of that required for combustion results in the same losses as in the case of nitrogen, and further, such excess oxygen is accompanied by additional nitrogen which accentuates the combustion loss. On the other hand, where sufficient oxygen is not supplied for complete combustion the loss due to the presence of the inert nitrogen is inappreciable as compared to that resulting from incomplete combustion. The figures of Table 26, indicating the difference in the amount of heat evolved in the burning of carbon to carbon monoxide and to carbon dioxide, represent the possible combustion losses resulting from incomplete combustion. This factor is discussed hereafter in connection with flue gas analyses and combustion losses.

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Steamer "Robert E. Lee," 4750 Shaft Horse-power, Babcock & Wilcox Boilers, Superheaters and Oil Burners
Owners, Old Dominion Steamship Company, New York, N. Y.

FLUE GASES AND THEIR ANALYSIS

With the data given in the chapter on "Combustion," formulæ for the amount of air theoretically required for the combustion of different fuels may be readily developed. The amount of air actually supplied for combustion, however, cannot be suitably or reliably measured or weighed in commercial practice, and such amount can only be determined from the analysis of the products of combustion, ordinarily known as flue, or funnel, gases. As indicated, it is the relation of these products of combustion to the amount of air theoretically required for a given fuel that is the measure of the efficiency of combustion.

Apparatus for Flue-gas Analysis—In the ordinary commercial gas analysis, the proportionate parts by volume of carbon dioxide, carbon monoxide, and oxygen are determined, the difference between the sum of these constituents and 100 per cent being assumed as nitrogen. Where accuracy carried, perhaps, beyond the known limitations of the ability in boiler practice to obtain true average gas samples is required, the gas sampling apparatus also determines the percentage by volume of hydrogen and methane, the latter representing the most probable hydrocarbon content of the flue gases.

For the ordinary analysis of flue gases the Orsat apparatus, illustrated in Fig. 10, is most commonly used. The burette *A* is graduated in cubic centimeters up to 100, and is surrounded by a water jacket to prevent any change in temperature from affecting the density of the gas being analyzed.

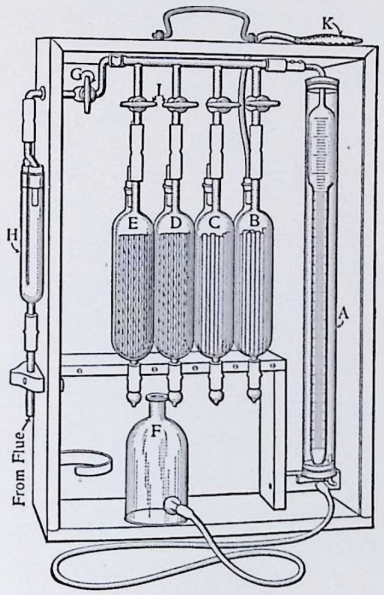
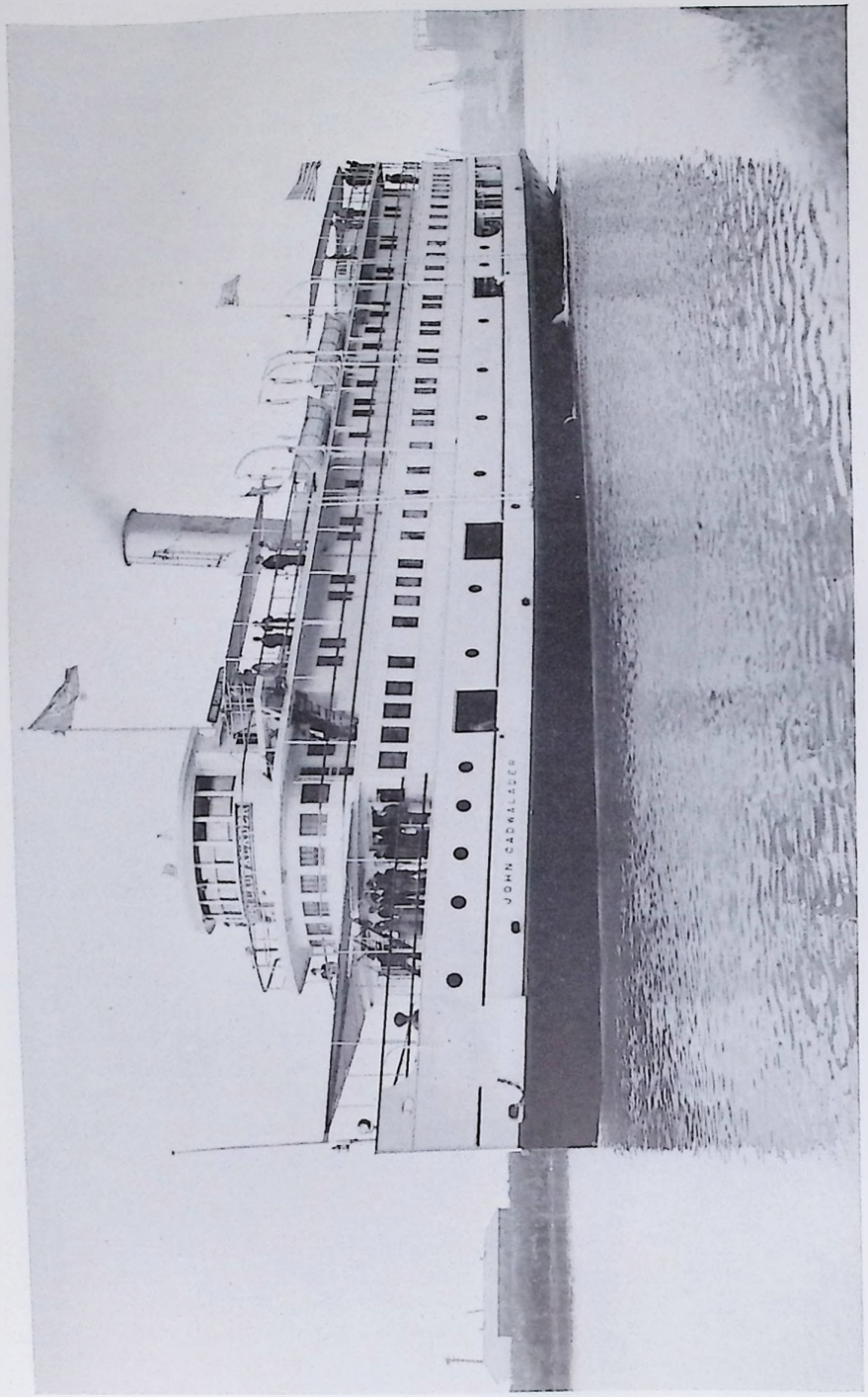


Fig. 10. Orsat Apparatus

For accurate work, it is advisable to use four pipettes, *B, C, D, E,* the first containing a solution of caustic potash for the absorption of carbon dioxide, the second an alkaline solution of pyrogallol for the absorption of oxygen, and the remaining two an acid solution of cuprous chloride for absorbing the carbon monoxide. Each pipette contains a number of glass tubes. to which some of the solution clings, thus facilitating the absorption of the gas. In the pipettes *D* and *E*, copper wire is placed in these tubes to re-energize the solution as it becomes weakened. The rear half of each pipette is fitted with a rubber bag, one of which is shown at *K*, to protect the solution from the action of the air. The solution in each pipette should be drawn up to the mark on the capillary tube.

The gas is drawn into the burette through the U-tube *H*, which is filled with spun glass or similar material to clean the gas. To discharge any air or gas in the apparatus, the cock *G* is opened to the air and the bottle *F* is raised until the water in the burette reaches the 100 cubic centimeters mark. The cock *G* is then turned so as to close the air opening and allow gas to be drawn through *H*, the bottle *F* being lowered for this purpose. The gas is drawn into the burette to a point below the

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Steamer "John Cadwalader," 2000 Indicated Horse-power, Babcock & Wilcox Boilers and Oil Burners
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zero mark, the cock G then being opened to the air and the excess gas expelled until the level of the water in F and in A is at the zero mark. This operation is necessary in order to obtain the zero reading at atmospheric pressure.

The apparatus should be carefully tested for leakage, as well as all connections leading to it. Simple tests can be made; for example, if after the cock G is closed, the bottle F is placed on top of the frame for a short time and again brought to the zero mark, the level of the water in A is above the zero mark, then a leak is indicated.

Before taking a final sample for analysis, the burette A should be filled with gas and emptied once or twice, to make sure that all the apparatus is filled with the new gas. The cock G is then closed and the cock I in the pipette B is opened, and the gas driven over into B by raising the bottle F. The gas is drawn back into A by lowering F, and when the solution in B has reached the mark in the capillary tube, the cock I is closed and a reading is taken on the burette, the level of the water in the bottle F being brought to the same level as the water in A. The operation is repeated until a constant reading is obtained, the number of cubic centimeters being the percentage of CO_2 in the flue gases.

The gas is then driven over into the pipette C and a similar operation is carried out. The difference between the resulting reading and the first reading gives the percentage of oxygen in the flue gases.

The next operation is to drive the gas into the pipette D, the gas being given a final wash in E, and then passed into the pipette C to neutralize any hydrochloric acid fumes which may have been given off by the cuprous chloride solution, which, especially if it be old, may give off such fumes, thus increasing the volume of the gases and making the reading on the burette less than the true amount.

The process must be carried out in the order named, as the pyrogallol solution will also absorb carbon dioxide and the cuprous chloride solution will also absorb oxygen.

The Orsat apparatus is ordinarily used for taking what may be called "snap" samples of gas at fixed intervals during the period of a boiler's operation, though it may be used in analyzing a continuous gas sample. Where an analysis is extended beyond the limitations of the Orsat apparatus as to constituent gas content, it is usually made of a continuous sample and a Hemphill apparatus is used.

While the errors resulting from the proper use of flue-gas analyses in the computation of combustion data are probably well within the error of boiler testing as a whole, there are, however, sources of possible error in the actual making of analyses that must be guarded against. These may be summarized as follows:

1st. Care should be taken that the sample of gas for analysis is an average sample. This is the feature which should be most carefully watched and is perhaps the most difficult of achievement. No hard and fast rules can be laid down for the methods of obtaining such average sample and it is largely a question of common sense. The sample should be drawn from the main body of the gases at a location where the possibility of dilution through air infiltration is a minimum.

2nd. Absorption reagents should be reasonably fresh. Each reagent is capable of absorbing a definite amount of one of the constituent gases, this amount ordinarily being expressed in terms of volume of the absorbing medium, and a check should be kept on the total absorption. Where solutions are weak and absorption

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is not accomplished within a relatively short time, there is a tendency to accept the absorption as complete, which results in an inaccurate analysis.

3rd. There is a tendency, particularly in the case of inexperienced operators, to attempt to force the absorption. With reasonably fresh solutions, the gas should be brought into contact with the absorption tubes at least twice, and oftener as the solutions become weaker. In the case of oxygen, where, through attempting to force the rapidity of the analysis, absorption is not complete, erroneous results as to both oxygen and carbon monoxide content will be obtained, since the absorption reagent for the latter will also absorb oxygen.

4th. Analyses should be completed. Too frequently it is assumed that the carbon dioxide content alone, or the carbon dioxide and oxygen content, is sufficient, but often the efficiency seemingly indicated by a high carbon dioxide content alone would be more than offset by the fact that appreciable amounts of carbon monoxide were present and not analyzed.

Various tables have been published as indicative of the losses due to decreasing percentages of carbon dioxide in the flue gases. Such tables, if based on the combustion of carbon alone, would be an approximately correct measure of such loss, but with the range of analyses of different fuels no table of carbon dioxide, as representing combustion losses, may be accepted as indicative of the extent of such loss.

TABLE 29

VARIATION IN CO_2 WITH DIFFERENT PERCENTAGES OF EXCESS AIR AND DIFFERENT FUELS

Constituent	Class of Fuel					
	Coal*	Wood*	Oil*	Natural Gas†	By-product Gas†	Blast Furnace Gas†
C	79.86	50.31	84.00	. . .	. . .	. . .
H_2	5.02	6.20	12.70	1.82	53.00	3.50
O_2	4.27	43.08	1.20	0.35	. . .	. . .
N_2	1.86	0.04	1.70	3.40	12.10	58.60
S	1.18	. . .	0.40	. . .	. . .	. . .
Ash	7.81	0.37	. . .	. . .	. . .	. . .
CO	. . .	. . .	. . .	0.45	6.00	25.40
CO_2	. . .	. . .	. . .	0.22	0.75	12.50
CH_4	. . .	. . .	. . .	93.33	28.15	. . .
C_2H_4	. . .	. . .	. . .	0.25	. . .	. . .
H_2S	. . .	. . .	. . .	0.18	. . .	. . .
Excess Air Per Cent	Percentage of CO_2 Corresponding to Different Amounts of Excess Air					
0	18.43	20.10	15.40	11.65	9.36	25.08
20	15.29	16.72	12.69	9.54	7.67	22.97
40	13.06	14.31	10.79	8.07	6.49	21.20
60	11.40	12.51	9.38	6.99	5.63	19.67
80	10.11	11.12	8.30	6.17	4.97	18.38
100	9.09	10.00	7.45	5.52	4.48	17.20

*Analysis by weight. †Analysis by volume.

The variation in percentage of carbon dioxide with different amounts of excess air with different fuels is shown in Table 29; and if a minimum of excess air, provided combustion is complete, is the measure of efficient combustion, the range of such combustion conditions, with different fuels, as shown by the carbon dioxide content of the products of combustion, is wide.

Combustion Formulæ—When combustion is complete, regardless of the amount of air supplied for combustion, the only products that can result from the combustion of any fuel are carbon dioxide, oxygen, sulphur dioxide or sulphur trioxide, water vapor and nitrogen. Since commercial gas analyses indicate only the first two constituents, they are in reality only a measure of the completeness of combustion of the carbon content of the fuel.

Air Required for Combustion—With carbon, hydrogen and sulphur, either free or in combination, the only combustible elements of the fuels used for steam generation, the amount of air theoretically required for complete combustion may be readily computed from the values of Table 27. This amount may be expressed as:

Pounds air required per pound fuel =

$$11.52C + 34.56\left(H - \frac{O_2}{8}\right) + 4.32S \qquad (15)$$

or reduced to simpler form:

Pounds air required per pound fuel =

$$34.56\left(\frac{C}{3} + (H_2 - \frac{O_2}{8}) + \frac{S}{8}\right) \qquad (16)$$

where C, H_2, O_2, and S represent the percentages by weight of carbon, hydrogen, oxygen and sulphur, as determined by ultimate analysis. These formulæ, as in the case of all combustion formulæ, assume that the oxygen constituent of the fuel is free to unite with the hydrogen to form water vapor, such assumption leading to an error in the computation of the amount of air required for combustion that is negligible. The constants of the formulæ are taken directly from Table 27.

Products of Combustion—The weight of the products of combustion resulting from the theoretically perfect combustion of any fuel has been indicated by example, and may be directly computed from the values of Table 27. To emphasize such values, it may be stated again that for perfect combustion the weight is:

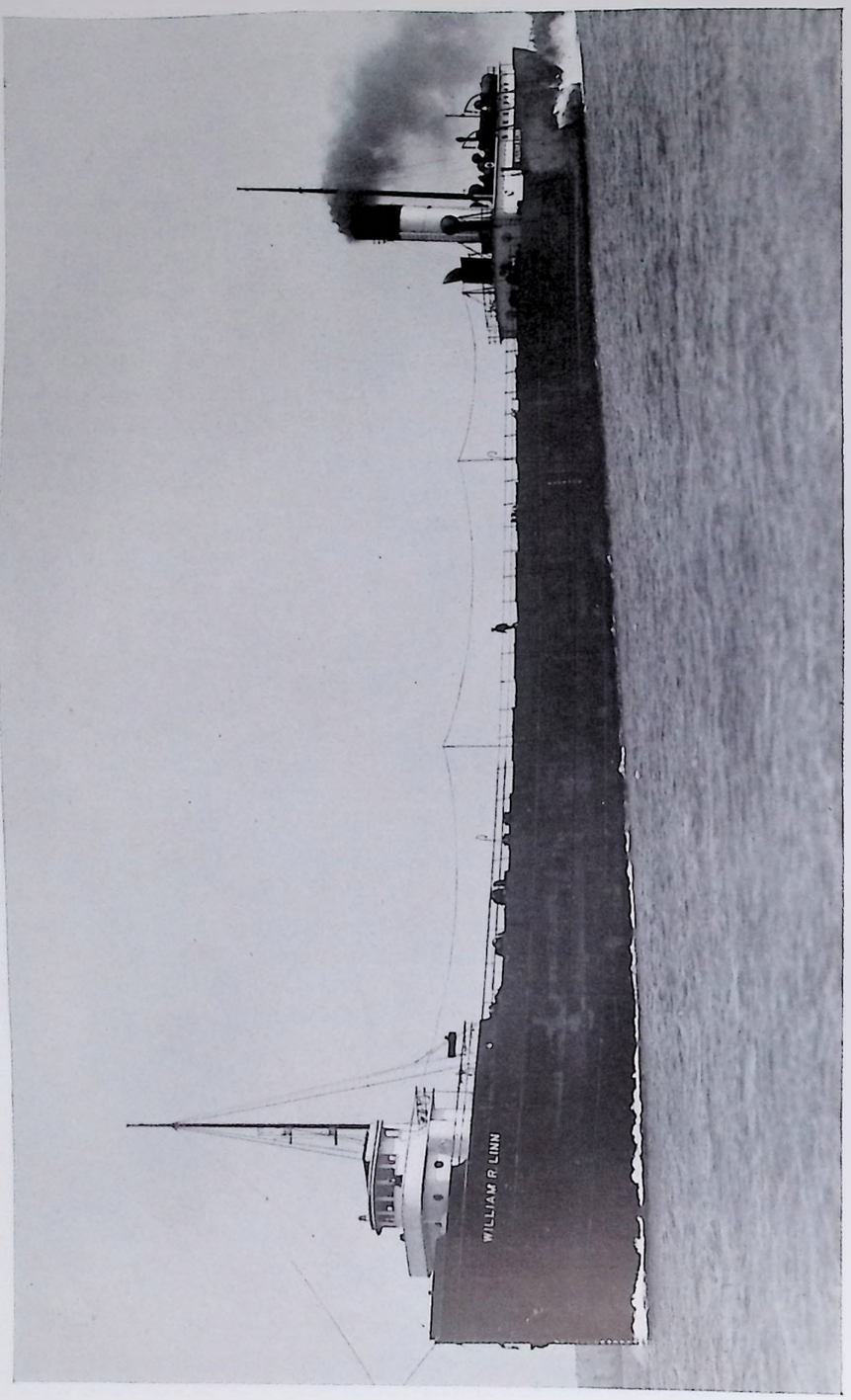
Products of combustion, pounds, per pound of fuel:

$$CO_2 = 3.667C$$
$$H_2O = 9\left(H_2 - \frac{O_2}{8}\right) + H_2O$$
$$SO_2 = 2S$$
$$N_2 = 8.85C + 26.56H_2 + 3.32S + N_2$$

The weights of the theoretical products of combustion as thus determined may be expressed as percentage weights and from such percentages may be readily transformed into percentages by volume.

Combustion Data from Flue-gas Analysis—While a flue-gas analysis may be used in connection with the analysis resulting from the perfect combustion of any fuel for the determination of combustion losses, its generally accepted use

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Steamer "William R. Linn," 2000 Indicated Horse-power. Babcock & Wilcox Boilers
Owners, Pittsburgh Steamship Company, Cleveland, O.

is its application in a formula which is assumed to give directly the weight of dry gas per pound of carbon burned, this weight being used in combustion loss computations. This formula is:

$$\frac{11CO_2 + 8O_2 + 7\,(CO + N_2)}{3(CO_2 + CO)} \qquad (17)*$$

where the symbols represent the percentages by volume of the constituents of the gas analyzed.

The principal assumption of this formula is that the analysis as determined is of dry gas. As a matter of fact, all flue gases as analyzed contain a greater or less amount of water vapor, depending upon the moisture content of the air supplied for combustion and of the fuel, and resulting from the burning of the hydrogen constituent of the fuel. From this latter alone, with certain fuels, the moisture content may be as high as 15 per cent of the total weight of flue gases. The presence of the moisture in the flue gases, together with the ordinary use of water as the displacement element in the commercial gas analysis apparatus, tends toward saturation of the gas analyzed. If such saturation exists, and from actual determinations such an assumption seems warranted, proportionate parts of the water vapor in the gases will be absorbed with the different other constituents and the resulting analysis may be assumed to be that of dry gas.

A further source of error arising from the use of formula *(17)* results from the presence of sulphur in a number of fuels. Such portion of the sulphur content as is burned appears ordinarily in the gases to be analyzed as sulphur dioxide, though it is absorbed with carbon dioxide and appears as such in the flue-gas analysis. Where the sulphur constituent of a fuel is low, the error arising from this source is so small as to be safely neglected, although the most accurate results are desired. Even where the sulphur content of the fuel is high, it is at best questionable whether in commercial boiler practice an attempt to make a sulphur correction is warranted. Any such correction would, of necessity, assume that all of the sulphur is burned to sulphur dioxide, whereas in practice some of the sulphur probably appears in compounds in the furnace refuse. Further, since the gases are in contact with the water used as a displacement agent in which sulphur dioxide is highly soluble, it is probable that some of the sulphur dioxide is absorbed before the gases come into contact with the absorption agents with which the analysis is accomplished.

Computations show that in burning a coal containing 70 per cent carbon, 5 per cent hydrogen, 7 per cent oxygen, 1 per cent nitrogen and 5 per cent sulphur, burned with as little as 10 per cent excess air, the sulphur dioxide appearing in the products of combustion, assuming all of the sulphur burned, is only 0.43 per cent by volume. For any smaller percentage of sulphur in the fuel, and 5 per cent represents the approximate maximum even in the case of the poorest fuels, the percentage by volume of sulphur dioxide in the flue gases would be appreciably less under the same conditions of combustion, and if the amount of excess air supplied is greater than 10 per cent, and such excess is probably far below the average minimum, the percentage would be correspondingly less. In view of the small percentage of sulphur dioxide appearing in a volumetric flue-gas analysis in what may be considered the extreme case of commercial combustion, any attempt to make a sulphur

* For derivation of this formula, see "Principles of Combustion in the Steam Boiler Furnace," The Babcock & Wilcox Company, 1920.

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correction in the ordinary gas analysis would appear to be a refinement not warranted from the standpoint of limits of error of such analysis.*

Formula *(17)*, as stated, gives the weight of dry products of combustion per pound of carbon burned. By multiplying this quantity by the actually *burned* carbon in one pound of fuel, the weight of dry products per pound of fuel may be determined, this expression being represented by the formula:

Weight of Dry Products per Pound of Fuel =

$$\frac{11CO_2 + 8O_2 + 7(CO + N_2)}{3(CO_2 + CO)}\,C \qquad (18)$$

It is necessary to emphasize here that the weight of carbon as given by an analysis of the fuel and the weight of carbon burned, *i. e.*, the weight by which the result



Turbo Electric Ferry Steamer "Peralta," 2600 Shaft Horse-power
Babcock & Wilcox Boilers, Superheaters and Oil Burners
Owner, Key System Transit Company, Oakland, Cal.

of the application of formula *(17)* is to be multiplied, is not of necessity the same. Obviously, if a coal containing 70 per cent carbon (or 0.70 pound of carbon per pound of coal) is burned in such manner that the resulting ash contains 20 per cent of unconsumed carbon, the value of C in formula *(18)* cannot be taken as 0.7 per pound of fuel. The variation in the weight of carbon as given by a fuel analysis and in the weight of carbon actually burned is best shown by example:

* A formula developed by Mr. C. H. Berry for sulphur corrections that gives result probably as accurate as any offered, where most accurate results are required, and where fuel and gas analyses are known to be accurate *and average*, is obtained by adding to the weight of carbon burned per pound of fuel ⅜ of the weight of sulphur burned, it being necessary to assume that all of the sulphur *is* burned. This formula may be expressed for the weight of dry products of combustion per pound of fuel (W).

W=Weight of combustion products per pound of C[by(17)]×C† + ⅜S†.

† Weight of C and S burned per pound of fuel.

Assume a coal containing 70 per cent carbon and 10 per cent *chemical* ash. If such coal is burned to show 20 per cent unconsumed carbon in the ash, the ash as determined by test is:

$$10.0 \div (1.0 - 0.2) = 12.5 \text{ per cent}$$

which, expressed in terms of one pound of fuel, consists of 0.10 pound of chemical ash and 0.025 pound of unconsumed carbon included in the total ash. The carbon per pound of coal *burned* would thus be $0.700 - 0.025 = 0.675$ pound, and it is this value that should be used in multiplying the weight of dry products per pound of carbon in the fuel, as given by formula *(17)*, for the determination of the weight of dry products of combustion per pound of fuel.

It is to be remembered that the unconsumed carbon content of the ash varies with every boiler test, even of a single boiler and stoker unit, and if but a single fuel analysis is made for a series of tests, a different unconsumed carbon correction must be made for each test.

Aside from the moisture content of the air supplied for combustion, which in terms of the total moisture in the ordinary flue gas is small, the total weight of moisture appearing in the products of combustion, resulting from the moisture in the fuel and from that due to the combustion of the hydrogen content, is a constant for any fuel, regardless of the amount of excess air supplied for combustion. The loss due to the moisture in the air supplied for combustion, except where every effort is made to distribute combustion losses, is ordinarily included in the unaccounted loss. Neglecting, then, the moisture in the air supplied for combustion, the total weight of water vapor per pound of fuel burned appearing in the products of combustion may be expressed:

$$\text{Weight of Water Vapor per Pound of Fuel} = 9H_2 + H_2O \qquad (19)$$

where H_2 and H_2O represent the weights of hydrogen and moisture per pound of fuel burned.

The sum of the dry products per pound of fuel burned from formula *(18)* and of the water vapor products from formula *(19)* evidently represents the weight of the total products of combustion per pound of fuel.

The total air supplied for combustion per pound of fuel burned must be represented by the total weight of products per pound, less the weight of fuel appearing in the flue gases. In the case of solid fuels, this weight is the unit weight of fuel less the total ash, *i. e.*, the ash by test and not the chemical ash. In the case of liquid or gaseous fuels, where the assumption is that all of the constituents pass up the funnel in the products of combustion, the weight of air supplied is the total weight of the flue gases per pound of fuel less one.

With an ultimate analysis available of the fuel burned, the weight of air actually supplied, as determined by this method, may be compared with the weight of air theoretically required, as given by the methods of Table 27, and the amount of excess air, expressed in percentage, may be computed.

Several formulæ have been offered for the computation of the weight of air supplied per pound of fuel. The most common of these is

$$\text{Pounds air per pound fuel} = \frac{3.032\,N_2}{CO_2 + CO}\,C \qquad (20)$$

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City of New Orleans Fireboat "Deluge," Equipped with Babcock & Wilcox Boilers and Oil Burners

where the symbols represent the percentages by volume of the constituents of the gas. This formula is based on a constant nitrogen content of one per cent in the fuel burned, and for solid and liquid fuels where the nitrogen percentage is reasonably close to the assumed value, results accurate within the limits of accuracy of commercial gas analyses may be determined by the use of this formula. With gaseous fuels, on the other hand, where the amount of nitrogen in the fuel itself is, under ordinary good combustion conditions, almost as great as in the air supplied for combustion, formula *(20)* is useless. The error in the case of a fuel like blast-furnace gas may be as great as 70 per cent.

It is frequently desirable to compute the percentage of excess air corresponding to a given amount of carbon dioxide with any fuel the analysis of which is given. A method of such determination has been developed that is, perhaps, best illustrated by example:

Let us assume a coal having an ultimate analysis as follows:

	Per cent
Carbon	66.9
Hydrogen	4.4
Oxygen	7.9
Nitrogen	1.2
Sulphur	3.3
Moisture	4.5
Ash	11.8

The products of perfect combustion per pound of coal, computed by the methods of the preceding chapter, will be:

	Pounds
CO_2	2.453
SO_2	0.066
O_2	0.000
CO	0.000
N_2	6.950
H_2O	0.441

which, assuming that the moisture content of the flue gases is proportionately absorbed, gives a dry gas analysis corresponding to a total weight of flue gas of 9.469 pounds.

The volumes of these different constituents under standard conditions are, from Table 19:

CO_2	2.453 × 8.103 =	19.877	cubic feet CO_2
SO_2	0.066 × 5.473 =	0.361	cubic foot SO_2
		20.238	cubic feet $CO_2 + SO_2$ *(a)*
N_2	6.950 × 12.809 =	89.023	cubic feet N_2
		109.261	cubic feet total *(b)*

The total volume of products of perfect combustion under standard conditions is, then, 109.261 cubic feet.

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If x equals the weight of excess oxygen supplied per pound of fuel, $3.32x$ will equal the weight of nitrogen accompanying such excess oxygen, or expressed volumetrically:

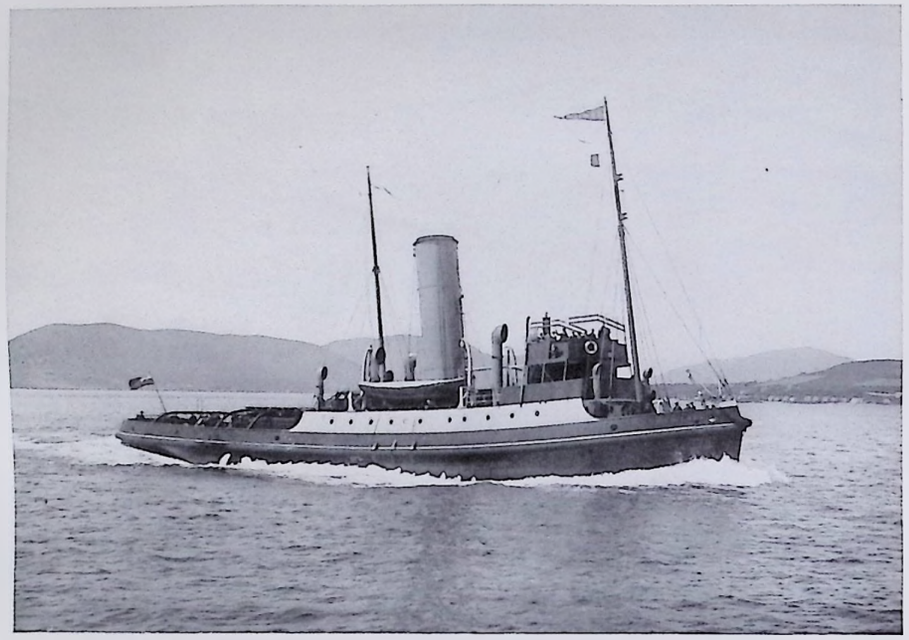
$$\text{Excess } O_2 = 11.209x \text{ cubic feet,}$$
$$\text{Excess } N_2 = 12.809 \times 3.32x \text{ cubic feet,}$$

and the total volume of excess air supplied is:

$$11.209x + (12.809 \times 3.32x) = 53.73x \text{ cubic feet.}$$

The percentage of carbon dioxide by volume is then:

$$\frac{a}{b + 53.73x}$$



Towboat "T. S. McEwen," 3000 Indicated Horse-power, Babcock & Wilcox Boilers
Owners, Cape Town Harbor Board

or if, for the coal assumed, combustion is such as to show 13.5 per cent of carbon dioxide, we have

$$\frac{20.238}{109.261 + 53.73\,x} = \frac{13.5}{100}$$

or

$$x = \frac{20.238 - (0.135 \times 109.261)}{0.135 \times 53.73}$$
$$= 0.7565$$

The weight of excess oxygen, x, corresponding to a gas analysis showing 13.5 per cent of carbon dioxide is thus 0.7565 pound, and the weight of excess nitrogen, $3.32x$, is 2.5116 pounds, or a total weight of excess air supplied for combustion of 3.2681 pounds. The total weight of air supplied for combustion per pound of fuel

is then

$$9.030 + 3.268 = 12.298 \text{ pounds}$$

and the percentage of excess air supplied

$$3.268 + 9.030 = 36.2 \text{ per cent.}$$

As has been stated, the amount of excess air supplied for combustion affects directly the loss in the flue gases leaving the boiler. This loss is given by the formula:

$$\text{Loss in B.t.u. per pound of fuel} = W \times 0.24(T - t) \qquad (21)$$

Where W = weight of dry products of combustion per pound of fuel,
T = temperature of flue gases,
t = temperature of atmosphere,
0.24 = the assumed specific heat of the flue gases (see below).

Obviously, for the combustion conditions just considered, the loss as computed from formula *(21)* would be 36.2 per cent greater with a percentage of carbon dioxide in the flue gases of 13.5 than if combustion had been perfect.

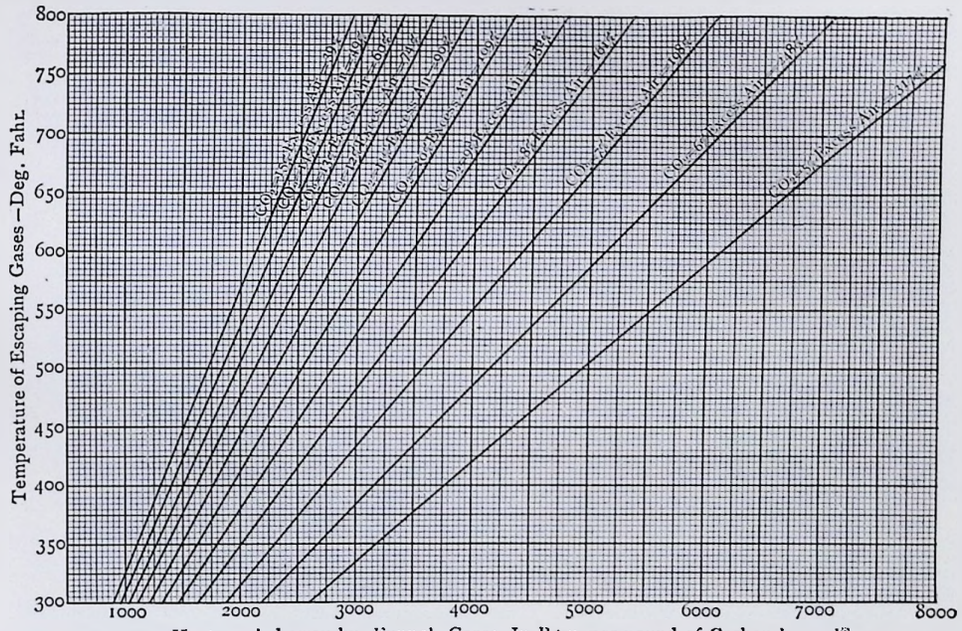
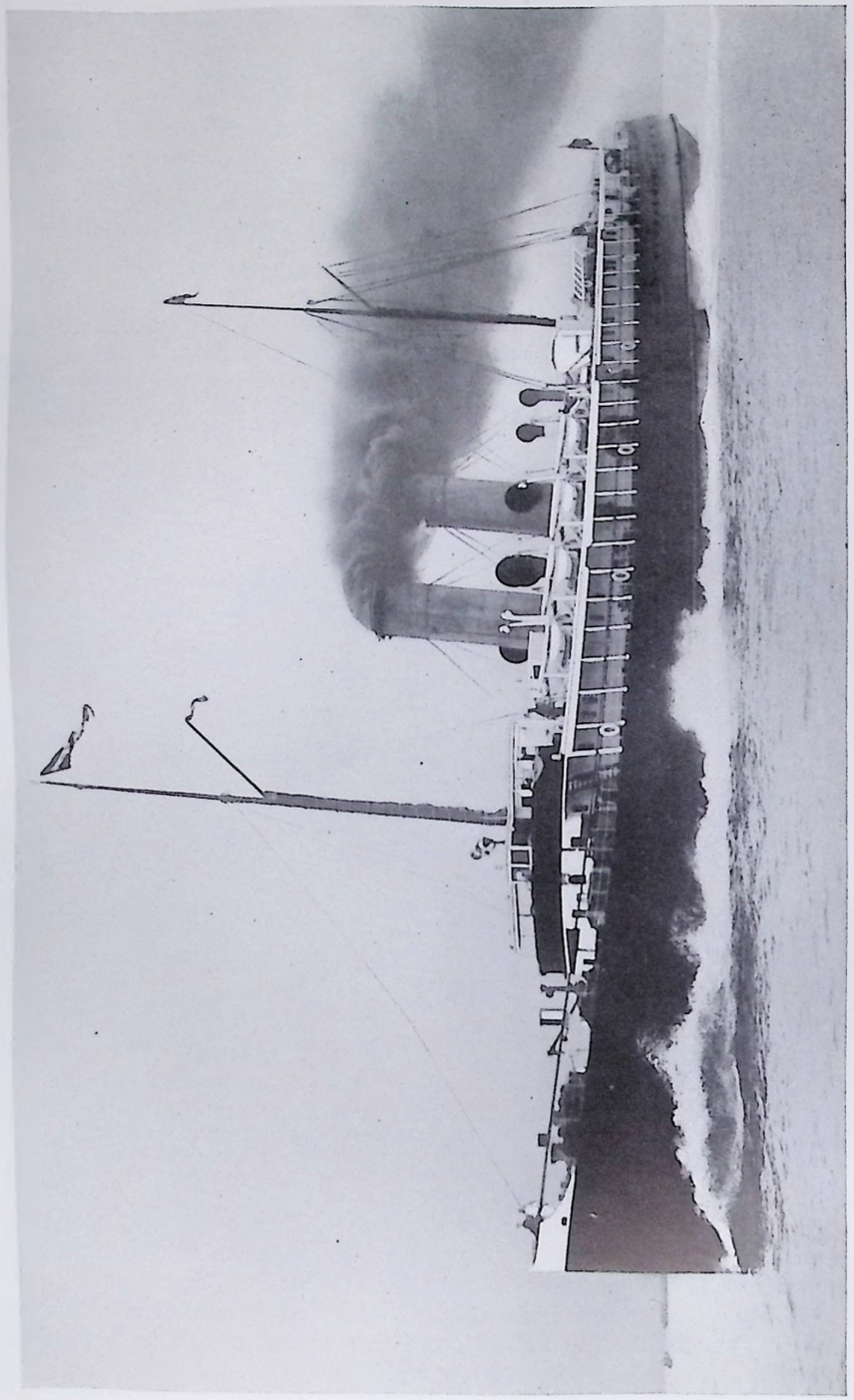


Fig. 11. Loss Due to Heat Carried Away by Funnel Gases for Varying Percentages of Carbon Dioxide. Based on Boiler Room Temperature = 80 Degrees Fahrenheit. Nitrogen in Flue Gas = 80.5 Per Cent Carbon Monoxide in Flue Gas = 0. Per Cent

Fig. 11 represents graphically the loss due to heat carried away in the dry flue gases for various percentages of carbon dioxide and different flue gas temperatures. The percentage loss depends upon the carbon content and heat value of the fuel. The nitrogen content of the flue gases has been assumed constant, regardless of variation in carbon dioxide. As a matter of fact, the percentage of nitrogen varies with different percentages of carbon dioxide, but such variation is small and, since Fig. 11 can be used only for approximate results, may be neglected.

* For loss per pound of coal, multiply by percentage of carbon in coal by ultimate analysis.

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STEAMER "PRINCESS ELIZABETH," 10,000 SHAFT HORSE-POWER, BABCOCK & WILCOX BOILERS: OWNERS, BELGIAN STATE RAILWAYS

The value 0.24 given in formula *(21)* as the specific heat of flue gases is rather an arbitrary value that custom rather than fact has brought into common use. The specific heat of the flue gases varies with the analysis of the gases and with the temperature, and where gas analyses and temperatures are known to be correct, for the most accurate results the specific heat should be computed for every set of conditions encountered in testing.

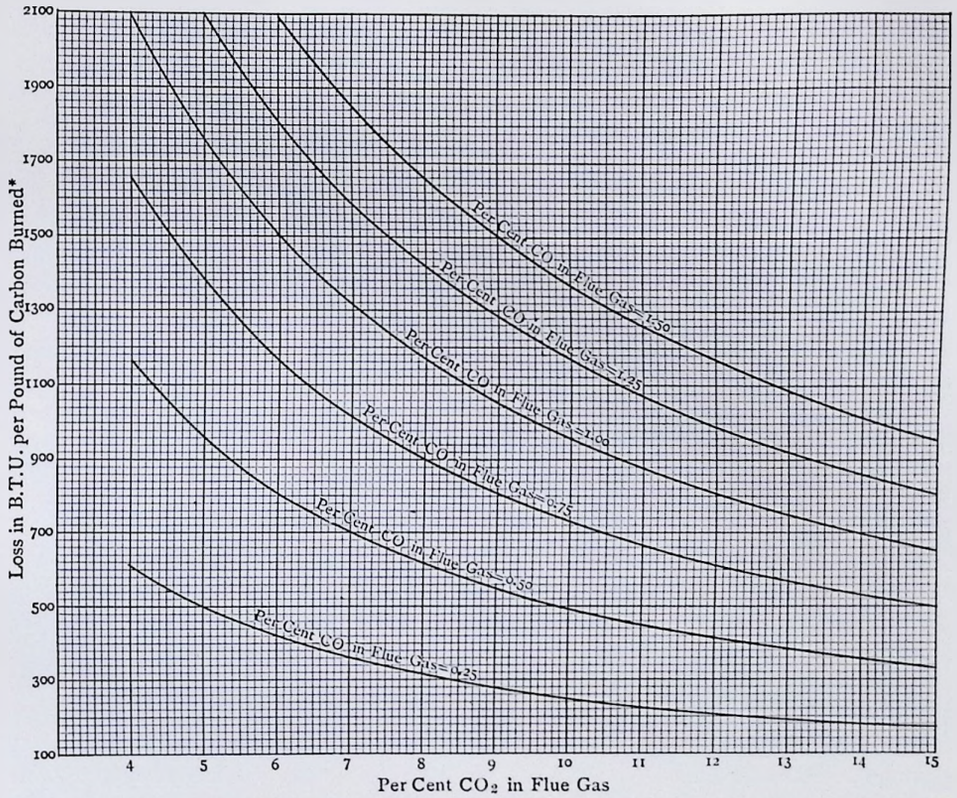


FIG. 12. LOSS DUE TO UNCONSUMED CARBON CONTAINED IN THE CO IN THE FLUE GASES

If combustion is not complete, and some part of the carbon constituent of the fuel is burned to carbon monoxide instead of carbon dioxide, the loss due to such incomplete combustion is very appreciable. Such loss is expressed by the formula:

Loss in B.t.u. per pound of fuel burned =

$$10{,}160\left(\frac{CO}{CO + CO_2}\right) C \qquad (22)$$

where, as before, CO and CO_2 are the percentages by volume of these constituents as shown by the flue-gas analysis, and C is the weight of carbon per pound of fuel

*For loss per pound of coal, multiply by percentage of carbon in coal by ultimate analysis.

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that is burned and appears in the flue gases. The value 10,160 represents the difference in the amount of heat evolved in the combustion of one pound of carbon to carbon monoxide and to carbon dioxide, as indicated by the heats of combustion as given in Table 26. Fig. 12 represents graphically the loss due to such carbon constituent in the fuel as is not completely burned and appears in the flue gases as carbon monoxide.

Unreliability of CO_2 Readings Alone—The possibility of error in the ordinary gas analysis has been indicated. It is perhaps well to add here a caution against taking a carbon dioxide reading, alone, as indicative of the efficiency of combustion. Ordinarily, a high carbon dioxide content in the flue gases is assumed as showing proper combustion conditions. Such an assumption is correct in so far as it indicates an amount of excess air approaching that theoretically necessary for combustion, but as the percentage of carbon dioxide increases, there is a tendency toward the presence of carbon monoxide in the flue gases, which an automatic carbon dioxide recorder does not show and which is frequently difficult to detect with an Orsat apparatus. For this reason, any automatic recording device showing carbon dioxide alone should be periodically checked by another method of gas analysis, and the solution for the absorption of carbon monoxide in the Orsat or other gas analysis apparatus should be kept fresh and known to be capable of absorption.

Any comparison of losses as indicated by flue-gas analyses showing comparatively low carbon dioxide, without carbon monoxide, as against a high carbon dioxide with carbon monoxide present, does, of course, depend upon the fuel burned. Roughly, it may be stated that with a gas temperature of 300 degrees leaving an air heater, the efficiency of combustion, as represented by 13 per cent carbon dioxide and no carbon monoxide, is as high as that indicated by a carbon dioxide content of 15 per cent with a carbon monoxide content of 0.25 per cent. Such a relation may be readily computed from the values of comparative losses as indicated in Figs. 11 and 12.

Conversion of Analyses—In combustion work, gas analyses are frequently given in terms of volume when analyses in terms of weight are required. To convert a volumetric analysis to one by weight, the percentage by volume of each constituent gas should be multiplied by its relative density, each product being divided by the sum of the products. To convert an analysis by weight to one in terms of volume, the percentage by weight of each constituent should be divided by its relative density, and each quotient so obtained be divided by the sum of the quotients. Since the molecular weights of the various gases bear the same relations to each other as the relative densities, these molecular weights may be used in transforming analyses instead of the relative densities.

Another method of converting volumetric analyses to analyses in terms of percentage by weight is through the use of the weights per cubic foot of the various constituent gases. The percentages by volume are multiplied by the weights per cubic foot, and each product is divided by the sum of the products. This method has an advantage in that it gives directly, in the sum of the products, the weights of the gas as a whole cubic foot.

