THE ENGINEERING RECORD SERIES

STEAM POWER PLANTS

THEIR

DESIGN AND CONSTRUCTION

BY

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INTRODUCTORY NOTE.

Frequently engineers and others in charge of a manufacturing business, be it a mill, factory or electric generating station, are called upon to design and purchase a steam power plant or parts of it when their knowledge of the machinery that goes into such a plant is more or less limited, and without being able to obtain the benefit of the advice of a competent consulting engineer. It is hoped that this book will be of special value to this class and of some value to all interested in steam power plant construction. Part of the text appeared in a series of articles in The Engineering Record and when the demand for them seemed to warrant their being published in book form they were thoroughly revised and considerable new matter added. A number of the illustrations have been selected from articles printed in The Engineering Record during the last two or three years descriptive of steam power plant construction. They are reprinted without the text that accompanied them, thinking they would be suggestive.

CONTENTS.

Chapter I.—The Design of Steam Power Plants.

No better service can be done the non-expert about to construct a steam plant than to advise him to engage at the outset of the project some capable engineer to design the plant and superintend its installation. In spite of the advantages of having work planned and carried out by such men, it will always be, probably, that a very considerable proportion of the work done will be constructed for one reason or another, by persons with a semi-technical training without the aid of the expert engineer. It is, therefore, proposed to give data and information which, it is hoped, will aid the engineer who cannot be called an expert in selecting the various kinds of machinery and apparatus that make up a steam power plant, to discuss specifications for these in a general way, and, in some instances, to outline the manner in which they should be purchased.

It is the practice of many engineers in steam-plant construction to invite bids on apparatus described very generally in a specification and intended to perform a service under conditions that are named, the idea of the engineer being to allow each bidder to proportion the parts of the apparatus he is to furnish and to quote a price on it. When bids are received under these conditions, it generally follows that there is a variation in the size of the machinery offered by different makers to do the same work, and the lowest in price may not be best adapted for the conditions. An engine, boiler or pump, in fact almost everything about a steam plant, may do the work required of it, but it may be so proportioned as to do it in a manner that is not best for the owner. For instance, while an engine may give the required power, its cylinders may be so small that it requires an excessively large amount of steam to run it, or a boiler may be so small that an abnormal amount of coal must be burned in order to generate the steam required. The expert engineer is, of course, able to detect and reject bids on deficient apparatus, yet when the size of the apparatus is fixed by the contractor it may happen that the purchaser, who is not an expert, will accept machinery which is not best adapted for the service that it is to perform, particularly if the too frequent custom of purchasing the apparatus lowest in price is followed.

There are other methods of buying apparatus for a steam plant. One is to go to a reputable manufacturer or contracting engineer and engage him to build the machinery wanted and pay what is asked for it. Such a contractor should not, however, be placed in competition with others if he is to design a plant to fill the requirements of the owner, for if this is done the contractor's interest is to design a cheaper plant than will be proposed by his competitor, whose bid is based on the plant designed by him, and this kind of competition sometimes results in inferior apparatus being supplied.

Another method of purchasing is to have the engineer state in his specification the dimensions of the apparatus wanted, permitting, however, a departure from these specified dimensions in order that a manufacturer can make use of his standard patterns, provided such a course is not detrimental to the purchaser's interest. When all manufacturers bid on the same basis their bids are lower and the purchaser is sure of getting a properly-proportioned machine, provided the engineer who prepares the plans and specifications is capable. Many believe the last two methods of procedure much preferable to the first. Latitude given an imprudent contractor, in the way of fixing the dimensions of the apparatus he is to supply, may, in a measure, be covered by demanding a guarantee as to efficiency, yet tests necessary to determine if the guarantees are fulfilled are expensive and are therefore generally omitted.

Location of Plant.—One of the first questions to be decided in the construction of a power plant is its location. This depends on several factors, the most important of which are, the ease with which the power may be transmitted from the generating source to the locations of the demand, the cost of delivering coal and removing ashes from the power house, and the availability of a supply of water for condensing purposes.

The first factor, the ease with which power may be transmitted from the generating source to the machines utilizing it, is the one that frequently decides whether an electric or belt and shafting system of transmission is to be used. It is, of course, impossible to give any general rule applicable to all cases, as each situation demands a thorough investigation of the cost of installing and operating both plants. In arriving at their relative costs of operation, the interest on the investments, the repairs, the fuel cost, the cost of attendance, supplies, etc., have to be reckoned. Generally speaking, it may be said that in any situation where the load on the power plant is practically constant throughout the working day, as in a textile mill, and where the power house may be located close to the lines of shafting so that belts may be carried from the engine to the shafting without the use of gearing, quarter-turn belts, etc., the belt and shafting system of transmission seems the most favored. If, however, the manufacturing establishment consists of a number of separate buildings in which lines of shafting are at different angles, so as to require several separate power plants or else a complicated system of belts or gears, the electric system possesses advantages as regards cost of operation that makes its adoption advisable. Again, the tendency in establishments where the work is similar to that of a large machine or bridge shop, where the tools are used intermittently, where one or two departments may run overtime, the electric system is rapidly gaining friends. The reason for this is not the difference in the operating expenses, which are slight, inasmuch as a machine shop requires a very small amount of power for its operation, but in the greater convenience and cleanliness of the electric system. One solution of the problem of transmitting power when shafts lie in directions that are not parallel is the rope drive. This has been used with the greatest success in many plants. Its low first cost and cost of repairs and the flexibility of the system make it well worth considering in many installations.

The cost of handling coal and removing ashes in a power house may be a considerable item in the operating expenses, and large plants are therefore usually equipped with coal-handling machinery. Some steam plants are so arranged that railway cars or coal cars may be run over a receiving hopper from which the coal is conveyed mechanically to storage bunkers over the boilers and chuted from the bunkers to the furnaces by gravity. If a plant is not to be provided with coal-handling machinery, it is well, if convenient, to provide a trestle so that cars may be run over and dump into a bunker opposite the furnace doors in order that the coal may fall by gravity through holes in the wall separating the bunker from the boiler room, on to the floor of the latter, in front of the furnace doors.

The value of water in sufficient quantity to condense the steam exhausted by the engines often determines the location of the power house. From 15 to 20 per cent. of the fuel used by a non-condensing engine will be saved if it is operated with a condenser. Each pound of steam exhausted by an engine requires a supply of 30 to 35 pounds of water for the condenser, so it will be seen that the water needed for condensing purposes is often a considerable quantity. Sometimes when a power house cannot be located on the bank of a stream from which a supply for this purpose is available, a pipe or conduit can be laid from a river to a well near the power house, the grade of the conduit being below that of minimum low water. The injection pipe, as the pipe that conveys the water to the condenser is called, can then be run from this well to the condenser, which is usually located close to the engine. Because of the vacuum in the condenser, the water will rise in the injection pipe from a lower level to the condenser. It is not advisable, however, to attempt to lift the injection water over 20 feet. Where the lift would be slightly greater than this the condenser can be placed in a pit. Since the development of the cooling tower, an apparatus for cooling condensing water so that it can be used over and over again, condensing plants are not so dependent on an abundant water supply, as they were before this apparatus was perfected, hence the importance of locating a plant by a stream for condensing purposes is not as great as it was once.

Drawings.—Complete and accurate drawings of all of the details of a steam plant are a necessity. It is well to make assembled drawings showing the plant in plan and as many elevations as may be necessary to make the arrangement perfectly clear. Assembled drawings insure all parts of the plant fitting together properly, and prevent mistakes such as attempting to run steam pipes where a building column ought to be, and so on. If assembled drawings are to be made, the scale assumed should be sufficiently large to show the steam-pipe system. Three-eighths of an inch to the foot is about as small as can well be used. As soon as contracts are made for engine, boilers, feed-water heaters, pumps, etc., accurately dimensioned blue-prints showing the machinery in plan and elevation should be obtained from the contractors. Experience

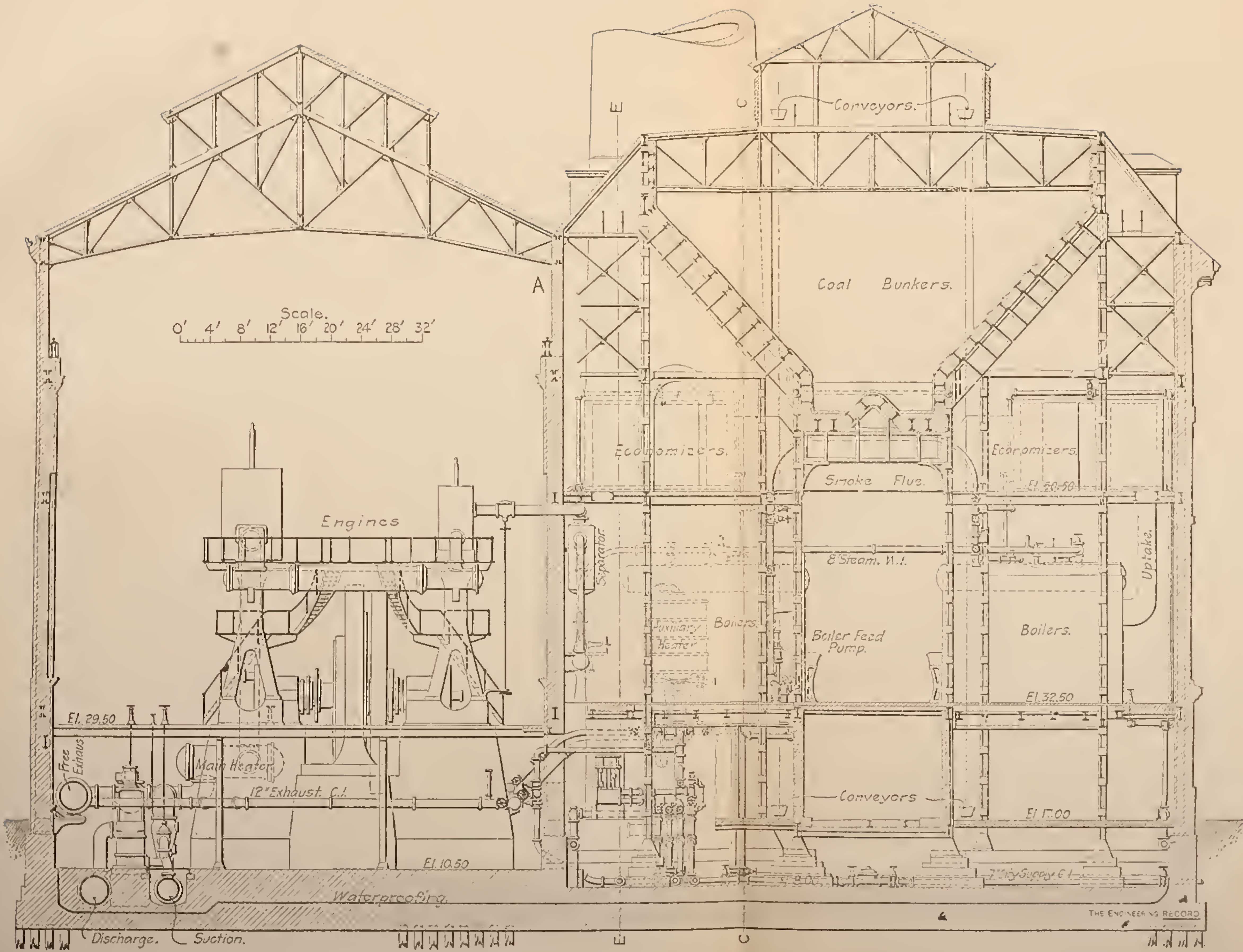


PLATE 1.—LINCOLN WHARF POWER STATION, BOSTON ELEVATED RAILWAY CO.

GEORGE A. KIMBALL, CHIEF ENGINEER; SHEAFF & JAASTED, CONSULTING ENGINEERS.

has shown that it is well, with some firms at least, in order to avoid delay, to get these blue-prints before the contracts are let. Contracts for the building, steel work, etc., should not be made until the machinery is contracted for, since it may not be possible to obtain the machinery which was contemplated at the time the building plans were made, and machinery then available may not fit. It is, of course, essential that the building be fitted to the machinery, not the machinery fitted to the building.

Type of Power House.—Generally the relative location of the engine and boiler rooms is determined by some local condition. Where it can be done, however, it is better to locate the engine and boilers in one building, with a wall between them, and to place them in parallel rows with the cylinders of the engine adjacent to the rear of the boilers. This is particularly the best arrangement if the plant is likely to be enlarged in the future. The steam pipe connecting the boiler and engine or engines is the most direct in this arrangement, and it can be most readily enlarged. If the engine and boiler houses are placed end to end and contain two or more boilers supplying one engine, the proper size of piping for such a plant will be inadequate if another engine is added at one end of the plant and more boilers installed at the other.

Where land is very expensive, boilers are sometimes placed in buildings on two or more floors. Ordinarily, however, the boiler-room floor is usually on the level with the outside ground, while the engine-room floor with large engines is invariably higher, usually from 6 to 12 feet or more, depending on the height of the engine foundations. Sometimes an engine is installed where the engine-room floor is on the ground level. When this is done, it is necessary to construct a pit for the condenser, if such an auxiliary is used, also pipe trenches and, if the engine is a large one, a pit for the fly-wheel. It perhaps ought to be stated at this time that a condenser ought always to be below the engine, as the pipe leading to the condenser cannot rise at any point without introducing a dangerous element into the power plant. The arrangement that necessitates the construction of pits ought to be avoided, if it is possible, as it is difficult to construct a water-tight pit for the fly-wheel and condenser, because the vibration of the engine is apt to crack the lining of the pit and allow water to enter, if there is any in the soil. The trenches for the exhaust piping have to be covered and the piping is not nearly so accessible when in a covered trench as it is when in the basement usually provided under engine rooms.

Building.—A building for a power house should, if possible, be constructed of fireproof materials. Brick walls with steel trusses supporting a wooden roof covered with tar and gravel, or with some form of fireproof construction, is the usual construction. The brick walls sometimes carry the roof trusses and tracks for traveling cranes, and again these are supported by steel columns resting on the foundations and imbedded in the brick walls, the latter, however, carrying only their own weight. The building should be designed by an architect or structural engineer. The construction of the building can be done by bridge shops making a specialty of constructing buildings of this character and supplying the steelwork for them.

The buildings should be of such a size that the machinery in them is not cramped. When there are several machines in an engine room it should be remembered when locating them that it is sometimes necessary to stop a machine very quickly. It is well, therefore, to place all machinery in such a position that it is readily accessible to the man or men in charge of it. Provision should be made in planning and constructing a power house for bringing the machinery into the building after it is erected, and a door of sufficient size to admit the largest part of a machine must be provided. Ample room must be left around the horizontal steam and water cylinders to be able to remove piston rods if it should be necessary, without removing the cylinder from its foundations. Enough space must be left between the foundations of adjacent engines and between foundations and engine-room walls to allow a man to get between them to reach the foundation bolts. In a boiler room, there must be a clear space in front of the boilers at least as wide as the boiler tubes are long. The distance between the rear of the boilers and the wall need not be greater than five or six feet, or enough to allow a wheelbarrow to be placed opposite the soot door in the rear of the boiler setting, if such a door is provided.

Foundations.—The character of the soil underlying the site for a power house should be carefully examined in order that the foundations can be so planned as to keep the load imposed by the buildings and machinery within the safe limit. For ordinary one-story buildings where the loads are not excessive, holes should be

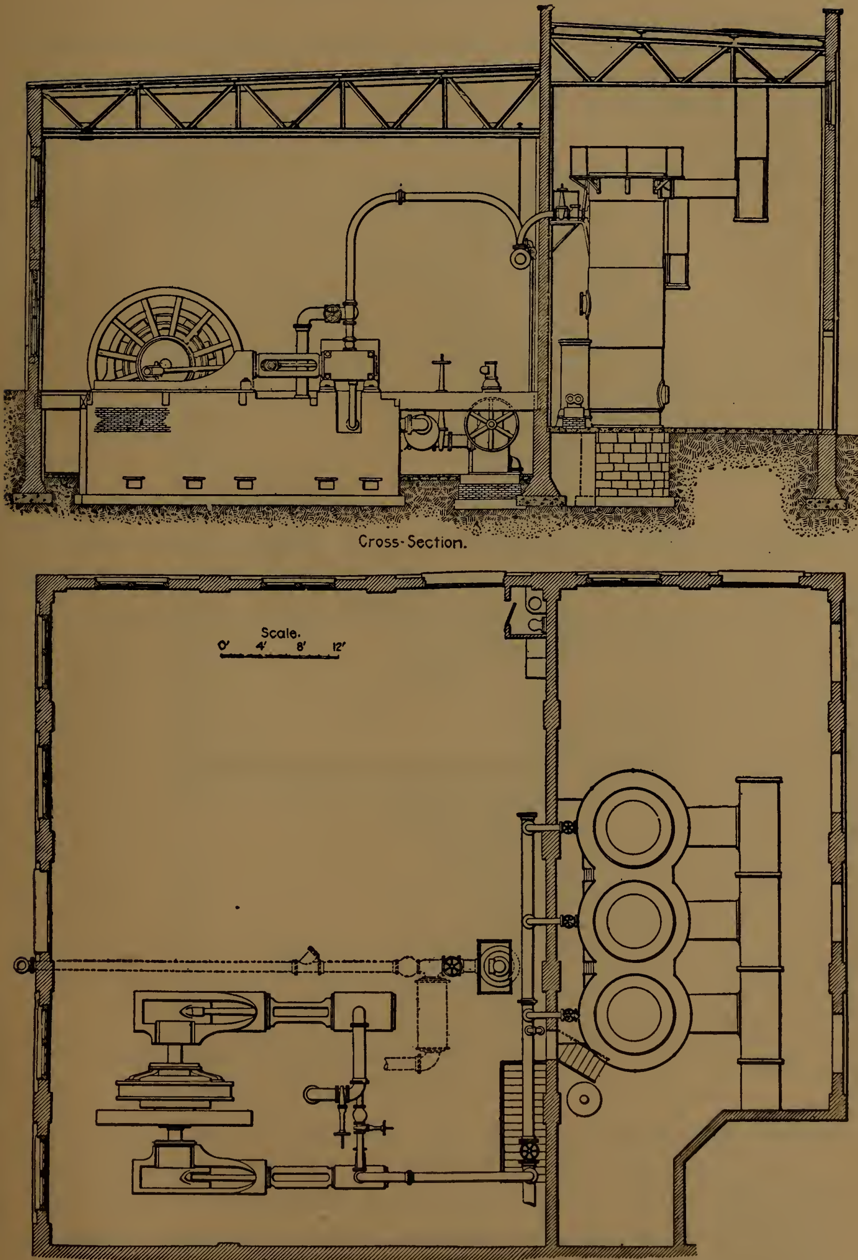


FIGURE 1.—EI-CTRIC POWER STATION, DESIGNED BY DEAN & MAIN.

dug at numerous points over the site and the character of the soil determined. As the magnitude of the work increases more care should be taken. In important work a competent specialist in

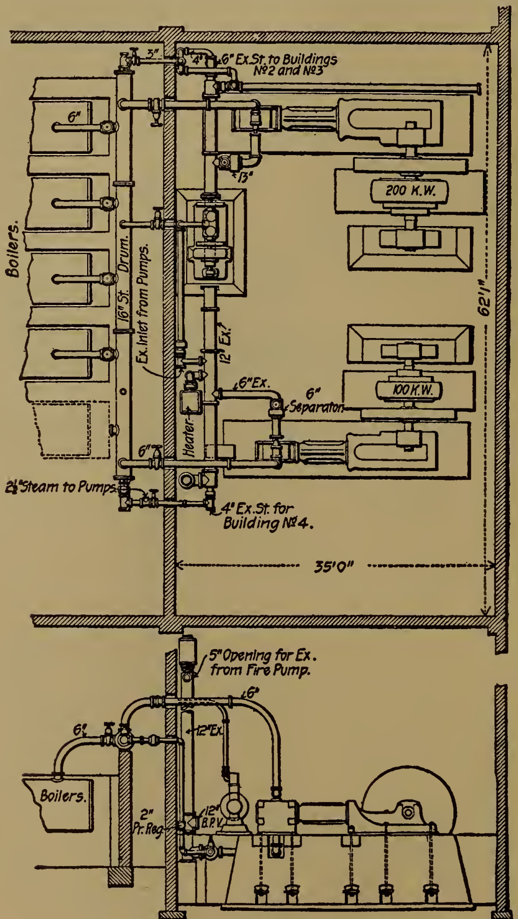


FIGURE 2.—PIPING IN POWER HOUSE; LOCKWOOD, GREENE & CO., ENGINEERS.

foundation work should be consulted. As to the bearing capacity of different soils, the New York Building Code states: "Different soils, excluding mud, at the bottom of the footings shall be deemed to safely sustain the following loads to the superficial foot, namely: Soft clay, one ton per square foot; ordinary clay and sand together, in layers, wet and springy, two tons per square foot; loam, clay or fine sand, firm and dry, three tons per square foot; very firm, coarse sand, stiff gravel or hard clay, four tons per square foot."

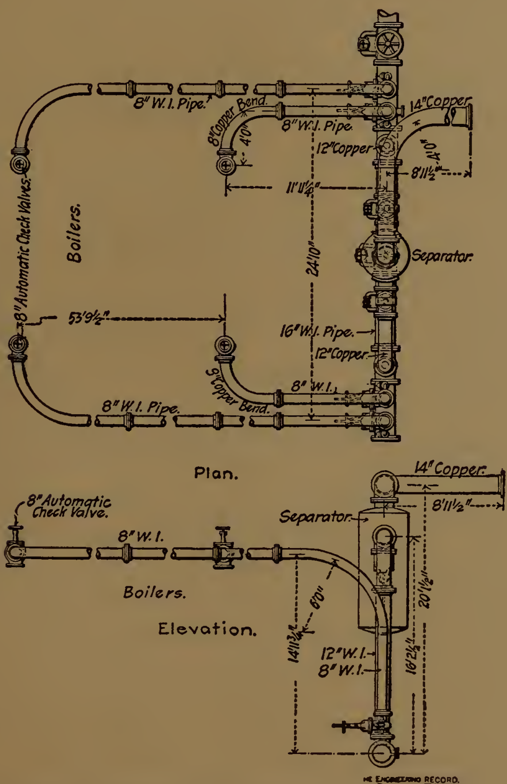


FIGURE 3.—SECTION MAIN STEAM PIPING, LINCOLN WHARF STATION.

If loose rock is found it should be removed, solid rock should be dressed off in steps with vertical risers and horizontal treads so that the pressure will be exerted everywhere in a vertical direction. Solid rock will stand almost any load that can be imposed upon it. If soil of low bearing power is found, piling is usually resorted to. Piles may be of spruce or hemlock at least 5 inches in diameter at the point and 10 inches in diameter at the butt for piles 20 feet or less in length, and 6 inches at the point and 12 inches at the butt for piles over 20 feet in length. The bearing power of piles not driven to rock or hardpan or similar firm material may be calculated by the Wellington formula, in which the safe bearing power in tons is equal to twice the weight of the hammer in tons multiplied by the height of the fall in feet divided by one plus the penetration of pile under the last blow in inches. From a knowledge of the total load to be carried and the load that each pile will support, the number of piles necessary under a weight to be supported can be calculated. If the soil is not firm, the bearing power of a pile should be taken much less than that given by the formula, in order to allow for decrease in strength due to vibrations of the machinery. To avoid decay the piles should be cut off at the level of the ground water or not more than one foot above it.

It is getting to be the practice in power-house construction where piles are used, to saw off the heads of the piles to a uniform grade, then excavate the materials between the heads of the piles for a depth of a foot or so and lay a bed of concrete sometimes several feet in thickness between the heads of the piles and over their tops. An 8-foot bed of concrete over piles on 30-inch centers was used as a foundation for the 96th Street power house of the Metropolitan Street Railway Company in New York City. If the soil underlying a power house site is found to possess too low a bearing power for the foundations of the engines and boilers to be constructed directly on it, concrete beds may be laid so as to distribute the load. Sometimes a concrete bed is laid under the engine foundations and another under the boilers. Again the entire site is covered with a bed of concrete and the wall footings and machinery foundations built directly upon it.

Engine Foundations.—These are almost invariably constructed by the owner, the engine builder furnishing the drawings. The latter generally consist of accurately dimensioned drawings showing the foundations in plan and one or two elevations and also a drawing of a board template which has to be made for locating the foundation bolts. A hole is bored in the template where each bolt is located, and the template is supported over the place where the foundation is to be constructed, at such a height that the foundation bolts may be suspended in the position that they will finally occupy, by passing them through the holes in the template, the nut on the upper end holding them in position. The foundation is then built around the bolts, leaving holes about them an inch or two greater in diameter than the bolts themselves, so that the latter may be moved slightly, to pass through the holes in the engine bed plate, should an error occur in locating the position of the bolts or the holes in the engine bed.

Foundations for engines are sometimes constructed on a very thick bed of concrete, as has been explained. This is not always necessary, and where good loam and clay are known to exist for some distance below the surface of the ground, the earth may be leveled off and the foundation commenced on a layer of concrete just thick enough to give a good bearing. If loose rock or poor earth is found, this should be removed and the excavation filled with concrete, in which loose stones, old bricks, etc., may be imbedded, care being taken to put them in layers alternating with the concrete, which should be so placed and rammed as to make sure that there are no voids. This should be continued up to the level at which the regular brick or concrete foundations are to commence. There seems to be no good reason for using a pure concrete bed if the earth is of good quality and the load imposed upon it not excessive. Engine foundations should be of brick laid in cement mortar made of one part Portland cement to two parts clean sharp sand or of concrete. Good concrete is obtained by mixing one part good Portland cement, two parts sand and four parts broken stone, the latter small enough to pass through a 2-inch ring. Concrete should be laid in layers not over 6 inches thick, each layer being thoroughly rammed before the one above is put down. Such a foundation has to be surrounded with board walls or forms to suitably hold the concrete while it is being laid. A Portland cement concrete foundation ought to stand at least two months, and one of brick in Portland cement one month before it is loaded, whenever possible to spare the time, but if suitably proportioned and carefully watched it can be safely loaded much sooner, especially if the concrete is made as dry as possible.

Chapter II.—Proportioning Steam Boilers.