As an example of these methods of conversion, assume a flue gas having a volumetric analysis as follows:

	Per Cent
Carbon Dioxide	13.5
Carbon Monoxide	0.1
Oxygen	5.9
Nitrogen	80.5

(a) Conversion by relative densities:

Gas	*Relative Density*		*Product*				*Weight Per Cent*
13.5	11	=	148.5	÷	759.9	=	19.54
0.1	7	=	0.7	÷	759.9	=	0.09
5.9	8	=	47.2	÷	759.9	=	6.21
80.5	7	=	563.5	÷	759.9	=	74.16
			759.9				100.00

(b) Conversion by weights:

Gas	*Volume per Cubic Foot*		*Weight per Cubic Foot*		*Weight Pound*				*Weight Per Cent*
CO_2	0.135	×	0.12341	=	0.016660	÷	0.084847	=	19.64
CO	0.001	×	0.07806	=	0.000078	÷	0.084847	=	0.09
O_2	0.059	×	0.08921	=	0.005263	÷	0.084847	=	6.21
N_2	0.805	×	0.07807	=	0.062846	÷	0.084847	=	74.06
					0.084847				100.00

The weight per cubic foot of the flue gas as determined by this second method is thus 0.08485 pound.

The slight discrepancy in the two sets of weight percentages is due to the fact that in the first computations relative densities based on approximate instead of accurate molecular weights are used.

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Municipal Ferryboats "Manhattan," "Brooklyn," "Bronx," "Queens," and "Richmond," 3800 Indicated Horse-power, Speed 16.9 Miles
Babcock & Wilcox Boilers; Owner, City of New York

COAL AND ITS COMBUSTION

Formation of Coal—All coals are of vegetable origin and are the remains of prehistoric forests. They are the result of the transformation of woody fiber and other vegetable matter by the elimination of oxygen and hydrogen in proportionately larger quantities than the removal of carbon, or a devolatilization of the original matter. The causes and methods of the first stage of transformation are not exactly defined but were probably due to the action of bacterial ferment in transforming cellulose into peaty matter. Subsequent changes were the results of great pressures and temperatures which, by destructive distillation, resolved the organic matter into its invariable ultimate constituents, carbon, hydrogen, oxygen and other substances in varying proportions. The factors of time, depth of beds, disturbances of beds and introduction of foreign matter resulting from such disturbances, have produced the variation in the degree of evolution from vegetable fiber to hard coal. While age is an important factor in the degree of evolution, it is not, as ordinarily considered, the primary factor in transformation. This primary factor is the presence or absence of disturbance in the rocky crust of the earth's surface during the period of transformation, either in the field where the coal was being formed, or close to such field. Lignites represent the coals of most recent formation and are considered, to an extent correctly, low in the rank of coals because of this fact. On the other hand, there are coal beds in the Western States, where disturbances of the earth's crust were great, which are of the same age as the lignitic beds of other sections of the country and still are representative of the highest rank coals of that portion of the country.

The amount of foreign and incombustible matter found in coal depends upon two factors in its formation: the degree of disturbance of the bed in the formative stage as affected by the introduction of rock, sand, etc., and the conditions which permitted mineral-bearing waters to circulate through the beds in process of formation. Geologically, these conditions are difficult, if not impossible, of determination, and the grade of coal cannot be foretold simply from the area in which it is to be mined.

Composition of Coal—The analyses of coal and the methods of analyses are discussed in detail in the following chapter. For an understanding of the classification of coals, however, analyses as ordinarily given must be briefly considered here. The uncombined carbon of coal is known as the fixed carbon. A certain percentage of the total carbon constituent is combined with hydrogen and, in combination, burns as a hydrocarbon. This, together with the other gaseous substances driven off by the application of heat, forms that portion of the coal known as volatile matter. The fixed carbon and the volatile matter constitute the combustible. The oxygen and nitrogen contained in the volatile matter are not combustible, but custom has applied this term to that portion of the coal free from moisture and ash, thus including oxygen and nitrogen. The other important substances entering into the composition of coal are moisture and the refractory earths which form the ash. The ash varies in different coals from 3 to 30 per cent and the moisture from 0.75 to 45 per cent of the total weight of the coal, depending upon the grade and the locality in which it is mined.

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TABLE 29A

TESTS OF BABCOCK & WILCOX MARINE BOILERS USING COAL FUEL

Date	August, 1922		June, 1910		
Location	S. S. "Crescent City"		U. S. S. "Wyoming" Test Boiler, Bayonne, N. J.		
Type of Babcock & Wilcox marine boiler	4-inch tube		2-inch tube		
Duration, hours	9	9	24.06	24	3
State of weather	Clear	Clear	Clear	Partly Clear	Clear
Type of draft	Natural	Natural	Natural	Forced	Forced
Method of firing	Hand	Hand	Hand	Hand	Hand
Kind of fuel	Semi-Bituminous (Detour)		Pocahontas, Hand Picked		
Total heating surface, square feet	5366	5366	2571.39	2571.39	2571.39
Grate surface, square feet	140	140	57.89	57.89	57.89
Average Pressures					
Boiler steam pressure by gauge, pounds			202.0	201.6	200.4
Starboard	234.4	236.4			
Port	233.8	236.8			
Barometer	29.22	29.26	30.06	29.83	29.62
Average Temperatures—Degrees Fahrenheit					
Fireroom	85.23	90.7	93.7	94.0	106.0
Hotwell	124.0	124.0			
Feed water leaving heater	184.33	195.4	211.6	204.1	194.4
Boiler exit gases at uptake	532.0	538.0	491.0	628.0	659.0
Fuel					
Weight of coal as fired, pounds	26691.0	27598.5	21200.0	57700.0	12200.0
Moisture in coal, per cent	2.49	2.30	0.88	1.06	0.75
Weight of dry coal consumed, pounds	26043.4	26963.7	21013.0	57088.0	12108.0
Weight of ash and refuse, pounds	2000.0	1945.5	1261.0	2477.0	1035.0
Weight of combustible consumed, pounds	24043.4	25018.2	19752.0	54611.0	11073.0
Refuse in dry coal, per cent	7.68	7.21	6.00	4.34	8.55
Fuel Per Hour					
Coal consumed per hour, pounds	2965.6	3049.6	881.0	2404.0	4066.0
Dry coal consumed per hour, pounds	2893.5	2996.0	873.0	2379.0	4036.0
Combustible consumed per hour, pounds	2669.6	2780.0	821.0	2275.0	3691.0
Coal consumed per hour per square foot G. S., pounds	21.18	21.78	15.22	41.53	70.24
Dry coal consumed per hour per square foot G. S., pounds	20.66	21.38	15.08	41.10	69.72
Combustible consumed per hour per square foot G. S., pounds	19.08	19.83	14.18	39.30	63.77
Coal per hour per square foot H. S., pounds	.55	.57	.343	.935	1.581
Dry coal per hour per square foot H. S., pounds	.54	.55	.340	.925	1.569
Combustible per hour per square foot H. S., pounds	.50	.52	.319	.885	1.435

TESTS OF BABCOCK & WILCOX MARINE BOILERS USING COAL FUEL—Continued

Water					
Total weight of water to boilers, pounds	242711.	246743.	228095.	612698.	106504.
Factor of evaporation	1.0812	1.0698	1.052	1.060	1.069
Total evaporation from and at 212° Fahrenheit, pounds	262419.	263965.7	239956.	649460.	113853.
Water Per Hour					
Water evaporated per hour, pounds	26967.9	27415.9	9480.	25530.	35501.
Equivalent evaporation from and at 212° F., pounds	29157.7	29329.5	9974.	27060.	37951.
Equivalent evaporation from and at 212° F. per sq. ft. G. S., pounds	208.27	209.49	172.	468.	656.
Equivalent evaporation from and at 212° F. per sq. ft. H. S., pounds	5.44	5.47	3.88	10.52	14.76
Economic Results					
Water actually evaporated per pound of coal as fired, pounds	9.09	8.99	10.76	10.62	8.73
Equivalent evaporation from and at 212° F. per lb. of coal as fired, lbs.	9.83	9.62	11.32	11.26	9.33
Equivalent evaporation from and at 212° F. per lb. of dry coal, lbs.	10.08	9.78	11.42	11.38	9.40
Equivalent evaporation from and at 212° F. per lb. of combustible, lbs.	10.92	10.54	12.15	11.89	10.28
Efficiency: Boiler including grate	71.31	71.37	72.55	72.28	59.73
Flue Gas Analysis—Top 3rd Pass					
Carbon dioxide (CO_2)	12.50	12.28	13.20	13.60	11.60
Oxygen (O_2)	5.48	6.28	4.20	3.90	5.07
Carbon monoxide (CO)	0.00	0.00	0.50	0.80	1.09
Nitrogen (N_2) (by difference)	82.02	81.44	82.10	81.70	82.24
Ultimate Analysis of Dry Coal—Per Cent					
Carbon	75.88	75.88	87.51	87.51	87.51
Hydrogen	5.41	5.41	4.74	4.74	4.74
Oxygen and nitrogen	11.74	11.74	3.67	3.67	3.67
Ash	6.45	6.45	3.38	3.38	3.38
Sulphur	.52	.52	.70	.70	.70
Heat value per pound of dry coal, B. t. u.	13718.	13297.	15273.	15273.	15273.
Combustible in ash, per cent	38.86	48.10	31.93	60.99	70.72

Heat Balance	B. t. u.	Per Cent	B. t. u.	Per Cent	B. t. u.	Per Cent	B. t. u.	Per Cent	B. t. u.	Per Cent
Heat absorbed by boiler	9779.	71.31	9491.	71.37	11081.00	72.55	11040.00	72.28	9122.00	59.73
Loss due to moisture in fuel	30.	.22	29.	.22	10.84	.07	13.72	.09	9.71	.06
Loss due to moisture formed by burning of hydrogen	607.	4.42	606	4.55	521.00	3.41	547.00	3.58	548.00	3.59
Loss due to heat carried away in dry gases	1661.	12.10	1669.	12.56	1503.00	9.84	1909.00	12.50	2134.00	13.97
Loss due to incomplete combustion of carbon	0.	0.00	0.	0.00	317.00	2.08	479.00	3.14	711.00	4.65
Loss due to unconsumed carbon in ash	435.	3.17	507.	3.82	280.00	1.83	386.00	2.53	883.00	5.78
Loss due to radiation and unaccounted losses	1206.	8.78	995.	7.48	1560.16	10.22	898.28	5.88	1865.29	12.22
Total	13718.	100.00	13297.	100.00	15273.00	100.00	15273.00	100.00	15273.00	100.00

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A large percentage of ash is undesirable, as it not only reduces the heat value of the coal but chokes the air passages in the furnace and in the fuel bed, thus preventing the rapid combustion necessary for high efficiency. If the sulphur content of the coal is high, it unites with the ash to form a fusible slag or clinker which will cause furnace difficulties.

Moisture in coal may be more detrimental than ash. It absorbs heat in evaporation and in being brought to the furnace temperature, and thus reduces such temperature. With some classes of coal, however, a certain amount of water produces a mechanical action that assists in combustion and leads to better results.

Classification of Coal—The U. S. Geological Survey classifies coal by "ranks," a term used to designate differences in coal in various states of transformation from lignite to anthracite, such changes being marked by a decrease in the moisture, oxygen and volatile matter, and usually an increase in fixed carbon, sulphur, and possibly ash. The Survey uses the term "grade" in distinguishing between two coals as to ash and sulphur content. A high-grade coal would be one relatively pure and a high-rank coal one high in the scale of coal or one in which devolatilization had occurred to a high degree.

The second Pennsylvania Geological Survey used a method of coal classification based on what it termed "fuel ratios," *i. e.*, the quotient of the fixed carbon divided by the volatile matter as given by a proximate analysis of the ash-free coal. The U. S. Geological Survey, believing it to be impossible to classify all ranks of coal by chemical composition, makes use of a modification of the fuel ratio basis for high-rank coals, and supplements chemical composition of the lower ranks by physical characteristics.

The ranks of coal recognized by the U. S. Geological Survey are lignite, sub-bituminous, bituminous, semi-bituminous, semi-anthracite and anthracite. While the chemical composition of any rank may vary considerably, the figures of Table 30 indicate the approximate variation among the different ranks. Since the ash content is a matter of grade rather than rank, the analyses are given on an ash-free basis.

TABLE 30

CHEMICAL COMPOSITION AND HEAT VALUE OF VARIOUS RANKS BASED ON ASH-FREE COAL

Rank	Fixed Carbon Per Cent	Volatile Matter Per Cent	Moisture Per Cent	Heat Value B.t.u.
Lignite	37.8	18.8	43.4	7400
Sub-bituminous	42.4	34.2	23.4	9720
Low-rank Bituminous	47.0	41.4	11.6	12880
Medium-rank Bituminous	54.2	40.8	5.0	13880
High-rank Bituminous	64.6	32.2	3.2	15160
Low-rank Semi-bituminous	75.0	22.0	3.0	15480
High-rank Semi-bituminous	83.4	11.6	5.0	15360
Semi-anthracite	83.8	10.2	6.0	14880
Anthracite	95.6	1.2	3.2	14440

The coals generally used in marine practice are bituminous and semi-bituminous. These come mostly from what the U. S. Geological Survey calls the Appalachian region, though in some cases coal from states bordering on the Great Lakes is burned.

Bituminous Coal—This is the lowest rank to which the Geographical Survey gives a fuel ratio, this being as a maximum 3, and in general the volatile and fixed carbon being approximately the same. This description does not, however, definitely classify bituminous coal, since the same fuel ratio is found in some of the coals of lower rank. It differs from lignite and sub-bituminous coal primarily in the effect of weathering, being only slightly affected by exposure except over extended periods.

Bituminous coal represents the largest division of classified coals and is widest in range of chemical composition. Numerous attempts have been made to subdivide this rank, but none of the schemes advanced has ever been generally accepted. The possibility or lack of possibility of coking is a classification frequently attempted, but since the chemical composition of coal has apparently no definite relation to its coking property, such a classification fails. In this rank are included the so-called gas coals, cannel coal and scattered local groups known to the trade as "block" or "splint" coals. Inasmuch as the bituminous rank includes all grades of coal, the logical subdivision from the commercial standpoint would appear to be a classification by heat value, say above or below 12,500 B.t.u. per pound.

Semi-bituminous—Mr. Marius B. Campbell, of the U. S. Geological Survey, refers to the term used for this rank of coal as "exceedingly unfortunate, as literally it implies that this coal is half the rank of bituminous, whereas it is . . . really super-bituminous." The Survey defines this coal definitely as having a fuel ratio of from 3 to 7, the narrowest limits of any rank. The high ratio of fixed carbon allows nearly smokeless combustion to be accomplished, and the better grades have a higher heat value than coal of any other rank. Semi-bituminous coal is friable and in transportation the size of individual pieces becomes relatively small. Burned commercially for steam generation, provided the size is relatively uniform, this fact is of but little importance. While some coal of this rank is found in scattered districts of the West, it is primarily an Eastern field coal.

The Appalachian region contains the greatest amount of high-rank coal in the United States and probably in the world. Beginning near the northern boundary of Pennsylvania, in the western portion of the State, it extends southwestward through West Virginia, touching Maryland and Virginia on their western borders, passing through southeastern Ohio, eastern Kentucky and central Tennessee, and ending in western Alabama, 900 miles from its northern extremity.

Along the eastern margin of the Appalachian region are a number of isolated natural fields known as "outliers." The best known of these are Broadtop, in south central Pennsylvania, Georges Creek and Upper Potomac in Maryland and West Virginia, Lookout Mountain in Georgia and Alabama, and Cahaba in Alabama.

The coal of the Appalachian region is generally of high rank but of appreciable variation in chemical composition, the fixed carbon increasing and the volatile matter decreasing in a west to east direction. The semi-bituminous coals are found on the eastern margin of this region.

There is no classification of bituminous or semi-bituminous coal as to size that holds good in all localities. The American Society of Mechanical Engineers suggests the following grading:

Eastern Bituminous Coals—

(A) Run of mine coal; the unscreened coal taken from the mine.

(B) Lump coal; that which passes over a bar-screen with openings 1¼ inches wide.

(C) Nut coal; that which passes through a bar-screen with 1¼-inch openings and over one with ¾-inch openings.

(D) Slack coal; that which passes through a bar-screen with ¾-inch openings.

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Steamer "Herman Winter," 2300 Indicated Horse-power, Babcock & Wilcox Boilers
Owners, Eastern Steamship Lines, Inc., Boston, Mass.

Combustion of Coal—The composition of coal varies over such a wide range, and the methods of firing have to be altered so greatly to suit the various coals and the innumerable types of furnaces in which they are burned, that any instructions given for the handling of different fuels must of necessity be of the most general character. For each kind of coal there is some method of firing which will give the best results for each individual set of conditions. General rules can be suggested, but the best results can be obtained only by following such methods as experience and practice show to be the best suited to the specific conditions.

The question of draft is an all important factor. If this be insufficient, proper combustion is impossible, as the suction in the furnace will not be great enough to draw the necessary amount of air through the fuel bed and the gases may pass off only partially consumed. On the other hand, an excessive draft may cause losses due to the excess quantities of air drawn through holes in the fire. Where coal is burned, however, there are rarely complaints from excessive draft, as this can be and should be regulated by the boiler damper to give only the draft necessary for the particular rate of combustion desired.

As the variation in character of bituminous coals is so great, any rules set down for their handling must be general. The difficulties in burning bituminous coals with economy and with little or no smoke increase as the content of fixed carbon in the coal decreases. It is their volatile content which causes the difficulties and it is essential that the furnaces be designed to handle properly this portion of the coal. The fixed carbon will take care of itself, provided the volatile matter is properly burned.

William Kent, in his "Steam Boiler Economy," described the action of bituminous coal after it is fired as follows: "The first thing that the fine fresh coal does is to choke the air spaces existing through the bed of coke, thus shutting off the air supply which is needed to burn the gases produced from the fresh coal. The next thing is a very rapid evaporation of moisture from the coal, a chilling process, which robs the furnace of heat. Next is the formation of water-gas by the chemical reaction, $C + H_2O = CO + 2H$, the steam being decomposed, its oxygen burning the carbon of the coal to carbonic oxide, and the hydrogen being liberated. This reaction takes place when steam is brought in contact with highly heated carbon. This also is a chilling process, absorbing heat from the furnaces. The two valuable fuel gases thus generated would give back all the heat absorbed in their formation if they could be burned, but there is not enough air in the furnace to burn them. Admitting extra air through the fire door at this time will be of no service, for the gases being comparatively cool cannot be burned unless the air is highly heated. After all the moisture has been driven off the coal, the distillation of hydrocarbons begins, and a considerable portion of them escapes unburned, owing to the deficiency of hot air and to their being chilled by the relatively cool-heating surfaces of the boiler. During all this time great volumes of smoke are escaping from the chimney, together with unburned hydrogen, hydrocarbons and carbonic oxide, all fuel gases, while at the same time soot is being deposited on the heating surface, diminishing its efficiency in transmitting heat to the water."

To burn these gases distilled from the coal, it is necessary that they be brought into contact with air sufficiently heated to cause them to ignite, that sufficient space be allowed for their mixture with the air, and that sufficient time be allowed for their complete combustion before they strike the boiler heating surfaces, since these

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Steamer "Crescent City," 1600 Indicated Horse-power, Babcock & Wilcox Boilers
Owners, Pittsburgh Steamship Company, Cleveland, O.

surfaces are comparatively cool and will lower the temperature of the gases below their ignition point. The air drawn through the fire by the draft suction is heated in its passage and heat is added by radiation from the hot brick surfaces of the furnace, the air and volatile gases mixing as this increase in temperature is taking place. Thus in most instances is the first requirement fulfilled. The elements of space for the proper mixture of the gases with the air and of time in which combustion is to take place, should be taken care of by sufficiently large combustion chambers.

As stated, the difficulty of burning bituminous coals successfully will increase with the increase in volatile matter. This percentage of volatile matter will affect directly the depth of coal bed to be carried and the intervals of firing for the most satisfactory results. The variation in the fuel over such wide ranges makes it impossible to state definitely the thickness of fires for all classes, and experiment with the class of fuel in use is the best method of determining how that particular fuel should be handled. The following suggestions, which are not to be considered in any sense hard and fast rules, may be of service for general operating conditions for hand firing:

Semi-bituminous coals, such as Pocahontas, New River, Clearfield, etc., require fires from 10 to 14 inches thick; fresh coal should be fired at intervals of 10 to 20 minutes and sufficient coal charged at each firing to maintain a uniform thickness. Bituminous coals from the Pittsburgh Region require fires from 4 to 6 inches thick, and should be fired often in comparatively small charges. Kentucky, Tennessee, Ohio and Illinois coals require a thickness from 4 to 6 inches.

In general, as thin fires are found necessary, the intervals of firing should be made more frequent and the quantity of coal fired at each interval smaller. As thin fires become necessary, due to the character of the coal, the tendency to clinker will increase if the thickness be increased over that found to give the best results.

In hand firing the "spreading" method is ordinarily used. But little fuel is fired at one time and is spread evenly over the fuel bed from front to rear. Where there is more than one firing door, the doors should be fired alternately. The advantage of alternate firing is that the whole surface of the fire is not blanketed with green coal, and steam is generated more uniformly than if all doors were fired at one time. Again, a better combustion results due to the burning of more of the volatile matter directly after firing than where all doors are fired at one time.

The tendency in hand-fired practice is to work bituminous coal less than formerly. A certain amount of raking and slicing may be necessary, but in general the less the fire is worked the better the results.

Mechanical stokers have been used in marine practice only to a very limited extent, due to the headroom available preventing satisfactory furnace heights being used. For similar reasons, at the present time, so far as is known, pulverized coal has not been burned under marine boilers except experimentally. It is possible that developments will occur in both of these fields of combustion methods which will enable such methods to be used satisfactorily in marine work.

Smoke—From the standpoint of meeting the requirements of city or community ordinances, the question of smoke in marine practice is not of importance, as in stationary work. The question, however, as indicating and affecting combustion, is of the greatest importance.

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Smoke and Efficiency of Combustion*—Though there is perhaps no phase of combustion that has been so fully discussed as that which results in the production of smoke, the common understanding of the loss from this source is at best vague and based in part, at least, on misconception. For this reason a brief consideration of smoke is included here, regardless of the amount of data on the subject available elsewhere.

Of the numerous and frequently unsatisfactory definitions of smoke that have been offered, that of the Chicago Association of Commerce Committee in its report on "Smoke Abatement and the Electrification of Railway Terminals in Chicago" is perhaps the best. This report defines smoke as "the gaseous and solid products of combustion, visible and invisible, including .... mineral and other substances carried into the atmosphere with the products of combustion."

From the standpoint of combustion loss, it is necessary to lay stress on the term "visible and invisible." The common conception of the extent of loss is based on the visible smoke, and such conception is so general that practically all, if not all, smoke ordinances are based on the visibility, density or color of escaping stack gases. As a matter of fact, the color of smoke, which is imparted to the gases by particles of carbon, cannot be taken as an indication of the stack loss. The invisible or practically colorless gases issuing from a stack may represent a combustion loss many times as great as that due to the actual carbon present in the gases, and but a small amount of such carbon is sufficient to give color to large volumes of invisible gases which may or may not represent direct combustion losses. A certain amount of color may also be given to the gases by particles of flocculent ash and mineral matter, neither of which represents a combustion loss. The amount of such material in the escaping gases may be considerable where stokers of the forced draft type are used and heavy overloads are carried.

The carbon or soot particles in smoke from solid fuels are not due to the incomplete combustion of the fixed carbon content of the fuel. They result rather from the non-combustion or incomplete combustion of the volatile and heavy hydrocarbon constituents, and it is the wholly or partially incomplete combustion of these constituents that causes smoke from all fuels, solid, liquid or gaseous.

If the volatile hydrocarbons are not consumed in the furnace and there is no secondary combustion, there will, of course, be a direct loss resulting from the non-combustion of these constituents. While certain of these unconsumed gases may appear as visible smoke, the loss from this source cannot be measured with the ordinary flue-gas analysis apparatus and must of necessity be included with the unaccounted losses.

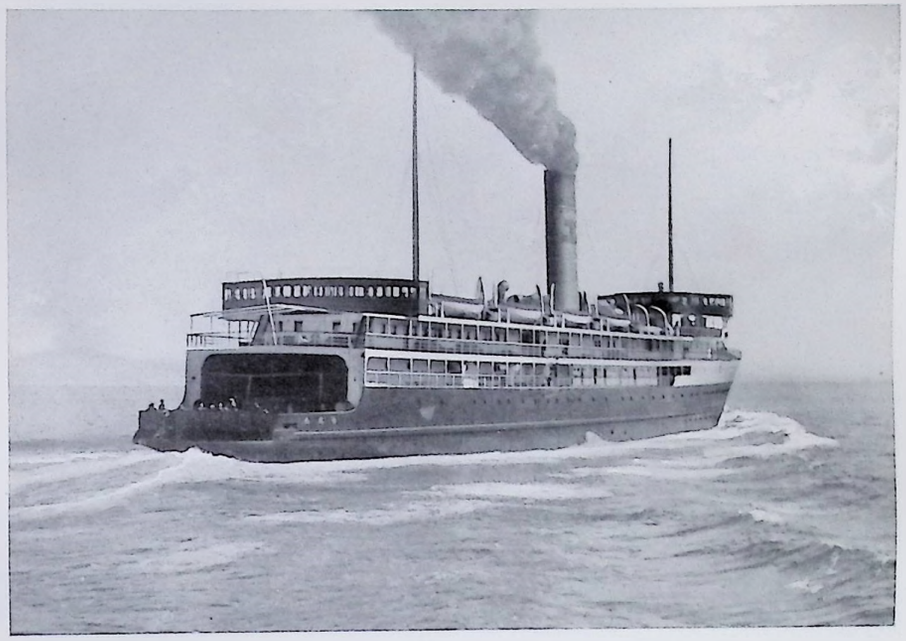
Where the combustion of the hydrocarbon constituents is incomplete a portion of the carbon component ordinarily appears as soot particles in the smoke. In the burning of hydrocarbons, the hydrogen constituent unites with oxygen before the carbon; for example, in the case of ethylene (C_2H_4):

$$C_2H_4 + 2O = 2H_2O + 2C$$

If, after the hydrogen is "satisfied," there is sufficient oxygen present with which that carbon component may unite, and temperature conditions are right, such combination will take place and combustion will be complete. If, on the other hand, sufficient oxygen is not present, or if the temperature is reduced below the combining temperature of carbon and oxygen, the carbon will pass off unconsumed as soot, regardless of the amount of oxygen present.

The direct loss from unconsumed carbon passing off in this manner is probably rarely in excess of one per cent of the total fuel burned, even in the case of the densest smoke. The loss due to unconsumed or partially consumed volatile hydrocarbons, on the other hand, though not indicated by the appearance of the gases issuing from a funnel, may represent an appreciable percentage of the total fuel fired.

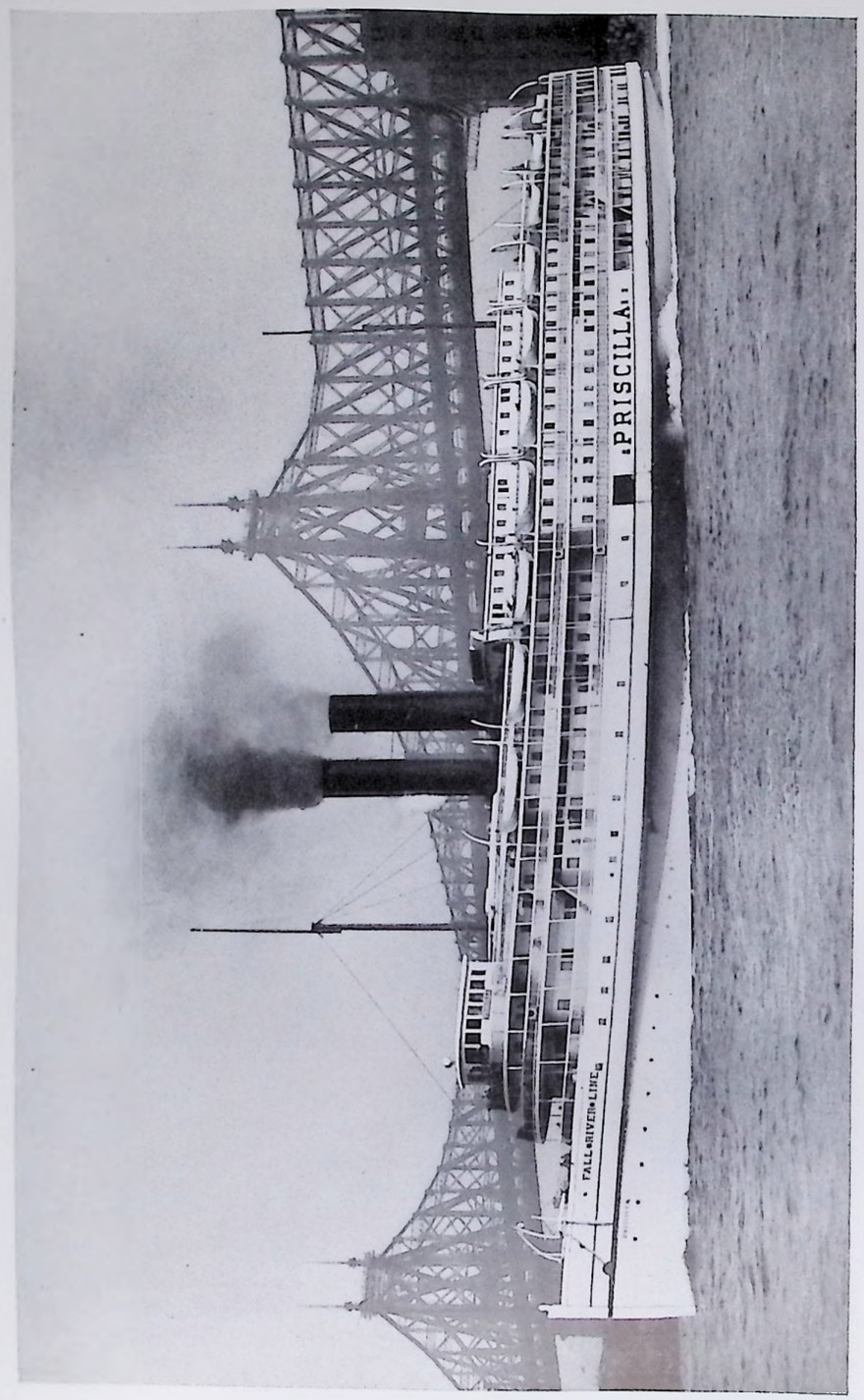
* From "Principles of Combustion in the Steam Boiler Furnace," by A. D. Pratt; published by The Babcock & Wilcox Company.



Ferry Steamers "Tsugaru Maru" and "Matsumae Maru," 6000 Shaft Horse-power.
Babcock & Wilcox Boilers
Owners, Imperial Japanese Government Railways

While the loss represented by the visible constituents of smoke leaving a funnel may ordinarily be considered negligible, there is a loss due to the presence of unconsumed carbon and tarry hydrocarbons in the products of combustion which, while not a direct combustion loss, may result in a much greater loss in efficiency than that due to visible smoke. These constituents adhere to the boiler heating surfaces and, acting as an insulating layer, greatly reduce the heat-absorbing ability of such surfaces. From the foregoing it is evident that the funnel losses indicated by smoke, whether visible or invisible, result almost entirely from improper combustion. Assuming a furnace of proper design and fuel-burning apparatus of the best, there will be no objectionable smoke where there is good combustion. On the other hand, a smokeless funnel is not necessarily indicative of proper or even of good combustion. Large quantities of excess air in diluting the products of combustion naturally tend toward a smokeless funnel, but the possible combustion losses corresponding to such an excess air supply have been shown.

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Steamer "Priscilla" 9000 Indicated Horse-power, Babcock & Wilcox Boilers
Owners, The New England Steamship Company, New York, N. Y.

ANALYSIS AND HEATING VALUE OF FUELS

THE heating value of a fuel may be determined either by the direct measurement of the heat evolved during combustion in a calorimeter, or by computation based on a chemical analysis of the fuel. The former method is decidedly preferable where accuracy is desired and should be used wherever possible.

A fuel analysis may be reported in two different forms. A proximate analysis, which is the more common, determines only the percentages of moisture, fixed carbon, volatile matter and ash, without determining the ultimate composition of the volatile matter. Such an analysis is possible only with solid fuels.

An ultimate analysis reduces the fuel to its elementary constituents of carbon, hydrogen, oxygen, nitrogen, sulphur, ash and moisture. Such an analysis requires the services of a competent chemist and laboratory equipment that should include a calorimeter for the direct determination of the heat value of the fuel. On the other hand, there are apparently many instances in which an ultimate fuel analysis is available without a heat value for the particular fuel used, and for this reason the methods of computation of heat value are given here. The accuracy of a computed as compared with a calorimetric heat value is given hereafter, and is appreciably greater than the heat value computed from a proximate analysis. The methods employed in an ultimate analysis may be found in books on engineering chemistry, or in Technological Paper No. 8 of the Bureau of Mines.

An ultimate analysis, while resolving the fuel into its elementary constituents, does not reveal how these may have been combined in the fuel as analyzed. The manner of their combination undoubtedly has a direct effect upon the calorific value of the fuel, since fuels having almost identical ultimate analysis show a difference in heating values as determined by calorimetric test.

The relative values of the constituents of solid fuels, as reported in a proximate or ultimate analysis, are indicated by Table 31, taken from the analyses made in connection with the coal testing plant of the United States Geological Survey at the Louisiana Purchase Exposition, St. Louis.

Proximate and ultimate fuel analyses are reported on both a wet and a dry fuel basis. Inasmuch as the latter is the basis ordinarily accepted for comparative results, it would appear to be the proper basis on which results should be given. When an analysis is given on the wet basis, it may be readily converted to the dry basis by dividing the percentage of the different constituents by one minus the percentage of moisture, reporting the moisture content separately. This method, which is the same for either a proximate or an ultimate analysis, is represented by an example of the latter:

	Moist Fuel	*Dry Fuel*
C	83.95	84.45
H	4.23	4.25
O	3.02	3.04
N	1.27	1.28
S	0.91	0.91
Ash	6.03	6.07
	100.00	100.00
Moisture	0.59	0.59
	100.00	

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Marine Steam (1928)
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Dredge "Geyser," Equipped with Babcock & Wilcox Boilers
Owners, Stewart, Kerbaugh, Shanley Co.

TABLE 31

SHOWING RELATION BETWEEN PROXIMATE AND ULTIMATE ANALYSES OF COAL

State	Field or Bed	Mine	Proximate Analysis		Ultimate Analysis					Common to Proximate and Ultimate Analyses	
			Volatile Matter	Fixed Carbon	Carbon	Hydrogen	Oxygen	Nitrogen	Sulphur	Ash	Moisture
Ala.	Horse Creek	Ivy Coal & Iron Co., No. 8	31.81	53.90	72.02	4.78	6.45	1.66	.80	14.29	2.56
Ark.	Huntington	Central C. & C. Co., No. 3	18.99	67.71	76.37	3.90	3.71	1.49	1.23	13.30	1.99
Ill.	Pana or No. 5	Clover Leaf, No. 1	37.22	45.64	63.04	4.49	10.04	1.28	4.01	17.14	13.19
Ind.	No. 5, Warrick Co.	Electric	41.85	44.45	68.08	4.78	7.56	1.35	4.53	13.70	9.11
Ky.	No. 11, Hopkins Co.	St. Bernard, No. 11	41.10	49.60	72.22	5.06	8.44	1.33	3.65	9.30	7.76
Pa.	"B" or Lower Kittanning	Eureka, No. 31	16.71	77.22	84.45	4.25	3.04	1.28	.91	6.07	.56
Pa.	Indiana Co.		29.55	62.64	79.86	5.02	4.27	1.86	1.18	7.81	2.90
W.Va.	Fire Creek	Rush Run	22.87	71.56	83.71	4.64	3.67	1.70	.71	5.57	2.14

Calculations from an Ultimate Analysis—The first formula for the calculation of heating values from the composition of a fuel as determined from an ultimate analysis is due to Dulong, and this formula, slightly modified, is the most commonly used today. Other formulæ have been proposed, some of which are more accurate for certain specific classes of fuel, but all have their basis in Dulong's formula, the accepted modified form of which is:

Heat units in B.t.u. per pound of dry fuel =

$$14{,}600C + 62{,}000\left(H - \frac{O}{8}\right) + 4000S \quad (23)$$

where C, H, O, and S are the proportionate parts by weight of carbon, hydrogen, oxygen and sulphur.

Assume a coal of the composition given. Substituting in this formula (*23*):

Heating value per pound of dry coal:

$$= 14{,}600 \times 0.8445 + 62{,}000\left(0.0425 - \frac{0.0304}{8}\right) + 4000 \times 0.0091 = 14{,}765 \text{ B.t.u.}$$

This coal, when tested in a calorimeter, showed a heat value of 14,843 B.t.u. per pound, and from a comparison of these values the accuracy of computation as against calorimeter determination is indicated. The investigations of Lord and Haas in this country, Mahler in France, and Bunte in Germany, all indicate that the use of Dulong's formula gives results which approximate calorimeter tests closely, except in the case of cannel coal, lignite, peat and wood. The difference with these fuels unquestionably results from the manner of combination of the hydrocarbons in the fuel, previously discussed.

A number of methods of computing heat values of solid fuels based on proximate analyses have been offered. William Kent's method is based upon the relation between heat value and fixed carbon in the combustible matter, while Goutal gives the fixed carbon a fixed value and the heat value of

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the volatile matter is considered as a function of its percentage referred to combustible. These two methods give results that approximate each other, but the accuracy, as compared with calorimetric determinations, is less than that of the use of Dulong's formula, particularly with fuels containing less than 60 per cent of fixed carbon in the combustible matter. Where no method of determining the heat value of the fuel other than the proximate analysis is possible, perhaps the use of the isothermal-isocalorific proposed by the Bureau of Mines Technical Paper No. 93, "Graphic Studies of Ultimate Analyses of Coal," will be found helpful.

High and Low Heat Values—The heat value of a fuel, as defined, is known as the "higher" heat value and is ordinarily accepted as the standard in this country. In the case of fuel containing hydrogen, and this includes practically all fuels in commercial use, there is another value known as the "lower," "net" or "available" heat value, in the determination of which an attempt is made to allow for the latent heat recovered in the condensation of the water vapor formed in the combustion of hydrogen. For example, in the calorimetric determination of the heat value of a fuel containing hydrogen, the products of combustion are cooled to approximately the temperature of the original mixture, say 62 degrees Fahrenheit. In cooling the products to this temperature, the water vapor formed by the combustion of hydrogen is condensed, and the result, expressed in B.t.u., after being corrected for sulphur and like factors, *i. e.*, the higher heat value, includes the latent heat of water vapor given up in such condensation.

If the lower value be represented by H_p' and the weight of water produced per pound of fuel by w, the lower heat value may be determined from

$$H_p' = H_p - wr \qquad (24)$$

where H_p equals the higher heat value and r is a factor which varies with the percentage of hydrogen in the fuel, the amount of air or oxygen used in combustion, the moisture in the air and the temperature to which the products of combustion are cooled in the calorimeter. Too frequently r is simply taken as the latent heat of steam, either at 32 degrees or 212 degrees, though in calorimetric work neither of these temperatures is likely to occur.

With the lower heat value so defined, the difference between the higher and the net value will obviously be the total heat of the steam or water vapor as it escapes, less the sensible heat of an equivalent weight of water at the temperature of the fuel and of the oxygen before combustion takes place.

The lower heat value is in common use in Great Britain and in most foreign countries. In this country, the higher value is almost universally accepted, and this is the standard recommended by The American Society of Mechanical Engineers.

Any attempt to make use of the lower heat value introduces a source of possible error in the proper temperature for use in computation, and advocates of the use of this value are not in entire agreement as to the proper methods of such computations.

To sum up, a theoretically perfect absorption of heat after combustion would condense all of the vapor formed in the burning of hydrogen, and since the efficiency of any apparatus is based upon the performance of a theoretically perfect machine, it appears only logical to charge against the apparatus what would be secured from the theoretically perfect. Further, in the report of the performance of any apparatus, a heat balance offers a method of determining and expressing any loss due to the burning of hydrogen, and no such test or performance report can be accepted as

reliable unless accompanied by a heat balance or by data from which a heat balance may be computed.

Proximate Analysis—The proximate analysis of a fuel, as stated, reports the percentage by weight of moisture, fixed carbon, volatile matter and ash. The laboratory sample of fuel as obtained during the boiler trial,* reduced to about one quart, should be carefully sealed and shipped for analysis. The method of such an analysis is as follows:

Moisture. The moisture content is determined in one of two ways. In the first method, a differentiation is made between surface moisture as represented by that driven off by air drying (1) and inherent moisture, while the second (2) determines only total moisture.

(1) The sample is dried in an oven at a temperature of from 30 to 35 degrees Centigrade (86 to 98 degrees Fahrenheit), through which a current of warm air is drawn, until a constant weight of the sample is obtained. Such loss represents the air-drying loss, and is assumed to correct for difference in conditions between the time the sample is taken and that of this moisture determination.

The sample, after this drying, is quickly pulverized, quartered, and a portion reduced to 60 mesh. One gram of this portion is used for the determination of the residual or inherent moisture, by drying the sample in a shallow dish placed in a glycerine-jacketed drying oven through which dry air at a temperature of 105 degrees Centigrade (221 degrees Fahrenheit) is drawn. By this method, the loss so determined represents the loss at 105 degrees Centigrade, while the sum of the air drying loss and this loss represents the total moisture loss.

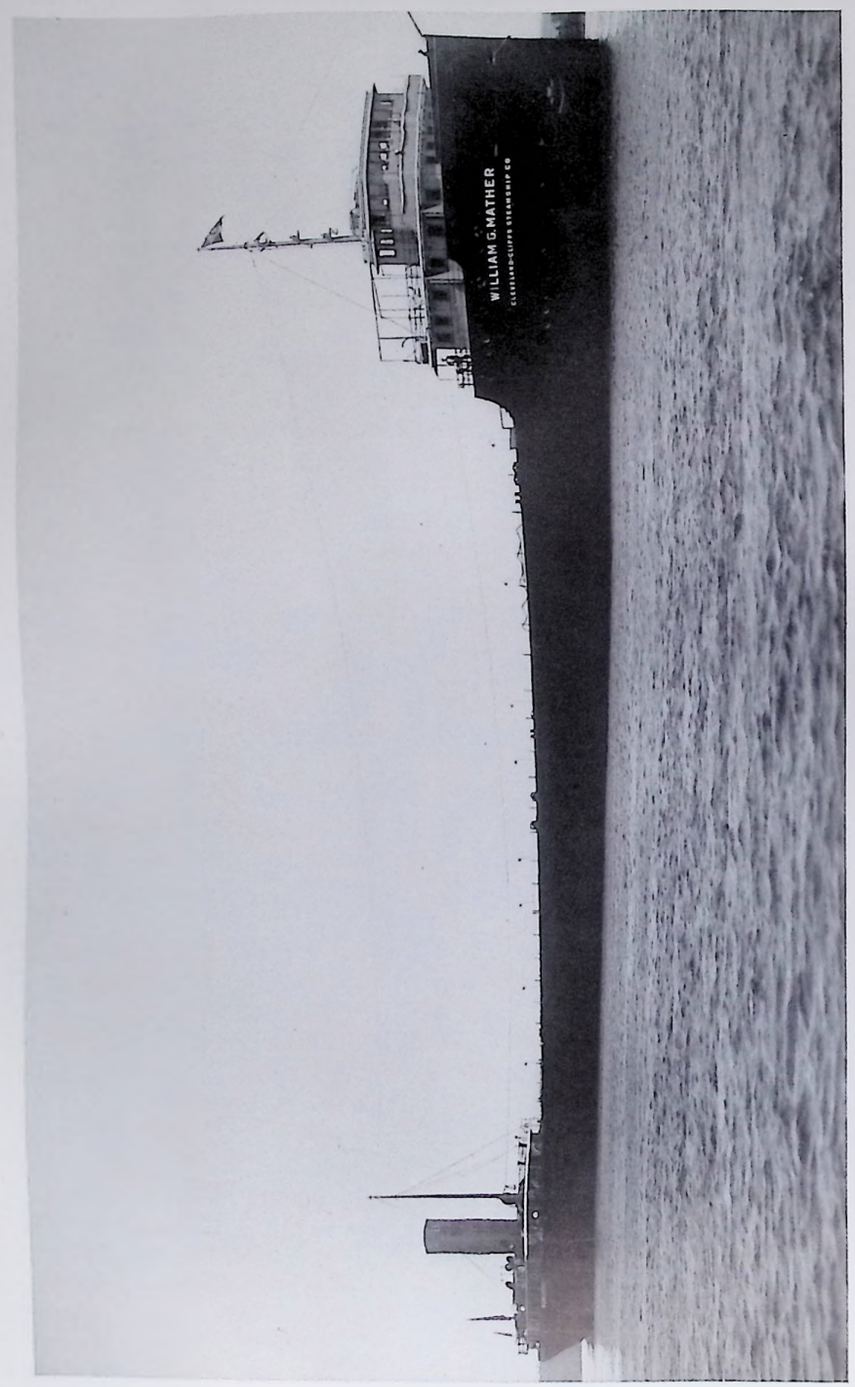
This method is that recommended by the Bureau of Mines, but if the results of a boiler test are to be reported, as has been recommended, on a dry fuel basis, it has no advantage over the second method.

(2) The entire sample is dried at a temperature of 105 degrees Centigrade (221 degrees Fahrenheit) to constant weight, the loss in weight representing moisture. The dry sample is then crushed, quartered through a riffle, and a portion of it ground to 60 mesh in an Abbé ball mill using flint pebbles. This sample is used for the analysis.

Volatile Matter. The determination of the volatile constituent of the coal is best accomplished by the use of an electric furnace of special design, the temperature of which can be maintained constant and which will give a crucible temperature of 950 degrees Centigrade (1742 degrees Fahrenheit). The temperature should be controlled by means of a thermocouple, the tip of which almost touches the bottom of the crucible. The sample is placed in a 10-cubic centimeter capsule-covered platinum crucible and placed in the furnace described, the heating period being seven minutes. The crucible is then removed from the furnace, cooled and weighed. If the sample used in this determination was of the fuel air dried, as described under the first method of moisture determination, the loss in weight represents the volatile matter plus the moisture at 105 degrees Centigrade, while if the sample was completely dried, as described in the second method of moisture determination, the loss in weight represents directly the volatile content.