Heating Surface Necessary.—The function of a steam boiler is to transmit to the water it contains as much of the heat generated by the combustion of fuel as possible. Each square foot of heating surface in the boiler can transmit only a certain amount of heat when the highest economy is being realized. By increasing the supply of heat a greater amount is transmitted and consequently a greater amount of water is evaporated by each square foot of heating surface, but with the increase, the same percentage of the heat generated is not utilized. The reason is that after a certain rate of evaporation is reached, the maximum capacity of the metal in the boiler plates to transmit the heat is more nearly reached, and, hence, a larger percentage of the heat of the fuel passes up the chimney. In other words, it is possible to evaporate a certain amount, say three pounds, of water per square foot of heating surface in a given time and to utilize a certain percentage, say 80 per cent., of the heat in the fuel, 20 per cent. going to waste. It is also possible to burn more coal and evaporate say five pounds of water per square foot of heating surface in the same time, but when doing so a greater percentage, perhaps 30 per cent., of the heat of the fuel is lost. The selection of the proper amount of heating surface for a steam boiler is, therefore, a very important matter.

The best efficiency, under ordinary working conditions, with most boilers, is obtained when evaporating about 3 pounds of water per square foot of heating surface per hour, from a feed temperature of 212 degrees Fahrenheit into steam at atmospheric pressure. This is equivalent to allowing nearly 12 square feet of heating surface per boiler horse-power. Some water-tube boilers with certain coals seem to give a good result when the heating surface per horse-power is a little less than that stated. Most boilers, sometimes, and quite frequently in fact, attain a high efficiency when the rate of evaporation is considerably higher than that given. Such high results with most boilers are usually attained, however, when all of the many conditions affecting the efficiency are such as to produce a good result. Many of these conditions are not so favorable when a boiler is operated in ordinary service. For instance, if the boiler surfaces are not clean, owing perhaps to the accumulation of scale or soot, the efficiency of the heating surface will be more or less impaired. For this and other reasons, the writer believes it is well to provide ample heating surface for the work to be done, for not only will such a course result in a saving of fuel at ordinary rates of evaporation, but it will make it possible to run a boiler considerably above its rating and still maintain a fair efficiency.

Value of a Boiler Horse-Power.—According to the American Society of Mechanical Engineers' standard, a boiler to develop one horse-power must raise 30 pounds of water from a temperature of 100 degrees Fahrenheit to the temperature of steam at 70 pounds pressure, and evaporate it into steam at the pressure. This is equivalent to evaporating 34½ pounds of water at a temperature of 212 degrees into steam at atmospheric pressure or "from and at 212 degrees" as it is sometimes called. The term horse-power is frequently used when estimating the capacity of steam boilers, and the custom of buyers was formerly to ask for boilers of a certain horse-power. This is an improper way to buy boilers unless the amount of heating surface per horse-power is closely examined. If bids are called for boilers of a given horse-power one bidder might offer a boiler with ample heating surface, while another might offer a boiler with much less. Both might develop the required horse-power, but the one with deficient heating surface might do it only at an increased cost for fuel, as explained in a previous paragraph. In saying this it should be borne in mind that the efficiency of various classes of surfaces varies according to their location and arrangement, but this variation in first-class boilers is confined to narrow limits which can only be detected by the expert.

Advantages of Types of Boilers.—Barrus, in his excellent work on "Boiler Tests," states "that the economy with which different types of boilers operate depends more upon their proportions and the conditions under which they work, than upon their types; and, moreover, that when these proportions are suitably carried out and when the conditions are favorable, the different types of boilers give substantially the same result." So much for the side of efficiency. As to safety, the water-tube boiler is, of course, far superior to boilers of the fire-tube type. In fact, water-tube boilers seldom explode, although the tubes sometimes burst and injure those firing them. Water-tube boilers also possess an advantage in that they contain less water than the shell type, as usually designed, and consequently steam can be raised in them more quickly. More water can probably be evaporated per square foot of heating surface in a water-tube boiler than in one having fire tubes, on account of a larger passage for the gases, and pos-

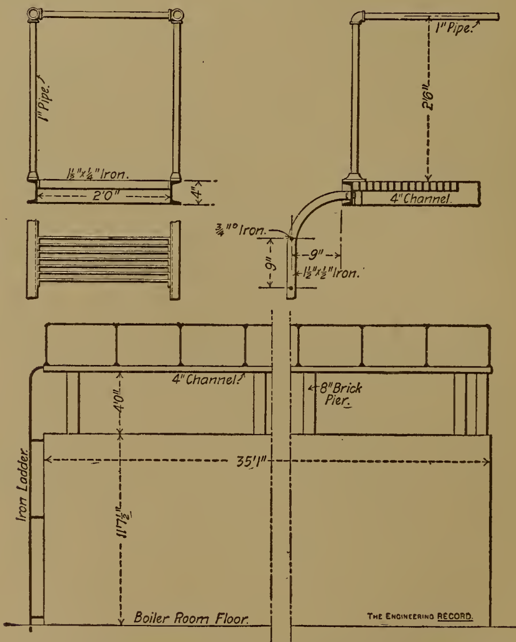


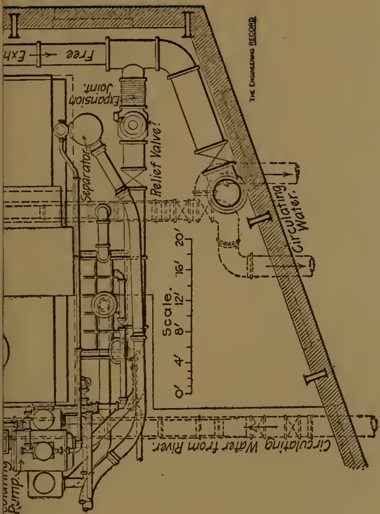
Figure 4.—Iron Walk Over Boilers, Designed by Sheaff & Jaasted.

sibly on account of a better circulation of water in contact with the heating surfaces. The efficiency of the two types is about the same, the difference depending more upon the proportions and management, as Mr. Barrus states, than upon the types. Water-tube boilers have increased in favor very rapidly in the past few years, particularly for electric work, where large amounts of steam are wanted suddenly, and sometimes without previous warning. In spite of the fact that the water-tube boiler possesses advantages over it, the fire-tube boiler, particularly the horizontal return tubular boiler, is still used to a very considerable extent on account of its lower cost. An internally-fired boiler, such as the Manning, locomotive, Galloway or Lancashire boiler, all of which are of the fire-tube type possesses advantages over the brick-set boilers in that there is not the air leakage into the furnaces, reducing the efficiency, that is often found in the boilers which require a brick setting. Furthermore, they do not need the costly brickwork. Internally-fired boilers are, on the other hand, unless provided with large furnaces, very poorly adapted for burning bituminous coals, which require large combustion chambers, in order that the flame from the burning coal cannot come in contact with the relatively cooler surfaces of the boiler and thus become cooled to such an extent that combustion ceases and the gases pass to the chimney unconsumed and consequently wasted. For this same reason, horizontal tubular boilers must be raised a good deal higher above the grate if bituminous coal is to be burned than if anthracite is used. It is held by some engineers that bituminous coal gives the best results if it is burned in a furnace entirely separate from the boilers, so that the radiant heat from the fires will heat the fire brick lining, or checkerwork, which is sometimes introduced in the combustion chamber, to such an extent that the gases coming in contact with this highly heated brickwork are consumed before reaching the boiler. Horizontal return tubular boilers are seldom used for pressures as high as 150 pounds.

Division of Heating Surface in Units.—The first problem connected with the design of a boiler plant is to determine accurately the maximum number of pounds of steam that will be used by the various engines, pumps, and other parts of the plant which will have to be supplied. As has been said, one square foot of heating surface should be allowed in most boilers for every 3 pounds of water to be evaporated into steam from and at 212 degrees in an hour's time. With this proportion and sufficient draft and grate surface to burn the necessary amount of fuel, a boiler can easily be forced 33 1/3 per cent. over this capacity and maintain a good efficiency. It ought to be stated here, however, that the maximum evaporation of a boiler is limited mainly by the amount of coal which can be burned upon the grate. If the draft is sufficient a good boiler can develop a horse-power upon one-half of the surface recommended. By dividing the total number of pounds of steam that are to be evaporated from and at 212 degrees per hour by 3, the amount of heating surface may be obtained with sufficient accuracy.

The next step is the subdivision of this heating surface into the proper number of boilers. This is of considerable importance, for careful study may result in much saving in the first cost and in the cost of operation. For instance, if boiler capacity equivalent to evaporating 28,800 pounds of water from and at 212 degrees an hour is required, 9,600 square feet of heating surface will be needed. If each square foot of heating surface may be overloaded 33 1/3 per cent., it is evident that if the 9,600 square feet were divided among four boilers, one boiler might be shut down for repairs or cleaning, which is frequently necessary, and the other three run at 33 1/3 per cent. overload and still evaporate 28,800 pounds of water per hour. If the total heating surface was divided into three boilers, each of 3,200 square feet of heating surface, two might not be able to run the plant alone, so a fourth or spare boiler would have to be supplied. This would manifestly be a poor division of power, as the money spent on the spare boiler would represent so much capital lying idle most of the time. The frequency with which a boiler is shut down for repairs or cleaning depends upon the attention given it and the character of the feed water.

Importance of Proper Grate Surfaces.—To evaporate a given amount of water into steam it is necessary to generate a certain amount of heat by the combustion of fuel. The factors controlling the amount of heat generated are, the kind of coal, the amount of grate surface the boiler contains, and the draft. Ample grate surface therefore is highly desirable, particularly when boilers are to be forced. The draft affects the rate of combustion, as the amount of coal burned on each square foot of grate surface in an hour's time is usually called. Of the factors mentioned, that pertaining to the kind of coal to be used should be determined by an engineer before designing a boiler plant. He should investigate the cost of the various fuels available at the locality at which the boiler plant is to be constructed, and with these data and a knowledge of the relative evaporative power of the different fuels, he can determine in advance the cheapest kind of coal, the evaporative power and cost of each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Separator
Relief Valve
Circulating Water
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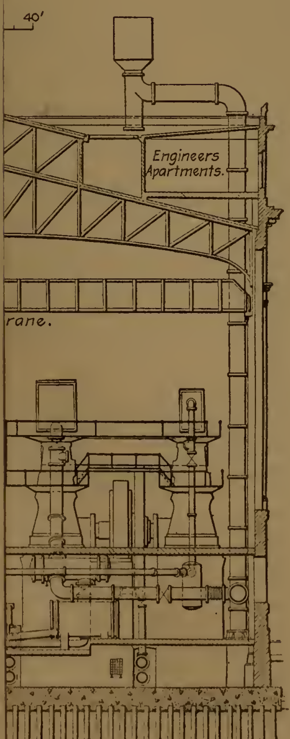
Circulating Water from River
Pump
THE ENGINEERING RECORD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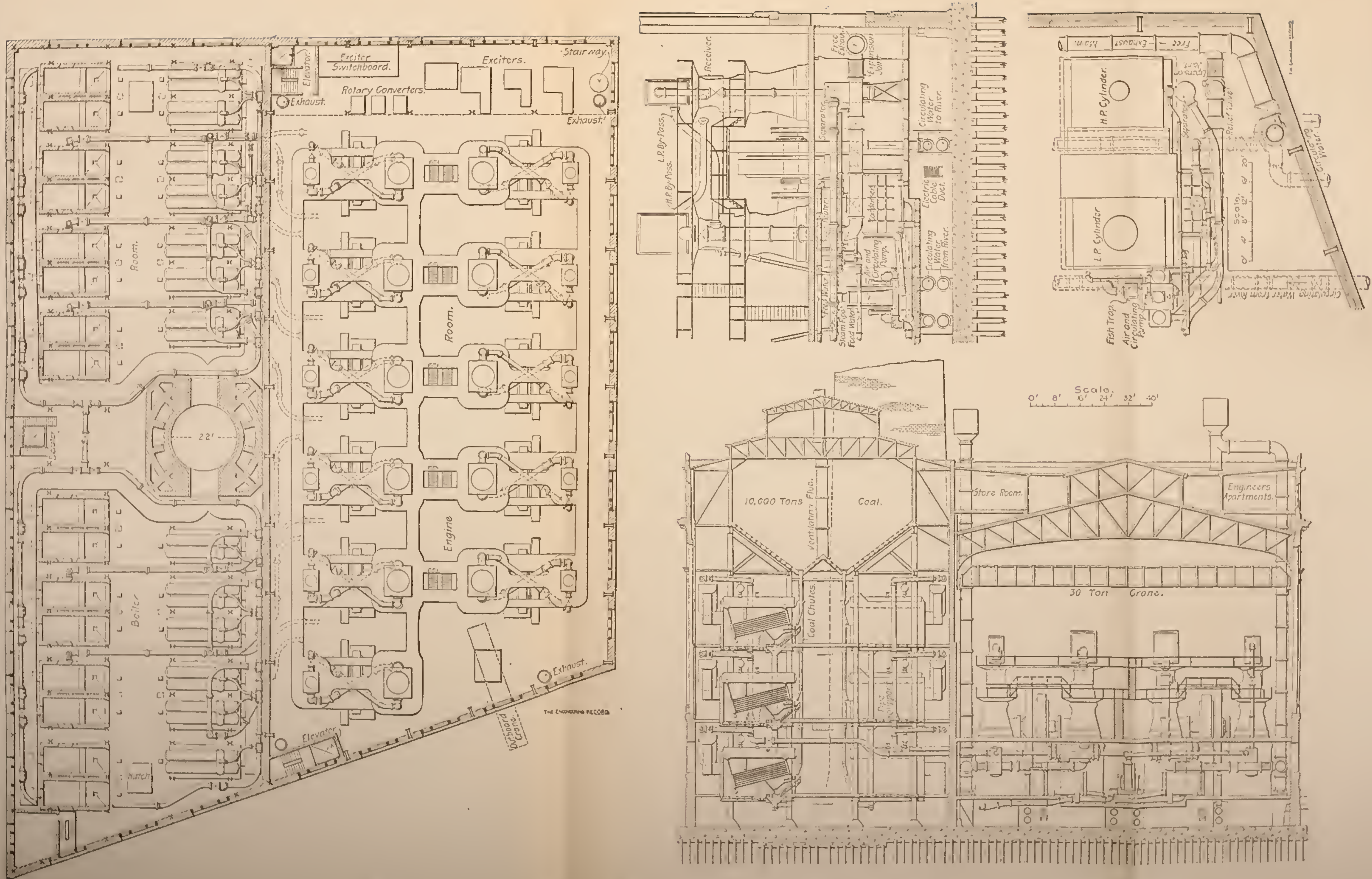

PLATE 2.—NINETY-SIXTH STREET POWER STATION, METROPOLITAN STREET RAILWAY COMPANY, NEW YORK.

M. G. STARRETT, CHIEF ENGINEER; F. S. PIERSON, CONSULTING ENGINEER.

being considered, and construct the boilers so that it can be used. Many plants have been somewhat handicapped by the failure of their designer to consider this subject. Some coals, besides having less heating power per pound than others, cannot be burned at as high a rate of combustion on account of peculiar properties they possess. For instance, with the finer sizes of anthracite coal, it is impossible, unless mechanical draft is used, to burn more than a limited amount on a grate, as the particles pack together so closely that the air cannot get through the bed of fuel in sufficient quantity to permit a rapid combustion. Again a coal high in ash and sulphur is limited in the rate at which it can be burned. Of course, if only a limited amount of fuel can be burned on a grate, the grate has to be made large in proportion to the heating surface, and when it is intended to burn a low-grade fuel, provision for a large grate ought to be made in the first place. If a plant is designed to burn a low-grade fuel, and it is desired to change to a fuel of better grade, this can be done by reducing the size of the grate by bricking off a portion of it. With a better fuel less coal would be burned, and it would probably be burned at a higher rate; consequently so large a grate would not be needed. It might seem that it would not be necessary to reduce the size of the grate to burn less fuel, but, although less fuel could be burned per square foot of grate by reducing the draft, yet it would probably not be good practice to do so, as it is necessary to burn some coals at a fairly high rate of combustion to secure the best result. Too slow combustion results in the partial burning of the gases, and this causes a loss. If it is certain, before a plant is constructed, that there never will be a desire to operate it with a lower grade of fuel than that for which it is designed, it would be foolish to provide larger grates than are necessary for this fuel, as boilers so constructed are frequently more expensive both in cost of construction and land occupied. A high rate of combustion is undoubtedly the best for many coals for the reason that the gases are much more thoroughly consumed when the furnace temperature is high.

Proportioning Grate.—The ratio of the grate to heating surface varies with the kind of coal and the amount burned, as was explained in an earlier paragraph. From a knowledge of the number of pounds of water to be evaporated, and the amount one pound of a given kind of coal will evaporate under ordinary conditions, the quantity of coal which must be burned in a given time can be calculated. If one knows the number of pounds of coal that must be burned, and the amount of coal that should be burned on each square foot of grate surface, in a given time for the normal rate of evaporation, the amount of grate surface can be determined.

It remains, then, to provide sufficient draft, either by a fan or chimney, to produce not only the proper rate of combustion for ordinary demands, but also a higher one which will enable the boiler to operate with such overload as may be thought necessary. The relative evaporative power of the better grades of a number of different coals is shown in the table, Pocahontas coal being placed at 100. These figures are approximate and should be used with some caution. The relative evaporation for the different coals shows what might be expected from the better grades of each kind of coal mentioned when fired by a good fireman under ordinary every-day conditions. The variation from these figures which might result from a difference in the quality of the coal from the same mine would be greatest with the Western coals, less with the Pennsylvania bituminous, and least with the semi-bituminous group. The anthracite, particularly the small sizes, might vary considerably on account of an abnormal amount of impurities present in the coal. The figures given in the table can be exceeded when all the conditions are favorable.

RELATIVE VALUE OF STEAM COALS.

Kind of coal.	Relative evaporative power.	Pounds of water that one pound of coal will evaporate into steam from and at 212 deg. Fahr., under ordinary conditions.	Pounds of coal per square foot of grate per hour.	Ratio of heating to grate surface.
Pocahontas, W. Va.*	100	9.5	15	45
Youghiogheny, Pa.†	92.5	8.7	17	48
Hocking Valley, O†	80	7.6	18	45
Big Muddy, Ill.†	80	7.6	20	50
Mt. Olive, Ill.†	67.5	6.4	20	45
Lackawanna, Pa.,‡ broken	87	8.5	15	35
Lackawanna, Pa.,‡ No. 1. buckwheat	73	7.5	13	32
Lackawanna, Pa.,‡ rice	63	7.0	12	30

*Semi-bituminous; †Bituminous; ‡Anthracite.

The table also shows about the amount of coal which should be burned per square foot of grate per hour under ordinary con-


6" Suction for Pumps
Strainer
14" Free Exhaust
6"
3" Suction.

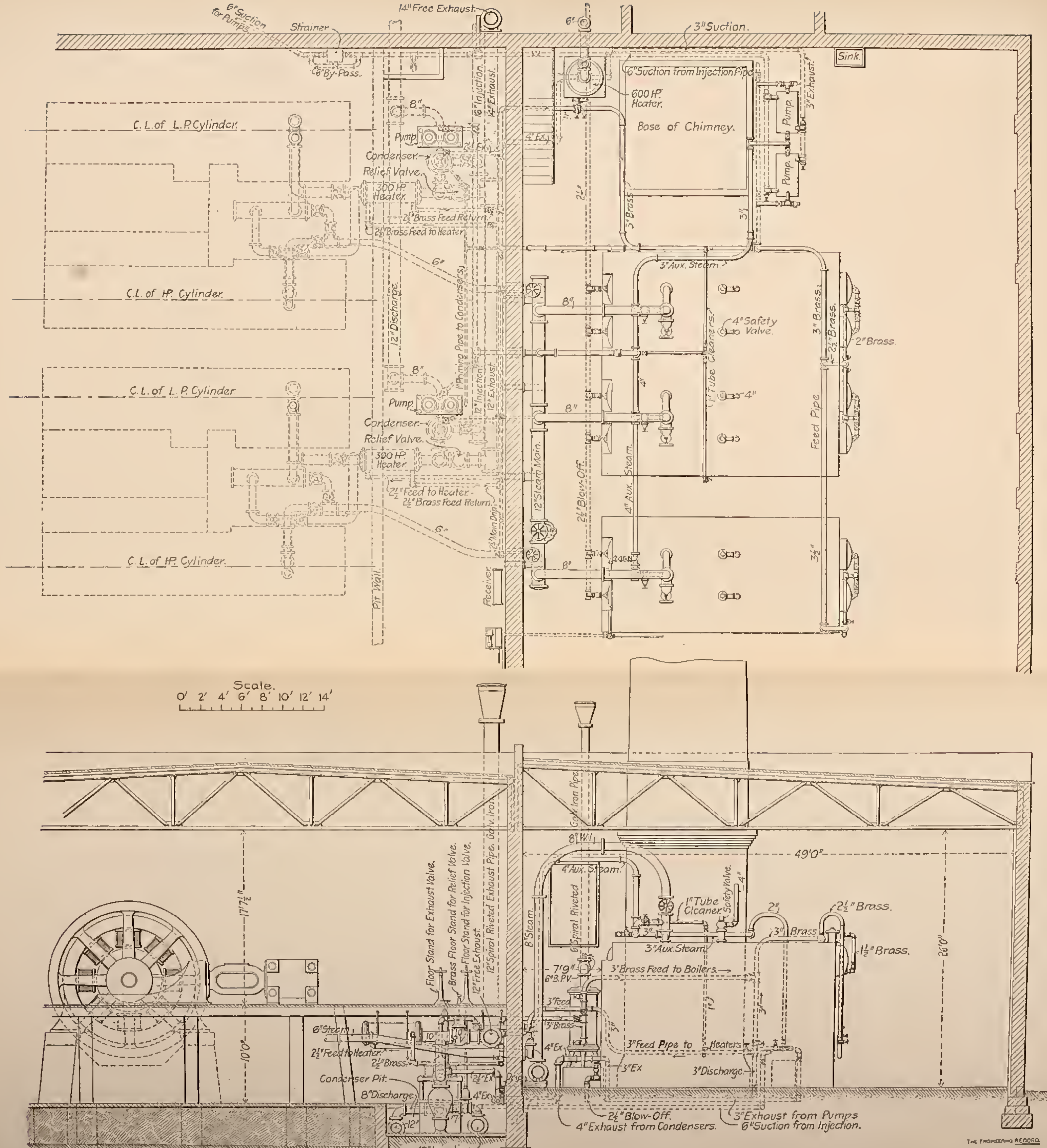

PLATE 3.—POWER STATION, HARTFORD AND SPRINGFIELD STREET RAILWAY COMPANY

E. H. KITFIELD, ENGINEER.

ditions; also, the ratio of heating to grate surface necessary for the boiler to develop its rated capacity when burning about the amount of coal stated and when each pound of coal is evaporating the quantities of water given in the table. Some authorities of considerable experience with Illinois coals advise higher rates of combustion than are recommended in the table for the best results. A grate proportioned in accordance with the data given can easily be reduced in size if found desirable. It is proposed to provide sufficient draft, that equivalent to the pressure of 0.5 inch of water, to run easily a boiler proportioned according to the data given in the table at one-third over its rating. Five-tenths of an inch of draft might not be sufficient with the rice size of anthracite, if it is of poor quality, to enable a boiler proportioned according to the data in the table to operate with an overload of one-third. With the rice size of anthracite mechanical draft ought to be used. It is well to have a maximum draft of 0.6 inch of water available for the poorer Illinois coals.

It is undoubtedly true that a better result will be obtained by higher rates of combustion than those given in the table, but if these are increased for the normal working of a boiler it will be necessary to have available a very intense draft, nothing less perhaps than that created by a fan, in order that there may be sufficient reserve draft to operate the boiler at much of an overload. In other words, if the rates of combustion are taken higher than those given for ordinary service, the draft must be made correspondingly greater to provide a reserve capacity in evaporative power. The high rates of combustion possible with mechanical draft are conducive to high furnace efficiency, but that is a subject which will be discussed later.

Coal.—In an earlier paragraph attention was called to the necessity of proportioning boilers to burn the fuel which will be the cheapest. It seems well, therefore, to show in a general way some of the properties and the relative values of different steam coals. Such information can be misleading on account of the variations that exist not only in different coals, but in coal mined from the same seam. Nevertheless an approximate relation can be given for the better grades of several different coals that are typical of the kinds most used. What the poorer grades of coal of some of these kinds will do, it is impossible to predict. The steam coals used in the eastern and middle parts of the United

States may be divided into anthracite, bituminous and semi-bituminous classes. A coal is classified in these groups according to the relative proportions of fixed carbon and volatile hydro-carbons that it contains. The hydro-carbons are those gases given off by certain coals when they are heated moderately. Semi-bituminous coals contain less than 25 per cent. hydro-carbons and bituminous coals 25 to 60 per cent. The former are the best steam coals for the reason that when the hydro-carbons are more than 20 per cent. of the fuel composition, the heat value of the fuel becomes less, for then the hydro-carbons contain more or less oxygen while with less than 20 per cent. hydro-carbon, the volatile gases are mostly hydrogen and the coal therefore has a higher heat value. The percentage of hydro-carbons in anthracite coal is very small.

Semi-Bituminous Coals.—This group contains the finest steam coals mined in the United States. They aré found mainly in Virginia, West Virginia and Maryland. The ash varies from 3 to 8 per cent., while the coals contain about 14,500 heat units per pound. The coals of this group are much more uniform in heating power and evaporation than those of any other, but there is a variation in some of them owing to the fact that care is not taken to exclude impurities which affect their heat value. This group of coals includes the Pocahontas, New River, Cumberland (George's Creek), and Clearfield varieties. The value is about in the order named, Pocahontas and New River probably being the most constant in quality. Placing the evaporative power of Pocahontas and New River at 100, none of the other coals would be hardly less than 95.

Bituminous Coals.—These are found in Pennsylvania, Ohio, Kentucky, Tennessee, Indiana, Illinois, Missouri and other states. They differ widely in heating power, not only one coal from another, but a great difference is also found in coal from the same mine. The bituminous coals are divided into two classes, caking and non-caking. The Indiana, Illinois and Missouri coals are of the caking variety, which on burning becomes pasty and forms into lumps that greatly impede the fire unless broken up. Certain western coals have to be burned at a comparatively high rate of combustion, about 20 pounds of coal per square foot of grate per hour, otherwise it is difficult to keep the fire from going out. The Pennsylvania coals are much the best for steam purposes, and the Ohio coals are usually better than those found farther west.