* For methods of sampling see chapter on Efficiency and Capacity of Boilers.

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Steamer "William G. Mather," 3000 Indicated Horse-power, Babcock & Wilcox Boilers and Superheaters
Owners, Cleveland-Cliffs Steamship Company, Cleveland, O.

Ash. If the first method of moisture determination has been used, the ash constituent is determined from the same sample used in such determination and not from that used for volatile matter. If the second moisture method has been used it is best to take a fresh sample for ash. The sample, in a crucible, is placed in a cool muffle furnace and the temperature gradually raised to 750 degrees Centigrade (1382 degrees Fahrenheit) and ignited. The ash should be occasionally stirred, and the temperature maintained until a constant weight is reached.

Fixed Carbon. The fixed carbon is the difference between 100 and the sum of the ash and volatile matter on a dry basis, determined as described.

Table 32 gives the proximate analysis and calorific value of a number of representative coals found in the United States*.

Calorimetric Determination of Heat Value (Solid and Liquid Fuels)—While an ultimate or a proximate analysis is useful in the determination of the general characteristics of a fuel and may be used for an approximate determination of its heating value, wherever the accurate determination of the efficiency of a steam generating unit is involved, such heat value should be determined by means of a fuel calorimeter. In such an apparatus the fuel is completely burned under oxygen pressure, the heat evolved in combustion absorbed by water, and the amount of such heat calculated from the rise in water temperature.

There are a number of fuel calorimeters available for heat value determinations, which vary somewhat in design and construction, but which in principle of operation and manipulation are essentially the same. The following description of the parts and operation of a calorimeter applies in general to any of the standard types.

Calorimeter Bomb. The combustion bomb is made of steel, lined with material which is not attacked by nitric and sulphuric acid or other products of combustion. The bomb is equipped with a small pan attached internally, in which the sample of the fuel to be tested is placed, and with two connections to which are attached leads, passing through the jacket cover to a source of current by which ignition of the sample tested is accomplished. Actual ignition is brought about by the use of a fuse wire passing through the fuel sample in the pan, such wire completing the electric circuit.

Calorimeter Bucket. The bucket is a vessel in which the bomb is placed. It is filled with a definite weight of water, the rise in temperature of which, when the sample is burned, is the measure of the heat evolved.

Calorimeter Jacket. The calorimeter is equipped with an outer water jacket which has a cover to prevent air currents from reaching the calorimeter. The jacket is either a plain jacket filled with water at approximately room temperature, or a so-called adiabatic water jacket. Where the former is used, a radiation correction must be made in the computation of results from the observed data.

Where an adiabatic water jacket is used, radiation is eliminated by raising the temperature of the jacket water at the same rate as the temperature of the water in the calorimeter increases after combustion of the fuel sample. With the temperature of the water in the jacket kept the same as that of the water in the calorimeter, as the temperature of the latter increases following ignition of the sample, there is no exchange of heat between the calorimeter bucket and its surrounding jacket and the

* Bulletins Nos. 22, 123 and 193 of the Bureau of Mines give many thousands of analyses of mine and car samples collected between 1904 and 1919, together with descriptions of the beds where mined.

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TABLE 32

APPROXIMATE COMPOSITION AND CALORIFIC VALUE OF CERTAIN TYPICAL AMERICAN COALS

State	County	Field, Bed or Vein	Mine	Size	Moisture	Proximate Analysis (Dry Coal)			B.t.u. Per Pound Dry Coal	Authority
						Volatile Matter	Fixed Carbon	Ash		
		SEMI-BITUMINOUS								
Md.	Alleghany	Big Vein, George's Crk.			3.50	21.33	72.47	6.20	14682	B. & W. Co.
Md.	Alleghany	George's Creek			3.63	16.27	76.93	6.80	14695	B. & W. Co.
Md.	Alleghany	George's Creek			2.28	19.43	77.44	6.13	14793	B. & W. Co.
Md.	Alleghany	George's Creek	Ocean No. 7	Mine run	1.13	. . .	. . .	. . .	14451	B. & W. Co.
Md.	Alleghany	Cumberland			1.50	17.26	76.65	6.09	14700	. . . . .
Md.	Garrett		Washington No. 3.	Mine run	2.33	14.38	74.93	10.49	14033	U. S. Geo. S*
Pa.	Bradford		Long Valley		1.55	20.33	68.38	11.29	12965	. . . . .
Pa.	Tioga		Antrim		2.19	18.43	71.87	9.70	13500	. . . . .
Pa.	Cambria	"B" or Miller	Sonman Shaft C. Co.		3.40	20.70	71.84	7.46	14484	N. Y. Ed. Co.
Pa.	Cambria	"B" or Miller	Henrietta		1.23	18.37	75.28	6.45	14770	So. Eng. Co.
Pa.	Cambria	"B" or Miller	Penker		3.64	21.34	70.48	8.18	14401	B. & W. Co.
Pa.	Cambria	"B" or Miller	Lancashire		4.38	21.20	70.27	8.53	14453	B. & W. Co.
Pa.	Cambria	Lower Kittanning	Penn. C. & C. Co. No. 3	Mine run	3.51	17.43	75.69	6.88	14279	U. S. Geo. S.
Pa.	Cambria	Upper Kittanning	Valley	Mine run	3.40	14.89	75.03	10.08	14152	B. & W. Co.
Pa.	Clearfield	Lower Kittanning	Eureka	Mine run	5.90	16.71	77.22	6.07	14843	U. S. Geo. S.
Pa.	Clearfield		Ghem	Mine run	3.43	17.53	69.67	12.80	13744	B. & W. Co.
Pa.	Clearfield		Osceola		1.24	25.43	68.56	6.01	13589	B. & W. Co.
Pa.	Clearfield	Reynoldsville			2.91	21.55	69.03	9.42	14685	B. & W. Co.
Pa.	Clearfield	Atlantic-Clearfield		Mine run	1.55	23.36	71.15	5.94	13963	Whitham
Pa.	Huntington	Barnet & Fulton	Carbon	Mine run	4.50	18.34	73.06	8.60	13770	B. & W. Co.
Pa.	Huntington		Rock Hill	Mine run	5.91	17.58	73.44	8.99	14105	B. & W. Co.
Pa	Somerset	Lower Kittanning	Kimmelton	Mine run	3.09	17.84	70.47	11.69	13424	U. S. Geo. S.
Pa.	Somerset	"C" Prime Vein	Jenner	Mine run	9.37	16.47	75.76	7.77	14507	P. R. R.
W. Va.	Fayette	New River	Rush Run	Mine run	2.14	22.87	71.56	5.57	14959	U. S. Geo. S.
W. Va	Fayette	New River	Loup Creek		0.55	19.36	78.48	2.16	14975	Hill
W. Va.	Fayette	New River		Slack	6.66	20.94	73.16	5.90	14412	B. & W. Co.
W. Va.	Fayette	New River		Mine run	2.16	17.82	75.66	6.52	14786	B. & W. Co.
W. Va.	Fayette	New River	Rush Run	Mine run	0.94	22.16	75.85	1.99	15007	B. & W. Co.
W. Va.	McDowell	Pocahontas No. 3	Zenith	Mine run	4.85	17.14	76.54	6.32	14480	U. S. Geo. S.
W. Va.	McDowell	Tug River	Big Sandy	Mine run	1.58	18.55	76.44	4.91	15170	U. S. Geo. S.
W. Va.	Mercer	Pocahontas	Mora	Lump	1.74	18.55	75.15	6.30	15015	U. S. Geo. S.
W. Va.	Mineral	Elk Garden			2.10	15.70	75.40	8.90	14195	B. & W. Co.
W. Va.	McDowell	Pocahontas	Flat Top	Mine run	0.52	24.02	74.59	1.39	14490	B. & W. Co.
W. Va.	McDowell	Pocahontas	Flat Top	Slack	3.24	15.33	77.60	7.07	14653	B. & W. Co.
W. Va.	McDowell	Pocahontas	Flat Top	Lump	3.63	16.03	78.04	5.93	14956	B. & W. Co.
		BITUMINOUS								
Ala.	Bibb	Cahaba	Hill Creek	Mine run	6.19	28.58	55.60	15.82	12576	B. & W. Co.
Ala.	Jefferson	Pratt	Pratt No. 13		4.29	25.78	67.68	6.54	14482	B. & W. Co.
Ala.	Jefferson	Pratt	Warner	Mine run	2.51	27.80	61.50	10.70	13628	U. S. Geo. S.
Ala.	Jefferson		Coalburg	Mine run	0.94	31.34	65.65	3.01	14513	B. & W. Co.
Ala.	Walker	Horse Creek	Ivy C. & I. Co. No. 8	Nut	2.56	31.82	53.89	14.29	12937	U. S. Geo. S.
Ala.	Walker	Jagger	Galloway C. Co. No. 5	Mine run	4.83	34.65	51.12	14.03	12976	U. S. Geo. S.

*U. S. Geological Survey

State	County	Field, Bed or Vein	Mine	Size	Moisture	Proximate Analysis (Dry Coal)			B.t.u. Per Pound Dry Coal	Authority
						Volatile Matter	Fixed Carbon	Ash		
Ill.	Christian	Pana	Penwell Col.	Lump	8.05	43.67	49.97	6.36	10900	Jones
Ill.	Franklin	No. 6	Benton	Egg	8.31	34.52	54.05	11.43	11727	U. S. Geo. S.
Ill.	Franklin	Big Muddy	Zeigler	¾ inch	13.28	31.97	57.37	10.66	12857	U. S. Geo. S.
Ill.	Jackson	Big Muddy			4.85	31.55	62.19	6.26	11466	Breckenridge
Ill.	La Salle	Streator			8.40	41.76	51.42	6.82	11727	Breckenridge
Ill.	La Salle	Streator	Marseilles	Mine run	12.98	43.73	49.13	7.14	10899	B. & W. Co.
Ill.	Macoupin	Nilwood	Mine No. 2	Screenings	13.34	34.75	44.55	20.70	10781	B. & W. Co.
Ill.	Macoupin	Mt. Olive	Mine No. 2	Mine run	13.54	41.28	46.30	12.42	10807	U. S. Geo. S.
Ill.	Madison	Belleville	Donk Bros.	Lump	13.47	38.69	48.07	13.24	12427	U. S. Geo. S.
Ill.	Madison	Glen Carbon		Mine run	9.78	38.18	51.52	10.30	11672	Bryan
Ill.	Marion		Odin	Lump	6.20	42.91	49.06	8.03	11880	Breckenridge
Ill.	Perry	Du Quoin	Willis	Mine run	7.22	33.06	53.97	12.97	11352	U. S. Geo. S.
Ill.	Sangamon		Pawnee	Slack	4.81	41.53	39.62	18.85	10220	Jones
Ill.	St. Clair	Standard	Nigger Hollow	Mine run	14.39	32.90	44.84	22.26	11059	B. & W. Co.
Ill.	St. Clair	Standard	Maryville	Mine run	15.71	38.10	41.10	20.80	10999	B. & W. Co.
Ill.	Williamson	Big Muddy	Daws	Mine run	8.17	34.33	52.50	13.17	12643	U. S. Geo. S.
Ill.	Williamson	Carterville or No. 7	Carterville		4.66	35.65	56.86	7.49	12286	Univ. of Ill.
Ill.	Williamson	Carterville or No. 7	Burr	Nut, Pea and Sl.	11.91	33.70	55.90	10.40	12932	B. & W. Co.
Ind.	Brazil	Brazil	Gartside	Block	2.83	40.03	51.97	8.00	13375	Stillman
Ind.	Clay		Louise	Block	0.83	39.70	52.28	8.02	13248	Jones
Ind.	Green	Island City		Mine run	6.17	35.42	53.55	11.03	11916	Dearborn
Ind.	Knox	Vein No. 5	Tecumseh	Mine run	10.73	35.75	54.46	9.79	12911	B. & W. Co.
Ind.	Parke	Vein No. 6	Parke Coal Co.	Lump	10.72	44.02	46.33	9.65	11767	U. S. Geo. S.
Ind.	Sullivan	Sullivan No. 6	Mildred	Washed	16.59	42.17	48.44	9.59	13377	U. S. Geo. S.
Ind.	Vigo	Number 6	Fontanet	Mine run	2.28	34.95	50.50	14.55	11920	Dearborn
Ind.	Vigo	Number 7	Red Bird	Mine run	11.62	41.17	46.76	12.07	12740	U. S. Geo. S.
Ky.	Bell	Straight Creek	Str. Ck. C. & C. Co.	Mine run	2.89	36.67	57.24	6.09	14362	U. S. Geo. S.
Ky.	Hopkins	Bed No. 9	Earlington	Lump	6.89	40.30	55.16	4.54	13381	Ky. State Col.
Ky.	Hopkins	Bed No 9	Barnsley	Mine run	7.92	40.53	48.70	10.77	13036	U. S. Geo. S.
Ky.	Hopkins	Vein No. 14	Nebo	Pea and Slack	8.02	31.91	54.02	14.07	12448	B. & W. Co.
Ky.	Johnson	Vein No. 1	Miller's Creek	Mine run	5.12	38.46	58.63	2.91	13743	U. S. Geo. S.
Ky.	Mulenburg	Bed No. 9	Pierce	Pea and Slack	9.22	33.94	52.18	13.88	12229	B. & W. Co.
Ky.	Pulaski		Greensburg		2.80	26.54	63.58	9.88	14095	N. Y. Ed. C.
Ky.	Webster	Bed No. 9		Pea and Slack	7.30	31.08	60.72	8.20	13600	B. & W. Co.
Ky.	Whitley		Jellico	Nut and Slack	3.82	31.82	58.78	9.40	13175	B. & W. Co.
Ohio	Athens	Hocking Valley	Sunday Creek	Slack	12.16	34.64	53.10	12.26	12214	. . . . .
Ohio	Belmont	Pittsburg No. 8	Neff Coal Co.	Mine run	5.31	38.78	52.22	9.00	12843	U. S. Geo. S.
Ohio	Columbiana	Middle Kittanning	Palestine		2.15	37.57	51.80	10.63	13370	Lord & Haas
Ohio	Coshocton	Middle Kittanning	Morgan Run	Mine run		41.76	45.24	13.00	13239	B. & W. Co.
Ohio	Guernsey	Vein No. 7	Little Kate		6.19	33.02	59.96	7.02	13634	B. & W. Co.
Ohio	Hocking	Hocking Valley		Lump	6.45	39.12	50.08	10.80	12700	Lord & Haas
Ohio	Hocking	Hocking Valley			2.60	40.80	47.60	11.60	12175	Jones
Ohio	Jackson	Brookville	Superior Coal Co	Mine run	7.59	38.45	43.99	17.56	11704	U. S. Geo. S.

APPROXIMATE COMPOSITION AND CALORIFIC VALUE OF CERTAIN TYPICAL AMERICAN COALS—CONTINUED

State	County	Field, Bed or Vein	Mine	Size	Moisture	Proximate Analysis Dry (Coal)			B. t. u. Per Pound Dry Coal	Authority
						Volatile Matter	Fixed Carbon	Ash		
Ohio	Jackson	Lower Kittanning	Superior Coal Co.	Mine run	8.99	41.43	50.06	8.51	13113	U. S. Geo. S.
Ohio	Jackson	Quakertown	Wellston	¾ inch	3.38	35.26	54.18	7.56	12506	Hill
Ohio	Jefferson	Pittsburg or No. 8	Crow Hollow	¾ inch	4.04	40.08	52.27	9.65	13374	U. S. Geo. S.
Ohio	Jefferson	Pittsburg or No. 8	Rush Run No. 1	¾ inch	4.74	36.08	54.81	9.11	13532	U. S. Geo. S.
Ohio	Perry	Hocking	Congo	Slack	6.41	38.33	46.71	14.96	12284	B. & W. Co.
Ohio	Stark	Massillon		Nut and Slack	6.67	40.02	46.46	13.52	11860	B. & W. Co.
Ohio	Vinton	Brookville or No. 4	Clarion	Slack	2.47	42.38	50.39	6.23	13421	U. S. Geo. S.
Pa.	Allegheny	Pittsburgh 3rd Pool			1.77	32.06	57.11	10.83	13205	Carpenter
Pa.	Allegheny	Monongahela	Turtle Creek	¾ inch	1.75	36.85	53.94	9.21	13480	Lord & Haas
Pa.	Allegheny	Pittsburgh	Bertha	Slack	2.61	35.86	57.81	6.33	13997	U. S. Geo. S.
Pa.	Cambria		Beach Creek	Mine run	3.01	32.87	55.86	11.27	13755	B. & W. Co.
Pa.	Cambria	Miller	Lincoln	Slack	5.39	30.83	61.05	8.12	13600	B. & W. Co.
Pa.	Clarion	Lower Freeport		Lump	0.54	35.93	57.66	6.41	13547	Whitham
Pa.	Fayette	Connellsville		Screenings	1.85	28.73	63.22	7.95	13775	B. & W. Co.
Pa.	Greene	Youghiogheny			1.25	32.60	54.70	12.70	13100	P. R. R.
Pa.	Greene	Westmoreland			11.12	31.67	55.61	12.72	13100	
Pa.	Indiana		Iselin	Mine run	2.70	29.33	63.56	7.11	14220	B. & W. Co.
Pa.	Jefferson		Punxsutawney	Mine run	3.38	29.33	64.93	5.73	14781	B. & W. Co.
Pa.	Lawrence	Middle Kittanning			0.70	37.06	56.24	6.70	13840	Lord & Haas
Pa.	Mercer	Taylor			4.18	32.19	55.55	12.26	12820	B. & W. Co.
Pa.	Washington	Pittsburgh	Ellsworth		2.46	35.35	58.46	6.19	14013	U. S. Geo. S.
Pa.	Washington	Youghiogheny	Anderson		1.00	39.29	54.80	5.91	13729	Jones
Pa.	Westmoreland	Pittsburgh	Scott Haven	¾ inch	4.06	32.91	59.78	7.31	13934	B. & W. Co.
Tenn.	Campbell	Jellico		Lump	1.80	37.76	62.12	1.12	13846	U. S. Navy
Tenn.	Claiborne	Mingo			4.40	34.31	59.22	6.47		U. S. Geo. S.
Tenn.	Marion		Etna		3.16	32.98	56.59	10.43		
Tenn.	Morgan	Brushy Mt.			1.77	33.46	54.73	11.87	13824	B. & W. Co.
Tenn.	Scott	Glen Mary No. 4	Glen Mary		1.53	40.80	56.78	2.42	14625	Ky. State Col.
Va.	Henrico		Gayton		0.82	17.14	74.92	7.91	14363	B. & W. Co.
Va.	Lee	Darby	Darby	1½ inch	4.35	38.46	56.91	4.63	13939	U. S. Geo. S.
Va.	Lee	McConnel	Wilson	Mine run	3.35	36.35	57.88	5.77	13931	U. S. Geo. S.
Va.	Wise	Upper Banner	Coburn	3½ inch	3.05	32.65	62.73	4.62	14470	U. S. Geo. S.
Va.	Rockingham		Clover Hill			31.77	57.98	10.25	13103	
Va.	Russel	Clinchfield			2.00	35.72	56.12	8.16	14200	
Va.		Monongahela	Bennmont			32.00	59.90	8.10	13424	Carpenter
W. Va.	Harrison	Pittsburg	Ocean	Mine run	2.47	39.35	52.78	7.87	14202	U. S. Geo. S.
W. Va.	Harrison		Girard	Nut, Pea and Sl.		36.66	57.49	5.85	14548	B. & W. Co.
W. Va.	Kanawha	Winifrede	Winifrede		1.05	32.74	64.38	2.88	14111	Hill
W. Va.	Kanawha	Keystone	Keystone	Mine run	2.21	33.29	58.61	8.10	14202	U. S. Geo. S.
W. Va.	Logan	Island Creek		Nut and Slack	1.12	38.61	55.91	5.48	14273	Hill
W. Va.	Marion	Fairmont	Kingmont		1.90	35.31	57.34	7.35	14198	U. S. Geo. S.
W. Va.	Mingo	Thacker	Maritime		0.68	31.89	63.48	4.63	14126	Hill
W. Va.	Mingo	Glen Alum	Glen Alum	Mine run	3.02	33.81	59.45	6.74	14414	U. S. Geo. S.
W. Va.	Preston	Bakerstown			4.14	29.09	63.50	7.41	14546	U. S. Geo. S.
W. Va.	Putnam	Pittsburg	Black Betsy	Bug Dust	7.41	32.84	53.96	13.20	12568	B. & W. Co.
W. Va.	Randolph	Upper Freeport	Coalton	Lump and Nut	2.11	29.57	59.93	10.50	13854	U. S. Geo. S.

necessity for a cooling or radiation correction is obviated. This is of importance in scientific calorimetric work as tending toward increased accuracy, while in commercial work it is of advantage in that it eliminates the necessity of thermometer readings prior to ignition and following the maximum temperature of calorimeter water after combustion, which are necessary in the computation of the cooling correction.

The temperature of the cooling jacket is changed with the temperature of the calorimeter, either by means of an electric current or by an adjustable hot and cold water supply. A form of adiabatic water jacket developed by Dr. Farrington Daniels, of the Worcester Polytechnic Institute, utilizes the water itself as the heating medium, and by passing an electric current through such water its temperature is raised simultaneously with that of the calorimeter. With this form of jacket there is no lag due to the use of heating coils or necessity for stirring jacket water, since it is heated uniformly throughout the jacket.

Stirring Device. In order that temperature changes in the calorimeter may be represented by consistent thermometer readings, the water in the calorimeter must be stirred continuously, and for this purpose a motor-driven screw or turbine device should be used. The speed of stirring should be such that the temperature of the calorimeter water is not increased more than 0.01 degree Centigrade during a period of 10 minutes before firing, and such speed should be kept constant and not irregular.

Thermometers. The accuracy of a calorimeter depends largely upon the accuracy of the thermometers with which changes in temperature are measured. For such work, either special calorimetric thermometers or Beckman thermometers graduated to 0.01 or 0.02 degree Centigrade should be used. All thermometers should have a Bureau of Standards calibration certificate, and readings should be corrected in accordance with the corrections given by such certificate.

Oxygen. The oxygen used to bring about rapid and complete combustion should be free from all combustible matter. The amount should be not less than 5 grams per gram of coal, which amount corresponds to a pressure of approximately 20 atmospheres for the larger bombs or 30 atmospheres for small bombs. Completeness of combustion is indicated by the absence of sooty matter on the interior of the bomb after the test.

Firing Wire. The wire used for ignition may be either of iron or platinum. If the former, it should be of about No. 34 B & S gauge, and not more than 10 centimeters (preferably 5 centimeters) should be used at a time. A correction of 1600 calories per gram weight of iron burned must be deducted from the observed number of calories.

Calorimeter Standardization. During every calorimetric test, the immersed parts of the calorimeter are carried through the same range of temperature as the water in the calorimeter bucket. The weight of water, the temperature of which would be increased one degree for an increase of one degree in the temperature of the combined parts of the immersed calorimeter, is known as the "water equivalent" of the calorimeter. Expressed in another manner, the water equivalent is the amount of heat absorbed by the immersed calorimeter for one degree rise in the water contained therein. While the water equivalent can be computed from the weight of the various parts of the apparatus and the specific heat of the different materials entering into the construction of such different parts, it is best determined by the combustion of standard combustion samples supplied by the Bureau of Standards. The required

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City of Baltimore Fireboat "Torrent," Equipped with Babcock & Wilcox Boilers

water equivalent, as determined by such method, is equal to the weight of the sample multiplied by its heat of combustion per gram and divided by the corrected rise in calorimeter water temperature. The methods to be followed in the burning of these standard combustion samples as to amount of water, oxygen, firing wire, radiation correction, etc., are the same as for the combustion of solid or liquid fuel samples.

*Manipulation of Calorimeter.** (1) The sample, ground as described in the methods of making proximate analyses, should be thoroughly mixed in the containing bottle and approximately one gram taken out and weighed in the pan or crucible in which it is to be burned.

(2) Bomb Preparation. The ignition wire, if of iron, should be weighed and, whether of iron or platinum, connected to the platinum terminals. This wire must be so placed that it is in contact with the fuel sample, but should not touch the fuel pan. The wire should be clean. Approximately 0.5 cubic centimeter of water is placed in the bottom of the bomb to saturate with moisture the oxygen used for combustion and for titration after the test.

(3) Oxygen. The oxygen from the supply tank should be admitted to the bomb slowly, to prevent blowing the sample from the fuel pan or crucible. The pressure, indicated by a gauge forming part of the calorimeter equipment, should be from 20 to 30 atmospheres, depending upon the size of the bomb.

(4) Calorimeter Water. The calorimeter bucket should be filled with the amount of water corresponding to the water equivalent of the particular apparatus in use. Such amount naturally must be that determined in the standardization of the calorimeter.

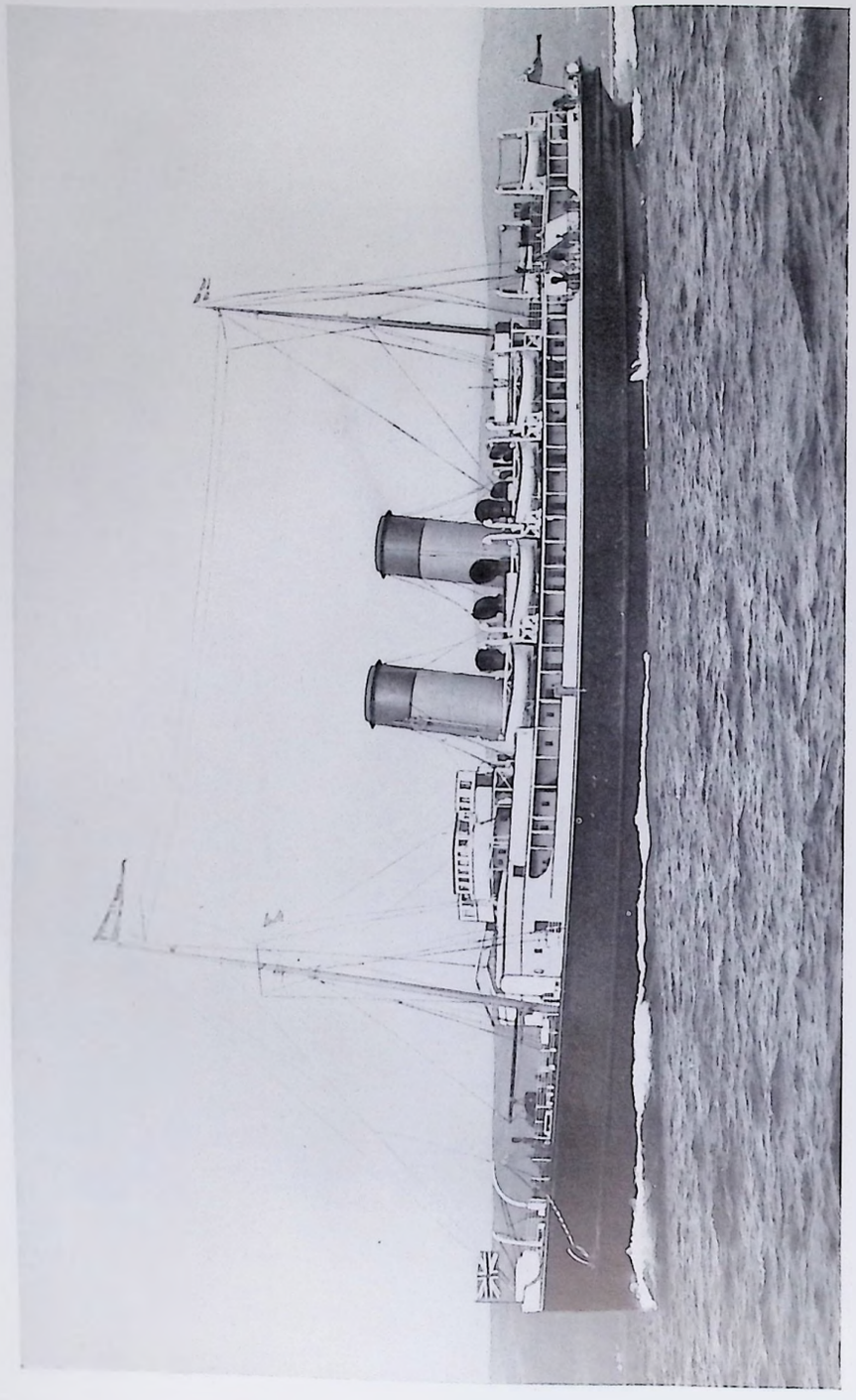
(5) Temperature. The initial temperature in the calorimeter should be such that the maximum temperature attained after combustion will not be more than one degree (and preferably not more than 0.5 degree) above that of the jacket. Under such temperature conditions, the radiation correction, while necessary, is small and tends toward accuracy in the final computation of heat value.

(6) Firing Current. The current used for firing, obtained from dry or storage cells, must be of a voltage lower than that which would result in arcing between firing connections. The circuit should be closed by a switch, which is kept in contact not over two seconds. An ammeter in the circuit or some other means should be used for the indication of the burning out of the firing wire.

(7) Observation Methods. The combustion bench in which the sample for testing has been placed, as described, is placed in the calorimeter bucket, the cover of the water jacket placed, and the stirring apparatus and thermometers so placed as not to be in contact with the bomb or bucket. The stirrer is then started, and after thermometer readings have become steady, temperatures are read for a period of five minutes at one-minute intervals. The charge is then fired by means of the switch, the exact time of firing being noted. Temperature observations after firing are observed at intervals that vary with the method used in the determination of the radiation loss. When the temperature of the calorimeter has reached its maximum, a second series of readings is taken over a five-minute period, at one-minute intervals, to determine the final cooling or radiation loss.

* Some of the methods of manipulation given apply only to calorimeters not equipped with adiabatic water jacket. Where a difference in the two types occurs they are discussed hereafter.

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STEAMER "ANGLIA," 14,000 SHAFT HORSE-POWER, BABCOCK & WILCOX BOILERS
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(8) Titration. After combustion, the bomb is opened. If incomplete combustion is indicated by traces of unconsumed combustible or sooty deposit, the test should be discarded. If combustion has been completed, the bomb should be rinsed with the water placed within it before the test and other water placed in the bomb, and such water titrated with a standard alkali solution (one cubic centimeter = 0.02173 gram HNO_3 = 5 calories), using methylorange to determine the amount of sulphuric acid formed. A correction of 230 calories per gram of nitric acid should be deducted from the total heat observed, and an additional correction of 1300 calories per gram of sulphur in the coal should be made for the excess of heat of formation in the combustion of sulphur dioxide, as occurring in the combustion of sulphur in practice. over that of aqueous nitric acid occurring in a calorimetric test.

Test Observations and Methods of Computation—The following methods of observation of thermometer readings during a calorimetric test, and those of computing heat values from such observations, are those recommended by a joint committee of the American Society for Testing Materials and the American Chemical Society, 1917. The observations are:

(1) The rate of rise in temperature in degrees per minute (r_1) for five minutes before firing.

(2) The time *a,* at which the last temperature reading is made prior to firing.

(3) The time *b,* when the rise in temperature has reached six-tenths of the total rise after firing. Such temperature may generally be determined by adding to the temperature observed before firing 60 per cent of the expected temperature rise and noting the time when this temperature is reached.*

(4) The time *c* of temperature measurements when the temperature change of the calorimeter water has become uniform, some five minutes after firing; and

(5) The final rate of cooling, r_2, as represented by temperature change per minute for the five-minute period after firing.

The rate r_1 is to be multiplied by the time *(b-a)* expressed in minutes and tenths of a minute, and this product added (or, if the temperature at the time *a* was falling, subtracted) to the thermometer reading taken at the time *a*. The rate r_2 is to be multiplied by the time *(c-b)* and this product added (subtracted if the temperature was rising at the time *c* and later) to the thermometer reading taken at the time *c*. The difference in the two temperatures so determined, with proper calibration corrections made, gives the total rise in temperature resulting from combustion. Such rise multiplied by the water equivalent of the calorimeter gives the total amount of heat, expressed in calories, liberated. This latter result corrected for the heat of the formation of nitric acid and sulphuric acid, as determined by titration, and for the heat of combustion of the firing wire, when this is included, is divided by the weight of the sample to determine the heat of combustion in calories per gram. Calories per gram multiplied by 1.8 give the heat in B.t.u. per pound.

The results determined from the above computations should be given in calories per gram or B.t.u. per pound of *dry fuel,* the methods of conversion from fuel as fired to a dry basis being the same as described for the conversion of ultimate or proximate fuel analyses.

* In practice the difference in temperature before firing and that when six-tenths of the temperature rise has occurred *(b-a)* is approximately a constant, and the value of *b* need be determined only occasionally.

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The method of computation of the heat value of a fuel from observed data is illustrated in the following example: Assume:

Water equivalent of calorimeter, 2550 grams.

Weight of full sample tested, 1.0535 grams.

Approximate rise of calorimeter temperature expected, 3.2 degrees Centigrade.

Six-tenths of approximate rise, 1.9 degrees Centigrade.

Thermometer readings as given below.

$r = 15.272 - 15.244 = 0.028 \div 5 = 0.0056$ degree per minute,

$(b - a) = 10:272 - 10:26 = 1.2$ minutes.

The corrected initial temperature is then:

$15.276 + 0.0067 = 15.283$ degrees,

$r_2 = 18.500 - 15.493 = 0.007 \div 5 = 0.0014$ degree per minute,

$(c - b) = 10:310 - 10:272 = 3.8$ minutes.

The correct final temperature is:

$18.497 + (0.0014 \times 3.8) = 18.502$ degrees.

Time of Observation	*Thermometer Readings Degrees Centigrade*	*Corrected Temperature from Bureau of Standards Certificate*
10:21	15.244	.....
10:22	15.250	.....
10:23	15.255	.....
10:24	15.261	.....
10:25	15.266	.....
10:26 *(a)*	15.272	15.276
10:27.2 *(b)*	17.200*	.....
10:31 *(c)*	18.500	18.497
10:32	18.498	.....
10:33	18.497	.....
10:34	18.498	.....
10:35	18.494	.....
10:36	18.493	.....

* The initial temperature *a* is 15.272 degrees; 60 per cent of the expected rise is 1.9 degrees. The reading for observation is thus 15.27 + 1.9 = 17.2 degrees.

The total rise in temperature, as represented by the corrected initial and final readings, is 18.502 — 15.283 = 3.219 degrees, and from the water equivalent of the calorimeter, the heat value less a deduction of heat evolved in the combustion of the firing wire and that from titration of, say, 7 calories, is, for the sample tested,

$$(2550 \times 3.219) - 7 = 8202 \text{ calories.}$$

The heat value per gram is this value divided by the weight in grams of the sample, or $8202 \div 1.0535 = 7785$ calories or, expressed in B.t.u. per pound, $7785 \times 1.8 = 14{,}103$ B.t.u.

The methods of operation of the calorimeter equipped with the ordinary water jacket, described above, in general hold for a calorimeter equipped with an adiabatic jacket. With this type of jacket the water in it must be originally at a temperature lower than that of the calorimeter, and then be brought to the latter temperature by use of the electric or other means of temperature adjustment. The temperature of the jacket water should be held within 0.1 degree of that of the calorimeter and after

ignition, this is accomplished by adjustment of such regulating device as is furnished with the calorimeter. When the temperature of the calorimeter, after ignition of the fuel sample, approaches the maximum, such maximum may be accurately determined by readings at sufficiently short intervals, and, because of the absence of radiation loss, no further readings are necessary, the firing and maximum temperature indicated being the only temperatures necessary for the computation of the heat evolved in combustion of the sample, though, of course, the corrections due to firing wire, products of combustion other than those found in practice, etc., must be made.

Where the water available for use in an adiabatic jacket of the Daniels type is "soft," and for that reason a poor electrical conductor, a small amount of soluble salts should be added to the water, preferably by passing the water for the jacket through a container of such salts before it enters the jacket.*

* For more detailed methods of analysis and determination of heat value, it is suggested that the complete report of the joint committee of the American Society of Testing Materials and the American Chemical Society on the "Methods for the Analysis of Coal," the Bureau of Mines Technical Paper No. 76, "Notes on the Sampling and Analyzing of Coal," and the same Bureau's Technical Paper No. 8, "Methods of Analyzing Coal and Coke," be studied.



Installing One of the Eight Babcock & Wilcox Boilers on the Steamer "Priscilla," at the Yards of the New England Steamship Company, Newport, R. I.

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Steamer "Maloio," 25,000 Shaft Horse-power, Babcock & Wilcox Boilers, Superheaters and Oil Burners
Owners, Matson Navigation Company, San Francisco, Cal.

LIQUID FUELS AND THEIR COMBUSTION

FUEL oil is the only liquid fuel sufficiently abundant for use in steam generation. The meaning of the term, fuel oil, has changed as time has passed, not only with respect to source of supply, but as to the ingredients of the residue from the refining process for any particular oil, in consequence of improved methods of refining. In the early days of oil burning, when the demands for the refined products were of limited extent, much of the oil burned was "crude oil" or oil in its natural state as it came from the well. Today, with the marked advance in the methods of refining and in the demand for the products of such refining, the oil in use for fuel purposes is the residue from crude oil after it has been more or less refined. Dr. Day of the United States Bureau of Mines defines fuel oil as including "all oils which are not salable for some other special purpose at a higher price than that which prevails for oils to be sold as fuel oils or to be burned under boilers."

While geologists are not in entire agreement as to the origin of petroleum, whether it be of animal, vegetable or mineral origin, the generally accepted opinion is that such origin is of organic nature. Crude oil, while varying widely in physical characteristics, may be broadly divided into three classes in accordance with predominant base, or the nature of the residue after distillation. These are: 1st, Paraffin base, which, in general, includes the lighter oils; 2nd, Asphaltic base, which ordinarily includes the heavier grade oils; 3rd, Mixed base, containing varying proportions of paraffin and asphaltic bases and including the broad class of intermediate grade oils.

While oil is found in varying quantities in practically every portion of the world, the largest supplies available today are from the United States, Mexico and Russia. The oil used in this country is entirely from our own fields and from Mexico.

The principal fields in this country from which oil is produced are:

1st. The Pennsylvania-Appalachian fields which produce light-grade crude of paraffin base.
2nd. The Lima, Indiana and Illinois fields, which produce for the most part oils similar to the Pennsylvania fields.
3rd. The Mid-continent fields, including those of Oklahoma, Kansas and part of Texas, which produce medium grades of crude oils, largely of asphaltic base, although some are of paraffin base.
4th. The Gulf fields, including the states bordering the Gulf of Mexico. These produce somewhat heavier grades of crude oil of asphaltic base.
5th. The California fields, producing medium-grade oils of asphaltic and mixed base.

All of the oils available for fuel purposes may be said to fall within the following grades:

1st. "Topped" crude, from which the light oils such as gasoline and kerosene have been removed.
2nd. Residue oils, or oils left after complete distillation, but from which water, coke and other foreign matter have been removed.
3rd. Distillates, such as gas oil, which may be mixed with the heavier reduced crudes and residuums to make them sufficiently fluid.
4th. Mixtures of the above in varying proportions.

Where oil fuel is to be used, there are certain physical characteristics that must be taken into consideration in the determination of the suitability of the available supply. These characteristics, together with the manner in which they affect suitability for use in steam generation, may be summarized as follows:

Specific gravity, or the ratio of weight of a given volume of oil to that of an equal volume of water. The specific gravity is of importance, particularly in the case of the

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lighter oils, since it has a bearing on the calorific value. In general, the higher the specific gravity, the lower the content of the lighter hydrocarbons and the lower the heat value of the oil per pound.

Specific gravity of oil fuels is ordinarily determined by a direct reading on the Baumé hydrometer scale and is expressed in degrees Baumé. In 1922–1923, an agreement was reached between the United States Bureau of Standards, the United States Bureau of Mines, and the American Petroleum Institute to call the degrees on the hydrometer scale used with petroleum and its products "degrees A. P. I." instead of degrees Baumé, the latter term being restricted to degrees on a hydrometer scale used with other liquids than petroleum. As the oils in common use become heavier, the

TABLE 33

SPECIFIC GRAVITY AND WEIGHT IN POUNDS PER GALLON OF FUEL OIL HAVING THE GIVEN BAUMÉ (A. P. I.) READINGS AT 60 DEGREES FAHRENHEIT

Degrees Baumé (A. P. I.)	Specific Gravity at 60 Degrees Fahrenheit	Pounds Per Gallon	Degrees Baumé (A. P. I.)	Specific Gravity at 60 Degrees Fahrenheit	Pounds Per Gallon
10.0	1.0000	8.331	22.5	0.9188	7.655
10.5	0.9965	8.302	23.0	0.9159	7.630
11.0	0.9930	8.273	23.5	0.9129	7.605
11.5	0.9895	8.244	24.0	0.9100	7.581
12.0	0.9861	8.215	24.5	0.9071	7.557
12.5	0.9826	8.186	25.0	0.9042	7.533
13.0	0.9792	8.158	26.0	0.8984	7.485
13.5	0.9759	8.130	27.0	0.8927	7.437
14.0	0.9725	8.102	28.0	0.8871	7.390
14.5	0.9692	8.074	29.0	0.8816	7.345
15.0	0.9659	8.047	30.0	0.8762	7.300
15.5	0.9626	8.019	31.0	0.8708	7.255
16.0	0.9593	7.992	32.0	0.8654	7.210
16.5	0.9561	7.965	33.0	0.8602	7.166
17.0	0.9529	7.939	34.0	0.8550	7.123
17.5	0.9497	7.912	35.0	0.8498	7.080
18.0	0.9465	7.885	36.0	0.8448	7.038
18.5	0.9433	7.859	37.0	0.8398	6.996
19.0	0.9402	7.833	38.0	0.8348	6.955
19.5	0.9371	7.807	39.0	0.8299	6.914
20.0	0.9340	7.781	40.0	0.8251	6.874
20.5	0.9309	7.755	41.0	0.8203	6.834
21.0	0.9279	7.730	42.0	0.8156	6.795
21.5	0.9248	7.705	43.0	0.8109	6.756
22.0	0.9218	7.680	44.0	0.8063	6.717

Baumé scale becomes more liable to error, but up to the present time its use has proven satisfactory for the purpose of comparing relative gravities. Table 33 gives the specific gravities corresponding to the Baumé scale, now also called the A. P. I. scale, together with the corresponding weights per gallon of oil.

Viscosity of oil is defined in a number of ways, but may be considered as the resistance of oil to flow or its resistance to internal movement. The viscosity of oil changes with its temperature and, accordingly, when recording viscosities, the corresponding temperature should also be given. While there are a number of viscosimeters available, The Babcock & Wilcox Company has adopted as standard in its oil-burning work the Engler apparatus. In this the viscosity is measured on a scale based on the ratio of the time required for 200 cubic centimeters of water at 68 degrees Fahrenheit to flow through an orifice to the time for the same volume of oil to pass through the same orifice.

Others of the available viscosimeters are the Saybolt Universal, the Saybolt Furol, the Redwood Standard, and the Redwood Admiralty. In reporting viscosity as determined by these viscosimeters, the results are given in terms of seconds of time taken for a given quantity of oil to flow through a calibrated orifice under standard conditions, the temperature of the oil being tested being taken at the time of test. The approximate relation between degrees Engler and the viscosities as determined by the other types of viscosimeters is given in Table 34.

TABLE 34

RELATION (APPROXIMATE) BETWEEN ENGLER DEGREES, SAYBOLT AND REDWOOD SECONDS AND KINEMATIC VISCOSITIES, AT THE SAME TEMPERATURE

FROM TESTS OF THESE INSTRUMENTS IN THE LABORATORIES OF THE TEXAS COMPANY

Engler Degrees	Saybolt Universal Seconds	Saybolt Furol Seconds	Redwood Standard Seconds	Redwood Admiralty Seconds	Kinematic Viscosity
2.5	83	13.9	74	. .	15.5
2.75	92	14.5	81	. .	17.5
3	101	15.2	88	. .	19.5
3.25	110	15.9	96	. .	21.5
3.5	118	16.5	104	. .	23.5
3.75	126	17.2	112	. .	25.3
4	135	18	119	. .	27.4
4.25	144	18.8	127	. .	29.3
4.5	152	19.5	134	. .	31.1
4.75	160	20.3	142	15.2	33.0
5	169	21	150	16	34.9
5.5	186	22.5	165	17.5	38.7
6	203	24	181	19	42.5
6.5	220	25.6	196	20.4	46.3
7	237	27.2	211	21.7	49.9
7.5	253	28.7	225	23.2	53.5
8	270	30.3	240	24.6	57

Viscosity is perhaps the most important factor in determining the suitability of available fuel oils, particularly where these are heavy. Since all oils must be heated for satisfactory handling in pumping and atomizing, it is necessary to determine to what extent the viscosity must be reduced and to what temperature the oil must be heated to accomplish such reduction. In general, the heavy oils have a high viscosity and the lighter oils a low viscosity. Oils of the same specific gravity do not necessarily have the same viscosity.

A formula developed* by the Research Laboratory of Applied Chemistry of the Massachusetts Institute of Technology gives a method of determining the pressure

* See "The Flow of Fluids through Commercial Pipe Lines," Robert E. Wilson, W. H. McAdams and M. Seltzer; Jour. Ind. & Eng. Chem. Vol. 14, p. 105.