Anthracite Coal.—This is mined chiefly in Pennsylvania, although quite a little is found in the far West. It is principally composed of pure carbon and its heat value is dependent mainly on the amount of earthy matter mixed with it when sold. The percentage of earthy matter is naturally greater in the smaller sizes. Anthracite coal is classified according to size into lump, broken, egg, stove, chestnut, pea, numbers 1 and 2 buckwheat, rice and barley. Pea coal is about as large as is usually used for steam purposes. The larger sizes of this coal, that is, the chestnut size and over, are about equivalent in evaporating power to Pittsburg bituminous coal. The smaller sizes require a very strong draft because the particles of coal, being small, pack together so that the air cannot get through the bed of fuel to cause rapid combustion. It is therefore impossible with natural draft to burn more than a very limited amount per square foot of grate, and it is inconvenient and costly to provide boilers with a sufficient grate to burn buckwheat or the smaller sizes with the draft due to an ordinary chimney. It is necessary, therefore, if this grade of fuel is to be used to construct a 125-foot, preferably 150-foot chimney, or to employ mechanical draft. Rice and the smaller sizes can hardly be burned without mechanical draft. With 0.5 inch of draft from 16 to 18 pounds of good clean buckwheat coal can be burned per square foot of grate per hour.

Chapter III.—Design of Horizontal Return Tubular Boilers and Boiler Specifications.

The horizontal return tubular boiler is used to a far greater extent than any other type in the United States at the present time and it is intended in this chapter to give some general rules for designing boilers of this type. Another well known boiler of the fire tube type is the Manning, which is a vertical boiler with unusually long tubes rising from a high combustion chamber surrounded by water. It has been used successfully to a considerable extent in New England. Various boilers of the fire-tube type of special design have been illustrated from time to time in technical papers, but the use of these boilers is so limited compared to the horizontal return tubular boilers that they will not be further alluded to. A sectional view of a typical setting for a horizontal return tubular boiler is given in Figure 5. The method of proportioning the different parts of a boiler of this type follows:

Thickness of Shell.—One of the most satisfactory rules for determining the thickness of shell necessary in boilers of the fire-tube type, is that used by the steam boiler inspection department of the City of Philadelphia. The rule is as follows:

Working pressure $= 2Tts \div Df$,

in which

$T =$ the ultimate tensile strength of the plate in pounds per square inch.

$t =$ the thickness of the plate in inches.

$s =$ the efficiency of the longitudinal joint.

$D =$ the diameter of the boiler in inches.

$f =$ the factor of safety.

The factor of safety is usually taken at 5. With a tensile strength of 60,000 pounds and a joint efficiency of 80 per cent. the rule shows that with a 9/16-inch plate, which is about as thick as is commonly used for the shell of an externally fired boiler on account of the resistance that a thicker plate offers to the transfer of heat at the girth seams of the boiler, the highest working

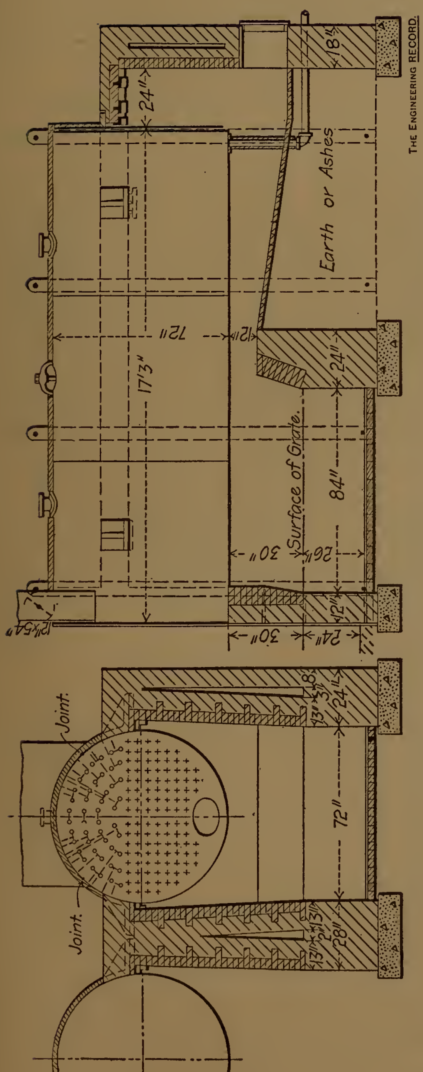

FIGURE 5.—TYPICAL SETTING OF HORIZONTAL RETURN TUBULAR BOILER.

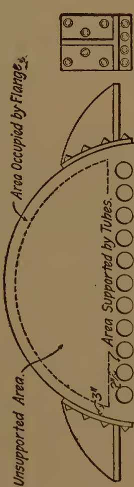

FIGURE 6.—UNSUPPORTED PART OF BOILER HEAD.

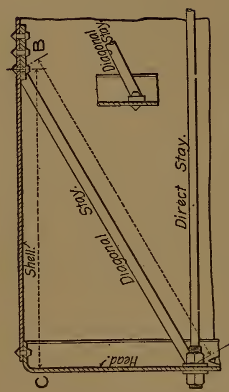

FIGURE 7.—DIAGONAL AND DIRECT STAYS.

pressure that can be carried in a boiler 6 feet in diameter is 150 pounds. A few horizontal return tubular boiler plants have been designed to carry a steam pressure of 150 pounds but it is better to use water-tube or internally fired fire-tube boilers if this or a higher pressure is to be carried unless the boilers are designed by an expert.

Braces.—Braces are used in horizontal tubular boilers to balance the pressure exerted on those parts of the heads of the boiler above and below the tubes and not close to the shell. If these parts are not braced, the pressure would cause the flat head to bulge outward. For a distance of 3 inches from the shell, and for a distance of 2 inches from the tubes, the head is usually considered to be sufficiently stiffened by the circumferential joint and the tubes not to need other bracing. Figure 6 shows one of the heads in a horizontal tubular boiler and the part which has to be braced is indicated. Braces used in this work are usually of two kinds, the direct or through stay or brace, which extends entirely

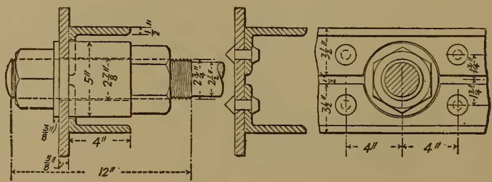

FIGURE 8.—CONNECTING DIRECT BRACES.

through the boiler and joins the heads together, and the diagonal or crow-foot brace. This latter is shown in Figure 7. The head is connected to the adjacent shell by it. The diagonal brace is preferred by the boiler inspection companies, as it is much easier to get inside of a boiler to inspect the interior with this brace than with the direct type. If the latter are used they should be sufficiently separated to allow a man entering the manhole opening to pass between them. The braces should be distributed uniformly over the area they are intended to support. Figure 5 shows the manner in which diagonal braces may be distributed over the head. Five direct braces are sometimes used, three being placed in a lower row and two above, all being symmetrically placed with reference to the center. When direct braces are used the heads are frequently further stiffened by channel iron or angle irons riveted to the heads as shown in Figure 8. If manholes are placed in one of the heads of the boiler the heads below the tubes should be braced.

In determining the number and size of braces the area in square inches of the surface to be stayed must be calculated and then the total pressure on this area. This latter divided by 7,500 will give the cross sectional area in square inches that must be provided in all of the braces combined, 7,500 pounds per square inch being the greatest strain to which a brace should be subjected. Cutting a thread on the end of the braces for the nuts, reduces the area of metal somewhat and this should be taken into account. Sometimes the ends of the braces are upset, that is, heated and hammered at the end so that their diameter is increased slightly where the thread is to be cut.

With diagonal braces the pull exerted is not perpendicular to the head of the boiler. Hence the area of the brace should be

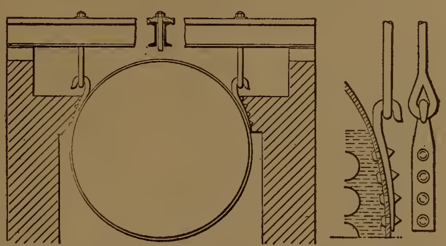

FIGURE 9.—METHOD OF SUSPENDING BOILER.

calculated by dividing the surface supported by 7,500 and then increasing the result by multiplying the quotient obtained by the length in inches of the brace divided by the distance in inches that the head is from the point where the brace is attached to the shell. Referring to Figure 7 the length of the brace is the distance A B and the distance from the head to the point where the brace is attached is the distance C B. The ratio A B : C B is the amount that the brace should be increased. For instance, if the surface to be stayed should require the pull of a brace 1 square inch and the length A B should be 48 inches and the length C B 40 inches, then the area of brace actually required would be $1 \times 48 \div 40 = 6 \div 5 = 1\ 1/5$ square inches.

Supporting Boilers.—For many years the practice has been to support boilers by riveting two pair of lugs on the sides of the shell, as in Figure 6, and supporting them on cast-iron plates built into the setting, rollers being placed under the pair of lugs in the rear, so that the movement due to expansion will be toward the rear. The principal objection to this plan is that the settlement of the walls of the setting, which is almost sure to occur, often throws almost the entire work of supporting the boiler on two lugs. A number of engineers suspend a boiler by riveting straps to the shell and passing hooked rods through them and through cast-iron plates resting on I-beams laid across the top of the set-

TABLE I.—TUBE SPACING AND HEATING SURFACE IN HORIZONTAL TUBULAR BOILERS.

Dia. boiler, in.	Tubes, Dia., in.	Tubes, Number.	Tubes, Internal area, sq. ft.	Heating surface per lin. ft. of boiler, sq. ft.	Size of manholes, inches.	Tubes. center to center. Horizontal, in.	Tubes. center to center. Vertical, in.	Center of boiler to center of upper row tubes, in.	Usual length of boiler, ft.
A.	B.				D.	E.	F.	G.	
48	3	46	1.94	43.67	9 x 14½	4¼	4	6½	10 to 12
50	3	52	2.19	48.69	9 x 14½	4¼	4	6¾	10 to 12
52	3	54	2.28	50.58	9 x 14½	4¼	4	7¼	10 to 12
54	3	60	2.53	55.60	9 x 14½	4½	4	7½	10 to 12
56	3	64	2.70	59.06	11 x 15	4½	4	8	10 to 12
58	3	70	2.95	64.09	11 x 15	4½	4	8¼	10 to 12
60	3	76	3.21	69.11	11 x 15	4½	4	8½	10 to 12
48	3½	34	1.97	38.68	9 x 14½	5	4½	6¼	12 to 14
50	3½	38	2.20	42.66	9 x 14½	5	4½	6½	12 to 14
52	3½	46	2.67	50.31	9 x 14½	5	4½	7¼	12 to 14
54	3½	47	2.73	51.53	9 x 14½	5	4½	7¼	12 to 14
56	3½	50	2.90	54.60	11 x 15	5	4½	7½	12 to 14
58	3½	52	3.02	56.74	11 x 15	5¼	4½	7¾	12 to 14
60	3½	56	3.25	60.71	11 x 15	5¼	4½	8¼	14 to 16
62	3½	60	3.48	64.72	11 x 15	5¼	4½	8½	14 to 16
64	3½	64	3.71	68.67	11 x 15	5½	4½	9	14 to 16
66	3½	70	4.06	74.49	11 x 15	5½	4½	9¼	14 to 16
54	4	36	2.75	46.17	11 x 15	6	5	7	16 to 20
56	4	38	2.90	48.59	11 x 15	6	5	7¼	16 to 20
58	4	40	3.05	50.99	11 x 15	6	5	7¾	16 to 20
60	4	47	3.59	58.63	11 x 15	6	5	8	16 to 20
62	4	49	3.74	61.04	11 x 15	6	5	8¼	16 to 20
64	4	51	3.89	63.45	11 x 15	6	5	8¾	16 to 20
66	4	56	4.27	69.00	11 x 15	6	5	9	16 to 20
68	4	62	4.73	75.59	11 x 15	6	5	9½	16 to 20
70	4	64	4.88	78.01	11 x 15	6	5	9¾	16 to 20
72	4	74	5.65	88.79	11 x 15	6	5	10¼	16 to 20

ting. Figue 9 shows this construction. This is one of the best methods. If it is used it is well to have an iron plate of generous size between the beams and the setting for the former to rest upon.

Tubes.—Tubes should not be so closely spaced in a boiler as to interfere with a proper circulation of water in them. Three, 3½ and 4-inch tubes are commonly used in horizontal tubular boilers, the last being generally used in large-size boilers, the size decreasing with the diameter of the boiler. Barrus states that a certain ratio of tube area to grate surface is necessary to give the best result with different fuels, a ratio of nine to one or ten to one being proper for anthracite coals and six to one or seven to one for bituminous coals, the difference being due to the large volume of gases with bituminous coals, which requires a large area in the tubes to pass the gases at the proper velocity for giving off their heat. Tubes should be placed in vertical rows and not staggered. They should be located as far apart as the number necessary to put in the boiler will permit; tor this will permit a better circulation, which is essential when the boiler is operating at high rates of combustion.

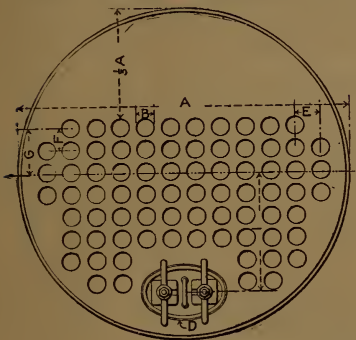

FIGURE 10.

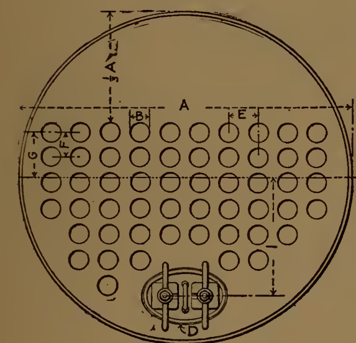

FIGURE 11.

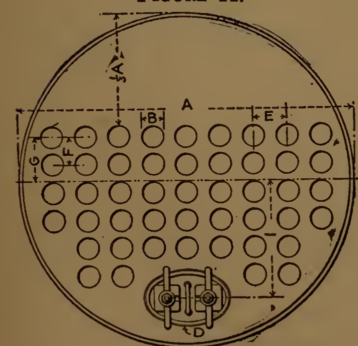

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FIGURE 12.

By means of Table I. and Figures 10, 11 and 12 the proper method of locating and spacing tubes in return tubular boilers may be found. The threc figures refer respectively to 60-inch boilers with 3, 3½ and 4-inch tubes. These data have been taken from Mr. W. M. Barr's work, "Boilers and Furnaces.". Figure 8 has also een taken from the same source. In using Table I. to determine the number and location of tubes for boilers of different diameters than those shown in the figure, a drawing of the tube sheet should be made on which the centers of the tubes can be located from the data given in the tables. Barr states that "the distance from the side of a tube to the inside of the boiler shell should not, in the case of a 36-inch boiler, be less than 2 inches, for a 48-inch boiler it should not be less than 2½ inches, and for a 50-inch boiler and larger, the distance should be not less than 3 inches, to secure good water circulation." Calculations have been made showing the heating surface per lineal foot of boiler for each of the different sizes given in the table, also the usual length of each. The tube area is also given and this may be useful in comparing the tube area with the size of grate. The method of

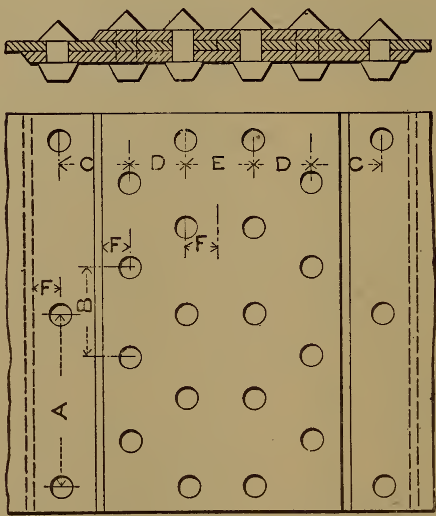


FIGURE 13.

determining the proper amount of heating and grate surface was explained in the preceding chapter.

Riveting.—In Tables II., III. and IV. there are given proportions of riveted joints recommended by the Hartford Steam Boiler Inspection & Insurance Company, which are published through the courtesy of that corporation. The efficiencies for the different joints, given in the table, are to be substituted in the formula for determining the thickness of shell plates. The dimensions to which the letters in Table IV. refer are shown in Figure 13. Butt joints are far superior to lap joints as there is not the tendency of the plate to crack on a line parallel with and near the rivets. The joints are designed for metal having a tensile strength of from 55,000 to 60,000 pounds, and the efficiency calculations are based upon the latter figure. The shearing strength

TABLE II.—DIMENSIONS OF DOUBLE-RIVETED STAGGERED SEAMS.

Thickness of sheet, inches.	Diameter of rivets, inches.	Diameter of rivet holes, inches.	Pitch, inches.	Lap, inches.	Distance between rows of rivets, inches.	Edge of sheet to pitch line, inches.	Efficiency.—Weakest part.
1/4	11/16	3/4	2 7/8	4 3/16	1 15/16	1 1/8	0.739
5/16	3/4	13/16	2 7/8	4 3/8	1 15/16	1 7/32	0.717
3/8	7/8	15/16	3 1/4	5	2 3/16	1 13/32	0.711
7/16	15/16	1	3 5/16	5 3/16	2 3/16	1 1/2	0.687
1/2	1	1 1/16	3.32	5 1/2	2 5/16	1 19/32	0.677
SINGLE RIVETED GIRTH SEAMS USED WITH ABOVE LONGITUDINAL SEAMS.							
1/4	11/16	3/4	2 1/16	2 1/4	....	....	0.545
5/16	3/4	13/16	2 1/8	2 7/16	....	....	0.494
3/8	7/8	15/16	2 3/8	2 13/16	....	....	0.49
7/16	15/16	1	2 7/16	3	....	....	0.466
1/2	1	1 1/16	2 1/2	3 3/16	....	....	0.449

TABLE III.—TRIPLE-RIVETED LAP JOINT.

Thickness of sheet, inches.	Diameter of rivets, inches.	Diameter of rivet holes, inches.	Pitch, inches.	Lap, inches.	Distance between rows of rivets, inches.	Edge of sheet to pitch line, inches.	
1/4	5/8	11/16	3	6 1/16	2	1 1/32	0.77
5/16	11/16	3/4	3 1/8	6 3/8	2 1/16	1 1/8	0.76
3/8	3/4	13/16	3 1/4	6 13/16	2 3/16	1 7/32	0.75
7/16	7/8	15/16	3 3/4	7 13/16	2 1/2	1 13/32	0.75
1/2	15/16	1	3 15/16	8 1/4	2 5/8	1 1/2	0.746
SINGLE RIVETED GIRTH SEAMS USED WITH ABOVE LONGITUDINAL SEAMS.							
1/4	5/8	11/16	2 1/16	2 1/16	....	....	0.456
5/16	11/16	3/4	2 1/8	2 1/4	....	....	0.419
3/8	3/4	13/16	2 1/8	2 7/16	....	....	0.412
7/16	7/8	15/16	2 3/8	2 13/16	....	....	0.42
1/2	15/16	1	2 1/2	3	....	....	0.398

TABLE IV.—TRIPLE RIVETED BUTT JOINT WITH DOUBLE WELT.

Thickness of sheet, inches.	Diameter of rivets, inches.	Diameter of rivet holes, in.	Thickness of strap, inches.	A.	B.	C.	D.	E.	F.	
5/16	11/16	3/4	1/4	6 1/4	3 1/8	2 3/8	2 1/8	2 1/2	1 1/4	0.88
3/8	3/4	13/16	5/16	6 1/2	3 1/4	2 1/2	2 3/16	2 7/16	1 7/32	0.75
7/16	7/8	15/16	3/8	6 3/4	3 3/8	2 3/4	2 1/4	2 13/16	1 13/32	0.86
1/2	15/16	1	7/16	7 1/2	3 3/4	3	2 3/8	3	1 1/2	0.866

SINGLE RIVETED GIRTH SEAMS USED WITH ABOVE LONGITUDINAL SEAMS.

Thickness of sheet, inches.	Diameter of rivets, inches.	Diameter of rivet holes, inches.	Pitch, inches.	Lap, inches.	
5/16	11/16	3/4	2	2 1/4	0.446
3/8	3/4	13/16	2	2 7/16	0.438
7/16	7/8	15/16	2 1/4	2 13/16	0.444
1/2	15/16	1	2 1/4	3	0.442

allowed to the rivet steel per square inch of section in single shear, when used in steel plates, is 38,000 pounds, and a rivet in double shear is considered to be equivalent to 85 per cent. additional. Rivet steel is allowed for shearing strength 45,000 pounds per square inch of section. In the calculations for the efficiency the rivet is assumed to fill the hole and the diameter of the rivet hole, not that of the rivet, is therefore used. The efficiency of the girth seams given in the tables need not be considered in determining the thickness of shells of boilers.

Boiler Settings.—Figure 5 shows a longitudinal and cross-section of popular form of setting for return tubular boilers. It should be constructed of hard red brick, laid in lime or cement mortar, and the entire setting ought to be lined with fire brick with every fifth course laid as headers, so that any part that might become damaged could be easily renewed without taking out the entire lining. Sometimes, to economize, the furnace as far as the bridge wall only is lined with fire brick. The fire brick should be laid in fire clay, using as little of it as possible. In the drawing the grate has a width equal to the diameter of the boilers and the side wall is battered so as to leave a space of 3 inches at the level where the setting closes into the boilers. The side walls are provided with air spaces, as shown, which are necessary to prevent the wall from cracking.

If fire brick, backed with common brick, is used the side walls have to be 13 inches thick and the air space should be about 2 inches wide at the bottom and diminish as its sides converge. Two 13-inch walls and a 2-inch air space make the entire thickness of the wall between the boilers 28 inches. This diminishes at the top as shown. It is sometimes difficult with low-grade fuels and natural draft to burn sufficient coal on the grate of a horizontal tubular boiler to obtain as high evaporation as might be needed, hence it is desirable, if these boilers are apt to be fired with such fuel, to place the boilers a little farther apart and make the side walls perpendicular and at a distance apart at the grate equal to the diameter of the boiler plus 6 inches. This makes the grate 6 inches wider and increases its surface materially. If a boiler is constructed in this way, it is a simple matter to diminish the size of the grate by bricking off the sides and rear. It is difficult to fire a grate more than 6 feet deep, although they are sometimes made 7 feet in depth, in large boilers. Furnaces over 5 feet wide should have two fire and two ash-pit doors.

The top of the bridge wall of the boiler is usually about 10 or 12 inches from the bottom of the shell, and the space behind may be filled with earth and paved with common brick or left empty. Curving this combustion chamber to conform with the shell only reduces its size, which is a disadvantage with bituminous coal and of no use with other kinds. The rear wall should contain an air space and be provided with a clean-out door about 16 x 20 inches. The wall should be located at a distance of 18 to 24 inches, depending on the size of the boiler, from the back head.

The back connection, that is, the connection between the rear wall and the head, is a source of more or less trouble on account of the expansion and contraction of the boiler, and the difficulty of making a joint that will remain tight. One method is to spring an arch across having one end resting on the wall and the other upon an angle iron riveted to the back head of the boiler. The arch consists of brick resting in an iron framework. The Hartford Steam Boiler Inspection & Insurance Company uses an arch somewhat similar to that shown in Figure 5.

The grate should be at a level of 24 inches above the floor and the shell from 28 to 30 inches above the grate with anthracite coals and from 36 to 42 inches if bituminous coals are to be burned, as the gases from the latter do not burn properly if the flame comes in contact with the cooler boiler surfaces. The floor of the ash pit should be paved with brick laid on edge and flushed with a thin cement mortar so as to be water tight. This floor is usually 3 to 6 inches below the floor of the boiler room.

An interesting horizontal tubular boiler is shown in Plate 4 and Figure 14. This boiler was built for a steam pressure of 185 pounds per square inch from the plans and specifications of Dr. E. D. Leavitt, consulting engineer. A full description of it appeared in The Engineering Record of May 18, 1901.

Boiler Specifications.—It was thought advisable to print one or two typical specifications for horizontal return tubular boilers and to outline a method to be pursued in purchasing water tube boilers, the difference in construction of boilers of the latter type making it necessary for the specification to be mainly a general description of what is wanted and what the boilers are to do.

In the first form of specification printed, which calls for the delivery and setting of a boiler, those clauses that relate to the prosecution of the work, the removal of rubbage after its completion, remuneration for a change in the plans, interpretation of drawings, etc., are omitted. The specification is divided into sections and in some cases comment or explanation follows a section. A form of specification for high-grade work follows:

Intent of Specifications.—It is the intent and purpose of these specifications to provide for the furnishing and installation of a complete boiler plant comprising the boilers, setting, grates, fronts, feed and blow-off piping, gauges, smoke breeching, flues, dampers, and such details as are hereinafter specified, all complete and erected ready to generate steam.