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U. S. Light Cruiser "Detroit," Equipped with Babcock & Wilcox Mechanical Atomizing Oil Burners

required to pump oil through a pipe of a given length and size and indicates the importance of viscosity in the consideration of the availability of a given oil as fuel. This formula is

$$P = \frac{0.323\ flsv^2}{D} = \frac{0.0538\ flsQ^2}{D^5} \qquad (25)$$

where

P = pressure, pounds per square inch
f = a friction factor dependent upon viscosity
l = length of pipe in feet
s = specific gravity of oil, Baumé
D = inside diameter (actual) of pipe, inches
Q = oil per minute, gallon
v = average velocity, feet per second

In this formula the effect of viscosity is, as stated, included in the value of f, the friction factor which is represented by the formula

$$f = cz \div Dvs$$

in which D, v, and s, have the values as given above, and
c = an experimentally determined coefficient
z = absolute viscosity of the oil in centipoises = Kinematic viscosity* multiplied by its specific gravity.

To make use of formula *(25)* it is necessary first to calculate the value of $Dvs \div z$, or its equivalent $0.408Qs \div Dz$, where Q and not v is known. To illustrate: The pressure required to deliver 50 gallons of oil of 200 Saybolt Universal viscosity and 20 degrees Baumé gravity per minute through 50 feet of 1½-inch pipe. The expression $0.408\ Qs \div Dz$ becomes

$$0.408 \times 50 \times 0.933 \div (1.61 \times 41.5) = 0.276$$

TABLE 35

VALUES OF THE FRICTION FACTOR, f, FOR OILS HAVING DIFFERENT ABSOLUTE VISCOSITIES, z; DIFFERENT SPECIFIC GRAVITIES, s; FLOWING IN DIFFERENT QUANTITIES, Q (IN GALLONS PER MINUTE), THROUGH IRON OR STEEL PIPE OF DIFFERENT DIAMETERS, D (IN INCHES ACTUAL NOT NOMINAL DIMENSION).

$\frac{0.408Qs}{Dz}$	f	Straight Pipe Equivalent of Each Elbow	$\frac{0.408Qs}{Dz}$	f	Straight Pipe Equivalent of Each Elbow
0.1	0.021	27 D	1	0.0098	23 D
0.15	0.016	27 D	1.5	0.0088	24 D
0.2	0.015	25 D	2	0.0082	24 D
0.25	0.014	24 D	2.5	0.0077	24 D
0.3	0.0135	24 D	3	0.0073	24 D
0.35	0.013	23 D	4	0.0069	25 D
0.4	0.0125	23 D	5	0.0065	25 D
0.45	0.012	23 D	6	0.0063	25 D
0.5	0.0117	23 D	7	0.0061	25 D
0.6	0.0112	23 D	8	0.0060	26 D
0.7	0.0108	23 D	9	0.0058	26 D
0.8	0.0104	23 D	10	0.0057	26 D
0.9	0.0101	23 D	15	0.0053	27 D

* Kinematic viscosity is a term introduced for convenience in the study of the calibration of viscosimeters and the influence of viscosity of fluid flow. It is equal to the absolute viscosity divided by the specific gravity. The value for use in formula *(25)* may be taken from Table 34 and is approximately equal to $0.216T - (180 - T)$ where T is the Saybolt University time in seconds.

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Table 35 gives for this value a friction factor, f, of 0.0138, and formula *(25)* for the conditions outlined, becomes numerically

$$P = \frac{0.0538 \times 0.0138 \times 50 \times 0.933 \times 50^2}{1.61^5} = 8 \text{ approx.}$$

To illustrate the effect of viscosity on pressure required for pumping, it may be stated that in forcing 20 gallons of oil of 20 degrees Baumé gravity, per minute, through 100 feet of 1-inch pipe (actual diameter 1.05 inches), approximately 20 pounds is required for an oil of Saybolt Universal viscosity of 100 seconds, 27 pounds for oil of 300 seconds viscosity, and 44 pounds for oil of 500 seconds viscosity.*

Flash Point—The temperature at which inflammable gas is driven from fuel oil is called its flash point. The oil is heated in a vessel, either open or closed to the atmosphere, and the flash point determined with a spark or flame as the distilled vapor ignites. The flash point as determined with the closed vessel, or the "closed cup" flash point, is lower than that as determined by the "open cup" method, and is ordinarily accepted as standard.

The flash point is of importance in connection with the storage of oil, and the minimum allowed by the regulations of most classification societies, the Steamboat Inspection Service and the Navy is 150 degrees "closed cup."

Fire Point—If the oil is gradually heated above the flash point, a temperature is ultimately reached at which sufficient gas is given off continuously to maintain a steady flame. This temperature is the "fire point" or "burning point" of the oil.

Specific Heat—The specific heat of fuel oil, which is of importance in the design of oil heaters, varies with the chemical composition, being higher as the hydrogen content is greater and lower as the carbon content is greater. The range of specific heat is from 0.4 in the case of California oils to 0.5 for the lighter Pennsylvania oils.

The specific heat of oils from various sources as given by Holde in his "Examination of Hydrocarbon Oils" is given in Table 36.

TABLE 36

SPECIFIC HEAT OF OIL

Crude Oil From	Specific Gravity	Specific Heat
Japan	0.862	0.453
Pennsylvania	0.810	0.500
Russia	0.908	0.435
California	0.960	0.398
Bustenari, Rumania	0.842	0.462

Chemical Composition—Liquid fuel is composed of carbon, hydrogen, oxygen, nitrogen, sulphur, silt, and moisture in varying proportions, the carbon and hydrogen being the main constituents. The heat value varies from 17,000 to 20,000 B.t.u. per pound, though the ordinary range of oil available for fuel purposes is from 17,500 to 19,000 B.t.u.

Typical analyses of fuel oils together with their heat values are given in Table 37.

Where no information as to the characteristics of an oil is available aside from its "gravity" as measured by a Baumé hydrometer, a rough approximation of its heat value may be obtained from the formula

B.t.u. per pound = 18650 + 40 (Baumé deg.—10).

* The values of 300 and 500 seconds are taken from a more extended table than Table 34.

From a consideration of the foregoing characteristics of oil, it is obvious that in making a selection of available oils for fuel purposes the primary factors are: 1st, maximum heat value per dollar of cost; 2nd, viscosity such as to require no undue heating for pumping and atomization; 3rd, minimum water, sulphur and sediment content; 4th, flash point such as will meet marine regulations.

TABLE 37

COMPOSITION AND CALORIFIC VALUE OF VARIOUS OILS

Kind of Oil	Per Cent Carbon	Per Cent Hydrogen	Per Cent Sulphur	Per Cent Oxygen	Specific Gravity	Degrees Flash Point	Per Cent Moisture	B.t.u. Per Pound	Authority
California, Coaling					.927	134		17117	Babcock & Wilcox Co.
California, Bakersfield					.975			17600	Wade
California, Bakersfield			1.30		.992			18257	Wade
California, Kern River					.950	140		18845	Babcock & Wilcox Co.
California, Los Angeles			2.56					18328	Babcock & Wilcox Co.
California, Los Angeles					.957	196		18855	Babcock & Wilcox Co.
California, Los Angeles					.977		.40	18280	Babcock & Wilcox Co.
California, Monte Christo					.966	205		18878	Babcock & Wilcox Co.
California, Whittier			.98		.944		1.06	18507	Wade
California, Whittier			.72		.936		1.06	18240	Wade
California	85.04	11.52	2.45	.99*			1.40	17871	Babcock & Wilcox Co.
California	81.52	11.51	.55	6.92*		230		18667	U. S. N. Liquid Fuel Board
California			.87				.95	18533	Blasdale
California					.891	257		18655	Babcock & Wilcox Co.
California			2.45		.973		1.50†	17976	O'Neill
California			2.46		.975		1.32	18104	Shepherd
Texas, Beaumont	84.6	10.9	1.63	2.87	.924	180		19060	U. S. N. Liquid Fuel Board
Texas, Beaumont	83.3	12.4	.50	3.83	.926	216		19481	U. S. N. Liquid Fuel Board
Texas, Beaumont	85.0	12.3	1.75	.92*				19060	Denton
Texas, Beaumont	86.1	12.3	1.60		.942			20152	Sparkes
Texas, Beaumont					.903	222		19349	Babcock & Wilcox Co.
Texas, Sabine					.937	143		18662	Babcock & Wilcox Co.
Texas	87.15	12.33	0.32		.908	370		19338	U. S. N.
Texas	87.29	12.32	0.43		.910	375		19659	U. S. N.
Ohio	83.4	14.7	0.6	1.3				19580	
Pennsylvania	84.9	13.7		1.4	.886			19210	Booth
West Virginia	84.3	14.1		1.6	.841			21240	
Mexico					.921	162		18840	Babcock & Wilcox Co.
Russia, Baku	86.7	12.9			.884			20691	Booth
Russia, Novorossick	84.9	11.6		3.46				19452	Booth
Russia, Caucasus	86.6	12.3		1.10	.938			20138	
Java	87.1	12.0		.9	.923			21163	
Austria, Galicia	82.2	12.1	5.7		.870			18416	
Italy, Parma	84.0	13.4	1.8		.786				
Borneo	85.7	11.0		3.31				19240	Orde

*Includes N. †Includes silt.

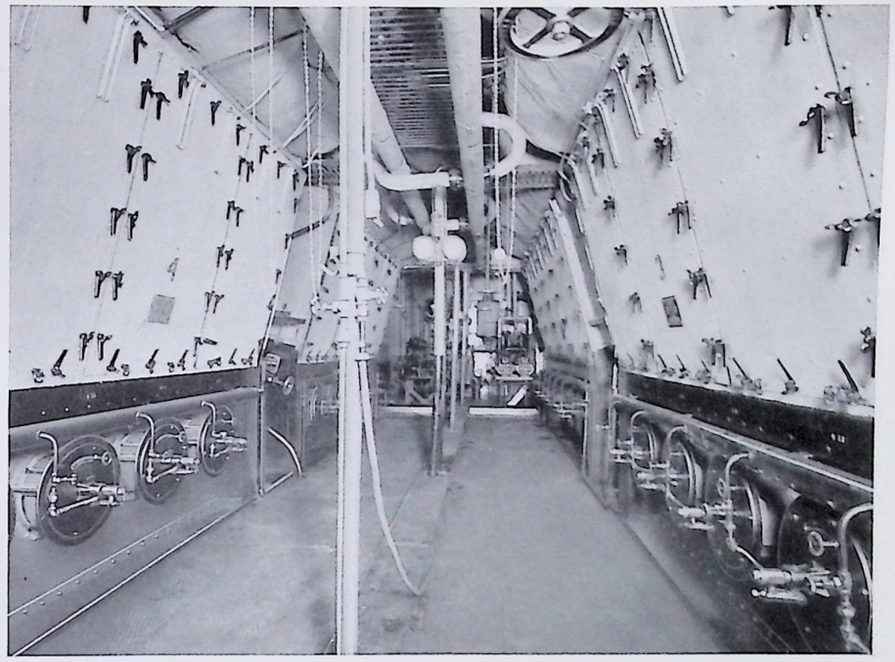
Oil Fuel *vs.* Coal—In considering boiler operation as a whole, oil fuel has many advantages over coal in the generation of steam. This is particularly true where

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U. S. S. "MAYFLOWER," EQUIPPED WITH BABCOCK & WILCOX MECHANICAL ATOMIZING OIL BURNERS

oil firing is compared with the hand firing of coal. Since, in marine practice, practically all coal-burning boilers are hand fired, the advantages in favor of oil are greater than in stationary work, where developments in stokers, pulverized fuel-burning equipment, and furnace design have been such as to minimize the advantages under numerous sets of conditions. It is, of course, possible that coal-burning equipment other than hand firing will be developed for marine work that will bring conditions to a point comparable with the best stationary practice and that many of the advantages of oil will be offset through other considerations.



FIREROOM OF STEAMER "ALAMEDA," ALASKA STEAMSHIP COMPANY
SHOWING ARRANGEMENT OF BABCOCK & WILCOX BOILERS AND CUYAMA DESIGN OIL BURNERS

EFFICIENCY—In the average marine fire room where oil is burned, the boiler efficiencies obtained in every-day operation are, without question, higher than those obtained, or perhaps obtainable, with hand-fired coal-burning boilers, the difference in favor of oil increasing as the rates of evaporation per square foot of heating surface increase. The higher efficiencies are due to the greater ease of obtaining the best combustion conditions, the absence of loss due to opening fire doors for coaling and cleaning fires and the absence of ash-pit loss, though this latter loss is at least partially offset by the loss in the burning of hydrogen in the case of oil.

The efficiencies and capacities that may be obtained when burning oil with mechanical burners, and the extent to which these efficiencies hold up at high rates of evaporation are indicated by the figures in Table 38.

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TABLE 38

PERFORMANCE OF BOILERS WITH BABCOCK & WILCOX MECHANICAL BURNERS

Thus far the efficiency of mechanical burners has been determined only in connection with the evaporative performance of the boiler to which the burners are attached. Babcock & Wilcox burners in connection with boilers built by The Babcock & Wilcox Company have shown the highest efficiencies thus far recorded, as compared with other burners on the same boilers. Columns 1 to 3 inclusive show tests on a Babcock & Wilcox Express Type Boiler at the U. S. Navy Fuel Oil Testing Plant, Philadelphia, Pa., while columns 4 to 6 inclusive show tests on a Babcock & Wilcox Header Type Marine Boiler at the Bayonne, N. J., Plant of the Company. Test Number 30–17 is particularly noteworthy, in view of the 76.15% efficiency obtained when evaporating 22.7 pounds of water per square foot of heating surface per hour, from and at 212° F., while burning oil at the rate of 1½ pounds per square foot of heating surface.

Test Number	30–14	30–20	30–17	MB 1	MB 2[a]	MB 1 2
Date	June 9, 1919	June 13, 1919	June 10, 1919	July 2, 1923	July 12, 1923	July 12, 1923
Duration of test, hours	3	2	2	3	3½	2
Type of boiler	B & W. Express	B. & W. Express	B. & W. Express	B. & W. Marine	B. & W. Marine	B. & W. Marine
Water-heating surface, square feet	7565	7565	7565	2140	2140	2140
Superheating surface, square feet	753	753	753	579	579	579
Furnace volume, cubic feet	751	751	751	230	230	230
Type of B. & W. burners used	Cuyama	Cuyama	Cuyama	Mayflower	Mayflower	Mayflower
Number of burners used	4	11	11	3	3	3
Steam Observations						
Steam pressure on drum, pounds per square inch gauge	296.1	295.9	295.6	205.3	211.9	217.6
Steam pressure below boiler stop, pounds per sq. inch gauge	290.1	272.9	245.5	204.0	204.0	190.4
Temperature of superheated steam, degrees Fahrenheit	465.3	483.6	484.5	497.7	550.2	634.0
Calorimeter temperature (lower thermometer) degrees F.	315.5	. . .	296.7	310.9	308.2	308.2
Calorimeter (inlet steam pressure), pounds per sq. inch gauge	294.1	. . .	281.6	. . .	. . .	. . .
Feedwater temperature at boiler, degrees Fahrenheit	197.6	197.2	198.0	184.0	200.8	169.3
Water actually evaporated, pounds per hour	55323	110663	153467	7248.3	14741.6	26653.
Fuel Observations						
Oil burned, pounds per hour	3726	7810	11355	512.7	1076.3	2244.5
Oil pressure at burners, pounds per square inch gauge	197.2	169.3	245.9	129.7	158.0	145.0
Oil temperature at burners, degrees Fahrenheit	119.6	120.0	120.0	248.0	254.1	260.7
Air Observations						
Dry bulb temperature, degrees Fahrenheit	89.8	100.3	90.5	108.3	105.4	108.5
Wet bulb temperature, degrees Fahrenheit	73.9	70.9	76.8	96.3	93.2	96.9
Fire-room temperature, degrees Fahrenheit	101.7	114.6	109.1	120.8	101.6	109.8
Stack temperature, degrees Fahrenheit	532	648	729	424.7	460.4	576.4
Fire-room pressure, inches of water	4.037	5.827	9.548	0	0	3.142
Air pressure in furnace, inches of water	.370	1.881	3.707	.0218	.02	1.583
Air pressure at base of stack, inches of water	–.093	+.046	+.242	. . .	. . .	¼–½
Smoke by Ringelmann scale	0–¼	0–¼	1½–2	0–¼	0–¼	. . .

PERFORMANCE OF BOILERS WITH BABCOCK & WILCOX MECHANICAL BURNERS—Continued

Flue Gas Analysis												
Carbon dioxide, per cent	13.02		12.44		13.03		13.9	12.6	13.8	13.4	13.2	12.4
Oxygen, per cent	3.27		4.40		3.32		2.2	3.7	2.2	2.6	2.3	3.9
Carbon monoxide, per cent	.11		.00		.09		0	0	0.	0.	0.	0.
Nitrogen by difference, per cent	83.60		83.16		83.56		83.9	83.7	84.0	84.0	83.5	83.7
Ultimate Analysis and Physical Tests of Fuel												
Carbon, per cent	86.00		85.90		85.90		84.17		84.17		84.17	
Hydrogen, per cent	11.70		11.40		11.60		11.06		11.06		11.06	
Sulphur, per cent	.65		.96		.65		2.95		2.95		2.95	
Nitrogen, per cent	.24		.29		.24		. . .		. . .		. . .	
Oxygen and undetermined, per cent	1.41		1.45		1.61		1.72		1.72		1.72	
Gravity, Baumé	26.1°		25.55[a]		25.55[a]		14° to 16°		14° to 16°		14° to 16°	
Calorific value, B. t. u. per pound	19300		19280		19300		18491		18645		18636	
Steam Results												
Superheat, degrees Fahrenheit	46.3		69.8		80.8		108.2		159.8		247.3	
Quality of steam leaving drum, per cent	99.28		. . .		98.29		99.80		99.64		99.71	
Factor of evaporation	1.1056		1.1187		1.1206		1.146		1.156		1.232	
Actual evaporation, per pound of fuel, pounds	14.848		14.169		13.515		14.138		13.697		11.875	
Equivalent evaporation, per pound of fuel, pounds	16.416		15.851		15.145		16.202		15.834		14.630	
Actual evaporation per square foot water H. S., lbs. per hour	7.313		14.628		20.286		3.387		6.889		12.455	
Equivalent evaporation per sq. ft. of water H. S., lbs. per hour	8.085		16.364		22.732		3.882		7.963		15.344	
Oil Results												
Oil burned per square foot of water H. S., pounds per hour	0.493		1.032		1.501		0.240		0.503		1.049	
Oil burned per cubic foot of furnace volume, pounds per hour	4.961		10.399		15.120		2.219		4.659		9.716	
Oil burned per burner per hour, pounds	931.5		710.0		1032.3		170.9		358.8		748.2	
Air Results												
Relative humidity, per cent	47.0		21.5		52.1		66		64		66	
Heat Balance	B. t. u.	P. C.	B. t. u.	P. C.	B. t. u.	P. C.	B. t. u.	P. C.	B. t. u.	P. C.	B. t. u.	P. C.
Heat absorbed by boiler	15930	82.54	15382	79.78	14697	76.15	15722	85.02	15365	82.41	14197	76.18
Loss by imperfect combustion	73	.38	0	0.0	60	.31	0	0.0	0	0.0	0	0.0
Loss with dry chimney gases	1703	8.82	2222	11.53	2453	12.71	1248	6.75	1390	7.46	1946	10.44
Loss with moisture from hydrogen in fuel	1296	6.72	1306	6.77	1374	7.12	1159	6.27	1195	6.41	1242	6.66
Loss with moisture in air	47	.24	41	.21	79	.41	86	.47	82	.44	133	.71
Radiation and error	251	1.30	329	1.71	637	3.30	276	1.49	613	3.28	1118	6.01
Total	19300	100.00	19280	100.00	19300	100.00	18491	100.00	18645	100.00	18636	100.00
Boiler, superheater and furnace efficiency, per cent	82.54		79.78		76.15		85.02		82.41		76.18	

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Steamer "American Legion," 12,000 Shaft Horse-power. Babcock & Wilcox Boilers, Superheaters and Oil Burners
Owners, Munson Steamship Line, New York, N. Y.

Capacity—Oil-fired boilers can be operated at rates of evaporation that cannot be approached by hand-fired, coal-burning units. Because of the ease of obtaining these, greater steaming capacity and higher speeds are practicable.

While it is true that much higher capacities are obtainable with oil than with coal hand-fired, it might be stated that in stationary practice the capacities obtainable and obtained with coal burned on stokers and in pulverized form are as high or even higher than obtained with oil in marine practice. In stationary work, the average capacities with stoker or pulverized fuel-fired boilers are probably appreciably higher than in oil-fired marine boilers, except in Navy work.

Labor—The saving in labor in the oil-fired boiler room as compared with the hand-fired coal boiler room is great because of the elimination of coal passers and reduction in number of firemen.

Control and Flexibility—The control of operating conditions with oil fuel is much simpler than in the case of hand-fired coal-burning boilers. This results from the greater ease of regulating combustion conditions to meet variations in load and to the greater ease of starting and stopping fires.

Up to the present time in marine practice with oil fuel, regulation of combustion conditions has been almost entirely by hand. While the number of marine installations having automatic regulation is small, compared with the great number of installations using hand regulation, the results obtained appear to justify the outlay for the automatic control.

Storage and Handling—Oil can be stored without danger of deterioration or of spontaneous combustion. While the latter may be prevented in the case of coal by the use of proper storage methods, all coal tends to disintegrate and deteriorate in heat value when stored, almost regardless of the methods used. The handling of oil from storage into the ship and to the oil-burning equipment is much more simple than the handling of coal, and there is no ash and refuse resulting from the former fuel that must be removed. Further, on shipboard, oil can be stored in spaces such as double bottoms that are otherwise unused.

Cleanliness—The cleanliness in handling and using oil as compared to that in handling coal and ashes is too obvious to need comment. The tendency of boiler heating surfaces to foul with properly operated oil-fired furnaces is, in general, not so great as with coal firing, regardless of operation. This leads to a decrease in labor required for blowing tubes, and to a somewhat better efficiency of heat absorption due to relatively cleaner exterior heating surfaces.

Bunker Space—A ton of coal occupies approximately 43 cubic feet of space, while a ton of oil occupies approximately 38 cubic feet, or about 10 per cent less. Based on relative heat values and increased efficiency in favor of oil fuel, weight for weight, the oil fuel will produce some 40 per cent more steam than coal. This represents a very appreciable increase in steaming radius for a given bunker capacity in favor of oil.

Smoke—With properly operated oil-fired furnaces, smoke can be practically eliminated; or, if necessary, combustion conditions may be made such that a dense smoke may be produced if desired, as in the case of naval vessels laying down a smoke screen.

Cost—Many tables have been prepared to show comparative costs of generating steam with oil and with coal. Such tables, however, are based either on the relative heating values of the two fuels without reference to the difference in efficiency, or take

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into consideration the variation in efficiency while maintaining a fixed calorific value of the fuels. There are so many factors entering into the cost of the generation of a given amount of steam, aside from the heat value of the fuel or the boiler efficiency shown, that no table of comparative costs can be anything but the most approximate guide, and in comparing costs each installation, with its individual operating characteristics, should be considered as a problem by itself. However, as a guide to a first approximation, and based on the fact that 200 gallons of oil equal one ton of coal, the following rule is interesting: "When the price of coal in dollars per ton is twice the price of oil in cents per gallon, the fuel cost is the same."*

Burning Oil Fuel—The requirements for the successful burning of oil fuel and the methods of meeting such requirements may be summarized as follows:

1st. The atomization of the oil must be thorough, this requirement being met by the selection of the proper type of burner.

2nd. When atomized, the oil must be brought into contact with a sufficient amount of air for combustion in a manner that assures a thorough mixture of oil and air, and at the same time the amount of air so supplied must be kept at a minimum. Proper methods of introducing air into the furnace and means of controlling this air enable this requirement to be met.

3rd. The furnace must be of such form as to maintain the temperatures essential for high efficiency, and built of a grade of refractory material that will stand up under high temperatures. Further, the form must be such as to assure complete combustion of the gases before they strike the boiler heating surfaces. This requirement is met by providing ample combustion space and length for flame travel.

4th. The burners must be so located relative to the boiler heating surfaces and must be so operated that there will be no localization of heat on certain portions of the surfaces, thus preventing trouble from overheating and blistering.

Oil Burners—Oil burners in use today are of two general classes, steam and mechanical atomizing. Because the steam used for atomizing in the former class is a direct loss, necessitating the evaporation of an amount of make-up equal to the steam consumption of the burner, and because of their limited capacity, steam atomizing burners are not practicable for marine work.

Mechanical Atomizing Oil Burners—Mechanical burners for atomization of the oil employ only the pressure of the oil as it comes to the burner. In practically all of the successful burners of this type, the oil is given a rapid whirling motion in a chamber at the burner tip and is liberated suddenly from a small orifice concentric with the axis of rotation. The oil is broken up into a very fine mist by the centrifugal force acting on it, this oil mist taking the form of a hollow cone. The individual minute particles of oil in this cone travel in straight lines which are elements of the cone and do not rotate after once being released from the atomizer unless the oil is picked up by the air for combustion which has been given a rotary motion in passing through the register. As it is customary in the majority of mechanical atomizing burners to admit the air in a rotating manner, the flame usually has a rotating appearance, leading to an impression held by many persons that the oil spray itself rotates independent of the air supply, which is not the case.

The Babcock & Wilcox Company manufacture several designs of mechanical oil burners, those in most general marine use being known as the Cuyama, the Lodi and the Mayflower designs, illustrated in Figs. 13 and 14.

In principle of design and operation these are alike, the main difference being in capacity, and in the case of the Mayflower burner, in the method of supplying air for

* W. M. McFarland in Presidential address to The Society of Naval Architects and Marine Engineers, 1913.

combustion. The Cuyama and Lodi burners are typical of mechanical oil burners as a class and a brief description will indicate their general features.

A conical casting is introduced through the boiler front wall and serves as an orifice for the introduction of oil and air into the furnace. To this is attached the register, which is fitted with adjustable automatic air doors. To the front of the register is attached a cast-iron cover plate, through the center of which passes a distance piece equipped at the inner end with a slightly conical impeller plate. The distance piece with the impeller plate and atomizer may be moved in or out on its axis relative to the cone, decreasing or increasing the area of the air passage around the impeller plate. The atomizer introduces the atomized oil through a central opening in the impeller plate, which has blades that give a whirling motion to the air entering the furnace around the burner tip. The cone through which the oil and air enter the furnace is also equipped with blades. These blades and the walls of the truncated cone direct such air as enters around the edge of the impeller plate along the axial lines of the burner. Where forced draft is used with mechanical burners, the register casting is within the draft duct, which is formed by the inner and outer front plates of the boiler. With the closed fire-room system of forced draft, only one front plate is necessary.

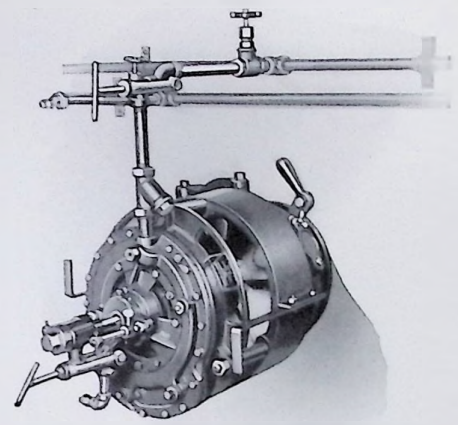


Fig. 13. Mayflower Type of Babcock Wilcox Mechanical Atomizing Oil Burner

Combustion conditions are controlled by adjustment of the boiler damper, the doors of the air registers, the position of the atomizer end relative to the cone, by oil pressure and temperature and by the air pressure supplied to the burners.

In principle of operation the Mayflower burner is as described above, the difference in design consisting of the omission of the air doors in the registers and the substitution of a turbine-driven fan which imparts to the entering air the same type and direction of rotation that is imparted by the air doors in the other designs. This design of burner is thus a complete forced-draft unit in itself, and as such does away with the necessity of large blowers, where a number of burners are used, and ducts occupying space in the fire-room. It has the further advantage of flexibility, since the air and oil are capable of fine adjustment over a wide range in individual burners. Also, when one or more of a number of burners on a boiler are not in use, no steam is used by those particular burners for providing air under pressure. In the case where air pressure is supplied in a double-front or closed fire-room, it is necessary to maintain this pressure with large blowers turning at practically normal speed, regardless of the number of burners in use.

Burner Capacity—Mechanical burners, as originally developed in marine work, were of relatively small size and the greatest quantity of oil that could be burned was approximately 300 pounds per hour per burner, even under the fire-room pressure

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Steam Yacht "Delphine II," 2800 Indicated Horse-power, Babcock & Wilcox Boilers and Oil Burners

available in naval work. It was desirable to increase the capacity of individual burners and to increase such capacity with a draft lower than that necessary with the early burners, without reducing the efficiency of combustion. It is along these lines that the development of mechanical oil burners has progressed.

Babcock & Wilcox mechanical burners now used in marine practice have burned a maximum of 4028 pounds of oil per burner per hour. This rate, however, was obtained with the fire-room under relatively high air pressure, such as is used in Navy practice, and with oil brought to the burner under very high pressure. For this maximum rate, the air pressure in the fire-room was 9 inches and the pressure of oil 340 pounds, neither of which would be practicable in merchant service.

As at present developed, the capacity of individual burners is dependent upon the pressure of the oil coming to the burner, the draft or air pressure available and the size of the burner tip. The capacities of the three Babcock & Wilcox burners described are probably typical of mechanical oil burners in general. While operating on natural draft, the Cuyama and small design of Mayflower burner have capacities up to 250 pounds of oil per burner per hour, while with a forced draft available of approximately 4 inches, this capacity can be increased to 800 pounds. The Lodi and larger size of Mayflower burner have, under natural draft conditions, capacities up to 400 pounds per burner per hour, while with a forced draft of approximately 4 inches, these are increased to 1500 pounds. When the Mayflower burner is operating on natural draft, the fan runs idly.

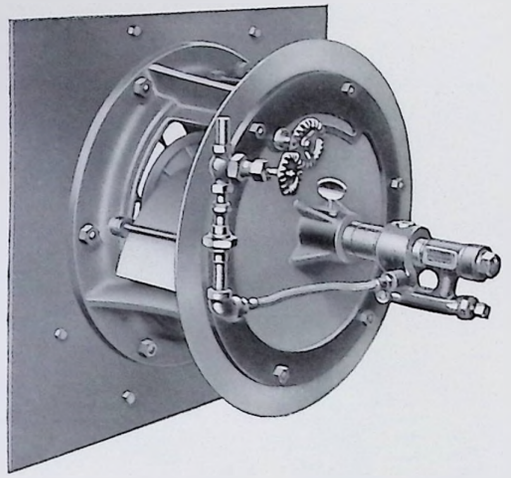


Fig. 14. Cuyama Type of Babcock & Wilcox Mechanical Atomizing Oil Burner

Oil Heating and Oil Pressures—To secure the best results from the standpoint of atomization, the oil should be heated to a point where the viscosity is from 3 to 5 degrees Engler. With the oils in common use this corresponds to a temperature of from 200 to 250 degrees Fahrenheit.

The oil pressure required at the burner varies with the capacity required, but the available pressure should not be less than 100 pounds. For high burner capacities, pressures up to 300 pounds are used.

Steam Consumption—The steam required for pumping and heating the oil will vary with different operating conditions, but ordinarily is from .75 to 1.5 per cent of the steam generated. This is not a total loss, however, as much of the heat in such steam may be returned to the system.

In the Mayflower design of burner the steam required to drive the fan turbine is approximately 1.5 to 2 per cent of the total generated, but here, too, the net loss is small as most of the heat is returned to the system in the exhaust.

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Furnace Design—With oil, as with any fuel, one of the most important factors in furnace design is furnace volume. For a given design of burner and a given size of atomizer tip, this volume controls the amount of oil that can be burned efficiently, and hence the amount of steam to be obtained from the heating surface set over the furnace in question. The space limitations on board ship make it necessary, for maximum rate of evaporation, to keep the furnace volume at the minimum which will permit the efficient burning of sufficient oil to secure this capacity. At lower capacities, of course, the furnace volume would then be generous. There is not an entire agreement among engineers as to what constitutes this minimum furnace volume. The United States Navy has set as a maximum combustion rate in Navy work 10.8 pounds of oil per cubic foot of furnace volume. This figure is probably conservative and may safely be used. There are, however, installations giving satisfactory service where, under test conditions, as high as 15 pounds of oil per cubic foot have been burned with good efficiency.

In ordinary practice, furnace volumes are made such as to give good efficiencies at normal load or cruising speed, and the burners so arranged as to allow high capacities under emergency conditions or at full speed.

While furnace volume sufficient for the required capacity is necessary, the form of the furnace is also important. This form should be such as to utilize most effectively the largest proportion of the volume, and to allow combustion to be completed before the boiler heating surfaces are encountered by the products of such combustion. Properly located, mechanical oil burners can utilize the maximum volume of a properly designed furnace. Because of the fineness of atomization and the methods employed for air admission, the combustion is completed within a short distance of the burner tip, enabling the second of the above requirements to be fulfilled.

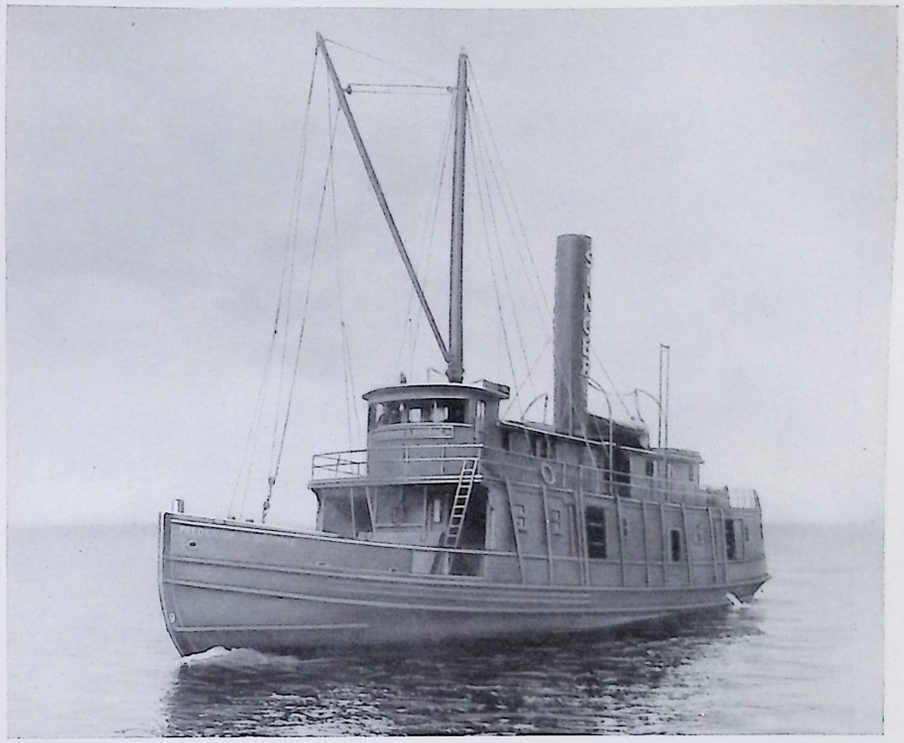
Burners should be so located relative to each other, that the flames from adjacent burners do not impinge on each other. Such impingement tends toward low efficiency and limits the amount of oil that can be burned per burner. They should be so located relative to the furnace walls and floor that there will be no impingement on the brickwork, or erosion will result. Further, with burners improperly located, there is a decided tendency for a carbon deposit to build up on the floor and walls, particularly the former.

Because of the high furnace temperatures developed with oil firing, refractories entering into the furnace construction should be of the highest grade. The furnace floor, which is subjected to the maximum temperature, needs particular attention. A floor construction which has given satisfactory service consists of two layers of split fire brick (1¼ inches thick each), of the best grade, 2½ inches high-grade calcined insulating brick, 1 inch of fire backing and a support of ¼-inch steel plate.

Operation of Burners—In starting up boilers equipped with oil burners, all burners and strainers should be thoroughly cleaned in order that there be no improper atomization resulting from an irregular flow of oil. Burners should be lighted individually with a torch made of oil-soaked waste or asbestos rope fastened to a stiff wire. No oil should be allowed to accumulate on the furnace floor as a result of poor atomization or explosions may occur when proper ignition is secured.

Upon ignition of mechanical burners, the air supplied should be reduced to a minimum, the limit to which such reduction may be extended being indicated by a pulsating action in the furnace or excessive smoke.

Good combustion conditions in the furnace are probably most readily indicated by the stack. A smokeless stack is not necessarily a sign of proper combustion, and the highest efficiency of combustion ordinarily exists if a slight, light gray haze issues from the stack.



Steam Lighter "Frederick G. Bourne," 500 Indicated Horse-power, Babcock & Wilcox Boiler and Oil Burners
Owners, Singer Manufacturing Company, New York, N. Y.

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Steamer "California," 17,000 Shaft Horse-power, Babcock & Wilcox Boilers and Superheaters
Owners, International Mercantile Marine Co. (Panama-Pacific Line), New York, N. Y.

FUNNELS AND DRAFT

THE height and diameter of a properly designed funnel depend upon the amount of fuel to be burned, its nature, and the design of the uptake, with its arrangement relative to the boiler or boilers. There are so many factors involved that as yet there has been produced no formula which is satisfactory in taking them all into consideration, and the methods used for determining funnel sizes are largely empirical. In this chapter a method sufficiently comprehensive and accurate to cover all practical cases will be developed.

Draft is the difference in pressure available for producing a flow of the gases. If the gases within a funnel be heated, each cubic foot will expand, and the weight of the expanded gas per cubic foot will be less than that of a cubic foot of the cold air outside the funnel. Therefore, the unit pressure at the funnel base due to the weight of the column of heated gas will be less than that due to a column of cold air. This difference in pressure, like the difference in head of water, will cause a flow of the gases into the base of the funnel. In its passage to the funnel the cold air must pass through the furnace or furnaces of the boilers connected to it, and it in turn becomes heated. This newly heated gas will also rise in the funnel and the action will be continuous.

The intensity of the draft, or difference in pressure, is usually measured in inches of water. Assuming an atmospheric temperature of 62 degrees Fahrenheit and the temperature of the gases in the funnel as 500 degrees Fahrenheit, and, neglecting for the moment the difference in density between the funnel gases and the air, the difference between the weights of the external air and the internal gases per cubic foot is 0.0347 pound, obtained as follows:

Weight of a cubic foot of air at 62 degrees Fahrenheit = 0.0761 pound
Weight of a cubic foot of air at 500 degrees Fahrenheit = 0.0414 pound
Difference = 0.0347 pound

Therefore, a funnel 100 feet high, assumed for the purpose of illustration to be suspended in the air, would have a pressure exerted on each square foot of its cross sectional area at its base of $0.0347 \times 100 = 3.47$ pounds. As a cubic foot of water at 62 degrees Fahrenheit weighs 62.32 pounds, an inch of water would exert a pressure of $62.32 \div 12 = 5.193$ pounds per square foot. The 100-foot funnel would, therefore, under the above temperature conditions, show a draft of $3.47 \div 5.193$ or approximately 0.67 inch of water.

The method best suited for determining the proper proportion of funnel and uptakes is dependent upon the principle that if the cross-sectional area of the funnel is sufficiently large for the volume of gases to be handled, the intensity of the draft will depend directly upon the height; therefore, the method of procedure is as follows:

1st. Select a funnel of such height as will produce the draft required by the particular character of the fuel and the amount to be burned per square foot of grate surface.

2nd. Determine the cross-sectional area necessary to handle the gases without undue frictional losses.

The application of these rules follows:

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Draft Formula—The force or intensity of the draft, not allowing for the difference in the density of the air and of the uptake gases, is given by the formula:

$$D = 0.52\,H \times P\left(\frac{1}{T} - \frac{1}{T_1}\right) \qquad (26)$$

in which

D = draft produced, measured in inches of water,
H = height of top of funnel above grate bars, in feet,
P = atmospheric pressure in pounds per square inch,
T = absolute atmospheric temperature,
T_1 = absolute temperature of funnel gases.

In this formula no account is taken of the density of the uptake gases, it being assumed that it is the same as that of air. Any error arising from this assumption is negligible in practice, as a factor of correction is applied in using the formula to cover the difference between the theoretical figures and those corresponding to actual operating conditions.

The force of draft at sea level (which corresponds to an atmospheric pressure of 14.7 pounds per square inch) produced by a funnel 100 feet high with the temperature of the air at 60 degrees Fahrenheit and that of the uptake gases at 500 degrees Fahrenheit is,

$$D = 0.52 \times 100 \times 14.7\left(\frac{1}{521} - \frac{1}{961}\right) = 0.67$$

For use in applying this formula it is convenient to tabulate values of the product

$$0.52 \times 14.7\left(\frac{1}{T} - \frac{1}{T_1}\right)$$

which we will call K, for various values of T_1. With these values calculated for assumed atmospheric temperature and pressure *(26)* becomes $D = KH$. *(27)*

For average conditions the atmospheric pressure may be considered 14.7 pounds per square inch, and the temperature 60 degrees Fahrenheit. For these values and various funnel temperatures K becomes:

Temperature Funnel Gases	*Constant K*
750	0.0084
700	0.0081
650	0.0078
600	0.0075
550	0.0071
500	0.0067
450	0.0063
400	0.0058
350	0.0053

Draft Losses—The intensity of the draft as determined by the above formula is theoretical and can never be observed with a draft gauge or any recording device. However, if the ashpit doors of the boiler are closed and there is no perceptible leakage of air through the boiler casings or uptakes, the draft measured at the funnel base will be approximately the same as the theoretical draft. The difference existing at

other times represents the pressure necessary to force the gases through the funnel against their own inertia and the friction against the sides. This difference will increase with the velocity of the gases. With the ashpit doors closed, the volume of gases passing to the funnel is a minimum and the maximum force of draft will be shown by a gauge.

As draft measurements are taken along the path of the gases, the readings grow less as the points at which they are taken are farther from the funnel, until in the boiler ashpit, with the ashpit doors open for freely admitting the air, there is little or no perceptible rise in the water of the gauge. The breeching, the boiler damper, the baffles and the tubes, and the coal on the grates all retard the passage of the gases, and the draft from the funnel is required to overcome the resistance offered by the various factors. The draft at the breeching of the boiler, where connection is made to the funnel or uptake may be 0.5 inch, while in the furnace directly over the fire it may not be over, say, 0.15 inch, the difference being the draft required to overcome the resistance offered in forcing the gases through the tubes and around the baffling.

One of the most important factors to be considered in designing a funnel is the draft required in the boiler furnace to burn the necessary amount of fuel. In the case of coal-fired boilers this represents the draft necessary to draw the air for combustion through the fuel bed, while with natural draft oil-fired boilers it represents the draft required to draw the combustion air through the air doors of the burners. In either case it may represent a large proportion of the total draft required. This is measured by the difference in furnace draft and, in the case of coal, ashpit draft, which, with properly designed ashpit doors, is zero. With oil it is the difference between furnace draft and fire-room pressure, which also is ordinarily zero. Where forced draft is used, either with coal or oil, it is measured by the sum of such pressure and furnace suction. Where forced draft is used, however, either with coal or oil, a draft suction in the furnace is ordinarily maintained such as will remove the products of combustion and protect the furnace brickwork.

Loss in Funnel—The difference between the theoretical draft as determined by formula *(26)* and the amount lost by friction in the funnel proper is the available draft, or that which the draft gauge indicates when connected to the base of the funnel. The sum of the losses of draft in the uptake, boiler and furnace must be equivalent to the available draft, and as these quantities can be determined from records of experiments, the problem of designing a funnel becomes one of proportioning it to produce a certain available draft.

The loss in the funnel due to friction of the gases can be calculated from the following formula:

$$\triangle D = \frac{fW^2CH}{A^3} \qquad (28)$$

$\triangle D$ = draft loss in inches of water,
W = weight of gas in pounds passing per second,
C = perimeter of funnel in feet,
H = height of funnel in feet,
f = a constant with the following values at sea level:
0.0015 for steel funnels, temperature of gases 600 degrees Fahrenheit.
0.0011 for steel funnels, temperature of gases 350 degrees Fahrenheit.
A = area of funnel in square feet.

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Excursion Steamer "State of Delaware," 2500 Indicated Horse-power, Babcock & Wilcox Boilers and Oil Burners
Owners, Wilson Line, Wilmington, Del.

This formula can also be used for calculating the frictional losses for uptakes, in which case, C = the perimeter of the uptakes in feet, H = the length of the uptakes in feet, the other values being the same as for funnels.

The available draft is equal to the difference between the theoretical draft from formula *(27)* and the loss from formula *(28)*, hence:

$$d' = \text{available draft} = KH - \frac{fW^2CH}{A^3} \qquad (29)$$

Height and Diameter of Funnels—From this formula *(29)* it becomes evident that a funnel of certain diameter, if it be increased in height, will produce the same available draft as one of larger diameter, the additional height being required to overcome the added frictional loss.

Combustion conditions may vary over such a wide range, resulting in corresponding variation in gas weights, even with a given fuel, that no hard and fast statement as to proper funnel diameter may be made. A method that has been found to give satisfactory results with hand-fired boilers is to allow 30 square feet of cross-sectional area per 10,000 square feet of boiler heating surface installed. With oil, where the weight of products of combination per horse-power developed is less, this area may be safely reduced to 25 square feet except in cases where very high rates of evaporation are required, in which case the 30 square feet should be maintained.

Losses in Uptakes—The loss of draft in uptakes due to friction and inertia can be calculated approximately from formula *(28)*, which was given for loss in funnels. It is to be borne in mind that C in this formula is the actual perimeter of the uptake and is least, relative to the cross-sectional area, when the section is a circle, is greater for a square section, and greatest for a rectangular section. The retarding effect of a square uptake is 12 per cent greater than that of a circular uptake of the same area, and that of a rectangular with sides as 1 and 1½, 15 per cent greater. Uptakes should be short and should have as near a square cross-section as possible. Abrupt turns are to be avoided, but as long easy sweeps require valuable space, it is often desirable to increase the height of the funnel rather than to take up more space in the boiler room. Short right-angle turns reduce the draft by an amount which can be roughly approximated as equal to 0.05 inch for each turn. The turns which the gases make in leaving the breeching of a boiler, in entering the uptake and in turning up into a funnel, should always be considered. The cross-sectional areas of the passages leading from the boilers to the funnel should be of ample size to provide against undue frictional loss. It is poor economy to restrict the size of the uptake and thus make additional funnel height necessary to overcome the added friction. The general practice is to make uptake areas the same or slightly larger than that of the stack; these should be, preferably, at least 20 per cent greater.

Loss in Boilers—The draft loss through the boiler proper depends upon the arrangement of the boiler heating surface and baffles, the class of fuel burned, the quantity of fuel burned, and the combustion conditions under which it is burned. A variation in any one of these factors with the others remaining constant might cause an appreciable variation in the draft loss through the boiler. These factors can enter into so many combinations that it is obvious that no statement of draft loss through boilers can be made that will be in any way comprehensive, and each combination of boiler, fuel and combustion conditions must be considered independently.

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Loss in Furnace—The draft loss in the furnace or through the fuel bed varies between wide limits. The air necessary for combustion must pass through the interstices of the coal on the grate. Where these are large, as is the case with broken coal, but little pressure is required to force the air through the bed; but if they are small, as with bituminous slack or small sizes of anthracite, a much greater pressure is needed. If the draft is insufficient, the coal will accumulate on the grates and a dead smoky fire will result, with the accompanying poor combustion; if the draft is too great, the coal may be rapidly consumed on certain portions of the grate, leaving the fire thin in spots and a portion of the grates uncovered, with the resulting losses due to an excessive amount of air.

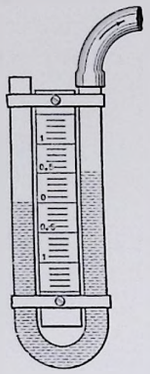


Fig. 15. U-tube Draft Gauge

With oil fuel, the furnace draft loss is that due to the drawing of combustion air through the air registers and varies only with the capacity at which burners are being operated.

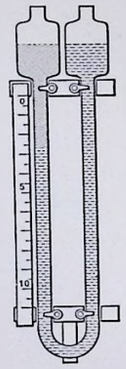


Fig. 16. Barrus Draft Gauge

Draft Required for Different Fuels—For every kind of fuel and rate of combustion there is a certain draft with which the best general results are obtained. A comparatively light draft is best with the free burning bituminous coals, and the amount to use increases as the percentage of volatile matter diminishes and the fixed carbon increases. Numerous other factors, such as the thickness of fires, the percentage of ash and the air spaces in the grates, bear directly on this question of the draft best suited to a given combustion rate. The effect of these factors can be found only by experiment. It is almost impossible, therefore, to show by one set of curves the furnace draft required at various rates of combustion for all of the different conditions of fuel, etc., that may be met.

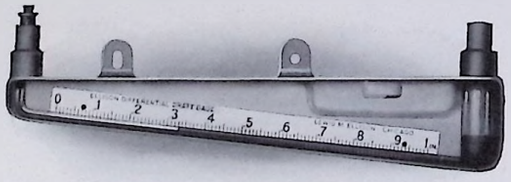


Fig. 17. Ellison Draft Gauge

Rate of Combustion — The amount of coal which can be burned per hour per square foot of grate surface is governed by the character of the coal and the draft available. The area of the grate and the ratio of this area to the boiler heating surface will depend upon the nature of the fuel to be burned, and the funnel should be so designed as to give a draft sufficient to burn the maximum amount of fuel per square foot of grate surface corresponding to the maximum evaporative requirements of the boiler.

With oil fuel, as indicated previously, combustion rates are dependent upon furnace volume.

Draft Gauges—The ordinary form of draft gauge, Fig. 15, which consists of a U-tube, containing water, lacks sensitiveness in measuring such slight pressure differences as usually exist, and for that reason gauges which multiply the draft indications are more convenient and are much used.

An instrument which has given excellent results is one introduced by Mr. G. H. Barrus, which multiplies the ordinary indications as many times as desired. This is

illustrated in Fig. 16, and consists of a U-tube made of one-half-inch glass, surmounted by two larger tubes, or chambers, each having a diameter of 2½ inches. Two different liquids which will not mix, and which are of different color, are used, usually alcohol colored red and a certain grade of lubricating oil. The movement of the line of demarcation is proportional to the difference in the areas of the chambers and the U-tube connecting them. The instrument is calibrated by comparison with the ordinary U-tube gauge.

In the Ellison form of gauge the lower portion of the ordinary U-tube has been replaced by a tube slightly inclined to the horizontal, as shown in Fig. 17. By this arrangement any vertical motion in the right-hand upright tube causes a very much greater travel of the liquid in the inclined tube, thus permitting extremely small variations in the intensity of the draft to be read with facility.



Fig. 18 Vernier Draft Gauge

The gauge is first leveled by means of the small level attached to it, both legs being open to the atmosphere. The liquid is then adjusted until its meniscus rests at the zero point on the left. The right-hand leg is then connected to the source of draft by means of a piece of rubber tubing. Under these circumstances, a rise of level of one inch in the right-hand vertical tube causes the meniscus in the inclined tube to pass from the point 0 to 1.0. The scale is divided into tenths of an inch, and the sub-divisions are hundredths of an inch.

The makers furnish a non-drying oil for the liquid, usually a 300 degrees test refined petroleum.

As this gauge would be affected by the inclination and motion of a ship, it is ordinarily not suitable for marine use.

A very convenient form of the ordinary U-tube gauge is shown in Fig. 18. This is a small modified U-tube with a sliding scale between the two legs of the U and with connections such that either a draft suction or a draft pressure may be taken. The tops of the sliding pieces extending across the tubes are placed at the bottom of the meniscus and accurate readings in hundredths of an inch are obtained by a vernier.

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STEAMER "HALEAKALA," 6240 INDICATED HORSE-POWER, BABCOCK & WILCOX BOILERS AND OIL BURNERS
INTER-ISLAND STEAM NAVIGATION CO., HONOLULU, T. H.

EFFICIENCY AND CAPACITY OF BOILERS

ASIDE from proper design and construction, the two factors of primary importance entering into the consideration of what constitutes a satisfactory boiler are its efficiency and that characteristic which is, perhaps ineptly, known as its "capacity." The present chapter deals with these factors only with a view to giving the terms a definite meaning as applied to steam-generating apparatus, together with the method of determining such values by actual test. The relation existing between efficiency and capacity is discussed in the following chapter on the "Selection of Boilers."

In the present consideration of the terms efficiency and capacity, the steam-generating apparatus as a whole is assumed, and may include a boiler alone, a boiler equipped with a superheater or a boiler and superheater with which an economizer or air heater is installed. In the following, a boiler is used as representing any combination of heat-absorbing surfaces, the methods of computation for the different combinations being the same.

EFFICIENCY—The term efficiency as specifically applied to a steam boiler is directly represented by the ratio of the total heat absorbed to the total heat in the fuel burned. Where the nature of the fuel is such that ash and refuse remain after combustion, such refuse, because of the limitations of fuel-burning apparatus, contains a certain amount of unconsumed combustible matter. Obviously, the boiler cannot properly be charged with the lack of absorption of the heat that such combustible matter would have evolved had it been burned; on the other hand, since the boiler user pays, in purchasing fuel, for the heat represented by this unconsumed combustible matter, it must be charged against the combined boiler, furnace and grate. The efficiency of the boiler as ordinarily given thus becomes in reality the efficiency of the boiler furnace, and grate or stoker, and is expressed:

$$\left.\begin{matrix}\text{Efficiency of boiler,}\\ \text{furnace, and grate}\end{matrix}\right\} = \frac{\text{Heat absorbed per pound of fuel}}{\text{Heat value per pound of fuel}} \qquad (30)$$

The efficiency as thus expressed is the same whether based on fuel as fired or on dry fuel. Any correction that would be made in changing from one basis to the other would appear in both numerator and denominator of *(30)* as the same value and would cancel.

A second efficiency basis, which is ordinarily assumed to be a measure of the efficiency of the boiler alone as distinguished from that of the boiler and grate, is the so-called efficiency based on combustible. While the difficulties arising from any attempt to separate efficiencies, discussed hereafter, must be recognized, common practice has established the use of the combustible basis as the ordinary efficiency that is assumed to make comparable the relative performance of boilers, irrespective of the grates or stokers installed, to an extent that makes advisable its inclusion here.

The heat in the products of combustion passed over boiler heating surface for absorption is due to the combustible portion of the fuel that is burned, regardless of what proportion of the total combustible constituent of the fuel this may represent. In determining the heat absorbed per pound of combustible burned, in order that any loss due to unconsumed combustible matter discharged with the ash and refuse shall not be charged against the boiler, the total combustible constituent of

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Steamer "Colonel James Pickands," 2500 Indicated Horse-power, Babcock & Wilcox Boilers and Superheaters
Owners, The Interlake Steamship Company, Cleveland, O.

the fuel must be corrected for such unconsumed combustible matter. On such a basis, the efficiency based on combustible is expressed:

$$\left.\begin{matrix}\text{Efficiency of boiler}\\ \text{and furnace}\end{matrix}\right\} = \frac{\text{Heat absorbed per pound combustible burned}}{\text{Heat value per pound combustible}} \qquad (31)$$

If the loss in unconsumed combustible in the ash could be eliminated, the efficiencies given by formulæ *(30)* and *(31)* would be the same. Thus, with liquid and gaseous fuels, if combustion is complete, there is no difference in the efficiencies as given on either basis.

Boiler performance is so directly related to grate or stoker performance that it is difficult and, where absolute accuracy is desired, impossible to separate the efficiency of the boiler from that of the combined unit of boiler and grate, even where proper correction is made for unconsumed combustible matter appearing in the ash and refuse.

Many methods have been proposed for a separation of boiler efficiency from that of the combined unit, including furnace and grate or stoker, and it is probable that this ground has been more thoroughly covered than any other feature of boiler testing. No method offered, however, has been able to make a separation of efficiencies in a manner such that the value for the efficiency of the boiler depends solely upon its design and is not influenced by the design of furnace and grate or stoker. In practically all of the proposed methods of separating efficiencies, that of the furnace and grate or stoker is based on the analysis of furnace or exit gas and the amount of unconsumed combustible in the ash. As has been indicated, one of the principal sources of loss in the operation of a steam-generating unit is in the amount of excess air supplied for combustion. There is no method of properly proportioning the loss from such excess air between boiler and stoker, though presumably it is largely due to the stoker. What is a more important factor in the difficulty of separation of efficiencies is that resulting from delayed or secondary combustion. This occurs in the passage of the gases through the heating surface of the boiler that cannot be considered as a part of the furnace, and which will occur under conditions of improper furnace operation regardless of the design of boiler. The effect of secondary combustion cannot be indicated by any analysis of furnace or exit-flue gas, or by an analysis of the ash, and shows directly against the boiler efficiency, while resulting from improper furnace or stoker design or operation. The factor of delayed combustion may affect the combined efficiency results as much as 10 per cent for the same analysis of flue gases leaving the boiler and the difficulty of separating the responsibility for such loss between boiler and furnace or grate is clear. On the subject of the difficulty of differentiation between boiler and combined steam-generation unit efficiency, Dr. D. S. Jacobus says, in a paper presented before The American Society of Mechanical Engineers in 1922:

> "Should it be possible to express accurately the furnace and stoker efficiency, a purchaser should obtain the same overall economy from his boilers with any arrangement of stokers and furnaces having a given efficiency. There is no method whereby this result could be insured and whereby the purchaser would be protected should he assume that the boiler and furnace efficiency would be an exact measure of the relative advantages of using one or another of the various stokers and furnaces in connection with his boilers."

The differentiation between the efficiency of boiler and furnace is of greater interest to the operator than to the boiler manufacturer, but from the standpoint of

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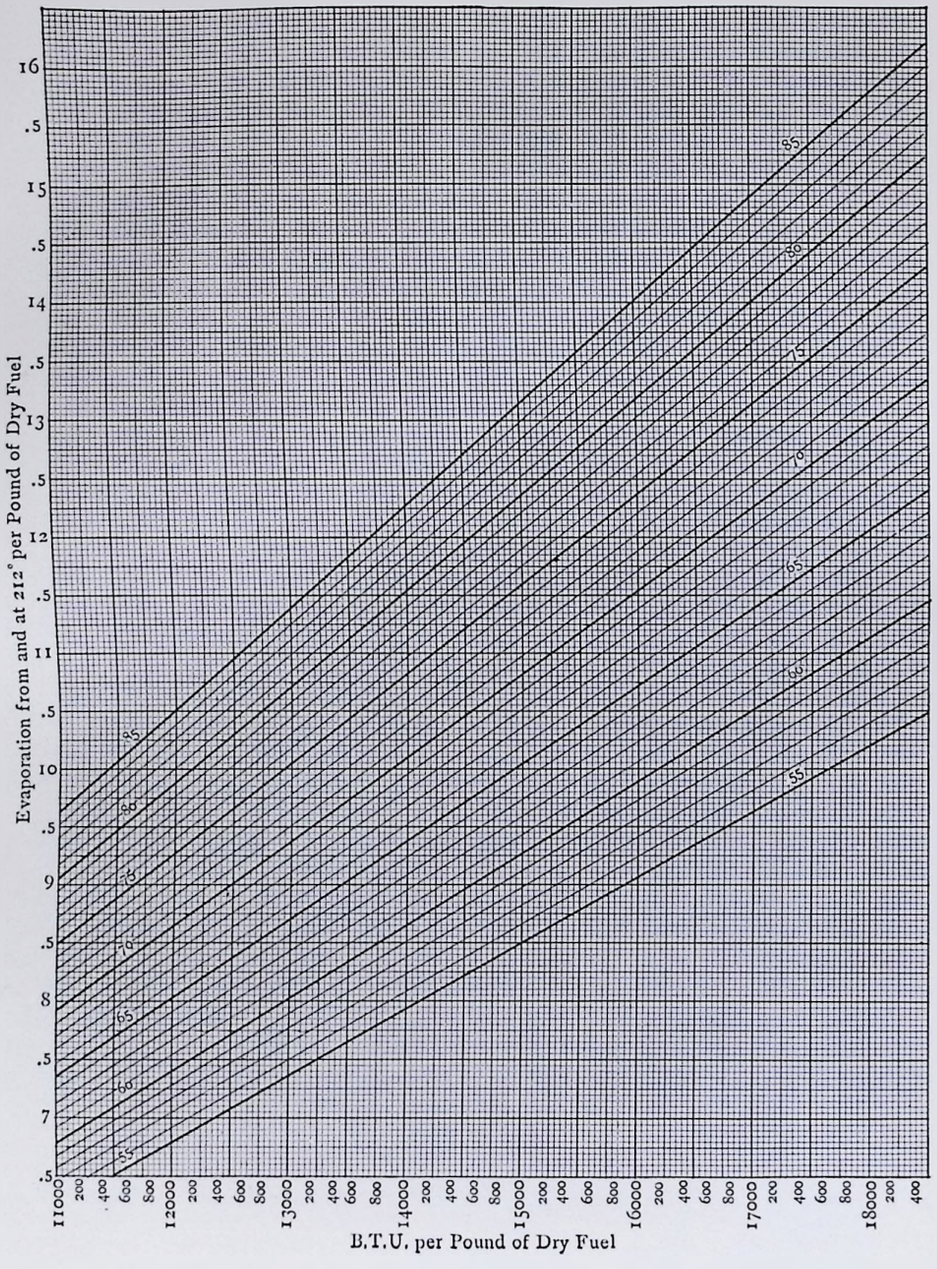


FIG. 19. EFFICIENCY CHART. CALCULATED FROM MARKS AND DAVIS' TABLES
DIAGONAL LINES REPRESENT PER CENT EFFICIENCY

both it emphasizes the importance of and necessity for co-ordination between furnace, grate or stoker, and boiler design, if the best results are to be obtained. If the boiler manufacturer is called upon to make a guarantee of performance of a combined unit, the design and operation of an element of which are not his responsibility, he must, as a matter of protection, make certain qualifications as to the performance of those elements of the combined unit which he does not furnish. Such qualifications are ordinarily expressed in terms of completeness of combustion in the boiler furnace, a minimum percentage of carbon dioxide and a maximum percentage of carbon monoxide in the flue gases, and, in the case of solid fuels, a minimum of unconsumed combustible matter in the discharged ash and refuse.

In connection with any attempt to allocate specifically losses in boiler-test work, either in the case of a combined unit or where it is endeavored to make a further segregation, the following quotation from The American Society of Mechanical Engineers' Test Code for Stationary Steam Generating Units is of interest:

> "Boiler testing should not be lightly undertaken by anyone who has not had some training under an experienced testing engineer, if reliable results are to be expected. The whole matter should be thoroughly understood, both theoretically and practically. Accurate tests depend very largely upon the care and faithfulness of the observers. It is much easier to make mistakes than is realized by those who are not familiar with practical testing.
>
> "The absolute accuracy of the results of a boiler test, even when conducted with the greatest care, is doubtful, but there is yet no possible basis upon which to determine what the probable limits of error might be. It is generally conceded, however, that there are several sources of indeterminate error, the more important of which are discussed below. The limits of accuracy of a test may very reasonably be taken to be within plus or minus 3 per cent.
>
> "One of the sources of probable error is the sampling of coal. Even when the greatest care is taken to obtain a representative sample, there may be an indeterminate error in ascertaining the heat value of the coal, even though the laboratory analysis is most reliable. With modern apparatus, these laboratory determinations should be substantially correct as regards the sample tested; but the question as to how truly the same represents the whole is always present and cannot be answered indubitably.
>
> "Another source of error is the moisture contained in the coal. As explained in the preceding paragraph, the sampling is more or less uncertain. It is contended by some that if the attempt is made to determine the moisture during the test, the methods of drying and weighing are unreliable; while others contend that though the moisture as determined in the laboratory is accurate so far as the sample delivered to the laboratory is concerned, this sample probably does not represent the bulk of the coal actually burned, since there must inevitably have been more or less loss of moisture during the collection, preparation and handling of the sample.
>
> "Similarly, it is problematical whether the samples collected for the determination of the moisture in steam and for gas analysis are representative of the bulk, although the testing of the samples obtained may be quite accurate.
>
> "It is not unusual for heat balances to be reported to the nearest B.t.u. and to the nearest one-tenth of 1 per cent. But the present state of the art of boiler testing does not provide means for attaining anything like this accuracy. In general, results should be reported only to the nearest significant figure. Reporting results of any kind in small units is likely to convey an erroneous idea as to the real accuracy of the figures.
>
> "It is, therefore, quite logical in the case of guarantee tests that a substantial compliance with the guarantee be accepted as full compliance therewith. A limit of tolerance should be agreed upon beforehand by the parties to the test. The amount of this tolerance might well bear some relation to the care exercised in arranging the details and in the conducting of the test."

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U. S. Battleship "Wyoming," 35,000 Indicated Horse-power, Babcock & Wilcox Boilers and Oil Burners

The computation of the efficiencies described may best be indicated by an example.

Assume the following data obtained from an actual boiler test:

(*a*) Steam pressure by gauge, 200 pounds.
(*b*) Feed temperature, 180 degrees.
(*c*) Superheat, 200 degrees.
(*d*) Total weight of fuel fired per hour, 3,761 pounds.
(*e*) Moisture in fuel, 5 per cent.
(*f*) Total ash and refuse per hour, 447 pounds.
(*g*) Total evaporation per hour, actual, 30,000 pounds.
(*h*) Heat value per pound of dry coal, 13,000 B.t.u.
(*i*) Heat value per pound of combustible, 14,444 B.t.u.

The factor of evaporation is, from Tables 16 and 14, and by formula *(4)*, 1.1972. The equivalent evaporation from and at 212 degrees is, therefore, 35,916 pounds per hour.

The total weight of dry fuel is $3{,}761 \times 0.95 = 3{,}573$ pounds per hour, and the evaporation from and at 212 degrees per pound of dry fuel is $35{,}916 \div 3{,}573 = 10.05$ pounds. The heat absorbed per pound of dry fuel is therefore $10.05 \times 970.4 = 9{,}752$ B.t.u. Hence the efficiency, by *(30)*, is $9{,}752 \div 13{,}000 = 75.00$ per cent.

The total combustible burned is $3{,}573 - 447 = 3{,}126$ pounds per hour, and the evaporation from and at 212 degrees per pound of combustible is $35{,}916 \div 3{,}126 = 11.49$ pounds. Hence the efficiency based on combustible is from *(31)*, $(11.49 \times 970.4) \div 14{,}444 = 77.19$ per cent.

If item (*h*) were expressed in terms of fuel as fired, and the evaporation from and at 212 degrees computed on such basis, the efficiency per pound of fuel would be found to be the same, whether the wet or dry basis were used.

For approximate results, the efficiency may be computed from a chart. Fig. 19 shows such a chart based on the evaporation and heat value per pound of dry fuel, and efficiencies may be read directly to within one-half of one per cent. The same chart may be used for efficiency determinations on the fuel as fired or the combustible basis, provided evaporations and heat values corresponding to such basis are used in the graphic computations.

The difference in the efficiency shown in a test of any combination of steam generating units, and 100 per cent efficiency, includes the unavoidable and avoidable losses arising in the conversion of the heat energy of the fuel to that transmitted to the steam as delivered by the generating unit.

Unavoidable losses result directly from the necessity of discharging the products of combustion at a temperature above that of the temperature of the fuel, regardless of the combination of heat-absorbing apparatus, and from the moisture and hydrogen content of the fuel. The avoidable losses, or rather those that to an extent can be controlled, result from the amount of air in excess of that necessary for perfect combustion that is heated to the temperature of the exit gases; the moisture accompanying such excess; unconsumed solid combustible matter discharged with the ash and refuse; unconsumed gaseous combustible matter discharged with the exit gases; and heat radiated from the setting of the unit.

These losses, segregated as far as available data and methods of determination permit, added to the boiler efficiency will total less than 100 per cent, the difference

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between such sum and 100 representing losses that cannot be charged against any listed items. The amount of such unaccounted loss is a measure of the accuracy of the test as a whole, and for this reason the tabulation of the efficiency of the unit and the determinable losses is known as a "heat balance." The items given in the ordinary heat balance may be listed as follows:

1st. Loss due to the moisture content of the fuel. Such moisture must be evaporated and superheated to a temperature represented in the computation of the loss by that of the exit gases. The loss is represented by the heat which would be developed by the cooling of the water vapor to the initial temperature of the *fuel* and condensing it to liquid water at that temperature. The amount of heat that would be so developed is the total heat of the water vapor at the exit-gas temperature minus that at the initial temperature, the low temperature state being taken as liquid water at fuel temperature.

2nd. Loss due to the water vapor formed in the burning of hydrogen. This loss results from the same causes as the first item and is represented by the difference in total heat of water vapor at exit temperature and liquid water at fuel temperature.

3rd. Loss due to the moisture in the air used for combustion. This moisture enters the furnace as water vapor and not as liquid water. The loss therefore is represented by the difference in total heat of the water vapor at exit-gas temperature and at the temperature of the air used in combustion. It should be noted that the low-temperature state is as vapor, and not as liquid water as in the case of items 1 and 2. The weight of moisture per pound of air supplied for combustion is determined from wet and dry bulb thermometer readings and psychrometric tables or charts. This loss is small and is frequently included with the unaccounted losses.

4th. Loss due to the heat carried away by the dry chimney gases. This loss results from the necessity of heating the dry products of combustion from the temperature of the air used in combustion to that of the exit gases. The weight of the dry products is the total weight including excess air.

5th. Loss due to unconsumed solid combustible matter. This loss is ordinarily based on the unconsumed combustible matter that is discharged with the furnace ash and refuse, but, for accuracy, should properly include such unconsumed combustible particles as may be carried over into the setting or up the stack. The latter portion of combustible can be determined only where the arrangement of the unit permits the collection of ash and cinders discharged in this manner, and is ordinarily not included in the heat balance. The solid combustible matter contained in the ash and refuse is usually assumed to be carbon, but where the percentage of combustible in the ash is high, the volatile constituent may be relatively large. The accuracy of the determination of this loss is primarily dependent upon the ash sampling.

6th. Loss due to unconsumed gaseous combustible matter discharged with the exit gases. This loss in the ordinary boiler trial is measured by the carbon monoxide constituent of the flue gases alone, though losses from unconsumed hydrocarbons may exist that are greater than that resulting from the presence of carbon monoxide. The ordinary gas analysis apparatus, however, does not carry the analysis further than the carbon monoxide constituent. The heat value used in the computation of this loss is that representing the difference in burning one pound of carbon to carbon monoxide and carbon dioxide.

7th. Radiation and unaccounted losses. While several methods of determining radiation losses have been offered, none is satisfactory, and such losses are ordinarily included with the unaccounted loss. As stated, this loss is represented by the difference between 100 per cent and the sum of the above items plus the boiler efficiency.

The methods of computing these losses, with an example in each instance, are given in the following:

(A). Loss due to moisture in the fuel. At temperatures below 1000 degrees and at vapor pressures below 3 pounds per square inch absolute, the total heat of saturated or superheated vapor depends only upon its temperature, and is given by the formula

$$H = 1{,}058.7 + 0.455T$$

where T is the actual temperature in degrees Fahrenheit. This formula is purely empirical, and the constant 0.455 is not to be confused with the specific heat of superheated steam.

The total heat of liquid water is

$$t - 32$$

where t is the temperature of the water.

The difference between these total heats is thus

$$[(1058.7 + 0.455T) - (t_f - 32)] = 1090.7 + 0.455T - t_f$$

The loss due to the moisture content of the fuel, therefore, is

$$\text{Loss, B.t.u. per pound} = W\,(1090.7 + 0.455T - t_f) \qquad (32)$$

where W = weight of moisture per pound of fuel

t_f = temperature of the fuel.

(B). Loss due to combustion of hydrogen. In combustion, one pound of hydrogen unites with eight pounds of oxygen to form water vapor. On the basis of (A), the loss from this source is then

$$\text{Loss, B.t.u. per pound} = 9H_2\,(1090.7 + 0.455T - t_f) \qquad (33)$$

where H_2 = weight of hydrogen per pound of fuel and T and t_f have the same values as in (A).

(C). Loss due to moisture in the air used for combustion. The difference in the total heat of this vapor at the temperatures of the exit gas and of the air for combustion is

$$[(1058.7 + 0.455T) - (1058.7 + 0.455t_a)] = 0.455\,(T - t_a)$$

where t_a = initial temperature of the air used for combustion.

The loss from this source is, then,

$$\text{Loss, B.t.u. per pound} = w \times m \times 0.455\,(T - t_a) \qquad (34)$$

where w = weight of air used for combustion per pound of fuel,

m = weight of moisture per pound of air,

(D). Loss in dry chimney gases. The weight of dry products of combustion per pound of fuel, W, has been shown by formula *(18)* to be

$$\frac{11CO_2 + 8O_2 + 7\,(CO + N_2)}{3\,(CO_2 + CO)}C$$

where C is the weight of carbon burned per pound of fuel and appearing in the products of combustion.

The loss from this source is, then,

$$\text{Loss, B.t.u. per pound} = W(T - t_a)0.24 \qquad (35)$$

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Steamer "City of Honolulu," Babcock & Wilcox Boilers and Oil Burners
Owners, Los Angeles Steamship Co., Wilmington, Cal.

where W = weight of dry products per pound of fuel,
T = temperature of exit gases,
t_a = temperature of air used for combustion,
0.24 = specific heat of products. (This value is not exact, as it depends upon temperature. It may be computed for actual temperatures if desired.)

(E). Loss due to unconsumed combustible in the ash. This loss may be expressed by

$$\text{Loss, B.t.u. per pound} = \frac{c \times C}{100} 14{,}600 \qquad (36)$$

where c = weight of ash per pound of fuel,
C = percentage of unconsumed combustible in the ash,
$c \times C$ = weight of unconsumed combustible in terms of total carbon per pound of fuel.

The value 14,600 is the heat value per pound of carbon, all of the combustible matter being assumed to be carbon. Such an assumption gives rise to an error, which, however, is so small as to be negligible.

(F). Loss due to unconsumed gaseous combustible matter. The weight of carbon monoxide in the exit gases is represented by the expression

$$\frac{CO}{CO_2 + CO}$$

where the symbols represent volumetric percentages. The loss from this source is

$$\text{Loss, B.t.u. per pound} = \frac{CO}{CO_2 + CO} \times C \times 10{,}160 \qquad (37)$$

where C = the weight of carbon burned and appearing in the products of combustion, per pound of fuel burned,
10,160 = the number of B.t.u. evolved in burning one pound of carbon *in carbon monoxide* to carbon dioxide.

(G). Radiation and unaccounted losses. If E is the heat absorbed by the steam-generating unit, expressed in B.t.u. per pound of fuel, or in terms of percentage, this loss is represented by

$$\text{Loss, B.t.u. per pound} = H.V. - (E + A + B + C + D + E + F) \qquad (38)$$

or

$$\text{Per cent loss} = 100 - (E + A + B + C + D + E + F) \qquad (38a)$$

the values of A, B, C, etc., being expressed in terms of B.t.u. or as percentages.

A heat balance may be given upon either a dry fuel or a fuel as fired basis, but care must be taken to have the numerical values used in the computation consistent with the basis used.

To illustrate the computation of the items of a heat balance as ordinarily reported, assume the data obtained in a boiler trial to be as given in Table 39. The heat balance as computed here is on a dry coal basis.

The heat absorbed per pound of dry coal is

$$11.12 \times 970.4 = 10{,}791 \text{ B.t.u.}$$

and the efficiency of the boiler

$$10{,}791 \div 14{,}000 = 77.08 \text{ per cent}$$

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STEAMER "ALAMEDA," 3000 INDICATED HORSE-POWER, BABCOCK & WILCOX BOILERS AND OIL BURNERS
OWNERS, ALASKA STEAMSHIP COMPANY, SEATTLE, WASH.

TABLE 39

BOILER TRIAL DATA

FOR COMPUTATION OF A HEAT BALANCE

Wet Bulb Thermometer, Degrees Fahrenheit		67
Temperature Boiler Room, Degrees Fahrenheit		73
Temperature Exit Gases, Degrees Fahrenheit		575
Ultimate Analysis Dry Coal	C, Per Cent	78.42
	H_2, Per Cent	5.56
	O_2, Per Cent	8.25
	N_2, Per Cent	1.09
	S, Per Cent	1.00
	Ash, Per Cent	5.68
Moisture in Coal, Per Cent		1.91
B.t.u. per Pound Dry Coal		14000
Ash and Refuse (Per Cent Dry Coal)		7.03
Unconsumed Carbon in Ash, Per Cent		31.50
Fuel Gas Analysis	CO_2, Per Cent	14.00
	O_2, Per Cent	5.50
	CO, Per Cent	0.42
	N_2, Per Cent	80.08
Evaporation from and at 212 Degrees per Pound Dry Coal, Pounds		11.12

(A). The moisture content of the coal, 1.91 per cent, in terms of dry coal, becomes

$$1.91 \div (100 - 1.91) = 1.95 \text{ per cent}$$

and the loss due to this moisture is, from (A),

$$0.0195\ [1090.7 + (0.455 \times 575) - 73] = 25 \text{ B.t.u.}$$

or

$$25 \div 14.000 = 0.18 \text{ per cent}$$

(B). Loss due to burning of hydrogen

$$9 \times 0.0556\ [1090.7 + (0.455 \times 575) - 73] = 640 \text{ B.t.u.}$$

or

$$640 \div 14.000 = 4.57 \text{ per cent}$$

(C). Loss due to moisture in air used for combustion.

From the wet and dry bulb thermometer readings and psychrometric tables, the weight of moisture in the air per pound of dry air supplied is 0.0127 pound.

The weight of dry air supplied per pound of dry coal is, from formula *(20)*

$$\frac{3.036 \times 80.08}{14.00 + 0.42}(0.7842 - 0.0221) = 12.849 \text{ pounds}$$

The weight of water vapor per pound of dry coal is

$$12.849 \times 0.0127 = 0.163 \text{ pounds}$$

and the loss from this source is

$$0.163 \times 0.48\ (575 - 73) = 39.3 \text{ B.t.u.}$$

or

$$39.3 \div 14.000 = 0.28 \text{ per cent}$$

(D). Loss in dry chimney gases. The weight of dry gases per pound of carbon burned, from formula *(17)*, is

$$\frac{11 \times 14.00 + 8 \times 5.50 + 7\ (0.42 + 80.28)}{3\ (14.00 + 0.42)} = 17.603 \text{ pounds}$$

and the weight per pound of fuel is

$$17.603\ (0.7842 - 0.0221) = 13.415 \text{ pounds}$$

The loss due to the heat carried away in the dry chimney gases is then

$$13.415 \times 0.24\ (575 - 73) = 1,616 \text{ B.t.u.}$$

or

$$1,616 \div 14,000 = 11.54 \text{ per cent}$$

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(E). Loss due to unconsumed combustible matter in the ash and refuse. This loss is

$$\frac{0.0703 \times 31.5}{100} 14{,}600 = 323.3 \text{ B.t.u.}$$

or

$$323.3 \div 14{,}000 = 2.31 \text{ per cent}$$

(F). Loss due to unconsumed gaseous combustible matter. This loss is

$$\frac{0.42}{14.00 + 0.42} 0.7621 \times 10{,}160 = 225.5 \text{ B.t.u.}$$

or

$$225.5 \div 14{,}000 = 1.61 \text{ per cent}$$

(G). Radiation and unaccounted loss. This, expressed in terms of heat value of the fuel burned, is

$$14{,}000 - (10{,}791 + 25 + 640 + 39.3 + 1{,}616 + 323.3 + 225.5) = 14{,}000 - 13{,}660 = 340 \text{ B.t.u.}$$

or, in terms of percentage loss,

$$340 \div 14{,}000 = 2.43 \text{ per cent}$$

The heat balance as ordinarily tabulated is shown in Table 40.

TABLE 40

HEAT BALANCE

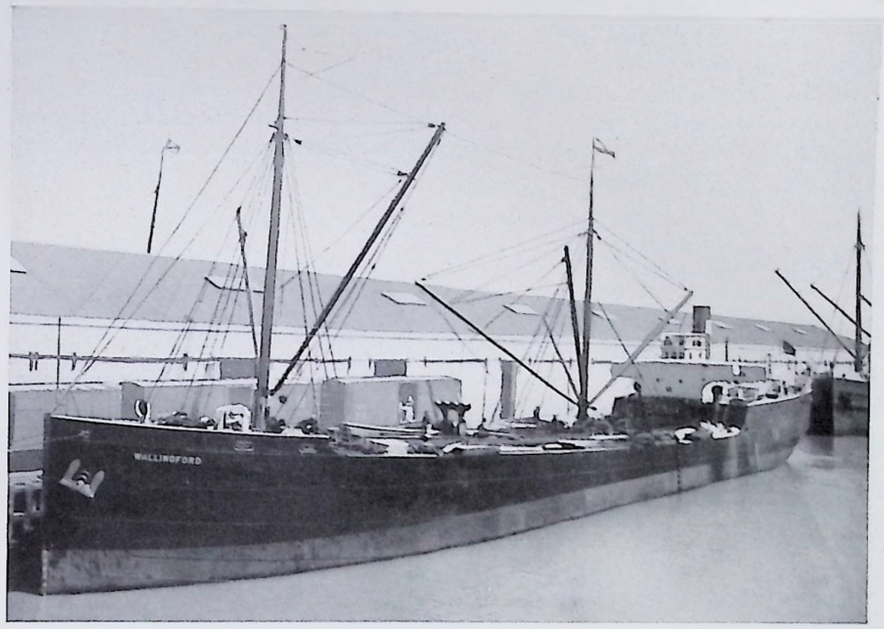
	B.t.u.	Per Cent
Heat absorbed by boiler	10,791	77.08
Loss due to moisture in coal	25	0.18
Loss due to moisture formed in burning H_2	642	4.58
Loss in dry chimney gases	1,616	11.54
Loss due to moisture in air	39	0.28
Loss due to incomplete combustion of C	227	1.62
Loss due to unconsumed C in ash	323	2.31
Radiation and unaccounted losses	337	2.41
Total	14,000	100.00

Application of the Heat Balance—A heat balance should be made in connection with any boiler test on which sufficient data for its computation are reported. This is particularly true where the boiler performance has been considered unsatisfactory, the distribution of the losses shown by the heat balance enabling any extraordinary loss to be detected. Even where accurate data for computing a heat balance are not available, such a computation based on certain assumptions is frequently of assistance in indicating unusual losses.

Where it is not practicable to weigh or measure the quantity of water evaporated or fuel burned, it is possible to determine the efficiency of the unit by a method of indirect heat balance. If a radiation and unaccounted loss is arbitrarily assumed, all the items of the ordinary heat balance with the exception of the boiler efficiency may be computed from fuel, ash and exit-gas analyses. The sum of these items and the assumed unaccounted loss deducted from 100 per cent give indirectly the efficiency

of the unit. It is to be recognized that this method can give results that are at best approximate, but with certain fuels and under conditions where fuel weighing is difficult, even approximate results may be of value.

Capacity—In stationary boiler practice in this country, the term capacity has come, through custom, to refer to boiler output expressed in terms of boiler horse-power. Stationary boilers have been given a "rated horse-power" based ordinarily on a given number of square feet of heating surface per horse-power, and overloads referred to as "per cents of rating." In view of present methods of stationary boiler room operation, capacity and per cent of rating as so expressed are entirely illogical, and there is a decided tendency to abandon the use of these terms as a measure of boiler output. As an indication of such tendency, the recommendations of the Committee on "Definitions and Values" of the Power Test Codes of the American Society



Steamer "Wallingford," 900 Indicated Horse-power, Babcock & Wilcox Boilers and Oil Burners; Owners, McCormick Steamship Company, San Francisco, Cal.

of Mechanical Engineers may be of interest. This committee has recommended that boiler output, instead of being expressed in terms of boiler horse-power be expressed:

(*a*) By the heat output in steam delivered by the unit per hour, as represented by the difference between the total heat in the steam delivered and in the feed water as fed to the unit.

(*b*) By the actual evaporation in pounds of steam per hour at the reported steam pressure and quality of temperature of steam, and the reported feed-water temperature, and,

(*c*) To do away with the objection resulting from (*a*) to the use of large numbers, that the total heat output be expressed in "units of output" such units being determined by the total heat output (*a*), divided by 1000.

In marine practice the term horse-power as applied to boilers has been almost universally used in expressing the horse-power developed by the prime movers which the boilers serve, and the output from such boilers as representing "capacity" is given

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in terms of evaporation per square foot of boiler-heating surface. This would seem to be an entirely logical method of expressing boiler output.

The efficiency of a boiler or of a steam generating unit as a whole, and the capacity, whether in terms of evaporation per square foot of heating surface, in terms of maximum rating, or in terms of maximum heat output for the particular combustion apparatus with which the boiler is installed, can be determined accurately only by means of a boiler test. Standard methods of conducting such tests have been recommended by the Subcommittee on "Test Code for (Stationary) Steam Generating Units" of the Power Test Codes Committee of the American Society of Mechanical Engineers, presented in 1923. A résumé of this code is given in the following pages, such items as do not affect the testing of marine boilers being omitted.

The code as presented makes many references to other sections of the complete Power Test Code, which are too long to be included here, and it is suggested that before undertaking serious testing work, not only the Code for Stationary Steam Generating Units but also the General Instructions, the Code on Definitions and Values, the Code on Instruments and Apparatus, and the Code on Fuels, as given in the complete Power Test Codes of the American Society of Mechanical Engineers, be carefully studied. The résumé following, however, gives the principal recommendations and methods to be followed.

OBJECT AND PREPARATIONS

Definitely determine the object of the test and keep it continuously in mind. Take the dimensions, note the physical conditions, examine for leakage, install the testing appliances, etc., and make preparations for the test accordingly.

Determine the character of the fuel to be used.

In guarantee tests with fuel of a specified calorific value or other characteristics, there should be a clear understanding as to the permissible variation from the specified characteristics.

* * * * * *

INSTRUMENTS AND APPARATUS

The instruments and apparatus required for boiler tests are:

(*a*) Scales for weighing coal, oil or other fuel, ashes, furnace refuse, etc.
(*b*) Graduated scales attached to the water glasses.
(*c*) Tanks or tanks and scales for volumetric or weight measurement of water.

* * * * * *

(*e*) Pressure gauges, thermometers, pyrometers and draft gauges.
(*f*) Calorimeter for determining the quality of the steam.
(*g*) Gas-sampling and analyzing apparatus.

As no two plants are alike, it is quite impracticable to do more than suggest how and where the various instruments should be located.

(*a*) Fuel-weighing apparatus should be near the point where the fuel is to be used, under direct observation of and convenient to the one in charge of the test. The weighing of refuse may be done at any convenient point.

(*b*) The water-glass scale should be so attached that the breaking of the glass will not disturb the scale.

(*c*) The water-weighing or measuring apparatus, if other than a meter, should be arranged in an easily accessible place and made as convenient as practicable. It should also be under the direct observation of the one in charge of the test.

(*e*) Pressure gauges should be away from any disturbing influences, such as extreme heat, and in a position to be easily read. Thermometers for determining atmospheric temperatures should be placed so as to give an average indication away from cold or hot drafts, etc.

A thermometer well for measuring feed-water temperatures should be as close to the boiler inlet as possible, and the pipe between boiler and thermometer should be well protected with heat-insulating covering.

Saturated-steam temperatures may be measured at any point in the steam pipe where the pressure is the same as that where the temperature is desired, care being taken that the well is not cooled by condensate. The temperature and pressure of superheated steam should be measured as close to the outlet of the superheater as possible, eliminating all losses by pressure drop and radiation between the superheater and thermometer.

Pyrometers of any type must have the part on which the heat impinges so located that the temperature which it is desired to measure is actually obtained. Much experience and good judgment are essential in order to obtain even approximately reliable readings.

Draft at the gas outlet is usually measured, if nowhere else. It is often desirable to know the draft at other points of the setting, in which cases the several locations should be chosen to suit the information wanted. Care should be exercised to see that the ends of the connecting pipes are not obstructed and that the open ends are not subject to other than static pressures. All joints of draft-gauge connections must be tight.

The sampling tube of the calorimeter should be as close to the point at which the quality of steam is desired as possible. All exposed piping in which condensation might influence the result must be well lagged. Ordinarily the quality of the steam leaving the boiler, or that entering the superheater, is desired.

Analysis of the exit gases is usually what is taken, but very frequently analyses are wanted from other points. Sample tubes must be so located that only the gas to be analyzed enters the tube and precautions taken against the possibility of inleakage of air anywhere in the line. The gas-analyzing apparatus should be in a readily accessible and protected place with good light and convenient facilities.

OPERATING CONDITIONS

Determine what the operating conditions and method of firing should be to conform with the object in view, and see that they prevail as nearly as possible throughout the trial.

The duration of tests to determine the efficiency of a steam-generating unit burning coal, either hand or stoker-fired, should preferably be 24 hours, but where operating conditions do not permit or other considerations make it advisable or necessary the length of test may be reduced to not less than 10 hours. When the rate of combustion is less than 25 pounds of coal per square foot of grate surface per hour, the test should be continued until a total of 250 pounds per square foot of grate area has been burned, except that in cases where a type of stoker is used that does not permit the quantity of fuel and (or) the condition of the fuel bed to be accurately estimated, the duration of test should not be reduced below that required to minimize the error.

In the case of a unit using pulverized fuel, the duration of test should be not less than 6 hours, and for liquid or gaseous fuel not less than 4 hours.

* * * * * *

In tests conducted under conditions where the service requires continuous operation night and day, with frequent shifts of firemen, the duration of test should be at least 24 hours. Likewise in such tests, either of a single boiler or of a plant of several boilers which operate regularly a certain number of hours and during the balance of the day are banked, the duration should be not less than 24 hours.

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STEAMER "PRINCESS MARIE JOSE," 15,000 SHAFT HORSE-POWER, BABCOCK & WILCOX BOILERS AND OIL BURNERS
OWNERS, BELGIAN STATE RAILWAYS

The duration of tests to determine the maximum evaporative capacity of a steam-generating installation, when the efficiency is not determined, should be not less than two hours, unless otherwise agreed upon.

STARTING AND STOPPING

Combustion, fuel, draft and temperature conditions, the water level, rate of feeding water, rate of steaming, and the steam pressure should be as nearly as possible the same at the end as at the beginning of the test.

To secure the desired equality of conditions with hand-fired boilers, the following method should be employed:

The furnace being well heated by a preliminary run at the same combustion rate that will prevail during the test and sufficiently long to thoroughly heat the setting, burn the fire low and thoroughly clean it, leaving enough live coal spread evenly over the grate (say, from 2 to 4 inches) to serve as a foundation for the new fire. Note quickly the thickness of the coal bed as nearly as it can be estimated or measured, also the water level, the steam pressure, and the time, and record the latter as the starting time. Fresh coal should then be fired from that weighed for the test, the ashpit thoroughly cleaned, and the regular work of the test proceeded with.

Before the end of the test the fire should again be burned low and cleaned in such a manner as to leave the same amount of live coal on the grate as at the start. When this condition is reached, observe quickly the water level, the steam pressure, and the time, and record the latter as the stopping time. If the water level is lower or higher than at the beginning, a correction should be made by computation, not by feeding additional water. Finally remove the ashes and refuse from the ashpit.

In a plant containing several boilers, where it is not practicable to clean them simultaneously, the fires should be cleaned one after the other as rapidly as may be, and each one after cleaning charged with enough coal to maintain a thin fire in good working condition.

After the last fire is cleaned and in working condition, burn all the fires low (say, 4 to 6 inches), note quickly the thickness of each, also the water levels, steam pressure, and time, which last is taken as the starting time. Likewise, when the time arrives for closing the test, the fires should be quickly cleaned one by one, and when this work is completed they should all be burned low the same as at the start, and the various final observations made as before stated.