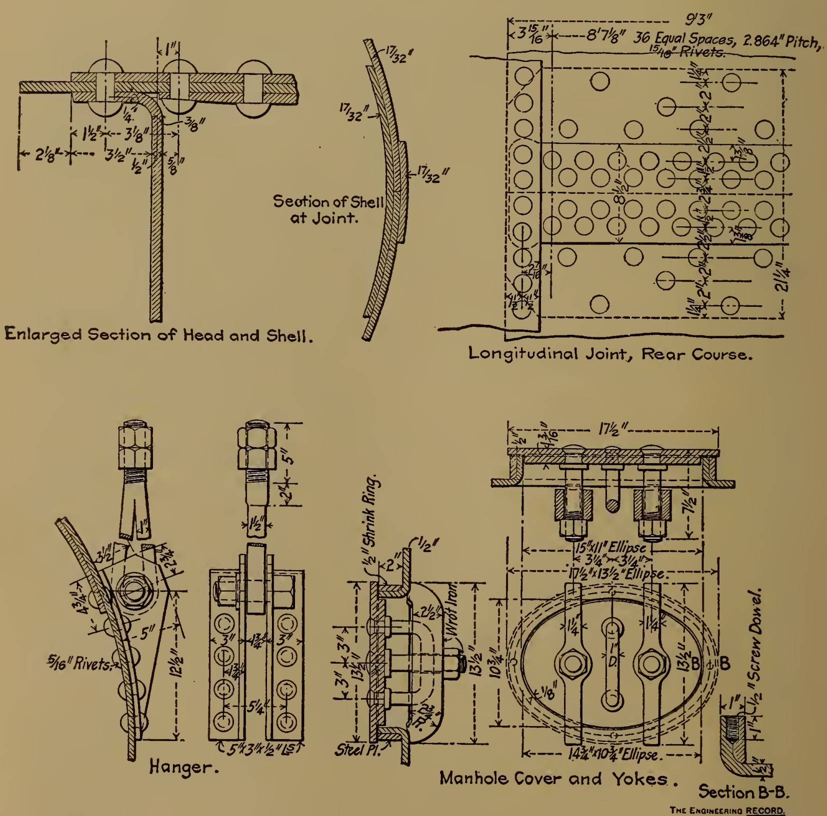


FIGURE 14.—DETAILS LEAVITT BOILER.

Materials.—The shells and heads of the boiler are to be constructed of open hearth steel having a tensile strength from 55,000 to 65,000 pounds per square inch, with a yield point not less than one-half the tensile strength and an elongation of 25 per cent. in eight inches. (See note below.) Test specimens cut from the plates must, when cold, be capable of being bent 180 degrees flat on themselves without fracture on the outer side of the bent portion, also, after being heated to a light cherry red and quenched in water of a temperature between 80 and 90 degrees Fahrenheit.

For the bending tests the test specimens shall be 1½ inches wide, if possible, and for all material ¾ inch or less in thickness, the test specimen shall be of the same thickness as that of the finished material from which it is cut; but for material more than ¾ inch thick, the bending specimen may be only ½ inch thick. One specimen for the cold bending and one for the quenched test to be furnished from each plate and marked for identification. Specimens for the tensile tests of the dimensions shown in the upper illustration in Figure 15 are to be cut from each plate marked for identification and given to the engineer.

Rivet steel shall have a tensile strength from 45,000 to 55,000 pounds per square inch, the yield point to be not less than one-half the tensile strength and the elongation 28 per cent. in eight inches. Rivet steel of the full size as rolled shall pass the same bending tests as specified for

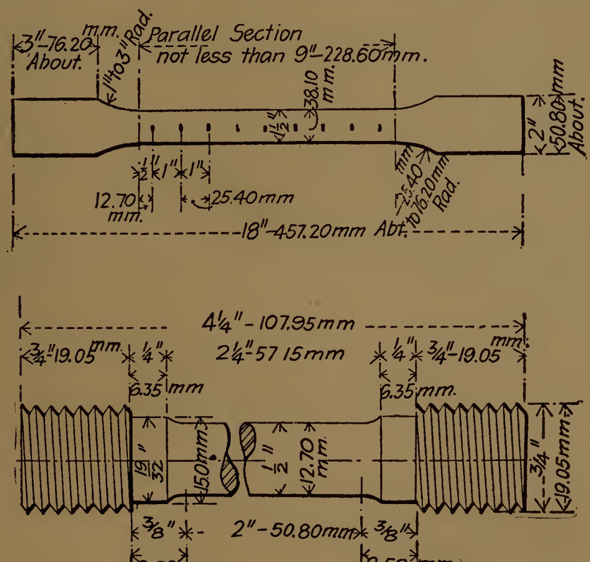


FIGURE 15.—DIMENSIONS TEST SPECIMENS.

shell plates. Two tensile test specimens shall be furnished from each melt of rivet rounds. In case any one of these develops flaws or breaks outside of the middle third of its gauged length, it may be discarded and another test specimen substituted therefor. Two cold bending specimens and two quenched bending specimens shall be furnished from each melt of rivet rounds.

NOTE.—The foregoing section upon materials is an abstract of the standard specification for boiler plate and rivet steel adopted June 29, 1901, by the American Section of the International Association for Testing Materials. Reference to the chemical properties of the steel has been omitted. A variation in the elongation with different thicknesses of plate is provided as follows: For each increase of ⅛ inch in thickness above ¾ inch, a deduction of one per cent. shall be made from the specified elongation. For each decrease of 1-16 inch in thickness below 5-16 inch a deduction of 2½ per cent. shall be made in the specified elongation.

Shell and Heads.—The shell plates are to be — inches thick, the heads — inches thick. The shell is to be in three courses (or two courses if the boiler is small enough), and the middle one must be of the smallest diameter. The edges of the plates must be planed before they are put together and caulking must be done with a round-nosed tool. Heads must be flanged to a radius not less than 1½ inches and they must be annealed after flanging. The flange must be free from cracks and flaws.

Riveting.—The girth seams are to be of the single-riveted lap-joint type, the pitch of the rivets to be — inches; lap — inches and rivets — inches in diameter. The longitudinal seams are to be placed well above the fire line, and to be of the butt type triple riveted with double welt, as shown in the drawing (which the engineer should supply). The rivet holes are to be drilled or punched ⅛ inch small and afterward drilled or reamed out to full size. A reamer must be used instead of the drift-pin. The burr is to be removed and the rivets driven by a hydraulic or pneumatic riveter as far as possible.

Note.—Some specifications intended for high grade work demand that the rivet holes shall be punched ⅛ inch small and the shell then rolled and the lapping ends bolted together while the holes are drilled out the full size. This insures that the holes in one end of the sheet shall come opposite those in the other end and much better riveting will be the result. Oftentimes the holes are punched out to the full size, but authorities unite in saying that the metal immediately surrounding a punched hole is weakened. Hence it is better to punch the holes small and drill or ream them out to full size.

Bracing.—The heads are to be braced to the shell by — radial solid crow foot braces, — on each head distributed as shown on drawing. The braces must be of — inch stay bolt iron at least 3 feet long, forged up without a weld and be connected to the shell by two — inch rivets. Each brace must be so located as to lie in a plane passing through the axis of the boiler.

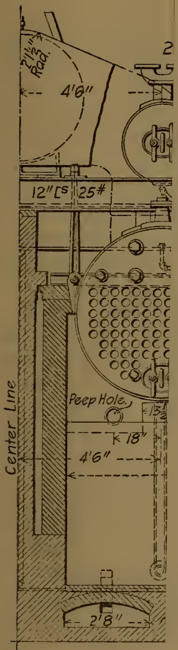
Note.—If through braces are used, their number and diameter should be given and their location shown on a drawing. If the ends of the braces are to be upset, or if channel or angle irons are to be riveted to the heads to stiffen them, it should be so stated, and their weight and dimensions specified, or shown by a drawing.

Tubes.—The tubes are to be — in number, — inches in diameter, — feet long, of the best charcoal iron lap welded or of steel. The tube holes in the head are to chamfered and the tubes are to be carefully expanded and beaded over at the ends. The tubes are to be located as shown in a drawing (which the engineer should supply).

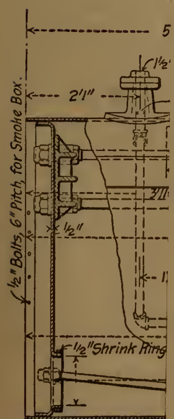
Manholes.—A pressed steel manhole frame, 11 by 15 inches, is to be riveted to the top of each boiler, with the long diameter girthwise (in a position indicated by the engineer), and another is to be riveted to the front head below the tubes. The usual pressed steel manhole plate, gasket, yoke and bolt is to be provided for each.

Hand-hole.—A 4 by 6 inch hand-hole is to be placed in the back head below the tubes. The plate, gasket, yoke and bolt is to be provided.

Nozzles.—Each boiler is to have two — inch cast-iron nozzles riveted to the boiler in the middle of the front and rear courses. The nozzles are to be drilled to fit the A. S. M. E. standard flange schedule.


2'1½" Rad.
4'6"
12" [s 25#
Peep Hole.
18"
4'6"
2'8"
Center Line

Secti


2'1"
½" Bolts, 6" Pitch, for Smoke Box.
½"
½" Shrink Ring

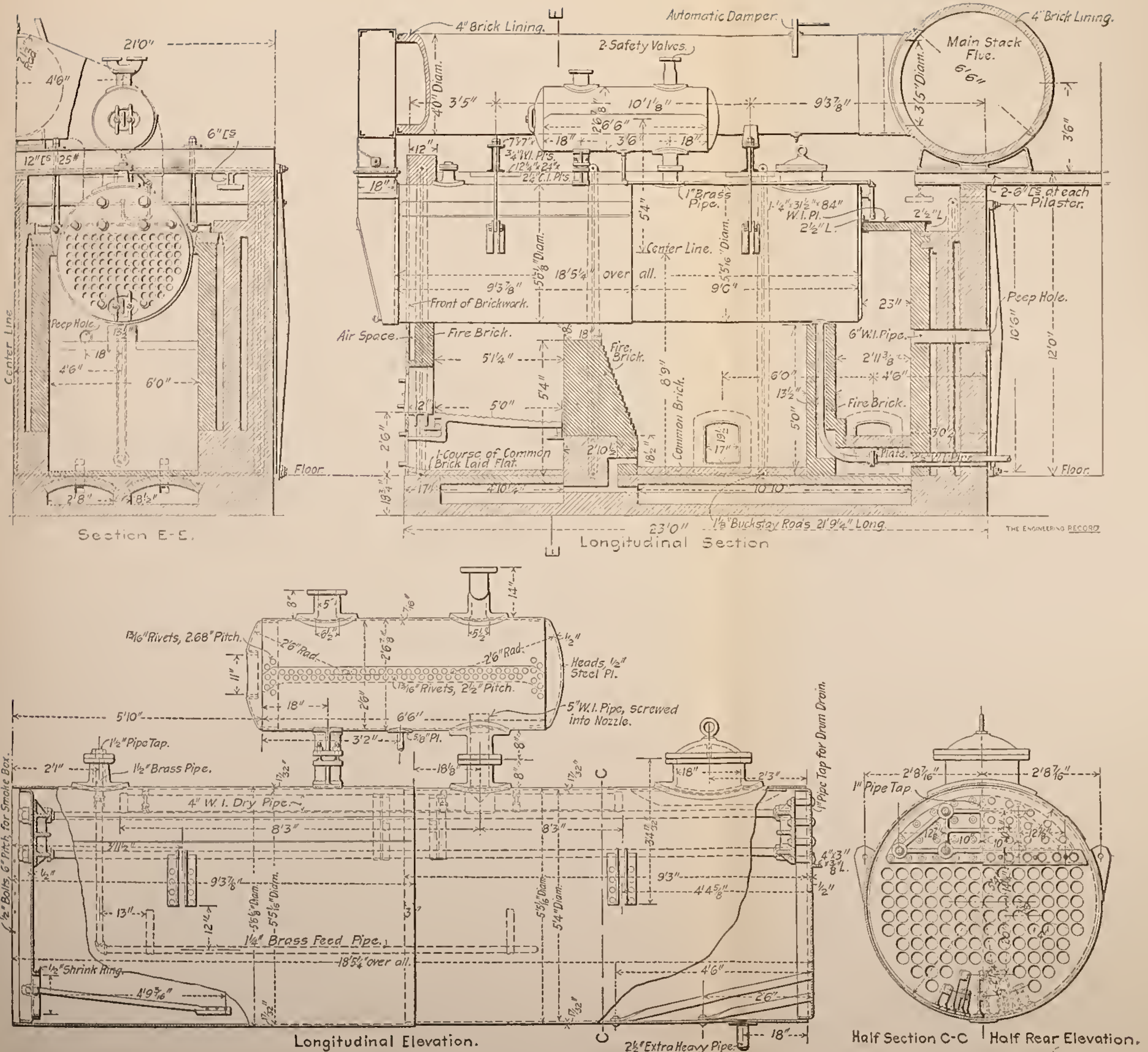


PLATE 4.—BOILERS AT PLYMOUTH CORDAGE COMPANY.

DR. E. D. LEAVITT, ENGINEER.

Lugs.—The boiler is to be supported on the brickwork by four heavy cast-iron lugs riveted to the shell, one pair to the front and the other to the rear course. The front lugs shall rest upon bearing plates. Three 1-inch rollers shall be placed under each rear lug between it and the bearing plates, which are also to be provided.

NOTE.—If the boiler is to be supported by straps from beams laid across the top of the setting, it should be specified and a drawing showing the detail of the method of hanging the boiler given.

Feed Pipe.—The contractor is to supply and connect a — inch steel (or brass) feed pipe that is to pass through a brass bushing in the front head of each boiler, at one side and 3 inches above the tubes and extending to the rear of the boiler.

Tools.—Each boiler is to be provided with a fire shovel, slice bar, hoe and poker. The boilers are to be provided with one tube scraper and the necessary hose and nozzle for blowing soot from the interior of the tubes.

Blow-off.—A blow-off connection is to be provided at the bottom of each shell at the back end. The — inch opening in the shell is to be tapped to receive a — inch blow-off pipe and the opening is to be reinforced by a — inch steel plate riveted to the shell. The blow-off pipe is to be protected by a fire-clay tube and run in the manner shown in the drawing to a blow-off main provided for in another contract. The boiler contractor is to supply a — inch blow-off cock of —— make for each boiler and the fire-clay tube for protecting the blow-off pipe.