In the case of a large boiler having several furnace doors requiring the fire to be cleaned in sections one after the other, the above directions pertaining to starting and stopping in a plant of several boilers may be followed.

To obtain the desired equality of conditions when a mechanical stoker is used, the above procedure should be modified as follows:

Stokers should be in operation at approximately the same rate that will prevail during the test for at least 12 hours before starting test. At the start and finish of the test, level the coal in stoker hopper. Make starting and stopping observations as in hand-fired tests.

(*a*) *Stokers of the Continuous-Dumping Type:* The desired operating conditions, *i. e.*, speed and stroke of coal-feeding mechanism, speed of grate, intensity of draft or blast, and rate of water feed should be maintained as nearly constant as possible for at least one hour, and preferably for two hours, before starting and stopping test.

(*b*) *Stokers of the Intermittent-Dumping Type:* Proceed as above except that stokers should be cleaned about one hour before starting and before stopping the test.

To obtain the desired equality of conditions when liquid fuel is used, the following method should be employed:

The boiler or boilers to be tested should be operated before the start of the test under the fuel, furnace, and combustion conditions which are to be maintained throughout the test for a period of not less than three hours. Fuel temperature, fuel pressure and draft conditions should be kept as nearly constant as possible during this period and throughout the test. The references above for hand-fired tests as to starting and stopping observations hold also for liquid fuel tests.

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U. S. Coast Guard Cutter "Haida," 2600 Shaft Horse-power, Babcock & Wilcox Boilers, Superheaters and Oil Burners

RECORDS

Readings of the instruments at 15-minute intervals are usually sufficient. If there are sudden and wide fluctuations, the readings should be taken at intervals of such frequency as may be necessary to determine the true nature of the variations.

The approximate quantity of fuel needed each hour should be determined, and if the quantity required and inconvenience due to lack of room are not too great, the whole amount should be delivered on the firing floor at the beginning of each hour. If any is left at the end of the hour, the quantity should be estimated. In this way the amount of fuel used per hour can be determined and the approximate results be followed from hour to hour.

If the whole amount cannot be delivered at the beginning of the hour, convenient quantities may be weighed out at appropriate intervals so as to come out about even at the end of the hour. When hopper scales are used, receiving coal from bunkers and discharging directly into furnace hoppers, hourly quantities may be roughly determined by estimating furnace conditions. These hourly quantities should be properly noted on the log sheet. They are taken as a matter of convenience and guide, but only the totals are to be used in the final calculation.

The records should be such as to ascertain also the approximate consumption of feed water each hour, and thereby determine the degree of uniformity of evaporation. The maintenance of a uniformity of evaporation is greatly facilitated by the use of some form of graphic recording steam meter, which should be so placed as to keep continuously before the operator the rate of evaporation. Since the indications of this meter are not used in any of the test calculations, extreme accuracy is not essential.

Quality of Steam—If the boiler does not produce superheated steam, the quality of the steam should be determined by the use of a throttling or separating calorimeter. If the boiler has superheating surface, the temperature of the steam should be determined by the use of a thermometer inserted in a thermometer well.

Ashes and Refuse—The ashes and refuse withdrawn from the furnace and ashpit during the progress of and at the end of the test should be weighed, so far as possible, in a dry state. If wet, the amount of moisture should be ascertained and allowed for, a sample being taken and dried for this purpose. This sample may also serve for analysis.

Ash sampling is at best subject to larger errors, and every precaution should be taken to insure as representative a sample as possible. If sufficiently hot to allow combustion to proceed, ash should be thoroughly quenched with water immediately after dumping into the ashpit. Enough time should elapse before weighing to permit the heat of the refuse to drive out most of this moisture. If possible, all the ash should then be passed through a crusher, thoroughly mixed, and reduced to a laboratory-size sample by successive quartering. If it is impracticable to crush all the ash, the clinkers and fines should be placed in separate piles, and each pile weighed and sampled separately, the two samples being combined in proportion to the relative weights of their respective piles.

Sampling Fuel—During the progress of the test fuel should be regularly sampled for purposes of analysis.*

* The matter of fuel sampling is of the greatest importance in the determination of results with the maximum degree of accuracy. It is recommended that engineers conducting boiler tests refer for methods of fuel testing, in the case of coal, to the standard methods of the American Society of Testing Materials and, in the case of oil, to the methods recommended by the Subcommittee of the American Society of Mechanical Engineers Power Test Codes on Liquid Fuels.

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Calorific Tests and Analyses of Fuels—The quality of the fuel should be determined by calorific tests and analyses of the fuel sample above referred to and of the furnace ash and refuse.

Analyses of Flue Gases—For approximate determinations of the composition of the flue gases, the Orsat apparatus or some modification thereof should be employed. Gas samples should preferably be taken continuously, but, if momentary samples are obtained, the analyses should be made as frequently as possible, noting the furnace and firing conditions at the time the sample is drawn. Where the firing is intermittent, gas samples should be taken at such intervals that the complete firing cycle will be covered by the average of individual readings. If the sample drawn is a continuous one, the intervals may be made longer. Where the unit includes an economizer, gas analyses should be made at both boiler and economizer outlet.

Smoke Observations—In tests requiring a determination of the amount of smoke produced, observations should be made regularly throughout the trial at intervals of five minutes, or if necessary one minute. For observations covering a period of one or more single firings with solid fuel, the intervals should be one-quarter minute or less.

Chart—In trials having for an object the determination and exposition of the complete boiler performance, the entire log of readings and data should be represented graphically.

It is recommended that a record be kept of the energy used by auxiliaries immediately connected with the steam-generating unit being tested and a specific note made thereof; no deductions, however, shall be made on the report forms or in computing results unless the object of the test so requires, in which case the report should specifically so state. This applies to steam or power used in driving stokers or other fuel-feeding apparatus, oil burners, fans, feed pumps, soot blowers, etc.

DATA AND RESULTS

The data and results should be reported in accordance with the tables given herewith, using the proper table for the class of fuel burned. If necessary, items of data not provided for may be added, or if certain items are not required such may be omitted as may conform to the object of the test. Unless otherwise indicated, the quantities should be the average of the observations.

DATA AND RESULTS OF TEST OF STEAM-GENERATING UNIT

GENERAL INFORMATION

1	Date of test	
2	Location of plant	
3	Owner	
4	Maker of boiler	
5	Test conducted by	
6	Object of test	

DESCRIPTIONS, DIMENSIONS, ETC.

7	Grate	
8	Superheater	
9	Economizer	
10	Draft	
11	Fuel	
12	Boiler heating surface	square feet
13	Superheating surface	square feet
14	Grate area	square feet
15	Volume of combustion space	cubic feet
16	Furnace, center of grate to nearest heating surface	feet
17	Furnace volume per square foot of heating surface	cubic feet

FUEL AND GAS ANALYSES AND DATA

Fuel, Approximate Analysis

18	Volatile matter	per cent
19	Fixed carbon	per cent
20	Ash	per cent
21	Moisture (as fired)	per cent
22	B.t.u. per pound (as fired)	
23	B.t.u. per pound, dry	
24	Fusion temperature of ash	degree Fahrenheit

Fuel, Ultimate Analysis

25	Carbon	per cent
26	Hydrogen	per cent
27	Oxygen	per cent
28	Nitrogen	per cent
29	Sulphur	per cent
30	Ash	per cent

Gas

31	Furnace-gas analysis: Per cent of CO_2 . . O_2 . . N_2 . . CO . . SO_2 . .	
32	Flue-gas analysis, boiler outlet: Per cent of CO_2 . . O_2 . . N_2 . . CO . . SO_2 . .	
32a	Flue-gas analysis, economizer outlet: Per cent of CO_2 . . O_2 . . N_2 . . CO . . SO_2 . .	
33	Dry gas per pound of fuel, furnace (as fired, dry)	pounds
34	Dry gas per pound of fuel, boiler outlet (as fired, dry)	pounds
34a	Dry gas per pound of fuel, economizer outlet (as fired, dry)	pounds
35	Dry gas per pound of fuel, theoretical	pounds
36	Air supplied per pound of fuel (as fired, dry)	pounds

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TEST DATA AND RESULTS

Pressures and Drafts

37	Moisture in air	pound per pound of air
38	Steam pressure by gauge:	
	Boiler	pounds per square inch
	Superheater outlet	pounds per square inch
39	Air pressure in ashpit zone	inch of water
40	Draft in furnace	inch of water
41	Draft under boiler damper	inch of water
42	Draft at economizer outlet	inch of water

Temperatures

43	Steam temperature	degrees Fahrenheit
44	Moisture in steam	per cent
45	Superheat	degrees Fahrenheit
46	Temperature of feed water entering boiler	degrees Fahrenheit
47	Temperature of feed water entering economizer	degrees Fahrenheit
48	Furnace temperature	degrees Fahrenheit
49	Temperature of gases leaving boiler	degrees Fahrenheit
50	Temperature of gases leaving economizer	degrees Fahrenheit
51	Temperature of air for combustion	degrees Fahrenheit

Hourly Quantities

52	Duration of test	hours
53	Fuel as fired per hour	pounds
54	Dry fuel per hour	pounds
55	Fuel as fired per hour per square foot of grate	pounds
56	Fuel as fired per hour per retort	pounds
57	Dry fuel per square foot of grate per hour	pounds
58	Dry fuel per retort per hour	pounds
59	Combustion space per pound of coal per hour (as fired, dry)	cubic feet
60	Refuse per hour	pounds
61	Actual water per hour	pounds
62	Factor of evaporation	
63	Equivalent evaporation per hour	pounds

Refuse

64	Percentage of refuse in fuel (as fired, dry)	per cent
65	Percentage of combustible in refuse	per cent
66	Carbon burned per pound of fuel (as fired, dry)	pounds

Evaporation

67	Actual evaporation per pound of fuel as fired	pounds
68	Equivalent evaporation per pound of fuel as fired	pounds
69	Equivalent evaporation per pound of dry fuel	pounds
70	Equivalent evaporation per square foot of boiler heating surface per hour	pounds

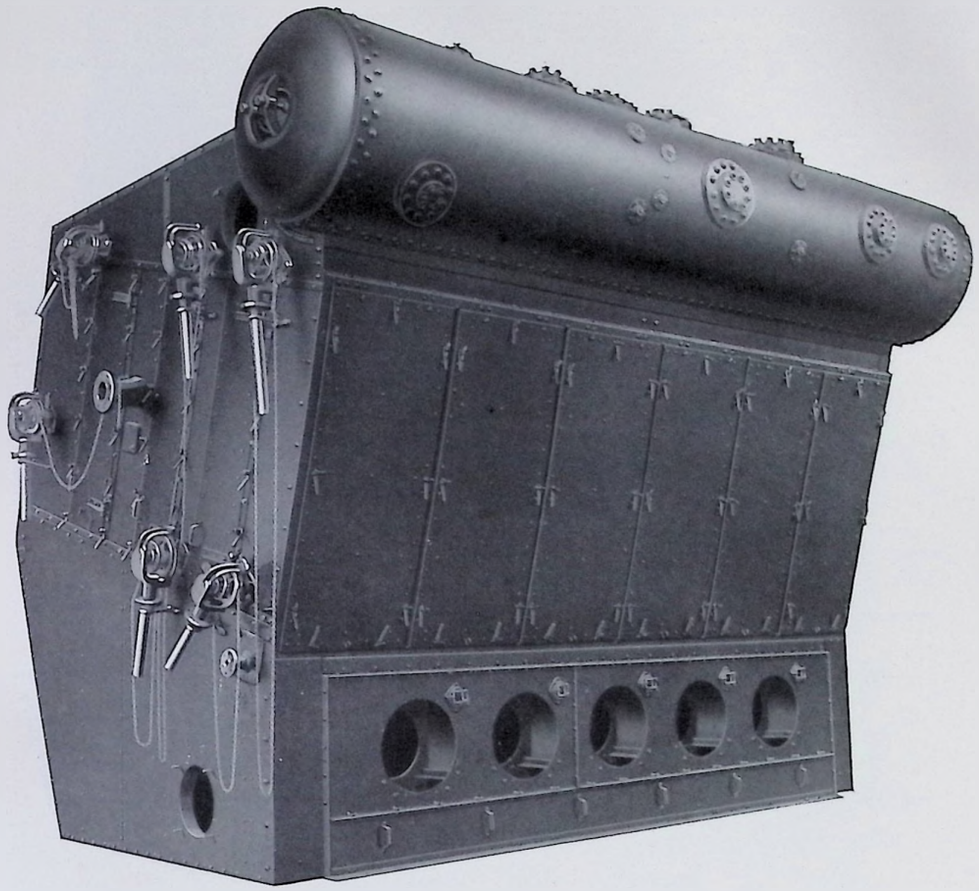
Efficiency

71	Efficiency of boiler, superheater, furnace and grate	per cent
72	Efficiency including economizer	per cent

Heat Balance (based on dry fuel—fuel as fired)

73	Heat absorbed by boiler and superheater	B.t.u.	per cent
74	Heat absorbed by economizer	B.t.u.	per cent
75	Loss due to moisture in fuel	B.t.u.	per cent
76	Loss due to moisture formed in burning H_2	B.t.u.	per cent
77	Loss due to moisture in air	B.t.u.	per cent
78	Loss due to chimney gases	B.t.u.	per cent
79	Loss due to incomplete combustion of carbon	B.t.u.	per cent
80	Loss due to unconsumed combustible in refuse	B.t.u.	per cent
81	Radiation and unaccounted for losses	B.t.u.	per cent

The form for tests with liquid fuel is similar to the foregoing, with such items added or omitted as may be necessary because of the methods of combustion used and different characteristics of the fuel.



One of Four Babcock & Wilcox Boilers, with Interdeck Superheaters and Cuyama Design Oil Burners, Built for the Steamer "Dixie" of the Southern Pacific Company

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Steamer "Caracas," 3500 Shaft Horse-power, Babcock & Wilcox Boilers, Superheaters and Oil Burners
Owners, Atlantic & Caribbean Steam Navigation Company (Red D Line), New York, N. Y.

THE SELECTION OF BOILERS WITH A CONSIDERATION OF THE FACTORS DETERMINING SUCH SELECTION

THE selection of steam boilers is a matter to which the most careful thought and attention may be well given. Within the last twenty years, radical changes have taken place in the methods and appliances for the generation of power. Such changes have been made largely in the prime movers, both as to type and size, and are perhaps best illustrated in central power plant practice. While the improvements in prime movers have been made, the steam-generating equipment has too often been considered more or less a standard piece of apparatus. When the fact is considered that the cost of steam generation is roughly from 65 to 80 per cent of the total cost of power production, it may be readily understood that the most fruitful field for improvements exists in the fire-room end of a ship's power plant.

The improvements to be secured in the fire-room results are not simply a matter of dictation of operating methods. The securing of perfect combustion, with the accompanying efficiency of heat transfer, while comparatively simple in theory, is difficult to obtain in practical operation. This fact is perhaps best exemplified by the difference between test results and those obtained in daily operation even under the most careful supervision. This difference makes it necessary to establish a standard by which operating results may be judged, a standard not necessarily that which might be possible under test conditions, but one which experiment shows can be secured under the very best operating conditions.

The study of the theory of combustion, draft, etc., as already given, will indicate that the question of efficiency is largely a matter of proper relation between fuel, furnace and boiler heating surface. While the possibility of a saving through added efficiency cannot be overlooked, the boiler design of the future must, even more than in the past, be considered particularly from the aspect of reliability and simplicity. A flexibility of operation is necessary as a guarantee of continuity of service.

For the intelligent selection of a boiler it is necessary to appreciate fully the meaning of the terms efficiency and capacity, as expressing rate of steam output, and their relations to each other.

The criterion by which the efficiency of a boiler installation is to be judged is the cost of the production of a definite amount of steam. Considered in this sense, there must be included in such efficiency, the simplicity of operation, flexibility and reliability of the boiler used. The items of repair and upkeep cost must also be taken into account. The governing factor in these items is unquestionably the type of boiler selected.

The factors entering into fire-room efficiency are so numerous that it is impossible to make a statement as to a means of securing the highest efficiency that will apply to all cases. Such efficiency is to be secured by the proper relation of fuel, furnace and boiler heating surface, actual operating conditions which allow the approaching of the potential efficiencies made possible by refinement of design, and a systematic supervision assisted by a detailed record of performance and conditions.

In considering efficiency, it is to be remembered that published boiler performance figures are ordinarily results obtained under test conditions which are usually better than those secured in every-day operation. Frequently under such circumstances it is

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possible to operate for the duration of the test under conditions as to furnace temperature, etc., that would not represent the best commercial efficiency in ordinary operation. For the highest thermal efficiency, the furnace temperature must be maintained at a maximum. Such a maximum temperature, however, combined with certain draft conditions allowable for the duration of a test, might lead to excessive furnace-brickwork upkeep if maintained in ordinary daily operation. In this case the highest thermal efficiency would not represent the maximum commercial efficiency. The closeness with which operating conditions approach test conditions is dependent upon the character of the fire-room supervision and the intelligence of the operating crew.

The bearing that the type of boiler has on the efficiency to be expected can only be realized from a study of the foregoing chapters.

Capacity—Capacity, as defined, is the total weight of steam which a definite amount of boiler heating surface will generate. Broadly speaking, in a properly designed boiler, the evaporative capacity of a certain amount of heating surface, that is, the total weight of steam such surface can deliver, is limited only by the amount of fuel that can be burned under the boiler. While such a statement would imply that the question of capacity to be secured was simply one of making an arrangement by which sufficient fuel could be burned under a definite amount of heating surface to generate the required amount of steam, there are limiting factors that must be weighed against the advantages of high capacity developed from small heating surfaces. Briefly stated these factors are as follows:

1st. Efficiency. As the capacity increases, there will, in general, be a decrease in efficiency, this loss above a certain point making it inadvisable to attempt to obtain a greater steam output from a given boiler. This loss in efficiency with increased rates of output is discussed later.

2nd. Furnace Volume Possible or Practicable. It has been stated that for high efficiencies with any fuel it is necessary that combustion be completed before the products of combustion encounter the boiler heating surfaces. Since the furnace volume limits the amount of fuel that can be burned while meeting this requirement, it follows that it limits the maximum output from a given boiler that can be obtained with good efficiency.

The question of increasing furnace volumes for purposes of high efficiency at high rates of output is much more difficult in marine than in stationary practice. Any increase in such volume in marine work means an increase, not only in space occupied, but in weight. It is essential, then, in the selection of a marine boiler that the furnace volume be kept to the minimum capable of giving the maximum output with reasonably high efficiencies. This necessitates, in such selection, a thorough consideration of the arrangement of boiler heating surface with reference to the furnace, and particularly the furnace form, which should be such as to utilize the volume supplied to the maximum extent.

3rd. Grate Ratio Possible or Practicable. All fuels have a maximum rate of combustion, beyond which satisfactory results cannot be obtained, regardless of draft available or which may be secured by mechanical means. Such being the case, it is evident that with this maximum combustion rate secured, the only method of obtaining added capacity will be through the addition of grate surface. There is obviously a point beyond which the grate surface for a given boiler cannot be increased. This is due to the impracticability of handling grates above a certain maximum size, to the

enormous loss in draft pressure through a boiler resulting from an attempt to force an abnormal quantity of gas through the heating surface and to innumerable details of design and maintenance that would make such an arrangement wholly impracticable.

4th. Feed Water. The difficulties that may arise through the use of poor feed water or that are liable to happen through the use of feed water of practically any source have already been pointed out. This question of feed is frequently the limiting factor in the capacity obtainable, for with an increase in such capacity comes an added concentration of such ingredients in the feed water as will cause priming, foaming or rapid scale formation. Certain waters which will give no trouble that cannot be readily overcome with the boiler run at ordinary ratings will cause difficulties at higher ratings entirely out of proportion to any advantage secured by an increase in the power that a definite amount of heating surface may be made to produce.

To recapitulate: Where capacity is required in a marine boiler, in the sense of high or reasonably high rates of output, the type of boiler selected will play a large part in its successful operation. A boiler must be selected with which there is possible a furnace arrangement that will give flexibility without undue loss in efficiency over the range of steam output desired. The heating surface must be so arranged that it will be possible to install in a practical manner sufficient grate surface at or below the maximum combustion rate to develop the amount of power required. The design of boiler must be such that there will be no priming or foaming at high rates of evaporation, and that any added scale due to such high rates of output may be easily removed. Certain boilers which will deliver commercially dry steam when operated at low rates of evaporation will prime badly when operated at high rates, and this action may take place with a water that should easily be handled by a properly designed boiler at any rate. Such action is ordinarily produced by the lack of a well-defined, positive circulation.

Relation of Efficiency and Capacity—The statement has been made that in general the efficiency of a boiler will decrease as the capacity is increased. Considering the boiler alone, apart from the furnace, this statement may be readily explained.

Presupposing a constant furnace temperature, regardless of the capacity at which a given boiler is run, to assure equal efficiencies at low and high ratings, the exit temperature in the two instances would necessarily be the same. For this temperature at the high rating to be identical with that at the low rating, the rate of heat transfer from the gases to the heating surfaces would have to vary directly as the weight or volume of such gases. Experiment has shown, however, that this is not true, but that this rate of transfer varies as some power of the volume of gas less than one. As the heat transfer does not, therefore, increase proportionately with the volume of gases, the exit temperature for a given furnace temperature will be increased as the volume of gases increases. As this is the measure of the efficiency of the heating surface, the boiler efficiency will, therefore, decrease as the volume of gases increases or the capacity at which the boiler is operated increases.

Further, a certain portion of the heat absorbed by the heating surface is through direct radiation from the fire. Again, presupposing a constant furnace temperature, the heat absorbed through radiation is solely a function of the amount of surface exposed to such radiation. Hence, for the conditions assumed, the amount of heat absorbed by radiation at the higher rates of output will be the same as at the lower rates, but in proportion to the total absorption will be less. As the added volume of gas does not increase the rate of heat transfer sufficiently to offset the constant absorp-

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City of New York Fireboat "John Purroy Mitchel"
Equipped with Babcock & Wilcox Boilers and Oil Burners

tion by radiation, there are therefore two factors acting toward the decrease in the efficiency of a boiler with an increase in the capacity. This relation will hold to a point where the radiation of heat from the boiler setting is proportionately large enough to be a governing factor in the total amount of heat absorbed.

The second reason given above for a decrease of boiler efficiency with increase of capacity, *viz.*, the effect of radiant heat, is, to a greater extent than the first reason, dependent upon a constant furnace temperature. Any increase in this temperature will affect enormously the amount of heat absorbed by radiation, as this absorption will vary as the fourth power of the temperature of the radiating body. In this way it is seen that but a slight increase in furnace temperature will be necessary to bring the proportional part, due to absorption by radiation, of the total heat absorbed, up to its proper proportion at the higher ratings. This factor of furnace temperature belongs rather to the consideration of furnace than of boiler efficiency. There is a point, however, in any furnace above which the result of forcing the combustion will be so poor as actually to reduce the furnace temperature and, therefore, the proportion of heat absorbed through radiation by a given amount of exposed heating surface.

Since it is thus true that the efficiency of the boiler considered alone will increase with a decreased capacity, it is evident that, if the furnace conditions are constant regardless of the load, the combined efficiency of boiler and furnace will also decrease with increasing loads.

From the foregoing brief consideration of efficiency and capacity and their relation to each other, it is evident that no recommendation can be made as to the number and size of boiler units that will hold for all cases. In general, it will be found best to install units of the largest possible size compatible with the size of the installation and operating conditions, with the total power requirements divided among such a number of units as will give proper flexibility of load, with such additional units for spares as conditions of cleaning and insurance against interruption of service warrants.

In closing the subject of the selection of boilers, it may not be out of place to refer to the effect of the builder's guarantee upon the determination of design to be used. Here, in one of its most important aspects, appears the responsibility of the manufacturer. Emphasis has been laid on the difference between test results and those secured in ordinary operating practice. That such a difference exists is well known and it is now generally realized that it is the responsible manufacturer who, where guarantees of performance are necessary, is conservative in giving figures which may readily be exceeded under test conditions, and which may be closely approached under the ordinary plant conditions obtaining in daily operation.

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Steamer "Plymouth," 5500 Indicated Horse-power, Babcock & Wilcox Boilers
Owners, The New England Steamship Company, New York, N. Y.

CARE AND OPERATION OF MARINE BOILERS

THE general subject of fire-room practice may be considered under two aspects. The first is that of broad economy, with a suggestion as to the methods to be followed in securing the most economical results with the apparatus at hand and procurable. The second deals rather with specific recommendations leading not only to economy but also to safety and continuity of service. Such recommendations are dictated from an understanding of the nature of steam generating apparatus and its operation as previously described in this book.

It has already been pointed out that the attention given in recent years to steam generating practice has resulted from a realization of the wide difference between the results obtaining in every-day operation and those theoretically possible. The amount of attention and regulation given the steam generating end of a ship's power plant, however, is ordinarily less than that given the balance of the ship's mechanical equipment, but it may be safely stated that this part of the plant gives the greatest assurance of return for proper attention. This statement probably applies to a greater extent in the smaller installations than in the larger ones where the necessity for fire-room economy is becoming more and more appreciated.

In the endeavor to increase fire-room efficiency, it is of the utmost importance that a standard be set by which average results are to be judged. With the theoretical efficiency obtainable varying so widely, this standard cannot be placed at the highest efficiency that has been obtained regardless of operating conditions. It is better to set it as the best obtainable result for each individual fire-room under its condition of installation and daily operation.

When such a standard has been established there should be an earnest effort to improve existing practice by raising it to this mark. Success in this effort will be proportioned to the degree of intelligent supervision of work in the fire-room. Such direction can only occur when there is full realization of the possibilities under best conditions and also actual results under all the conditions of current operation. What the installation is doing should be made a matter of continuous record so arranged that the results may be directly compared for any period or set of conditions, and where such results vary from the established standard, steps must be taken immediately to remedy the causes of such failure. Such a record is an important check on the losses in the fire-room.

As the size of the installation and the fuel consumption increase, such a comparison becomes almost a necessity. In such installations, the saving in even a fraction of one per cent of the fuel bill may represent a very appreciable annual saving, while the cost of proper supervision to secure it is small. The methods of supervision followed in the larger installations may be elaborate and complete. In the case of the smaller installations, similar methods may be carried out on a more moderate scale, with a corresponding fuel saving and with a relatively small increase in either fire-room organization or expense.

Within the past few years there has been a great increase in the practicability and reliability of the various classes of apparatus for securing the records of fire-room operation. Much of such apparatus is ingenious and, considering the work to be done, remarkably accurate. Even where the accuracy is known to be only within limits of, say, five per cent either way, the records obtained are of the greatest service in con-

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sidering comparative results. Some of the more desirable records, and the apparatus for securing them, are given below.

Inasmuch as the ultimate measure of the efficiency of the fire-room is the cost of steam generation, the most important records are those of the amount of steam generated and fuel consumed. Unfortunately it is practically impossible in marine practice to obtain an accurate measure of the latter, but records of temperatures, gas analyses, drafts and the like, serve as a check on such consumption and, by indicating the distribution of losses, lead to methods of remedying conditions where improvement is possible.

Fuel Records—Many devices for conveniently weighing coal have been developed for stationary work, but in marine practice, because of weight and space conditions, the use of such apparatus is not practicable, and the only weights that can be used are those of total purchases. While such weights are only of use in figuring efficiencies between times of coaling, the efficiency of combustion can be checked by other means described below.

With oil fuel, meters may be used, but these are not particularly satisfactory because of the necessity for frequent calibration and the taking of other data in connection with the actual meter readings. The method ordinarily used with oil fuel is that of tank soundings; obviously in the case of rough weather this method may give results considerably in error.

Water Records—Because of the difficulty of obtaining accurate fuel weights, the water records are not of as much use as a measure of cost of steam generation as they are a measure of prime mover steam consumption.

The apparatus for recording the evaporation of boilers is of two general classes: those measuring water as it is fed to the boiler and those measuring steam delivered. Of the first, the Venturi meter is the best known. Results obtained by such meters are reasonably accurate and they have the advantage over a steam-measuring device that they may be more readily calibrated. Of the steam-measuring devices, the one in most common use is the steam-flow meter. Provided the instruments are selected for a proper flow, they are of the greatest value in indicating steam consumption. Where such instruments are placed on the various engine-room lines, they will immediately indicate any excessive consumption for any one unit. With a steam-flow meter placed on each boiler outlet, it is possible to fix relatively the output of individual boilers, and, considered in connection with some of the "check" records described below, clearly to determine whether its portion of the total steam produced is up to the standard set for the over-all fire-room efficiency.

Flue-gas Analysis—The value of a flue-gas analysis as a measure of furnace efficiency has already been indicated. There are on the market a number of instruments by which a continuous record of the carbon dioxide in the flue gases may be secured and in general the results so recorded are accurate. The limitations of an analysis showing only CO_2 and the necessity of completing such an analysis with an Orsat, or like apparatus, and in this way checking the automatic device, have already been shown. Where such records are properly checked from time to time and are used in conjunction with a record of flue temperatures, the losses due to excess air or incomplete combustion and the like may be directly compared for any period. Such records lead to the control of excess air and also act as a check on individual firemen.

Where the size of the installation will not warrant the purchase of an expensive continuous CO_2 recorder, it is advisable to make analyses of samples for various conditions of firing and to install an apparatus whereby a sample of flue gas covering a period of, say, eight hours, may be obtained and such a sample afterwards analyzed.

Temperature Records—Flue-gas temperatures, feed-water temperatures and steam temperatures are all taken with recording thermometers, any number of which will, when properly calibrated, give accurate results.

A record of flue temperatures is serviceable in checking stack losses and, in general, the cleanliness of the boiler. A record of steam temperatures, where superheaters are used, will indicate excessive fluctuations and lead to an investigation of their cause. Feed temperatures are valuable in showing that the full benefit of the exhaust steam is being derived.

Draft Records—As has been stated, any fuel has some rate of combustion at which the most efficient results are obtained. Such rate corresponds to some definite draft suction or air pressure, and draft gauges, indicating the suction or pressure, are of assistance in maintaining an efficient rate of fuel burning. Draft gauges indicating the difference in suction or pressure in the furnace or the fire-room and the boiler uptake, *i.e.* the draft loss through the boiler, offer a method of approximating the rate at which a boiler is operating. Further, since the draft loss for a given rate of output is less through a boiler externally clean than through one whose surfaces are fouled, such a draft differential is a measure of the boiler's cleanliness.

The installation of draft gauges, preferably of the recording type, is strongly recommended.

An important point to be considered in the installation of all recording apparatus is its location. Thermometers, gas-sampling pipes, and draft gauges should be so located as to give as nearly as possible an average of conditions, the gases flowing freely over the ends of the thermometers, couples and sampling pipes. With the location permanent, there is no certainty that the samples may be considered average, but in any event comparative results will be secured which will be useful in maintaining operating standards. The best permanent location of apparatus will vary considerably with the design of boiler.

The second aspect of fire-room management, dealing with specific recommendations as to the care and operation of boilers, is dictated largely by the nature of the apparatus. Some of the features to be watched in considering this aspect follow.

Before placing a new boiler, or one which has been idle over an extended period, in service, a careful and thorough inspection should be made of the pressure parts, furnace, casings and supporting structure. Baffles should be examined, to see that they are in place and that their condition is such that there can be no gas leakage.

In the case of new boilers, green walls should be thoroughly dried out. To start this drying process, as soon as the brickwork is completed, the damper and the ashpit doors should be blocked open to maintain a current of air through the setting. Whenever possible this should be done for several days before firing. When ready for firing, wood should be used for a light fire, gradually building it up until the setting is thoroughly warmed. If the precautions discussed below have been followed, the regular fuel may then be fired and the boiler put on light load for several days and then put into regular operation.

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The following precautions apply both to new boilers and those that have been out of service for an appreciable length of time.

Internally the boiler should be inspected to assure the absence of dirt, waste, oil and tools. If there is oil or paint inside the boiler, soda ash should be placed in the drum, allowing one pound of soda ash for every ten square feet of heating surface for four-inch tube boilers and one pound soda ash for every twelve square feet of heating surface for two-inch tube boilers, the boiler filled with water to its normal level, and a slow fire started. After twelve hours the fire should be allowed to die out, the boiler cooled slowly, then opened and washed out thoroughly. This will remove oil and grease from the interior of the boiler and prevent foaming when the boiler is placed in service.

The combustion chamber and gas passages should be clean and in good repair.

Dampers, where installed, and fan drives should be operated and shown to be ready for service.

After the boiler is closed, the blow-off valves, water-gauge glass drains, gauge cocks and feed valves should be examined for proper operating condition and closed. The air valve and the valves to the steam pressure gauge should be opened and the boiler filled. The stem of the main stop valve should be eased without lifting it from its seat, to make sure that it is not stuck and to allow for expansion when heating up.

If coal is the fuel, a thin layer should be placed on the grate and a light fire started. In the case of oil, the damper, if installed, should be fully opened, and with a light suction in the furnace a torch should be placed near the burners to obtain quick ignition, and a light fire started by admitting the fuel slowly. With oil, care must be taken to avoid possible flare-backs, and a torch should be used to light all burners until the furnace has been brought up to its working temperature.

In oil burning installations the oil system and burners should be inspected before starting, thoroughly cleaned and shown to be in proper working condition.

The fires should be operated so that a steam pressure is not generated until approximately one-half hour before the boiler is to be cut into the line. In case of an emergency, steam may be raised in Babcock & Wilcox marine boilers in very much shorter time than that recommended, without undue stress or damage to the heating surfaces, but this rapid raising of steam should not be followed in regular service as it will appreciably shorten the life of the furnace brickwork. As an indication of the rapidity with which boilers of this type may be put on the line from a cold condition (water at 100 degrees Fahrenheit) tests have been made with oil fuel where a boiler has been cut in the line at 200 pounds pressure eight minutes and forty-five seconds after the fire has been lighted. When the water has become heated, the water level should be checked by the gauge cocks and the blow-off valves examined for leaks. When steam has escaped from the air valve for a few minutes, it may be closed.

Steam pressure should be raised slowly and when it approaches operating pressure, the boiler should be examined for leaks, the pressure gauge connections blown out sufficiently to assure cleanliness, the safety valve tested by hand, and the boiler given a bottom blow.

Where superheaters located above the boiler heating surface are installed, these should be thoroughly drained before the boilers are fired.

When fires are lighted under boilers fitted with superheaters located at the top of the boiler tube bank, the superheater drains should be opened, thus allowing any

water in the superheaters to run out followed by air and steam as soon as the latter is generated. They should be left open until a slight pressure has built up in the boiler, when they may be closed and then left closed until just before the boiler is cut in the line, when they should again be opened and left open until after the boiler is cut in.

In lighting off boilers fitted with superheaters located in the interdeck position, the drains at both the inlet and outlet of the superheater headers should be opened. After a slight steam pressure has built up in the boiler, the inlet drain may then be closed, the outlet drain then being left partially opened to maintain a slight flow of steam through the superheater. Just before the boiler is cut in the line both drains on the superheater should again be opened wide and allowed to remain so until after the boiler is cut in, when they may both be closed.

Before cutting the boiler into the line, the connections between it and the steam main should be thoroughly drained. The stop valve should be opened very slowly as pressure in the boiler reaches the pressure in the main, to allow these pressures to equalize. If the main stop valve is equipped with a by-pass, this should first be opened gradually and the pressure allowed to equalize before the main valve is opened. Where a non-return steam stop valve, or stop-and-check valve, is installed, the stop valve should be slightly opened when the pressure within the boiler is from 10 to 15 pounds of that in the main and the non-return valve allowed to operate automatically. After the pressures are equalized the valve should be opened fully.

In regular operation, the safety valve and steam gauge should be checked daily. If practicable, the steam pressure should be raised until the safety valve blows. If not, the valve should be gently raised from its seat by hand. If the valve sticks, the pressure should be lowered and the defect causing the sticking remedied. When the valves blow, the steam gauge should be examined. If it shows a pressure different from that at which the valves are set, the gauge should be tested. If such a test shows the gauge to be correct, the safety valve should be adjusted by some competent person. When safety valve springs become weak, care should be taken in screwing down that there is no restriction in the relieving capacity.

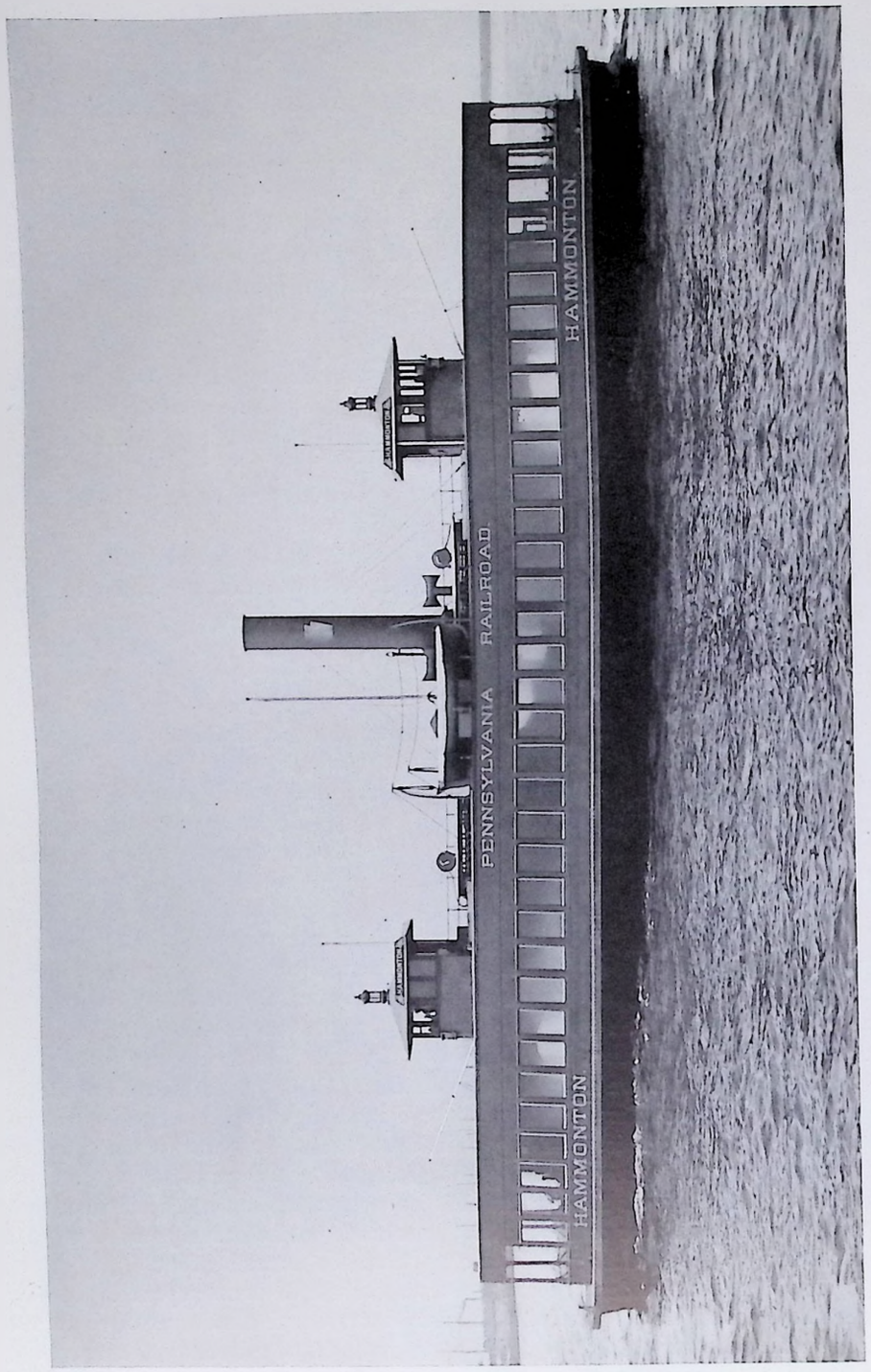
The connections to the steam-pressure gauge should be blown down when raising steam, when testing safety valves and gauge, and in case of foaming.

Water glasses should be kept clean and unscratched. Gauge glasses should be blown down thoroughly at least once each watch, and the height of water as indicated by the glass checked by the use of the gauge cocks. If steam is allowed to leak from the gauge glasses or its connections, the glass will not record a true water level.

Boilers should be blown down through the bottom blow valves at least once a day, and through the surface blow or scummer once a watch. Opening these valves wide and immediately closing them is ordinarily sufficient. Bottom blow valves should be used freely after boilers have been standing with banked fires or operating at very light loads. During such periods the circulation is less active than normally and since there is a greater opportunity for suspended matter to settle on the heating surfaces and in the mud-drum, blowing, after such intervals, should be more frequent.

In case of serious condenser leakage, it is not advisable to attempt to freshen the water in the boiler by blowing and pumping, at least not until the degree of concentration has reached approximately 11047 grains per gallon total solids, which corresponds to $\frac{5}{32}$ reading on the hydrometer scale. A high concentration is preferable to a con-

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Ferryboat "Hammonton," 1100 Indicated Horse-power, Babcock & Wilcox Boilers
Owners, Pennsylvania Railroad

tinuous renewal of water that may contain scale-forming matter, aside from the heat loss in blowing down. Further, pumping and blowing, by causing temperature changes of the pressure parts, may cause scale already deposited to crack off, with the consequent blistering or loss of tubes which results from the presence of loose scale.

When the concentration has reached the figures indicated above, the boiler should be completely emptied and refilled with fresh water if possible. This should preferably be done before the boiler is allowed to cool to atmospheric temperature, as the hot water retains more of the soluble materials in solution than does the cold water and removing the water from the boiler while it is hot will greatly reduce the amount of cleaning necessary after the boiler is opened up.

In a distinct case of low water, resulting either from carelessness or unforeseen operating conditions, it is essential to extinguish the fire in the quickest possible manner. The fuel supply and the air supply to the fuel should be shut off and, if not practicable to cover the fire with wet ashes, the fire should be drawn on to the fire-room floor and quenched. The feed valve should be closed and under no circumstance should water be fed to the boiler. The safety valves should not be opened nor the opening of the main steam valve be changed. In general, nothing should be done to alter the stresses acting on the boiler.

After the furnace has cooled, until it is black throughout, the main steam stop valve may be closed and the boiler taken off the line, following which a thorough examination should be made to determine what damage, if any, has been done before placing it back in service.

The efficiency and capacity of a boiler depend to an extent very much greater than is ordinarily appreciated upon its cleanliness internally and externally, and systematic cleaning should be included as a regular feature in the operation of any fire-room.

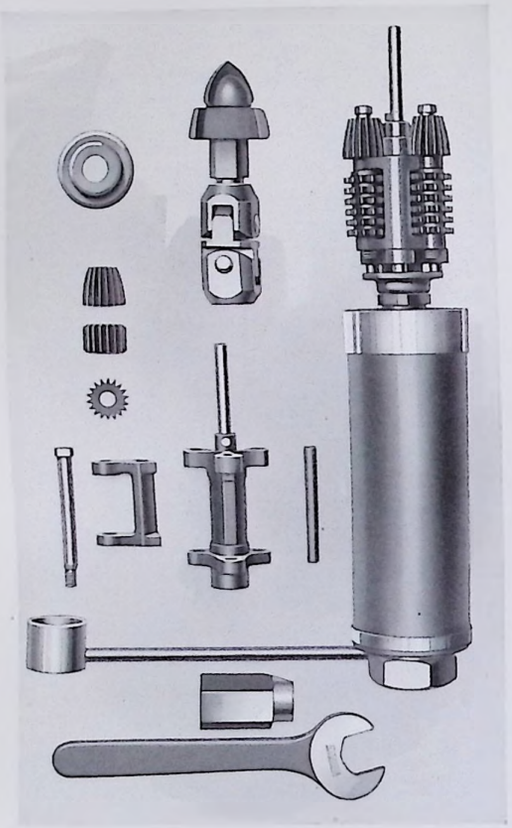
The outer surfaces of the tubes should be blown free from soot with automatic soot blowers or a hand lance at regular intervals. The frequency of such blowing depends upon the class of fuel burned, but should be at least once a day. The best guide for deciding when it is desirable to blow tubes is the temperatures recorded by the uptake gas pyrometers. An intelligent use of these uptake gas temperatures will make it possible to determine with accuracy just what the most economical blowing temperature for any particular installation will be, keeping in mind the amount of steam required each time the tubes are blown. Before blowing, the supply lines to the blowing elements or the hand lance should be thoroughly drained. Steam should not be used for dusting boilers not under pressure.

When permanent soot blowers are installed, care must be taken to see that the nozzles of the soot-blowing elements are maintained in their proper position relative to the boiler tubes. If these be improperly placed or become displaced in such a manner that there is a direct impingement of the steam on the tubes, erosion of the metal will result. In blowing with a hand lance, no one portion of the heating surface should be blown for too long a period or similar erosion will occur.

Boilers should be taken off the line at regular intervals for internal inspection, cleaning and repairs. In cutting out a boiler, the fuel supply should be stopped and, when the fuel is coal, the air supply shut off and the dampers, if installed, closed. When the boiler no longer requires any feed, the main steam valve should be shut. It is best to clean the heating surfaces externally before the fire has died out. The boiler and setting should be allowed to cool slowly and, when possible, should be allowed to stand

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twenty-four hours after the fires have been drawn before emptying and opening. Any attempt to hasten the cooling process by allowing cooler air to pass through the furnace will result in brickwork difficulties. While the boiler is off for cleaning, a careful inspection should be made externally and internally, and all leaks of steam, water, or air through the casings stopped. Internally the tubes should be cleaned from scale and sludge which will accumulate, due to the concentration of solids present in practically any boiler-feed water. This can best be accomplished by the use of an air or water-driven turbine. The illustration on this page shows a turbine cleaner which has been found to give satisfactory results. The wheel cutter head should be used except where the scale has been allowed to form to an excessive thickness.



Turbine Tube Cleaner, Showing Two Types of Head and Accessories

When the scale has been allowed to accumulate to an excessive thickness, the work of removing it is difficult. Where the scale is of sulphate formation its removal may be facilitated by filling the boiler with water in which there has been placed fifty pounds of soda ash to each 1000 gallons of water, starting a slow fire and allowing the water to boil for twenty-four hours without allowing any pressure on the boiler. It should then be cooled slowly, drained, and the turbine cleaner used immediately, as the action of the air tends to harden the scale. While the use of chemicals in feed water is permissible with a view to preventing the formation of scale, such an agent should not be introduced into the boiler, while it is in operation, with a view to softening or loosening any scale that may already be present in the boiler.

Aside from the aspect of efficiency and capacity, a clean interior of boiler heating surfaces insures protection from burning. In the absence of a blow-pipe action of the flames, it is impossible to burn a metal surface when water is in intimate contact with that surface. Any formation of scale on the interior surfaces of a boiler will keep the water from those surfaces and increase their tendency to burn. Particles of loose scale, which may have become detached, will lodge at certain points in the tubes and act at such points in the same manner as a continuous coating of scale except that the tendency to burn is localized. If oil is allowed to enter the boiler with the feed water, its action will be the same as that of scale, in keeping the water from the metal of the tubes and increasing their liability to burn.

It has been proved that a very large percentage of tube losses is due to the presence of scale which, in many instances, has been so thin as to be considered of no moment, and the importance of maintaining the interior of boiler heating surfaces in a clean condition cannot be emphasized too strongly.

If pitting or corrosion is noted, the parts affected should be carefully cleaned and painted with French zinc. The cause of such action should be determined immediately and steps taken to see that a proper remedy is applied.

When making an internal inspection of a boiler or when cleaning the interior of the heating surfaces, great care must be taken to guard against the possibility of steam entering the boiler from any other boilers on the line, through open blow-off valves or through the careless opening of the boiler stop valve. Bad cases of scalding have resulted from neglect of this precaution.

If a boiler is to remain idle for some time it is liable to deteriorate much faster than when in service. If the period for which it is to be laid off is not long it may be filled with water while out of service. The boiler should be thoroughly cleaned, internally and externally, and any accumulation of scale removed from the interior surfaces. It should then be filled with water to which about four buckets of soda ash have been added, a very light fire started to drive the air from the water, the fire then allowed to die out and the boiler pumped full. When boilers are idle and filled, the water within should be kept cold, as lukewarm water tends toward a maximum corrosion. This necessitates tight feed and steam connections.

If the boiler is to be out of service for an extended period, it should be emptied, cleaned and thoroughly dried. A tray of quicklime should be placed in the steam drum, the boiler closed up, the grates covered and a quantity of quicklime placed on these. Special care must be taken to prevent air, steam or water leaks into the casings and brickwork or onto the pressure parts, to obviate danger of corrosion.

When a boiler is to be out of service for a period of three months or more it should also be given a protective external coat of some material such as red lead, black japan or tar paint to prevent external corrosion.

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United States Battleship "Florida," 41,810 Shaft Horse-power, Babcock & Wilcox Boilers, Superheaters and Oil Burners

MARINE BOILER BRICKWORK

THE consideration of furnace linings for marine boilers may be divided into three parts, namely, furnace design, quality of brick and insulating material used, and workmanship of the installation.

Design—As has been indicated, the furnaces of marine boilers are of necessity much smaller than those commonly used in stationary practice. While such furnaces, which are rarely more than six feet in height from the floor pan to the boiler tubes, are not conducive to the best of combustion conditions, from the standpoint of brickwork difficulties, they have certain advantages over larger furnaces. Because of the limited height, there is but little compression exerted in the refractories of the lower and hotter portions of the walls, a factor that has a decided bearing on the life of refractories in higher settings.

Because of the relatively small areas of marine furnace walls, the problem of expansion is not so serious as in the case of larger furnaces; however, because of the rigid casing construction, this has to be given proper consideration. Further, because of the small refractory wall area compared with the large absorption area of the exposed tubes, the furnace temperatures developed are usually less than in the case of stationary units.

Where weight is of primary importance, as in the case of high-speed vessels, the furnace wall refractories are usually made 4½ inches thick, the brick being laid with a stretcher construction. Where weight is not so important a factor, as in the ordinary merchant marine service, the thickness of the refractory furnace lining is made 9 inches, the brick being laid with header construction and the joints broken between courses.

The refractory furnace lining is backed with a calcined diatomaceous earth insulating block or brick, ordinarily 2½ inches thick. Between the calcined insulating blocks and the steel casing there is placed a layer of light blocks composed of raw diatomaceous earth, this material being an excellent insulator, light in weight and superior in both respects to the calcined block, although inferior to the calcined block in its resistance to high temperatures. The thickness of this layer of insulating material is ordinarily made 2 inches with 4½-inch refractory furnace lining and 1 inch with the 9-inch brick lining.

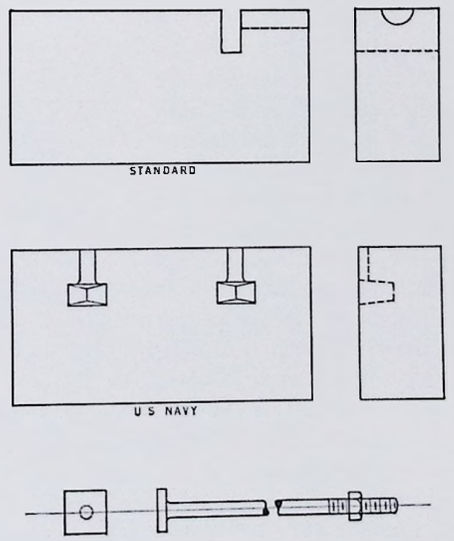


Fig. 20. Two Forms of Furnace Anchor Brick and Anchor Bolt

Because of the vibration, rolling, etc., to which marine boilers are subjected, it is customary to tie the refractory lining of the furnace on all sides to the boiler casing with tie bolts, to insure such linings remaining in place under all conditions. With the 4½-inch lining, it is customary to use one bolt for each two bricks, whereas with

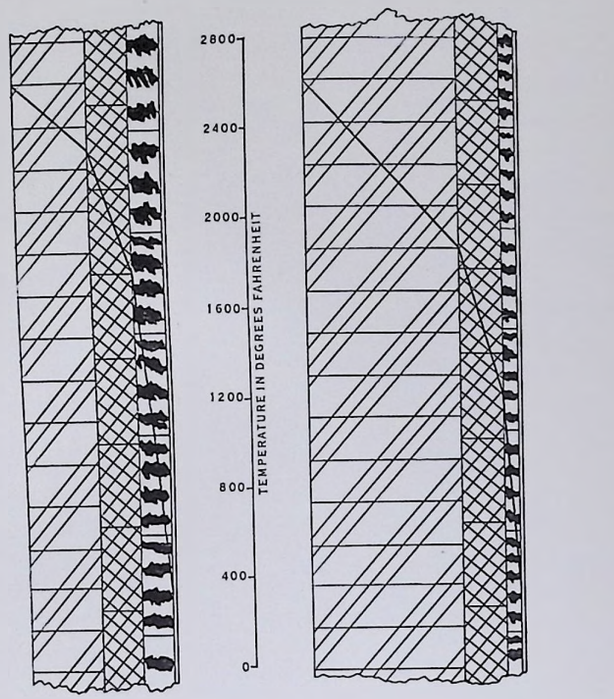
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the heavier 9-inch wall, the number of bolts per square foot of wall may be made appreciably less.

The heads of the anchor bolts are secured in the furnace brickwork lining, the threaded end extending through the insulating material and the steel casing and being fitted outside with a washer and nut.

The design of the bolt and forms of brick for the two thicknesses of wall used in Babcock & Wilcox marine practice are shown in Fig. 20.

It is essential that the bolts used for holding the refractory material in place be of high-grade heat-resisting metal. The heads of such bolts, imbedded in the furnace brickwork, are subjected to high temperatures and if any ordinary steel, heat-resisting metal-coated steel, or iron bolts are used, satisfactory results cannot be obtained. The heads of such bolts, under the temperatures to which they are exposed, fail, and the threaded shanks have no holding power. In the case of iron bolts, as the head fails, it is usual for the iron to combine with the silica in the brick to form a black silicate of iron which flows at a temperature of approximately 2000 degrees Fahrenheit and fluxes any brickwork with which it comes into contact.



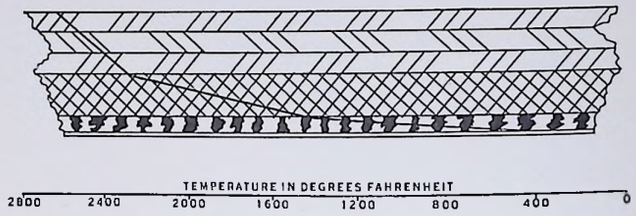


FIG. 21. TYPICAL FURNACE WALL AND FLOOR CONSTRUCTION WITH TEMPERATURE GRADIENTS

It has been found that the following materials give satisfactory service in this work: nickel containing approximately 4 per cent manganese, this alloy being used in either the cast or rolled form; chrome iron containing approximately 25 to 30 per cent chromium, this alloy to be used in the rolled form only, the material when cast being too brittle for the purpose; various combinations of chromium, nickel and iron, it being assumed that there are sufficient proportions of the two former materials in the alloy to assure a high heat-resisting product which, at the same time, is not too brittle.

In the case of oil-fired boilers, the floor construction is of as great importance as that of the furnace walls. A construction used in Babcock & Wilcox marine boilers which has given satisfactory results is as follows:

Immediately above the steel floor pan, 1 inch of uncalcined diatomaceous earth block is laid. Above this 2½ inches of calcined insulating brick is placed, which preferably should have been burned at a temperature in excess of 2300 degrees Fahrenheit. Above the insulating brick two, or sometimes three, layers of split brick (1¼ inches thick each) are laid with joints broken from layer to layer. An expansion space of 1/16 inch is left between the split bricks, and a similar space of ½ inch is left around the foot of all walls.

Typical Babcock & Wilcox marine boiler, wall and floor construction, together with temperature gradients through such constructions (assuming a furnace wall temperature of 2600 degrees Fahrenheit) are shown in Fig. 21.

QUALITY AND PROPERTIES OF REFRACTORIES AND INSULATING MATERIALS—The rate of driving modern marine boilers has greatly increased the severity of the conditions coming upon the furnace linings and thus has emphasized the importance, both from an operative and financial standpoint, of using only materials of the best quality. The serious effect of any failure while at sea is apparent. The limited stay in port will usually mean overtime work for repairs of any moment to the furnace linings with attendant increased cost. It is very evident, therefore, that it is true economy to use only the highest grade materials, which have a relatively long life with infrequent repairs and renewals.

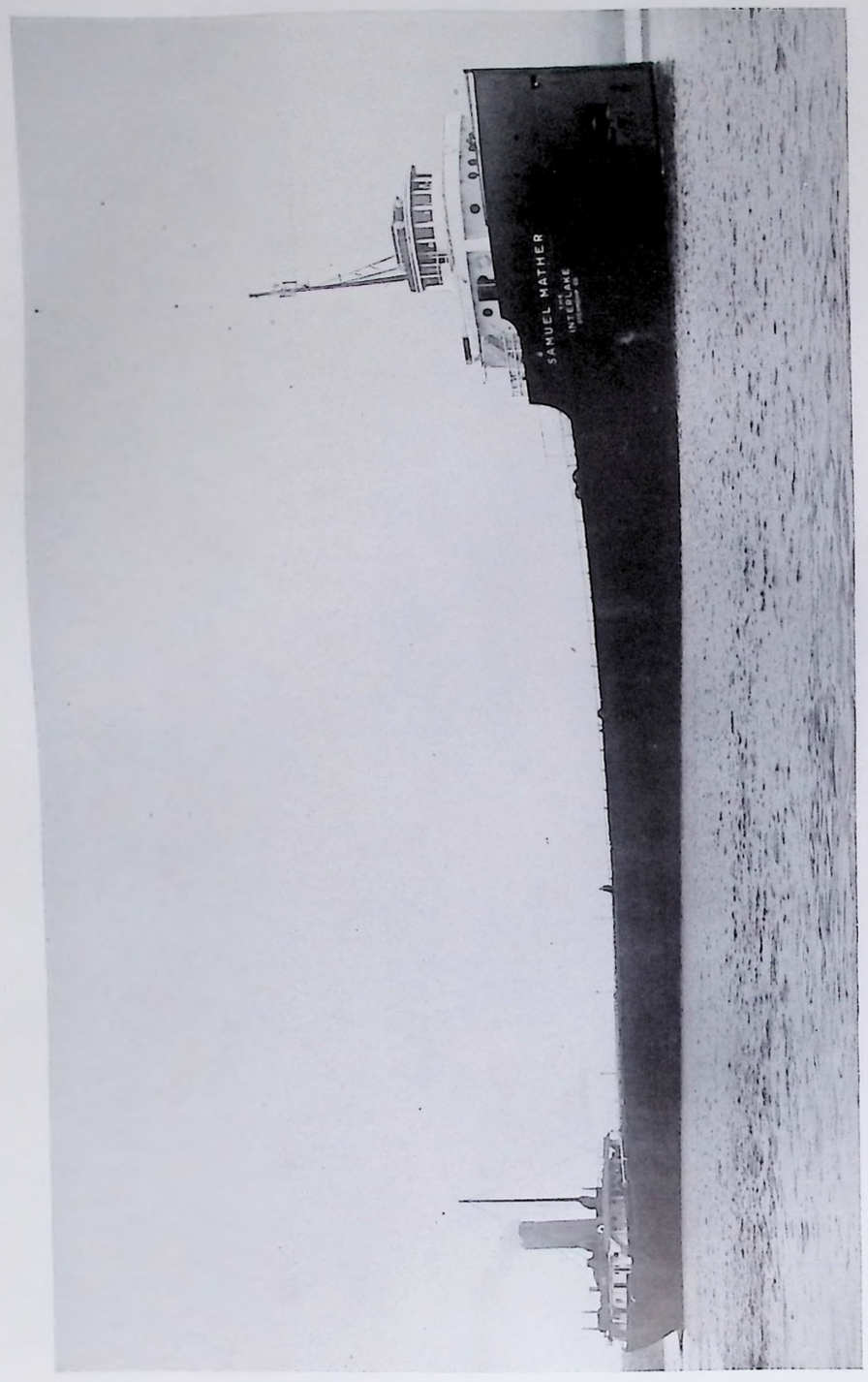
Assuming proper workmanship in the building of the furnace, and considering only the quality and properties of the refractory and insulating material entering into such construction, the causes of failure in marine furnaces may be said to be few. Such causes in the case of refractory material, and in the order of their relative importance are:

1. Spalling.
2. Fusion.
3. Expansion and Contraction.
4. Deformation from Load.
5. Slag Erosion and Clinker Adhesion.

It is to be noted that the relative importance of the causes of the failure of furnace linings in marine practice is somewhat different from the case of stationary boilers. This is due to the greater proportion of oil-fired boilers in the former than in the latter field.

Spalling—The term spalling, correctly used, is applied to the fracture or breakage of firebrick resulting from thermal shock. It should not be confused with cracking or breakage due to stresses applied from without the individual brick resulting from improper furnace design as regards allowance for expansion in the furnace structure as a whole, which is usually accompanied by a distortion of the brick. When true spalling occurs, the fractures take three distinct forms: The first result in the exposed ends of the firebrick breaking or chipping off, leaving a distinctly convex surface to the remaining end of the brick. This type of fracture is caused by shearing stresses developed from the thermal shock of rapid heating. The second is a straight fracture across the brick or across the corners caused by the thermal shock of rapid cooling. The third is a straight fracture across the width of the brick, but only a short distance

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Steamer "Samuel Mather," 2500 Indicated Horse-power, Babcock & Wilcox Boilers and Superheaters
Owners, The Interlake Steamship Company, Cleveland, O.

from its exposed end. This is caused by the stresses due to the advanced vitrification of the exposed end and the repeated application of thermal shock.

The most extreme conditions tending to cause failure from this cause are produced in mechanical burner, oil-fired furnaces. Such conditions result from the high temperatures developed and the flexibility of the burners. The effect of high temperatures is to produce a higher degree of vitrification of the exposed end of the brick and a less elastic structure. The effect of flexibility is an extremely high rate of heating and cooling. When burners are lighted in rapid succession, the furnace end of the brick is heated faster than the opposite end, because of its comparatively low conductivity, thus causing unequal expansion throughout the brick. The opposite occurs when burners are cut out rapidly, the furnace end cooling more rapidly, and thus causing more rapid contraction.

The action described sets up stresses which, after repeated application, fracture the firebrick. Recent investigations have shown that the shearing stress from rapid heating is applied along isothermal surfaces which results in the convex form of the nose of the brick remaining in the furnace when the spalled section has fallen out.

It has recently been shown that all firebrick material is in a plastic or semi-plastic state at a temperature as low as 1000 degrees Fahrenheit, and that the plasticity increases with increased temperatures. This explains the reason for spalling at low temperatures, since only at such temperatures are firebrick in a brittle state, sensitive to stresses produced by thermal shock.

The selection of a proper firebrick to resist spalling is a difficult problem. This is due to the fact that the standard methods of testing are necessarily made on new bricks and not on those that have been subjected to prolonged heat in service. Such prolonged heating tends to alter the homogeneity of the brick's structure, since one end is subjected to more intense heat than the other, and the result is a structure more sensitive to thermal shock. Due to this fact, many firebrick, that, in laboratory spalling tests show results that would indicate a tendency toward failure from spalling in service, actually stand up well under service conditions, because they may possess the property of undergoing only a slight or no structural change when subjected to prolonged heating.

While, as stated, the selection of a firebrick that will minimize the possibility of furnace failure resulting from spalling is a difficult problem, the following factors, having to do with the materials entering into the manufacture of the brick, the furnace design, and the actual boiler operation, if given proper consideration, will tend toward obviating trouble from this source.

The materials used in the manufacture of firebrick have a considerable bearing on the resistance of the brick to thermal shock. It has been rather conclusively proved that a brick having a low coefficient of expansion within the limits of the temperature range to which it will be exposed, will show a high resistance to failure through spalling. This property is inherent in the material used and cannot be altered to any appreciable extent by any method of manufacture. In general, a firebrick with the kaolin or mullite ratio of alumina to silica will have a more uniform and lower coefficient of expansion than those with a higher silica content.

A second factor in the spalling resistance of a firebrick, affected by the materials entering into its manufacture and the temperature of vitrification, is that of permanent volume change. Absence of such permanent change within the limits of service tem-

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peratures is a property required for good spalling resistance. This is due first, to the fact that the homogeneity of the brick is not altered, and second, without shrinkage, the joints between bricks will not be opened. If such shrinkage occurred the brick would be heated on five sides instead of only one. The change in homogeneity of the brick would tend to render it more sensitive to thermal shock, while the heating on more than one surface would increase the suddenness of such temperature shock. Both would result in a lower spalling resistance.

Insofar as furnace design affects spalling, it may be said that a properly insulated refractory furnace lining will appreciably reduce the possibility of failure from this cause. Proper insulation will result in a more uniform heating of such lining when starting up, and a more uniform cooling when the boiler is taken out of service, either of which conditions will reduce the thermal shock to which the refractories are subjected.

In oil-fired boilers, the tendency toward spalling difficulties may be minimized through proper operation of the individual burners. It is recommended in this class of firing that one burner, farthest from the furnace walls, be lighted first, and the boiler operated on this burner alone until the walls are brought to a temperature approaching that which they will ultimately reach. In shutting off burners the air registers and the dampers should be closed to give a maximum time for furnace cooling.

In coal-fired furnaces, troubles resulting from brickwork spalling are ordinarily of minor importance. This is due to the fact that the furnace temperature changes are much less rapid than with oil firing, and because the furnace linings usually have a coating of clinker or slag which lessens any temperature shock. In other instances, the walls may be covered by a glaze formed by a chemical combination of the impurities in the coal with certain constituents of the firebrick which will render the surface more plastic at lower temperatures and therefore less sensitive to thermal shock.

With any firebrick furnace lining, the least trouble will be experienced under continuous and uniform operating conditions. Such conditions, however, in many classes of marine service cannot be maintained or controlled, and, in the case of intermittent service in the merchant marine field or under maneuvering conditions in Navy practice, alternate heating and cooling of furnace brickwork cannot be avoided.

Fusion—Brickwork failure from fusion occurs in most instances in oil-fired furnaces operated at high rates of combustion, though it is occasionally found with certain other fuels having constituents that tend to aid the fluxing of the inside face of furnace linings. With oil-fired furnaces, failure from this source is of real importance, though even with oil fuel it is believed that the mineral constituents of the oil, deposited on the brickwork, cause it to flux rather than to fuse from temperature alone. The percentage of impurities such as iron oxide, sulphur, lime, etc., is small, but the amount of oil handled in a single furnace under present-day practice is large, and as such oil is finely atomized and brought into contact with the brickwork surfaces, the resulting effect may be serious. Further, these impurities are all violent fluxes at high temperatures. This theory is borne out by the fact that, in many oil-fired furnaces, brick which are of first-class fire-clay materials and ordinarily have a fusion point of 3000 or 3100 degrees Fahrenheit, apparently melt away although the maximum furnace temperature developed is not over 2800 or 2900 degrees Fahrenheit.

The fusion point of fire-clay materials is also affected by certain conditions of combustion and of draft in the furnace. Investigation has shown that a highly

reducing atmosphere or one which contains a high percentage of water vapor may lower the fusion temperature of firebrick as much as 200 degrees Fahrenheit. A plus pressure carried in the furnace heats the brickwork by convection through the pores of the brick as well as by conduction through its materials and, in addition, amplifies the effect of carbon monoxide present in a reducing atmosphere, and of water vapor, by causing their penetration of the brick to a greater depth. A suction or negative pressure in the furnace will not only eliminate the above cause of reduction of fusion point, but will also tend to cool the brickwork by the infiltration of air through the pores of the brick from the outside of the structure.

In addition to failure from actual fluxing of the brick as a whole, a partial melting of some of the constituents of the brick may result in failure, not through actual fusing, but through an alteration in structure that will cause failure from other sources such as spalling.

Thermal Expansion—Fracture of individual bricks or failure of entire furnace walls as a result of expansion are of more frequent occurrence than should be the case. Marine boiler casings are usually of rigid construction, and, unless provision is made in the design for brickwork expansion, there will be trouble. Sometimes, also, the brickwork is not properly laid up to allow for expansion. It is necessary that extremely thin joints and partly calcined or preshrunk mortar* should be used, and this makes a very compact structure with no internal opportunity for expansion. Hence, the importance of provisions in the design for expansion at the extremities of the walls.

The average fire-clay brick has a thermal expansion of from 0.5 to 1.0 per cent and such expansion is reversible. In general, the amount of expansion is in direct proportion to the silica content, though other properties such as porosity, degree of vitrification, etc., affect the coefficient of expansion to a lesser degree.

As has been indicated in the consideration of furnace design, care must be taken in installation of furnace linings to provide for both vertical and horizontal expansion of all walls and floor.

Failure from expansion may result in the bulging inward of an entire wall with its ultimate destruction by falling into the furnace, or by the pinching and breaking off of individual brick.

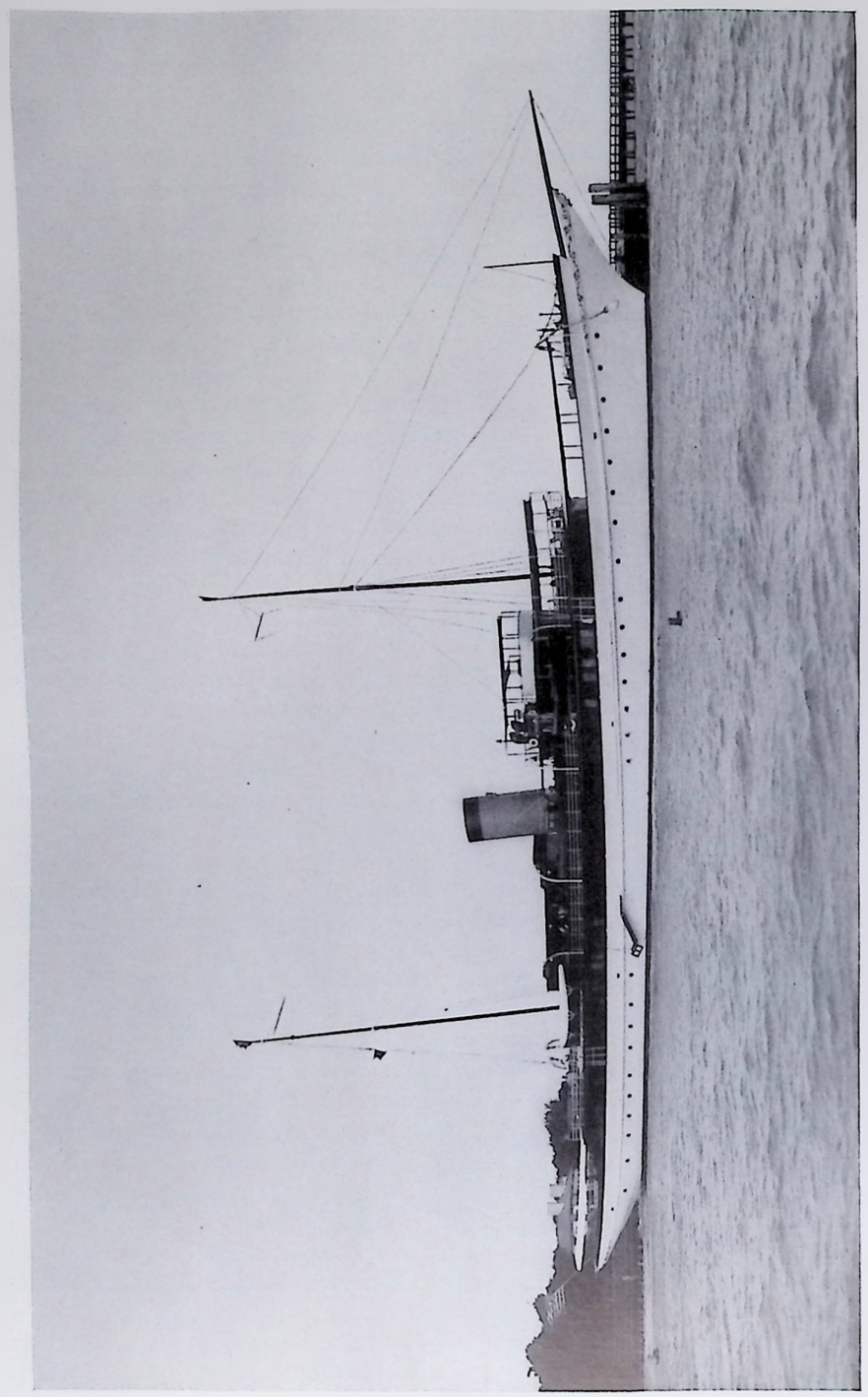
Deformation from Load—In marine practice, because of the limited heights of furnace walls, and the comparatively low temperatures in service, the ability of firebrick to carry loads at high temperatures is not, ordinarily, of primary importance. On the other hand, with a furnace insulated in such manner that the temperature of the lining throughout its thickness is little less than that of the furnace, even a load represented by a height of, say, 6 feet, may cause deformation. The average first quality firebrick will withstand a load of 25 pounds per square inch when heated throughout to approximately 2200 degrees Fahrenheit without deformation, but the load that can be carried is rapidly reduced as the temperature increases.

The capacity for load carrying is governed by the density of the brick structure and the temperature at which the brick is burned in manufacture. This temperature is limited by the materials used in such manufacture.

Slag Erosion and Clinker Adhesion—The mineral residues of most fuels are incombustible, but at sufficiently high temperatures they combine with firebrick ma-

* See "Workmanship," page 255.

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U. S. Naval Patrol Vessel "Nokomis," 2000 Indicated Horse-power, Babcock & Wilcox Boilers

terials, causing fusion at a temperature below the brick's original melting point. The slag formed by the combination is sometimes sufficiently fluid not only to flow over the face of the wall, thereby cutting and fluxing it away, but also to be drawn into the pores of the brick by capillary action. It will then flux the brick bond and carry the grog particles away by mechanical flotation. The rate at which brickwork failure from this source will occur is dependent upon the amount and violence of the fluxing constituents of the fuel and the temperatures at which the furnace is operated.

In marine practice, failure from furnace brickwork due to slagging is not of frequent occurrence, since the coals in ordinary use are of high grade, containing a minimum amount of fluxing constituents, and furnace temperatures developed are not sufficiently high to produce a very fluid mixture. Such failure, too, is not rapid, and is more often the result of attempts to remove slag from the lining than resulting from the actual slag formation. A dense brick with a smooth surface properly built into the wall with tight joints and smooth surfaces delays the formation of clinker and extends the length of periods necessary between removal of clinker by barring. In Babcock & Wilcox marine boilers that are to be coal fired, side-wall water-boxes, described earlier in this book, eliminate the trouble from this source.

The resistance of firebrick to coal-ash slag is primarily dependent upon the chemical constituents of the brick, though its structure is of some importance, a dense non-porous brick showing the greatest slag resistance.

Insulating Material—As stated in the discussion of furnace design, two classes of insulating material are ordinarily used behind the refractory furnace lining to keep the casing cool and the radiation loss at a minimum. The material placed immediately behind the firebrick must be able to withstand the high temperatures at this point, while the second, placed next to the casing, must have the highest possible insulating value.

Practically all of the high temperature insulating materials now available are made of calcined diatomaceous earth. Such material is chemically practically 100 per cent silica, and is in the form of microscopic closed air cells. The process of calcination preshrinks the material so that it will withstand higher temperature without further shrinkage than will the uncalcined raw earth.

The calcined insulator should possess the following properties in their order of importance:

1. Ability to withstand comparatively high temperatures without change of condition.
2. Insulating efficiency.
3. Strength.
4. Minimum weight.

The calcined insulator placed directly behind the refractory lining, if such lining be 9 inches thick, will, in marine boiler furnaces burning oil fuel, be called upon to withstand temperatures of from 1900 to 2300 degrees Fahrenheit, depending upon the efficiency of the insulation. In other words, as the efficiency of the insulation is increased and the heat passing through the walls is decreased, the materials in the wall structure are, at any given point in the cross-section of the wall, required to withstand higher temperatures. The limiting factors in the design of insulated walls are obviously the temperature limits which the materials entering into their construction can withstand.

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One of the temperature gradient diagrams shown in Fig. 21 illustrates this point. In the case of a refractory lining 9 inches thick, it will be noted that for a furnace temperature of 2800 degrees Fahrenheit, the temperature of such lining will be some 2600 degrees Fahrenheit, the lower temperature resulting from the "surface resistance" or "emissivity" of that face, or the resistance, independent of thickness, of the surface to the passage of heat waves. The temperature of the outer, or casing, end of the 9-inch refractory will be reduced to approximately 1900 degrees Fahrenheit, due to the insulating properties of the brick itself. Thus, the calcined insulating material backing this refractory brick lining must be capable of withstanding this temperature without change of structure. Because of the insulating properties of this backing of the refractory furnace lining (in case of 9-inch lining, 2½ inches of calcined material), the temperature at the outer or casing face of this primary insulator drops to approximately 1200 degrees Fahrenheit and it is this temperature which the raw uncalcined pressed block must stand.

Failure of calcined insulating material is ordinarily from shrinkage, though temperatures above certain limits will cause disintegration and result in a loss of insulating efficiency. Thus, if the calcined insulator shrinks at temperatures below the 1900–2300 degrees Fahrenheit range given above, the structure is so altered that the insulating efficiency is reduced and the temperature to which the uncalcined raw material is subjected is increased above its insulating limit and the entire combination as an insulating structure fails.

Insulating Efficiency—In the process of calcining diatomaceous earth, its coefficient of conductivity is increased with a corresponding loss in its insulating efficiency, but because of possible failure due to shrinkage, some of the insulating value of the raw uncalcined material must be sacrificed.

Strength—The ability of an insulator to maintain its manufactured size and shape in service is of importance. Because of the materials from which such insulating blocks are made, and because of the comparatively low temperatures at which such materials are calcined, the resulting available products are inherently mechanically weak. Failure due to lack of structural strength usually results in the disintegration of the insulating brick due to the movement of the furnace walls and the consequent settling of the insulating material in a powdered form, leaving the upper portion of the furnace lining unprotected by insulation. Whether such disintegration is the result of high furnace temperatures or from mechanical breakage has not been definitely established, but it is possible on board ship that vibration may account for this trouble.

Weight—The weight of the insulating material used is of importance in the case of high-speed vessels, but should not be considered to the detriment of other properties in merchant marine practice. It might be noted, however, that the insulating values of two or more representative insulating materials may be approximated from a comparison of their weights: that is, for the ordinary commercial materials available, the weight varies inversely with the insulating value, or, the lighter the weight, the higher such value.

Raw or Uncalcined Insulation—The relative importance of the properties of the raw or pressed insulator is as follows:

1. Insulating efficiency.
2. Shrinkage.
3. Strength.

Since this layer of insulation is farther away from the furnace, its insulating efficiency is of greater importance than its ability to withstand heat without shrinkage or disintegration. The ordinary raw or pressed block insulator available is made of diatomaceous earth and asbestos fiber or of mineral wool. As a class, these materials vary but little in their coefficient of conductivity, such coefficient running from 0.5 to 0.75 B.t.u. per hour per square foot of area, per inch of thickness per degree Fahrenheit difference in temperature.

The standard commercial block will not be affected under temperatures up to 1600 degrees Fahrenheit.

The strength of such blocks is ordinarily sufficient to eliminate danger of disintegration, though cases of failure from this cause are occasionally reported, the blocks breaking down into a fine powder.

Jointing Materials—The proper clays or cements for laying up furnace brickwork are of the utmost importance in the life of the furnace. The functions of such jointing materials are as follows:

1. To act as an air seal and cushion between bricks without shrinking, expanding or cracking.
2. To be equal in their refractory quality to the firebrick with which they are to be used.
3. To burn in service to a density sufficient to prevent the penetration of flames, slags, vapors, gases, etc.
4. To vitrify upon the application of heat.
5. To be of such character as not to disintegrate and fall from between joints.

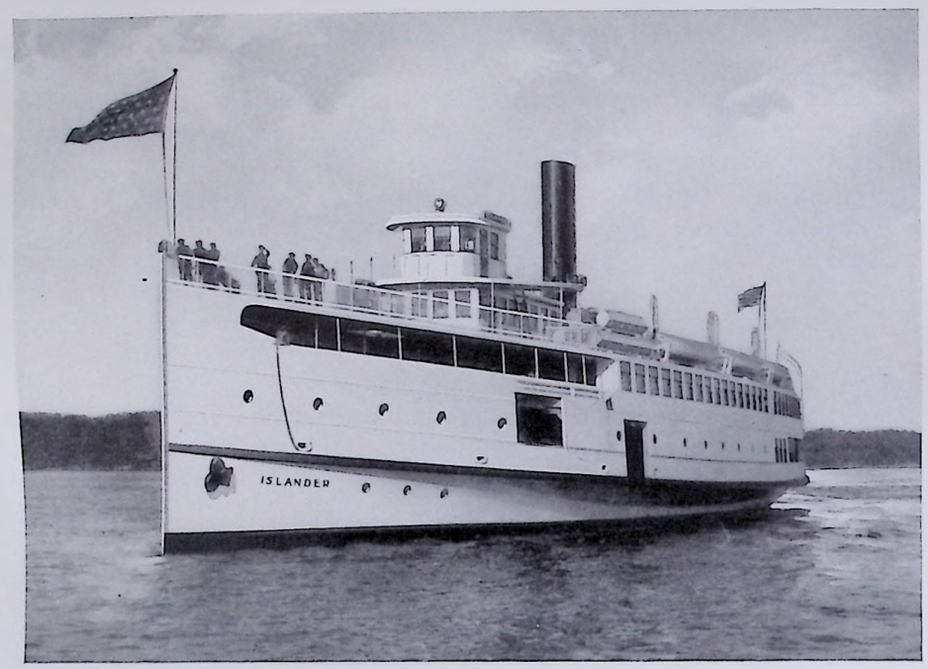
Jointing materials may be divided into two general classes, fire clay and special cements. While the latter class is so extensive that particular grades cannot be considered here, it is sufficient to state that provided they comply wholly with the requirements as outlined above, or with a minimum variation therefrom, they should be suitable for the service required. However, practically all high temperature cements have the common property of a strong bonding strength, and this property cannot be considered an advantage. This bonding strength results in a monolithic structure of the furnace lining which will tend to crack on expansion and contraction because of the non-uniform application of heat. On the other hand, the use of a jointing material without a high bonding strength will allow each individual brick to adjust itself to expansion and contraction without accumulating such action in one or more large cracks which may ultimately destroy the wall structure. Further, the large cracks that may result from the use of a high bonding material form a passage either for the infiltration of air to the furnace or the flow of hot gases to the insulating material, causing its disintegration or fusion.

In the past, common practice has ordinarily been to use as jointing material a plastic fire clay. With increased furnace temperatures as at present common, it was found that the use of the commercial grades of plastic fire clay caused trouble from shrinkage and in some cases from actual fusion. Such shrinkage caused a settling of the walls and an opening of the joints, exposing the firebrick to the action of the furnace gases on five faces and also exposed the insulating materials so that they disintegrated or melted. The fusion of the plastic fire clay sometimes caused it to become fluid and to actually flow from the joints and down the face of the wall, leaving open joints and fluxing and eroding brickwork.

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It is now becoming common practice to make a fire clay cement of a certain percentage, ordinarily about 75 per cent, of calcined flint clay or crushed brick and 25 per cent of the best available plastic clay. The calcined clay or crushed brick must be pulverized to a degree where 100 per cent will pass through a 20-mesh screen in order to allow dipping of the brick and its installation with brick-to-brick joints.

The best results are secured where the cement is ordered from the same manufacturer as that of the firebrick selected. The cement will then be of the same clays as the brick and will have approximately the same expansion, contraction, and refractoriness. A cement made up with calcined clay or crushed brick has a comparatively



STEAMER "ISLANDER," 1200 INDICATED HORSE-POWER, BABCOCK & WILCOX BOILERS
OWNERS, THE NEW ENGLAND STEAMSHIP COMPANY, NEW YORK, N. Y.

small amount of shrinkage because of the preshrunk ingredient. Further, plastic clay of high alumina content, and hence a higher fusion point, may be used, since it is not necessary to offset the shrinkage by means of silica.

PLASTICS—In order to cap the furnace walls for certain replace work, such as special oil-burner tile, peephole tile, etc., a so-called plastic firebrick is available. This material ordinarily consists of brick grog or calcined clay of approximately the size of the grog particles in standard fire clay brick, a certain percentage of plastic fire clay, and small percentage of silicate of soda to cause the mass used to air-set. Properly applied as a substitute for small or medium sized special shapes, plastic firebrick has given satisfactory service. It is ordinarily supplied in the proper condition for application, that is, in a condition that is moist enough to "ball in the hands" without being sticky. In applying, for best results, it should be carefully and thoroughly rammed into place with a hammer handle and then pounded with a mallet to assure the exclusion of all air.

A good plastic should be easy of application, possess a minimum shrinkage property, and be as refractory as the other firebrick materials used.

It is advocated by some manufacturers and engineers that the entire furnace lining be constructed of plastic firebrick. It would seem that the arguments advanced previously against the use of a jointing material possessing a high bonding strength, resulting in a monolithic structure of the furnace lining, would apply also to the use of an entirely plastic lining. On the other hand, a plastic is a definitely handy material in emergency cases for minor patching.

WORKMANSHIP—It is probable that a greater number of furnace brickwork and insulating material failures result from poor workmanship in laying up than from the use of inferior materials. It is good insurance of continuity of service, and resulting economy in such service, to employ only the best bricklayers specializing in this particular class of work.

While furnace practice cannot be fully covered here, a few suggestions, applying perhaps particularly to Babcock & Wilcox marine boiler work, may be of assistance.

With the pressure parts and the casing of the boiler installed, the first step in lining the furnace is the placing of the insulating materials in the walls. The uncalcined raw or pressed blocks should be laid against the steel casing, the joints between blocks being filled with the cement furnished with the blocks. Joints between blocks should be made as tight as possible and, if cutting of blocks is required, as little cutting as possible be done. Where it is difficult or impossible to cut blocks to fit small corners, an asbestos cement should be used.

The calcined insulator is then laid against the raw or pressed block, using for joints the cement supplied with this insulator. Care should be taken to break joints between the calcined insulator and the pressed blocks.

The firebrick walls are next laid, leaving the floor construction until last.

Firebrick should be dipped in a thin batter of a properly selected cement and pushed into place hard against the insulation, being tapped finally to a solid seat with a hammer handle. The thinnest possible joints, well filled with a cement having a low bonding property, will add to the life of the furnace brickwork. After each course is laid, a cement, such as recommended above, should be poured over the top surface to fill the vertical joints. Any excess cement should be cleaned off before the laying of the next course.

It is essential that the aligning of the anchor brick with the holes provided in the casing for the reception of the anchor bolts be carefully watched, in order that the bolts do not bind.

Nuts on the anchor bolts should be made only hand tight.

The laying of brick for expansion allowance and as regards headers, stretchers, and floor construction, has already been covered in the discussion of furnace design.

When the furnace firebrick lining is completed, it is necessary to cap the side walls and rear wall with plastic material, as described before. After such lining is completed, it is well to rub the walls with a brick, to remove surplus cement, and then wash the lining with a thin coat of such cement, brushing in all directions in order to properly fill all joints on the face of the walls.

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Steamer "Peter Stuyvesant," 2700 Indicated Horse-power, Babcock & Wilcox Boilers
Owners, Hudson River Day Line, New York, N. Y.

FLOW OF STEAM THROUGH PIPES AND ORIFICES

VARIOUS formulæ for the flow of steam through pipes have been advanced, all having their basis upon Bernoulli's theorem of the flow of water through circular pipes with the proper modifications made for the variation in constants between steam and water. The loss of energy due to friction in a pipe is given by Unwin (based upon Weisbach) as

$$E_f = \frac{f\,2\,v^2\,WL}{gd} \qquad (39)$$

where E is the energy loss in foot-pounds due to the friction of W units of weight of steam passing with a velocity of v feet per second through a pipe d feet in diameter and L feet long; g represents the acceleration due to gravity (32.2) and f the coefficient of friction.

Numerous values have been given for this coefficient of friction, f, which, from experiment, apparently varies with both the diameter of pipe and the velocity of the passing steam. There are no authentic data on the rate of this variation with velocity and, as in all experiments, the effect of change of velocity has seemed less than the unavoidable errors of observation, the coefficient is assumed to vary only with the size of the pipe.

Unwin established a relation for this coefficient for steam at a velocity of 100 feet per second,

$$f = K\left(1 + \frac{3}{10d}\right) \qquad (40)$$

where K is a constant experimentally determined, and d the internal diameter of the pipe in feet.

If h represents the loss of head in feet, then

$$E_f = Wh = \frac{f\,2\,v^2\,WL}{gd} \qquad (41)$$

$$\text{and } h = \frac{f\,2\,v^2\,L}{gd} \qquad (42)$$

If D represents the density of the steam or weight per cubic foot, and p the loss of pressure due to friction in pounds per square inch, then

$$p = \frac{hD}{144} \qquad (43)$$

and from equations (40), (42) and (43),

$$p = \frac{Dv^2L}{72gd} \times K\left(1 + \frac{3}{10d}\right) \qquad (44)$$

To convert the velocity term and to reduce to units ordinarily used, let d_1 = the diameter of pipe in inches = $12d$, and w = the flow in pounds per minute; then

$$w = 60v\frac{\pi}{4}\left(\frac{d_1}{12}\right)^2 D$$

$$\text{and } v = \frac{9.6w}{\pi d_1^2 D}$$

Substituting this value and that of d in formula (44),

$$p = 0.04839\,K\left(1 + \frac{3.6}{d_1}\right)\frac{w^2L}{Dd_1^5} \qquad (45)$$

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Some of the experimental determinations for the value of K are:

$K = 0.005$ for water (Unwin).
$K = 0.005$ for air (Arson).
$K = 0.0028$ for air (St. Gothard tunnel experiments).
$K = 0.0026$ for steam (Carpenter at Oriskany).
$K = 0.0027$ for steam (G. H. Babcock).

The value 0.0027 is apparently the most nearly correct, and substituting in formula *(45)* gives,

$$p = 0.000131 \left(1 + \frac{3.6}{d_1}\right) \frac{w^2 L}{D d_1^5} \qquad (46)$$

$$\text{or } w = 87 \left(\frac{p D d_1^5}{\left(1 + \frac{3.6}{d_1}\right) L} \right)^{1/2} \qquad (47)$$

Where w = the weight of steam passing in pounds per minute,
p = the difference in pressure between the two ends of the pipe in pounds per square inch,
D = density of steam or weight per cubic foot,*
d_1 = internal diameter of pipe in inches,
L = length of pipe in feet.

This formula is the most generally accepted for the flow of steam in pipes. Table 41 is calculated from this formula and gives the amount of steam passing per minute that will flow through straight smooth pipes having a length of 240 diameters from various initial pressures with one pound difference between the initial and final pressures.

To apply this table for other lengths of pipe and pressure losses than those assumed, let L = the length and d the diameter of the pipe, both in inches; l, the loss in pounds; Q, the weight under the conditions assumed in the table, and Q_1, the weight for the changed conditions.

For any length of pipe, if the weight of steam passing is the same as given in the table, the loss will be,

$$l = \frac{L}{240 d} \qquad (48)$$

If the pipe length is the same as assumed in the table but the loss is different, the quantity of steam passing per minute will be,

$$Q_1 = Q l^{1/2} \qquad (49)$$

For any assumed pipe length and loss of pressure, the weight will be,

$$Q_1 = Q \left(\frac{240 d\, l}{L} \right)^{1/2} \qquad (50)$$

Example: Find the weight of steam at 100 pounds initial gauge pressure, which will pass through a 6-inch pipe 720 feet long with a pressure drop of 4 pounds.

*D, the density, is taken as the mean of the density at the initial and final pressures.

Under the conditions assumed in the table, 287 pounds would flow per minute; hence, $Q = 287$, and

$$Q_1 = 287 \left(\frac{240 \times 6 \times 4}{720 \times 12} \right)^{1/2} = 234.3 \text{ pounds.}$$

Elbows, globe valves and a square-ended entrance to pipes all offer resistance to the passage of steam. It is customary to measure the resistance offered by such construction in terms of the diameter of the pipe. Many formulæ have been advanced for computing the length of pipe in diameters equivalent to such fittings or valves which offer resistance. These formulæ, however, vary widely and for ordinary purposes it will be sufficiently accurate to allow for resistance at the entrance of a pipe a length equal to 60 times the diameter; for a right-angle elbow, a length equal to 40 diameters; for a globe valve, a length equal to 60 diameters.

TABLE 41

FLOW OF STEAM THROUGH PIPES

LENGTH OF PIPE 240 DIAMETERS

Initial Gauge Pressure, Pounds per Square Inch	DIAMETER OF PIPE IN INCHES													
	¾	1.0	1½	2.0	2½	3.0	4.0	5.0	6.0	8.0	10.0	12.0	15.0	18.0	Density*
	WEIGHT OF STEAM PER MINUTE IN POUNDS WITH ONE POUND LOSS IN PRESSURE														
1	1.13	2.04	5.49	9.85	14.8	24.4	45.0	74.2	105	202	329	482	771	1127	0.0385
10	1.39	2.54	6.82	12.3	18.5	30.3	55.9	92.3	138	248	409	599	958	1404	0.0595
20	1.63	2.98	8.02	14.4	21.9	35.6	65.7	108	163	295	481	705	1128	1645	0.0823
30	1.84	3.37	9.07	16.2	24.5	40.2	74.1	122	183	333	542	795	1272	1858	0.1047
40	2.03	3.71	9.95	17.9	26.9	44.2	81.6	135	202	366	596	874	1388	2089	0.1267
50	2.19	4.02	10.8	19.4	29.2	47.9	88.3	146	219	397	646	948	1515	2214	0.1486
60	2.35	4.30	11.5	20.7	31.2	51.3	93.7	156	234	424	691	1014	1623	2372	0.1703
70	2.50	4.56	12.3	22.0	33.1	54.4	100	166	248	451	734	1075	1722	2517	0.1919
80	2.63	4.81	12.9	23.2	34.9	57.4	106	175	262	475	774	1135	1814	2653	0.2134
90	2.76	5.05	13.6	24.3	36.6	60.2	111	183	275	499	812	1192	1893	2782	0.2347
100	2.88	5.27	14.2	25.4	38.3	62.8	116	191	287	520	848	1243	1990	2908	0.2560
120	3.10	5.69	15.3	27.4	41.3	67.9	125	207	310	562	915	1342	2148	3138	0.2985
150	3.38	6.26	16.8	30.2	45.5	74.7	138	228	341	619	1007	1477	2361	3450	0.3616
200	3.88	6.96	19.1	34.3	51.5	84.8	156	258	387	703	1144	1677	2685	3919	0.4664
250	4.33	7.86	21.1	37.9	57.1	93.7	173	285	428	776	1264	1855	2962	4337	0.5695
300	4.68	8.56	23.0	41.2	62.1	102	188	310	466	845	1376	2019	3226	4717	0.6748
350	5.03	9.22	24.7	44.5	66.8	110	202	334	501	908	1478	2168	3467	5060	0.7788

*From Marks and Davis Tables.

The flow of steam of a higher toward a lower pressure increases as the difference in pressure increases to a point where the external pressure becomes 58 per cent of the absolute initial pressure. Below this point the flow is neither increased nor decreased by a reduction of the external pressure, even to the extent of a perfect vacuum. The lowest pressure for which this statement holds when steam is discharged into the atmosphere is 25.37 pounds. For any pressure below this figure, the atmospheric pressure, 14.7 pounds, is greater than 58 per cent of the initial pressure. Table 42, by D. K. Clark, gives the velocity of outflow at constant density, the actual velocity

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of outflow expanded (the atmospheric pressure being taken as 14.7 pounds absolute. and the ratio of expansion in the nozzle being 1.624), and the corresponding discharge per square inch of orifice per minute.

Napier deduced an approximate formula for the outflow of steam into the atmosphere which checks closely with the figures just given. This formula is:

$$W = \frac{pa}{70} \qquad (51)$$

Where W = the pounds of steam flowing per second,
p = the absolute pressure in pounds per square inch,
and a = the area of the orifice in square inches.

TABLE 42

FLOW OF STEAM INTO THE ATMOSPHERE

Absolute Initial Pressure per Square Inch Pounds	Velocity of Outflow at Constant Density Feet per Second	Actual Velocity of Outflow Expanded Feet per Second	Discharge per Square Inch of Orifice per Minute Pounds	Horse-Power per Square Inch of Orifice if Horse-Power =30 Pounds per Hour
25.37	863	1401	22.81	45.6
30.	867	1408	26.84	53.7
40.	874	1419	35.18	70.4
50.	880	1429	44.06	88.1
60.	885	1437	52.59	105.2
70.	889	1444	61.07	122.1
75.	891	1447	65.30	130.6
90.	895	1454	77.94	155.9
100.	898	1459	86.34	172.7
115.	902	1466	98.76	197.5
135.	906	1472	115.61	231.2
155.	910	1478	132.21	264.4
165.	912	1481	140.46	280.9
215.	919	1493	181.58	363.2

In some experiments made by Professor C. H. Peabody, on the flow of steam through pipes from ¼ inch to 1½ inches long and ¼ inch in diameter, with rounded entrances, the greatest difference from Napier's formula was 3.2 per cent excess of the experimental over the calculated results.

For steam flowing through an orifice from a higher to a lower pressure where the lower pressure is greater than 58 per cent of the higher, the flow per minute may be calculated from the formula:

$$W = 1.9AK\sqrt{(P-d)d} \qquad (52)$$

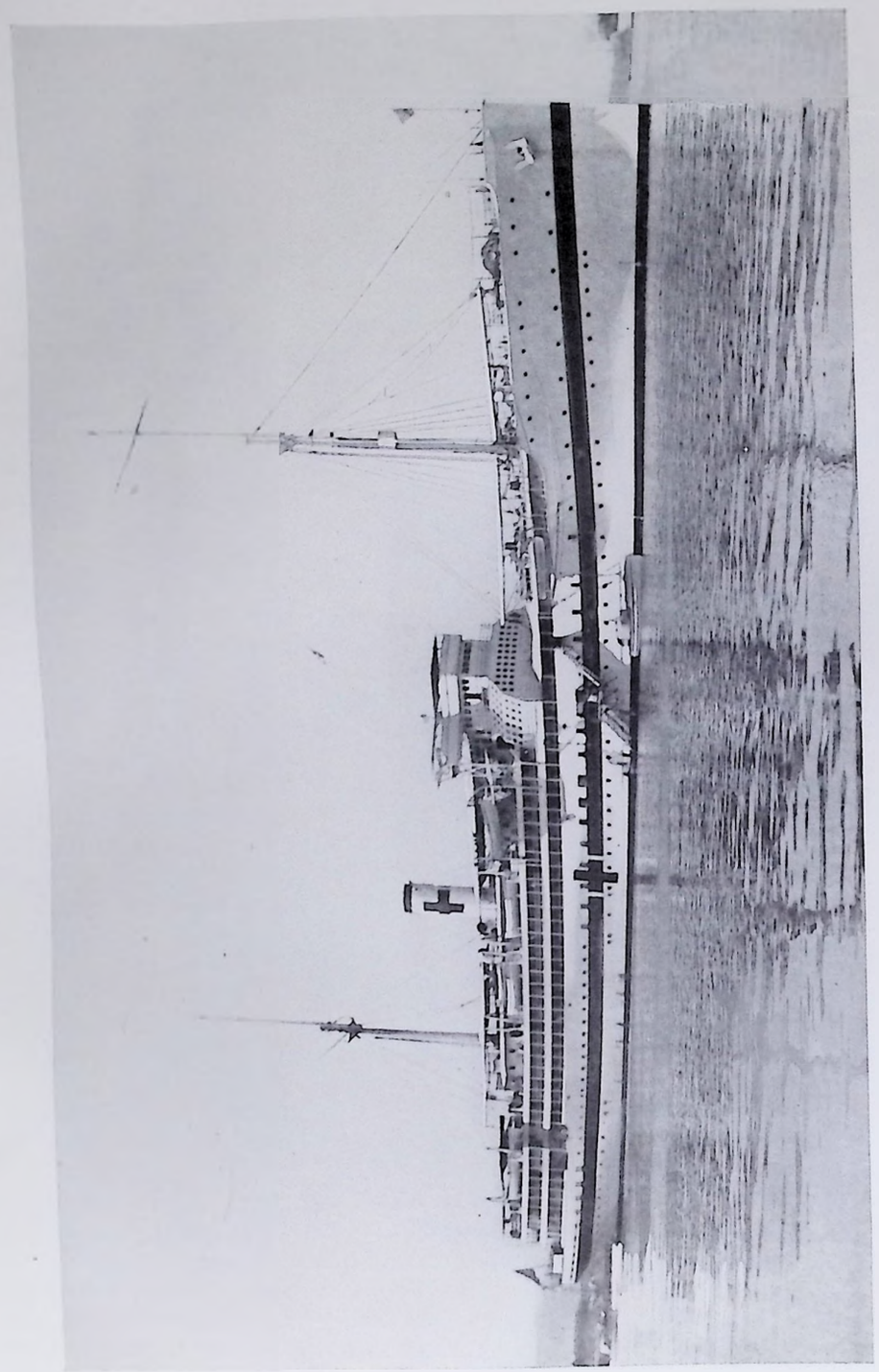
Where W = the weight of steam discharged in pounds per minute,
A = area of orifice in square inches,
P = the absolute initial pressure in pounds per square inch,
d = the difference in pressure between the two sides in pounds per square inch,
K = a constant = 0.93 for a short pipe, and 0.63 for a hole in a thin plate or a safety valve.

HEAT TRANSFER

THE rate at which heat is transmitted from a hot gas to a cooler metal surface over which the gas is flowing has been the subject of a great deal of investigation, from both the experimental and theoretical sides. A more or less complete explanation of this process is necessary for a detailed analysis of the performance of steam boilers. Such information at the present is almost entirely lacking and for this reason a boiler, as a physical piece of apparatus, is not as well understood as it might be. This, however, has had little effect in its practical development and it is hardly possible that a more complete understanding of the phenomena discussed will have any radical effect on the present design.

The amount of heat that is transferred across any surface is usually expressed as a product, of which one factor is the slope or linear rate of change in temperature and the other is the amount of heat transferred per unit difference in temperature in unit length. In Fournier's analytical theory of the conduction of heat, this second factor is taken as a constant and is called the "conductivity" of the substance. Following this practice, the amount of heat absorbed by any surface from a hot gas is usually expressed as a product of the difference in temperature between the gas and the absorbing surface into a factor which is commonly designated the "transfer rate." There has been considerable looseness in the writings of even the best authors as to the way in which the gas temperature difference is to be measured. If the gas varies in temperature across the section of the channel through which it is assumed to flow, and most of them seem to consider that this would be the case, there are two mean gas temperatures, one the mean of the actual temperatures at any time across the section, and the other the mean temperature of the entire volume of the gas passing such a section in any given time. Since the velocity of flow will of a certainty vary across the section, this second mean temperature, which is one tacitly assumed in most instances, may vary materially from the first. The two mean temperatures are only approximately equal when the actual temperature measured across the section is very nearly a constant. In what follows it will be assumed that the mean temperature measured in the second way is referred to. In English units the temperature difference is expressed in Fahrenheit degrees and the transfer rate in B.t.u. per hour per square foot of surface. Peclet, who seems to have been one of the first to consider this subject analytically, assumed that the transfer rate was constant and independent both of the temperature differences and the velocity of the gas over the surface. Rankine, on the other hand, assumed that the transfer rate, while independent of the velocity of the gas, was proportional to the temperature difference, and expressed the total amount of heat absorbed as proportional to the square of the difference in temperature. Neither of these assumptions has any warrant in either theory or experiment and they are only valuable in so far as their use determines formulæ that fit experimental results. Of the two, Rankine's assumption seems to lead to formulæ that more nearly represent actual conditions. It has been quite fully developed by William Kent in his "Steam Boiler Economy." Professor Osborne Reynolds in a short paper reprinted in Volume I of his "Scientific Papers" suggests that the transfer rate is proportional to the product of the density and velocity of the gas, and it is to be assumed that he had in mind the mean velocity, density and temperature over the section of the channel through which the gas was assumed to flow. Contrary

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U. S. Naval Hospital Ship "Relief," 5250 Indicated Horse-power, Babcock & Wilcox Boilers, Superheaters and Oil Burners

to prevalent opinion, Professor Reynolds gave neither a valid experimental nor a theoretical explanation of his formula and the attempts that have been made since its first publication to establish it on any theoretical basis can hardly be considered of scientific value. Nevertheless, Reynolds' suggestion was really the starting point of the scientific investigation of this subject and while his formula cannot in any sense be held as completely expressing the facts, it is undoubtedly correct to a first approximation for small temperature differences if the additive constant, which in his paper he assumed as negligible, is given a value.*

Experimental determinations have been made during the last few years of the heat transfer rate in cylindrical tubes at comparatively low temperatures and small temperature differences. The results at different velocities have been plotted and an empirical formula determined expressing the transfer rate with the velocity as a factor. The exponent of the power of the velocity appearing in the formula, according to Reynolds, would be unity. The most probable value, however, deduced from most of the experiments makes it less than unity. After considering experiments of his own, as well as experiments of others, Dr. Wilhelm Nusselt† concludes that the evidence supports the following formula:

$$a = b\frac{\lambda_w}{d^{1-u}}\left(\frac{wc_p\delta}{\lambda}\right)^u$$

Where a is the transfer rate in calories per hour per square meter of surface per degree Centigrade difference in temperature,

u is a physical constant, equal to 0.786 from Dr. Nusselt's experiments,

b is a constant which, for the units given below, is 15.90,

w is the mean velocity of the gas in meters per second,

c_p is the specific heat of the gas at its mean temperature and pressure in calories per kilogram,

δ is the density in kilograms per cubic meter,

λ is the conductivity at the mean temperature and pressure in calories per hour per square meter per degree Centigrade temperature drop per meter,

λ_w is the conductivity of the steam at the temperature of the tube wall,

d is the diameter of the tube in meters.

If the unit of time for the velocity is made the hour, and in the place of the product of the velocity and density is written its equivalent, the weight of gas flowing per hour divided by the area of the tube, this equation becomes:

$$a = .0255\ \frac{\lambda_w}{d^{.214}}\left(\frac{Wc_p}{A\lambda}\right)^{.786}$$

where the quantities are in the units mentioned, or, since the constants are absolute constants, in English units,

a is the transfer rate in B.t.u. per hour per square foot of surface per degree difference in temperature,

W is the weight in pounds of the gas flowing through the tube per hour,

A is the area of the tube in square feet,

d is the diameter of the tube in feet,

*H. P. Jordan, "Proceedings of The Institute of Mechanical Engineers," 1909.
†"Zeitschrift des Vereines Deutscher Ingenieure," 1909, page 1750.

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c_p is the specific heat of the gas at constant pressure,

λ is the conductivity of the gas at the mean temperature and pressure in B.t.u. per hour per square foot of surface per degree Fahrenheit drop in temperature per foot,

λ_w is the conductivity of the steam at the temperature of the wall of the tube.

The conductivities of air, carbonic acid gas and superheated steam, as affected by the temperature, in English units, are:

Conductivity of air . . . . . 0.0122 $(1 + 0.00132\,T)$
Conductivity of carbonic acid gas 0.0076 $(1 + 0.00228\,T)$
Conductivity of superheated steam 0.0119 $(1 + 0.00261\,T)$

where T is the temperature in degrees Fahrenheit.

Nusselt's formulæ can be taken as typical of a number of other formulæ proposed by German, French and English writers.* Physical properties, in addition to the density, are introduced in the form of coefficients from a consideration of the physical dimensions of the various units and of the theoretical formulæ that are supposed to govern the flow of the gas and the transfer of heat. All assume that the correct method of representing the heat transfer rate is by the use of one term, which seems to be unwarranted and probably has been adopted on account of the convenience in working up the results by plotting them logarithmically. This was the method Professor Reynolds used in determining his equation for the loss in head in fluids flowing through cylindrical pipes, and it is now known that the derived equation cannot be considered as anything more than an empirical formula. It, therefore, is well for anyone considering this subject to understand at the outset that the formulæ discussed are only of an empirical nature and applicable to limited ranges of temperature under conditions approximately the same as those surrounding the experiments from which the constants of the formula were determined.

It is not probable that the subject of heat transfer in boilers will ever be on any other than an experimental basis until the mathematical expression connecting the quantity of fluid which will flow through a channel of any section under a given head has been found and some explanation of its derivation obtained. Taking the simplest possible section, namely, a circle, it is found that at low velocities the loss of head is directly proportional to the velocity and the fluid flows in straight stream lines or the motion is direct. This motion is in exact accordance with the theoretical equations of the motion of a viscous fluid and constitutes almost a direct proof that the fundamental assumptions on which these equations are based are correct. When, however, the velocity exceeds a value which is determinable for any size of tube, the direct or stream-line motion breaks down and is replaced by an eddy or mixing flow. In this flow the head loss by friction is approximately, although not exactly, proportional to the square of the velocity. No explanation of this has ever been found in spite of the fact that the subject has been treated by the best mathematicians and physicists for years back. It is to be assumed that the heat transferred during the mixing flow would be at a much higher rate than with the direct or stream-line flow, and Professors Croker and Clement† have demonstrated that this is true, the increase in the transfer

* Heinrich Gröber—Zeit. d. Ver. Ing., March, 1912, December, 1912. Leprince-Ringuet—Revue de Mécanique, July, 1911. John Perry—"The Steam Engine." T. E. Stanton—Philosophical Transactions, 1897. Dr. J. T. Nicholson—Proceedings Institute of Engineers & Shipbuilders in Scotland, 1910. W. E. Dally—Proceedings Institute of Mechanical Engineers, 1909.

† "Proceedings Royal Society," Vol. LXXI.

being so marked as to enable them to determine the point of critical velocity from observing the rise in temperature of water flowing through a tube surrounded by a steam jacket.

The formulæ given apply only to a mixing flow and inasmuch as, from what has just been stated, this form of motion does not exist from zero velocity upward, it follows that any expression for the heat transfer rate that would make its value zero when the velocity is zero, can hardly be correct. Below the critical velocity, the transfer rate seems to be little affected by change in velocity and Nusselt,* in another paper which mathematically treats the direct or stream-line flow, concludes that, while it is approximately constant as far as the velocity is concerned in a straight cylindrical tube, it would vary from point to point of the tube, growing less as the surface passed over increased.

It should further be noted that no account in any of this experimental work has been taken of radiation of heat from the gas. Since the common gases absorb very little radiant heat at ordinary temperatures, it has been assumed that they radiate very little at any temperature. This may or may not be true, but certainly a visible flame must radiate as well as absorb heat. However this radiation may occur, since it would be a volume phenomenon rather than a surface phenomenon it would be considered somewhat differently from ordinary radiation. It might apply as increasing the conductivity of the gas which, however independent of radiation, is known to increase with the temperature. It is, therefore, to be expected that at high temperatures the rate of transfer will be greater than at low temperatures. The experimental determinations of transfer rates at high temperatures are lacking.

Although comparatively nothing is known concerning the heat radiation from gases at high temperatures, there is no question but what a large proportion of the heat absorbed by a boiler is received directly as radiation from the furnace. Experiments show that the lower row of tubes of a Babcock & Wilcox boiler absorb heat at an average rate per square foot of surface between the first baffle and the front headers equivalent to the evaporation of from 50 to 75 pounds of water from and at 212 degrees Fahrenheit per hour. Inasmuch as in these experiments no separation could be made between the heat absorbed by the bottom of the tube and that absorbed by the top, the average includes both maximum and minimum rates for those particular tubes and it is fair to assume that the portion of the tubes actually exposed to the furnace radiation absorb heat at a higher rate. Part of this heat was, of course, absorbed by actual contact between the hot gases and the boiler heating surface. A large portion of it, however, must have been due to radiation. Whether this radiant heat came from the fire surface and the brickwork and passed through the gases in the furnace with little or no absorption, or whether, on the other hand, the radiation were absorbed by the furnace gases and the heat received by the boiler was a secondary radiation from the gases themselves and at a rate corresponding to the actual gas temperature, is a question. If the radiation is direct, then the term "furnace temperature," as usually used, has no scientific meaning, for obviously the temperature of the gas in the furnace would be entirely different from the radiation temperature, even were it possible to attach any significance to the term "radiation temperature," and it is not possible to do this unless the radiation is what is known as "full radiation" from a so-called "black body." If furnace radiation takes place in this

* "Zeitschrift des Vereines Deutscher Ingenieure," 1910, page 1154.

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STEAMER "ITORORO," 850 INDICATED HORSE-POWER, BABCOCK & WILCOX BOILERS AND OIL BURNERS

manner, the indications of a pyrometer placed in a furnace are hard to interpret and such temperature measurements can be of little value. If the furnace gases absorb the radiation from the fire and from the brickwork of the side walls and in their turn radiate heat to the boiler surface, it is scientifically correct to assume that the actual or sensible temperature of the gas would be measured by a pyrometer and the amount of radiation could be calculated from this temperature by Stefan's law, which is to the effect that the rate of radiation is proportional to the fourth power of the absolute temperature, using the constant with the resulting formula that has been determined from direct experiment and other phenomena. With this understanding of the matter, the radiation absorbed by a boiler can be taken as equal to that absorbed by a flat surface covering the portion of the boiler tubes exposed to the furnace and at the temperature of the tube surface, when completely exposed on one side to the radiation from an atmosphere at the temperature in the furnace. With this assumption, if S' is the area of the surface, T the absolute temperature of the furnace gases, t the absolute temperature of the tube surface of the boiler, the heat absorbed per hour measured in B.t.u. is equal to

$$1600\left(\left(\frac{T}{1000}\right)^4-\left(\frac{t}{1000}\right)^4\right)S'$$

In using this formula, or in any work connected with heat transfer, the external temperature of the boiler heating surface can be taken as that of saturated steam at the pressure under which the boiler is working, with an almost negligible error, since experiments have shown that with a surface clean internally, the external surface is only a few degrees hotter than the water in contact with the inner surface, even at the highest rates of evaporation. Further than this, it is not conceivable that in a modern boiler there can be much difference in the temperature of the boiler in the different parts, or much difference between the temperature of the water and the temperature of the steam in the drums which is in contact with it.

If the total evaporation of a boiler measured in B.t.u. per hour is represented by E, the furnace temperature by T_1, the temperature of the gas leaving the boiler by T_2, the weight of gas leaving the furnace and passing through the setting per hour by W, the specific heat of the gas by C, it follows from the fact that the total amount of heat absorbed is equal to the heat received from radiation plus the heat removed from the gases by cooling from the temperature T_1 to the temperature T_2, that

$$E=1600\left(\left(\frac{T}{1000}\right)^4-\left(\frac{t}{1000}\right)^4\right)S'+WC(T_1-T_2)$$

This formula can be used for calculating the furnace temperature when E, t and T_2 are known, but it must be remembered that an assumption which is probably, in part at least, incorrect is implied in using it or in using any similar formula. Expressed in this way, however, it seems more rational than one proposed a few years ago by Dr. Nicholson,* where, in place of the surface exposed to radiation, he uses the grate surface and assumes the furnace gas temperature as equal to the fire temperature.

If the heat transfer rate is taken as independent of the gas temperature and the heat absorbed by an element of the surface in a given time is equated to the heat given out from the gas passing over this surface in the same time, a single integration gives

$$(T-t)=(T_1-t)\,e^{-\frac{Rs}{WC}}$$

*"Proceedings Institute of Engineers and Shipbuilders," 1910.

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where s is the area of surface passed over by the gases from the furnace to any point where the gas temperature T is measured, and the rate of heat transfer is R. As written, this formula could be used for calculating the temperature of the gas at any point in the boiler setting. Gas temperatures, however, calculated in this way are not to be depended upon, as it is known that the transfer rate is not independent of the temperature. Again, if the transfer rate is assumed as varying directly with the weight of the gases passing, which is Reynolds' suggestion, it is seen that the weight of the gases entirely disappears from the formula and as a consequence if the formula was correct, as long as the temperature of the gas entering the surface from the furnace was the same, the temperatures throughout the setting would be the same. This is known also to be incorrect. If, however, in place of T is written T_2 and in place of s is written S, the entire surface of the boiler, and the formula is re-arranged, it becomes:

$$R = \frac{WC}{S} \log_e \left(\frac{T_1 - t}{T_2 - t} \right)$$

This formula can be considered as giving a way of calculating an average transfer rate. It has been used in this way for calculating the average transfer rate from boiler tests in which the capacity has varied from an evaporation of a little over 3 pounds per square foot of surface up to 15 pounds. When plotted against the gas weights, it was found that the points were almost exactly on a line. This line, however, did not pass through the zero point but started at a point corresponding approximately to a transfer rate of 2. Checked out against many other tests, the straight line law seems to hold generally, and this is true even though material changes are made in the method of calculating the furnace temperature. The inclination of the line, however, varied inversely as the average area for the passage of the gas through the boiler. If A is the average area between all the passes of the boiler, the heat transfer rate in Babcock & Wilcox type boilers with ordinary clean surfaces can be determined to a rather close approximation from the formula:

$$R = 2.00 + 0.0014 \frac{W}{A}$$

The manner in which A appears in this formula is the same as it would appear in any formula in which the heat transfer rate was taken as depending upon the product of the velocity and the density of the gas jointly, since this product, as pointed out above, is equivalent to $W \div A$. Nusselt's experiments, as well as those of others, indicate that the ratio appears in the proper way.

While the underlying principles from which the formula for this average transfer rate was determined are questionable and at best only approximately correct, it nevertheless follows that, assuming the transfer rate as determined experimentally, the formula can be used in an inverse way for calculating the amount of surface required in a boiler for cooling the gases through the range of temperature covered by the experiments and it has been found that the results bear out this assumption. The practical application of the theory of heat transfer, as developed at present, seems consequently to rest on these last two formulæ, which from their nature are more or less empirical.

Through the range in the production of steam met with in boilers now in service, which in the marine type extends to as high as 12 to 15 pounds evaporation of water from and at 212 degrees Fahrenheit per square foot of surface, the constant 2 in the approximate formula for the average heat transfer rate constitutes quite a

large proportion of the total. The comparative increase in the transfer rate due to a change in weight of the gases is not as great consequently as it would be if this constant were zero. For this reason, with the same temperature of the gases entering the boiler surface, there will be a gradual increase in the temperature of the gases leaving the surface as the velocity or weight of flow increases and the proportion of the heat contained in the gases entering the boiler which is absorbed by it is gradually reduced. It is, of course, possible that the weight of the gases could be increased to such an amount, or the area for their passage through the boiler could be so reduced by additional baffles, that the constant term in the heat transfer formula would be relatively unimportant. Under such conditions, as pointed out previously, the final gas temperature would be unaffected by a further increase in the velocity of the flow and the fraction of the heat carried by the gases removed by the boiler would be constant. Actual tests of waste heat boilers, in which the weight of gas per square foot of sectional area for its passage is many times more than in ordinary installations, show, however, that this condition has not been attained and it will probably never be attained in any practical installation. It is for this reason that the conclusions of Dr. Nicholson in the paper referred to and of Messrs. Kreisinger and Ray in the pamphlet, "The Transmission of Heat into Steam Boilers," published by the Department of the Interior in 1912, are not applicable without modification to boiler design.

In superheaters the heat transfer is effected in two different stages; the first transfer is from the hot gas to the metal of the superheater tube and the second transfer is from the metal of the tube to the steam on the inside. There is, theoretically, an intermediate stage in the transfer of the heat from the outside to the inside surface of the tube. The conductivity of steel is sufficient, however, to keep the temperatures of the two sides of the tube very nearly equal to each other so that the effect of the transfer in the tube itself can be neglected. The transfer from the hot gas to the metal of the tube takes place in the same way as with the boiler tubes proper, regard being paid to the temperature of the tube which increases as the steam is heated. The transfer from the inside surface of the tube to the steam is the inverse of the process of the transfer of the heat on the outside and seems to follow the same laws. The transfer rate, therefore, will increase with the velocity of the steam through the tube. For this reason, internal cores are quite often used in superheaters and actually result in an increase in the amount of superheat obtained from a given surface. The average transfer rate in superheaters based on a difference in mean temperature between the gas on the outside of the tubes and the steam on the inside of the tubes is, if R is the transfer rate from the gas to the tube and r the rate from the tube to the steam:

$$\frac{R\,r}{R + r}$$

and is always less than either R or r. This rate is usually greater than the average transfer rate for the boiler as computed in the way outlined in the preceding paragraphs. Since, however, steam cannot, under any imagined set of conditions, take up more heat from a tube than would water at the same average temperature, this fact supports the contention made that the actual transfer rate in a boiler must increase quite rapidly with the temperature. The actual transfer rate in superheaters is affected by so many conditions that it has not been possible so far to evolve any formula of practical value.

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Steamer "Montgomeryshire," 3200 Shaft Horse-power, Babcock & Wilcox Boilers and Oil Burners
Owners, Royal Mail Steam Packet Company

MISCELLANEOUS TUBE AND PIPE DATA

TABLE 43

TUBE, WIRE AND SHEET-METAL GAUGES

(DIAMETERS OR THICKNESSES IN DECIMAL PARTS OF AN INCH)

Gauge No.	Birmingham Wire Gauge (B. W. G.): Stubs Iron Wire Gauge. *For steel tubes, wire and sheets*	U. S. Standard Gauge for Sheet Metal. *For iron and steel sheets.*	American Wire Gauge; Brown & Sharpe Gauge. *For copper wire.*	Steel Wire Gauge; Washburn & Moen and Roebling Gauge. *For steel wire.*	Stubs Steel Wire Gauge	British Imperial Standard Wire Gauge (S. W. G.).	Trenton Iron Company Gauge	Standard Birmingham Sheet and Hoop Gauge (B. G.)
0	0.340	0.312	0.325	0.3065	. . .	0.324	0.305	0.3964
1	0.300	0.281	0.289	0.2830	0.227	0.300	0.285	0.3532
2	0.284	0.266	0.258	0.2625	0.219	0.276	0.265	0.3147
3	0.259	0.250	0.229	0.2437	0.212	0.252	0.245	0 2804
4	0.238	0.234	0.204	0.2253	0.207	0.232	0.225	0.2500
5	0.220	0.219	0.182	0.2070	0.204	0.212	0.205	0.2225
6	0.203	0.203	0.162	0.1920	0.201	0.192	0.190	0.1981
7	0.180	0.188	0.144	0.1770	0.199	0.176	0.175	0.1764
8	0.165	0.172	0.128	0.1620	0.197	0.160	0.160	0.1570
9	0.148	0.156	0.114	0.1483	0.194	0.144	0.145	0.1398
10	0.134	0.141	0.102	0.1350	0.191	0.128	0.130	0.1250
11	0.120	0.125	0.091	0.1205	0.188	0.116	0.1175	0.1113
12	0.109	0.109	0.081	0.1055	0.185	0.104	0.105	0.0991
13	0.095	0.094	0.072	0.0915	0.182	0.092	0.0925	0.0882
14	0.083	0.078	0.064	0.0800	0.180	0.080	0.080	0.0785
15	0.072	0.070	0.057	0.0720	0.178	0.072	0.070	0.0699
16	0.065	0.062	0.051	0.0625	0.175	0.064	0.061	0.0625
17	0.058	0.056	0.045	0.0540	0.172	0.056	0.0525	0.0556
18	0.049	0.050	0.040	0.0475	0.168	0.048	0.045	0.0495
19	0.042	0.0438	0.036	0.0411	0.164	0.040	0.040	0.0440
20	0.035	0.0375	0.032	0.0348	0.161	0.036	0.035	0.0392

TABLE 44

STANDARD OPEN-HEARTH OR LAP-WELDED STEEL TUBES

External Diameter Inches	Birmingham Wire Gauge Number	Thickness Inch	Internal Diameter Inches	Circumference Inches		Transverse Area Square Inches		Square Feet of External Surface per Foot of Length	Length in Feet per Square Foot of External Surface	Nominal Weight Pounds per Foot
				External	Internal	External	Internal			
1	13	0.095	0.810	3.141	2.544	0.7854	0.5153	0.261	3.819	0.917
1¼	12	0.109	1.032	3.927	3.242	1.2271	0.8365	0.3272	3.055	1.327
1½	10	0.134	1.232	4.712	3.870	1.7671	1.1921	0.392	2.546	1.955
1½	9	0.148	1.204	4.712	3.782	1.7671	1.1385	0.392	2.546	2.137
1½	8	0.165	1.170	4.712	3.676	1.7671	1.0751	0.392	2.546	2.353
2	10	0.134	1.732	6.283	5.441	3.1416	2.3560	0.523	1.909	2.670
2	9	0.148	1.704	6.283	5.353	3.1416	2.2778	0.523	1.909	2.927
2	8	0.165	1.670	6.283	5.246	3.1416	2.1904	0.523	1.909	3.234
3¼	11	0.120	3.010	10.210	9.456	8.2958	7.1157	0.850	1.175	4.011
3¼	10	0.134	2.982	10.210	9.368	8.2958	6.9840	0.850	1.175	4.459
3¼	9	0.148	2.954	10.210	9.280	8.2958	6.8535	0.850	1.175	4.903
4	10	0.134	3.732	12.566	11.724	12.566	10.939	1.047	0.954	5.532
4	9	0.148	3.704	12.566	11.636	12.566	10.775	1.047	0.954	6.000
4	8	0.165	3.670	12.566	11.530	12.566	10.578	1.047	0.954	6.758
4	6	0.203	3.594	12.566	11.291	12.566	10.144	1.047	0.954	8.232

Dimensions are nominal and except where noted are in inches.

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TABLE 45

DIMENSIONS OF STANDARD WROUGHT-IRON AND STEEL PIPE

Size, Inches	Diameters Inches		Nominal Thickness Inch	Circumference Inches		Transverse Areas Square Inches			Linear Feet of Pipe per Square Foot of		Length of Pipe Containing One Cubic Foot, Feet	Nominal Weight per Foot Pounds		Number of Threads per Inch of Screw
	External	Approximate Internal		External	Internal	External	Internal	Metal	External Surface	Internal Surface		Plain Ends	Threaded and Coupled	
⅛	0.405	0.269	0.068	1.272	0.845	0.129	0.057	0.072	9.431	14.199	2,533.775	0.244	0.245	27
¼	0.540	0.364	0.088	1.696	1.144	0.229	0.104	0.125	7.073	10.493	1,383.789	0.424	0.425	18
⅜	0.675	0.493	0.091	2.121	1.549	0.358	0.191	0.167	5.658	7.747	754.360	0.567	0.568	18
½	0.840	0.622	0.109	2.639	1.954	0.554	0.304	0.250	4.547	6.141	473.906	0.850	0.852	14
¾	1.050	0.824	0.113	3.299	2.589	0.866	0.533	0.333	3.637	4.635	270.034	1.130	1.134	14
1	1.315	1.049	0.133	4.131	3.296	1.358	0.864	0.494	2.904	3.641	166.618	1.678	1.684	11½
1¼	1.660	1.380	0.140	5.215	4.335	2.164	1.495	0.669	2.301	2.767	96.275	2.272	2.281	11½
1½	1.900	1.610	0.145	5.969	5.058	2.835	2.036	0.799	2.010	2.372	70.733	2.717	2.731	11½
2	2.375	2.067	0.154	7.461	6.494	4.430	3.355	1.075	1.608	1.847	42.913	3.652	3.678	11½
2½	2.875	2.469	0.203	9.032	7.757	6.492	4.788	1.704	1.328	1.547	30.077	5.793	5.819	8
3	3.500	3.068	0.216	10.996	9.638	9.621	7.393	2.228	1.091	1.245	19.479	7.575	7.616	8
3½	4.000	3.548	0.226	12.566	11.146	12.566	9.886	2.680	0.954	1.076	14.565	9.109	9.202	8
4	4.500	4.026	0.237	14.137	12.648	15.904	12.730	3.174	0.848	0.948	11.312	10.790	10.889	8
4½	5.000	4.506	0.247	15.708	14.156	19.635	15.947	3.688	0.763	0.847	9.030	12.538	12.642	8
5	5.563	5.047	0.258	17.477	15.856	24.306	20.006	4.300	0.686	0.756	7.198	14.617	14.810	8
6	6.625	6.065	0.280	20.813	19.054	34.472	28.891	5.581	0.576	0.629	4.984	18.974	19.185	8
7	7.625	7.023	0.301	23.955	22.063	45.664	38.738	6.926	0.500	0.543	3.717	23.544	23.769	8
8	8.625	8.071	0.277	27.096	25.356	58.426	51.161	7.265	0.442	0.473	2.815	24.696	25.000	8
8	8.625	7.981	0.322	27.096	25.073	58.426	50.027	8.399	0.442	0.478	2.878	28.554	28.809	8
9	9.625	8.941	0.342	30.238	28.089	72.760	62.786	9.974	0.396	0.427	2.294	33.907	34.188	8
10	10.750	10.192	0.279	33.772	32.019	90.763	81.585	9.178	0.355	0.374	1.765	31.201	32.000	8
10	10.750	10.136	0.307	33.772	31.843	90.763	80.691	10.072	0.355	0.376	1.785	34.240	35.000	8
10	10.750	10.020	0.365	33.772	31.479	90.763	78.855	11.908	0.355	0.381	1.826	40.483	41.132	8
11	11.750	11.000	0.375	36.914	34.558	108.434	95.033	13.401	0.325	0.347	1.515	45.557	46.247	8
12	12.750	12.090	0.330	40.055	37.982	127.676	114.800	12.876	0.299	0.315	1.254	43.773	45.000	8
12	12.750	12.000	0.375	40.055	37.699	127.676	113.097	14.579	0.299	0.318	1.273	49.562	50.706	8

TABLE 46

DIMENSIONS OF EXTRA STRONG WROUGHT-IRON AND STEEL PIPE

Nominal Internal Diameter, Inches	Diameters Inches		Nominal Thickness Inches	Circumference Inches		Transverse Areas Square Inches			Linear Feet of Pipe per Square Foot of		Length of Pipe Containing One Cubic Foot, Feet	Nominal Weight per Foot, Plain Ends, Pounds
	External	Approximate Internal		External	Internal	External	Internal	Metal	External Surface	Internal Surface		
⅛	0.405	0.215	0.095	1.272	0.675	0.129	0.036	0.093	9.431	17.766	3,966.392	0.314
¼	0.540	0.302	0.119	1.696	0.949	0.229	0.072	0.157	7.073	12.648	2,010.290	0.535
⅜	0.675	0.423	0.126	2.121	1.329	0.358	0.141	0.217	5.658	9.030	1,024.689	0.738
½	0.840	0.546	0.147	2.639	1.715	0.554	0.234	0.320	4.547	6.995	615.017	1.087
¾	1.050	0.742	0.154	3.299	2.331	0.866	0.433	0.433	3.637	5.147	333.016	1.473
1	1.315	0.957	0.179	4.131	3.007	1.358	0.719	0.639	2.904	3.991	200.193	2.171
1¼	1.660	1.278	0.191	5.215	4.015	2.164	1.283	0.881	2.301	2.988	112.256	2.996
1½	1.900	1.500	0.200	5.969	4.712	2.835	1.767	1.068	2.010	2.546	81.487	3.631
2	2.375	1.939	0.218	7.461	6.092	4.430	2.953	1.477	1.608	1.969	48.766	5.022
2½	2.875	2.323	0.276	9.032	7.298	6.492	4.238	2.254	1.328	1.644	33.976	7.661
3	3.500	2.900	0.300	10.996	9.111	9.621	6.605	3.016	1.091	1.317	21.801	10.252
3½	4.000	3.364	0.318	12.566	10.568	12.566	8.888	3.678	0.954	1.135	16.202	12.505
4	4.500	3.826	0.337	14.137	12.020	15.904	11.497	4.407	0.848	0.998	12.525	14.983
4½	5.000	4.290	0.355	15.708	13.477	19.635	14.455	5.180	0.763	0.890	9.962	17.611
5	5.563	4.813	0.375	17.477	15.120	24.306	18.194	6.112	0.686	0.793	7.915	20.778
6	6.625	5.761	0.432	20.813	18.099	34.472	26.067	8.405	0.576	0.663	5.524	28.573
7	7.625	6.625	0.500	23.955	20.813	45.664	34.472	11.192	0.500	0.576	4.177	38.048
8	8.625	7.625	0.500	27.096	23.955	58.426	45.663	12.763	0.442	0.500	3.154	43.388
9	9.625	8.625	0.500	30.238	27.096	72.760	58.426	14.334	0.396	0.442	2.464	48.728
10	10.750	9.750	0.500	33.772	30.631	90.763	74.662	16.101	0.355	0.391	1.929	54.735
11	11.750	10.750	0.500	36.914	33.772	108.434	90.763	17.671	0.325	0.355	1.587	60.075
12	12.750	11.750	0.500	40.055	36.914	127.676	108.434	19.242	0.299	0.325	1.328	65.415

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TABLE 47

MAXIMUM ALLOWABLE WORKING PRESSURES FOR TUBES FOR WATER-TUBE BOILERS FOR DIFFERENT DIAMETERS AND GAUGES OF TUBES

(FROM THE BOILER CODE OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS)

Outside Diameter of Tube Inches	Birmingham Wire Gauge Number, and Thickness of Tube Wall												
	17	16	15	14	13	12	11	10	9	8	7	6	5
D	t= 0.058	t= 0.065	t= 0.072	t= 0.083	t= 0.095	t= 0.109	t= 0.120	t= 0.134	t= 0.148	t= 0.165	t= 0.180	t= 0.203	t= 0.220
½	434	686	938	1334	. .	. .	. .	. .	. .	. .	. .	. .	. .
¾	206	374	542	806	1094	. .	. .	. .	. .	. .	. .	. .	. .
1	. .	218	344	542	758	1010	. .	. .	. .	. .	. .	. .	. .
1⅛	. .	166	278	454	646	870	1046	. .	. .	. .	. .	. .	. .
1¼	. .	124	225	383	557	758	916	1118	. .	. .	. .	. .	. .
1½	. .	. .	146	278	422	590	722	890	1058	. .	. .	. .	. .
1¾	. .	. .	. .	203	326	470	583	727	871	1046	. .	. .	. .
2	. .	. .	. .	146	254	380	479	605	731	884	1019	. .	. .
2¼	. .	. .	. .	. .	198	310	398	510	622	758	878	1062	. .
2½	. .	. .	. .	. .	153	254	333	434	535	657	765	931	1053
2¾	. .	. .	. .	. .	117	208	280	372	464	575	673	824	935
3	. .	. .	. .	. .	. .	170	236	320	404	506	596	734	836
3¼	. .	. .	. .	. .	. .	. .	199	276	354	448	531	658	752
3½	. .	. .	. .	. .	. .	. .	167	238	310	398	475	594	681
3¾	. .	. .	. .	. .	. .	. .	139	206	273	355	427	537	619
4	. .	. .	. .	. .	. .	. .	. .	178	240	317	385	488	565
4½	. .	. .	. .	. .	. .	. .	. .	. .	186	254	314	406	474
5	. .	. .	. .	. .	. .	. .	. .	. .	142	204	258	340	402

$$P = \left(\frac{t - 0.030}{D}\right) 18000 - 250$$

Where P = Maximum allowable working pressure, pounds per square inch
t = Thickness of tube wall, inches
D = Outside diameter of tube, inches

NOTE—Maximum allowable working pressures for superheater tubes shall be the same as for boiler tubes.

TABLE 48

RADIATION FROM COVERED AND UNCOVERED STEAM PIPES

CALCULATED FOR 160 POUNDS PRESSURE AND 60 DEGREES TEMPERATURE

Pipe Inches	Thickness of Covering	½ inch	¾ inch	1 inch	1¼ inch	1½ inch	Bare
2	B.t.u. per lineal foot per hour	149	118	99	86	79	597
	B.t.u. per square foot per hour	240	190	161	138	127	959
	B.t.u. per square foot per hour per one degree difference in temperature	0.770	0.613	0.519	0.445	0.410	3.198
4	B.t.u. per lineal foot per hour	247	193	160	139	123	1085
	B.t.u. per square foot per hour	210	164	136	118	104	921
	B.t.u. per square foot per hour per one degree difference in temperature	0.677	0.592	0.439	0.381	0.335	2.970
6	B.t.u. per lineal foot per hour	352	269	221	190	167	1555
	B.t.u. per square foot per hour	203	155	127	110	96	897
	B.t.u. per square foot per hour per one degree difference in temperature	0.655	0.500	0.410	0.355	0.310	2.89
8	B.t.u. per lineal foot per hour	443	337	276	235	207	1994
	B.t.u. per square foot per hour	196	149	122	104	92	883
	B.t.u. per square foot per hour per one degree difference in temperature	0.632	0.481	0.394	0.335	0.297	2.85
10	B.t.u. per lineal foot per hour	549	416	337	287	250	2468
	B.t.u. per square foot per hour	195	148	120	102	89	877
	B.t.u. per square foot per hour per one degree difference in temperature	0.629	0.477	0.387	0.329	0.287	2.83

Covering—Magnesia, canvas covered.

For calculating radiation for pressure and temperature other than 160 pounds and 60 degrees, use B.t.u. figures for one degree difference.

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