Castings.—Each boiler is to be provided with a cast-iron flush front, cheek plates, mouth plates, one (or two) fire and one (or two) ash pit doors and clean-out door for rear wall, arch plates, grate bars, bearing bars, T bars for back connection, tie rods, wall braces, etc. The grate bars shall contain 50 per cent. air space and are to fit a grate — by — inches and to be suitable for burning —— coal. The fire doors are to be fitted with registers and perforated linings, the ash pit doors with registers, and the clean-out door a lining.

Fittings.—Each boiler to be furnished with the following fittings, which are to be connected: One water column upon which can be mounted a steam gauge, gauge cocks and gauge glass connected to the boiler by 1¼-inch pipe; three — inch gauge cocks; one — inch gauge glass; one — inch steam gauge of —— make and provided with stop cock and siphon.

Hydrostatic Test.—The boiler must be made sufficiently tight to stand without leaking when subjected to a hydrostatic test of a pressure 33 1-3 per cent. in excess of the normal working pressure of — pounds.

Setting.—The boilers are to be set in accordance with the accompanying drawing. The settings are to extend 6 inches below the floor level and are to be of the best brick laid in lime mortar and the entire setting is to be lined with the best quality of fire brick laid in fire clay and with every fifth course set as headers. The setting is to be closed in at the shell 5 inches above the center line of the boilers, except at the lugs where the setting closes in two courses below the lugs. The boilers are to be supported by lugs as previously stated and care must be taken that bearing plates are placed level and that rollers under the rear and middle lugs are placed perpendicular to the axis of the boilers. The foundations for the setting are to be laid by the contractor.

Smoke Flue and Chimney.—If an iron chimney is to be used, some believe, in small work, that it is best to put the boilers, smoke breechings and flue, damper and damper regulator and chimney in the boiler contract for the reason that all of them affect the operation of the boiler. The method of proportioning the chimney, etc., is discussed later. The boiler specification therefore should indicate the thickness and dimensions of the smoke flue. Provisions should be made for a damper in the uptake of each boiler that may be closed when the boiler is undergoing repairs. There should be a damper in the main flue and this should be under the control of an automatic damper regulator which opens or closes the damper according as the steam pressure is respectively below or above the normal. The smoke flue should have doors through which the soot may be removed when necessary.

The following specification for a horizontal return tubular boiler, without the setting, was prepared by Messrs. Dean and Main, a well known firm of mill engineers, and it is printed through their courtesy. The author believes it to be an excellent specification for the purpose intended. It will be noticed that it calls for the construction and delivery of the boilers only. Appended to the specification was a blue print reproduced in Figure 16.

SPECIFICATIONS FOR THREE 78-IN. HORIZONTAL RETURN TUBULAR BOILERS FOR—

Proposals.—Proposals will be received by Dean & Main, Exchange Building, Boston, Mass., for building and delivering three horizontal return tubular boilers of the following general dimensions:

Working pressure by gauge, lbs.	125
Inside diameter of end rings of shell, in.	78
Thickness of shell, in.	7/16
Length of tubes, ft.	18
Diameter of tubes, in.	3
Number of tubes	166
Width of grate, ft.	7
Length of grate, ft.	7
Height of center of boilers above floor, ft.	8.25
Distance from c. to c. of boilers, ft.	9
Height front of grate above floor, ft.	2.5
Height back of grate above floor, ft.	2
Thickness of heads, in.	1/2
Diameter of rivets, in.	7/8
Heating surface, sq. ft.	2,376
Grate area, sq. ft.	49

Contract Price.—The price is to be stated to cover the manufacture of the boilers and their delivery with the accessories furnished on cars.

Quality of Materials and Workmanship.—It is the intention that the materials and workmanship of the boilers shall be of the best. A representative of Dean & Main is to have the privilege of inspecting the boilers during their construction.

Plates.—The plates are to be of the best quality of open hearth acid or basic steel, having the following qualities:

Elastic limit per sq. in. not less than	30,000 lbs.
Ultimate strength per sq. in	52,000 to 60,000 lbs.
Elongation in eight inches not less than	27 per cent.
Sulphur not to exceed	0.35 per cent.
Phosphorus not to exceed	0.30 per cent.

The plates are to be free from laminations and surface defects, and also to stand cold and quench bending tests flat down without showing a sign

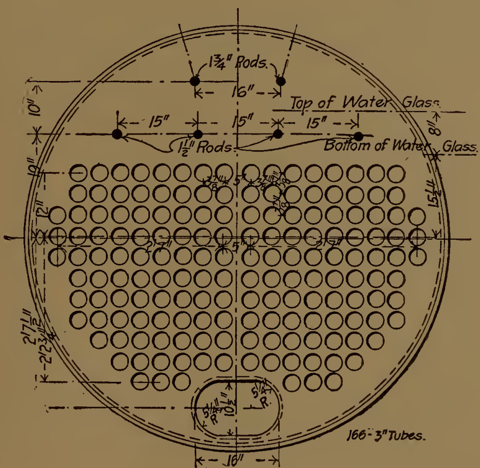


FIGURE 16.

of fracture. The tests are to be made by (name of testing laboratory) at the expense of the purchaser.

Joints.—The circular joints are to be single riveted and lapped. The longitudinal joints are to be butted and provided with inside and outside covering plates. The inside and outside covering plates are to be double riveted each side of the center of the joint, and the inside plate will be extended sufficiently beyond the outside on both sides of the joint to receive on each side two additional rows of rivets passing through it and the boiler shell. Thus there will be eight rows of rivets in each longitudinal joint. The outer rows of rivets will have double and triple the pitches of the other rows.

There will be one longitudinal joint in each plate. The boiler is to be made of three plates, and the joints of the two end plates will be above the water line on one side of the boiler and that of the center plate above the water line on the other side. If practicable it is preferred to have the seam near the bridge wall back of that wall, thus making the front plate wider than the others. The heads will be single riveted to the shell.

Riveting and Holes for Rivets.—The holes for rivets will be punched one-fourth inch small and drilled to size with all plates and covering plates in place. The riveting is to be done by a hydraulic machine. As little hand riveting as possible is to be done. The plates are to be fitted metal to metal before riveting, brought up to butt in contact, and are to be properly curved out to the ends. No filling pieces are to be used and no drifting is to be done.

Caulking.—Caulking is to be done with a round nose tool.

Tubes.—The tubes are to be of lap welded steel, and are to be beaded over the ends. A Prosser expander is to be used. The tube holes are to be rounded on each side to 1/16 in. radius, and not beveled as is customary. The tubes are to be located as shown on the enclosed blue print. (Figure 16.)

Manholes and Handholes.—There are to be two manholes in each boiler, one on top and the other in the front head below the tubes. The manhole on top is to be at one end of the boiler just far enough from the head to enable a man to get in feet first in either direction. This is to have a pressed steel frame, plate and yoke. The frame is to amply make up the area of plate cut from the shell and is to be double riveted thereto. The manhole in the head is to be located as shown on the accompanying print, is to be pressed on the plate and to have a pressed steel plate and yoke. There will be a 4 inch x 6 inch handhole in the back head.

Bracing.—Above the tubes there will be four longitudinal tie rods passing through the heads and upset at the ends, having a nut outside and inside the plates. There will be four braces from the heads to the shell riveted to the latter some five or six feet from the heads. These braces will be pinned to horizontal stiffening angles riveted to the heads. There will also be horizontal stiffening angles riveted to the heads above and below the longitudinal tie rods. Below the tubes the heads will be stiffened by several short diagonal braces riveted to heads and shell.

Steam Nozzle.—There will be one 7-inch steam nozzle midway between the heads of each boiler. It will be a steel casting double riveted to the shell, and drilled for 1⅛ inch bolts.

Steam Box and Dry Pipe.—Bolted up against the underside of the steam nozzle, or screwed into it, there is to be a 5 x 5 x 7-inch tee. Into each 5-inch branch there is to be a 5-inch wrought iron pipe, the lengths to be equal and determined by the distance from the steam nozzle to the manhole nozzle. The top of the pipe is to be about one inch below the shell and each branch is to be perforated on top with twenty-five 1-inch holes. The outer ends of the pipes are to be capped and the pipes strapped up to the shell. There are to be two ¼ inch drain holes in the bottom of the steam box.

Safety Valve Nozzle.—There will be no safety valve nozzle.

Feed Pipe Nozzle and Feed Pipe.—This will be on top of the boiler near the front head. The feed pipe will pass through it, branch to one side and pass to the other end of the boiler where it will discharge. The feed pipe will be 2-inch brass, iron size, and will be furnished with the boiler.

Blow-off Pipe.—This will be tapped into the bottom of the boiler, and will be 2½ inches in diameter.

Safety Plug.—This will be screwed into the back head in the customary position. A Lunkenheimer plug filled with pure Banca tin is to be used.

Smoke Box.—The smoke box will not be formed by the extension of the main shell but by bolting a ⅜ inch plate thereto, and having a smoke nozzle riveted to it. The size of the smoke nozzle will be 18 inches x 5 feet 6 inches. The smoke boxes will overhang. The smoke box head will be of cast iron secured to the shell air tight. The doors will likewise be of cast iron fitting against planed faces and having turned hinge pins and drilled holes. The doors are to be clamped to the head after locomotive practice.

Damper.—A damper with two plates with a rod between them is to be provided with a weighted lever and chain for its control.

Front.—The front is to be made of steel plate ½ inch thick.

Grates.—These will be furnished by the purchaser.

Fire and Other Doors.—Each boiler will have two large size fire doors, with door frames, planed to match each other, and to have turned pins and drilled holes. The back cleaning door is to be clamped to the frame, after locomotive practice, to have planed joints, turned pins and drilled holes for pins. It will be so made that it can be securely built in.

Supports.—The boilers will have a double set of supporting brackets at each end, eight for each boiler, the back ones having rolls.

Fittings.—The contractor will furnish, but not attach, the following fittings for each boiler:

One 10-inch ——— company's steam gauge graduated to 300 pounds, and numbered every 20 pounds.

One 5-inch ——— company's safety valve set to blow at 127 pounds.

One Reliance water column without safety device, and provided with two water glass fixture bosses on opposite sides at such distances apart as will accommodate an 8-inch exposed length of glass. There will be no gauge cock bosses.

Two ——— company's heavy pattern water glass fixtures with best Scotch glasses, 8-inch exposed length.

Four extra glasses of proper length.

Extra strong iron size brass piping for water column, all cut to length and threaded, and holes drilled and tapped therefor in shell.

No gauge cocks will be used.

Arch Bars.—Proper arch bars are to be supplied with each boiler.

Buck Stays and Rods.—These are to be furnished and the buck stays are to be of 8-inch I-beams.

Drawing.—The party receiving the contract is to furnish a first-class drawing of the boiler as built, and of the setting therefor.

Testing.—Each boiler will be tested at the works of the contractor to 190 pounds per square inch with water pressure and made tight under that pressure in the presence of a representative of Dean & Main.

Inspection.—The boilers will be built under the inspection of Dean & Main and the Mutual Boiler Insurance Company.

Reservation.—The right is reserved to reject any or all bids.

Terms of Payment.—Proposals are to state the desired terms of payment.

Delivery.—The proposal is to state the guaranteed time of delivery of the three boilers.

Specifications for water tube boilers.—An engineer's specification for water tube boilers can only be a very general statement of what is wanted, for the reason that such boilers vary greatly in design. The engineer can, however, describe what he wants and ask each bidder to supply information enabling him to understand thoroughly what each bidder proposes to furnish. The engineer should state in his specification the number of boilers he wishes to purchase, the amount of heating and grate surface each is to contain, the location of the plant for which they are wanted and the name of the owner or owners of the plant. If the boilers are to be erected and placed in running order by the

FIGURE 17.—BOILER ROOM PLYMOUTH CORDAGE COMPANY.

builder, it should be so stated. The engineer should state the kind of coal to be burned, the kind of service to be required of the boilers and the dimensions of the chimney and flues. It is well to furnish a sketch showing the location of the boilers, flues and chimney. If a mechanical stoker is to be used the boilermaker should be asked to furnish a boiler setting to suit. The engineer should require each bidder to furnish detail drawings showing all parts of the boiler and setting, in such a way as to show the facilities for cleaning and inspecting the parts and the manner of making renewals. He should ask for the physical qualities of the metals used; the material of which the tubes are made; the name of the manufacturer of the different fittings used, such as the safety valve, blow-off cocks, water and steam gauges; the character of feed piping, whether iron or brass.

For certain types of water-tube boilers with straight tubes slightly inclined from the horizontal and connecting with vertical or nearly vertical headers at the front and rear of the boilers, the headers connecting with combined steam and water drums, there are usually several different arrangements of tubes that will meet a specification calling for a certain heating surface. For instance, there may be sixty 18-foot tubes so arranged that there are ten tubes in width and six tubes in height or there may be twelve tubes wide and five tubes high. The latter arrangement would mean a wider boiler costing more, but giving a wider and consequently larger grate that might be needed with low-grade fuels. It is probable, the heating and grate surface and coal burned being the same, that a high narrow boiler of this type will give a better efficiency than a low wide one. If the boilermaker is to supply and erect the breechings, smoke flues, chimney dampers and damper regulator it should be stated in the specifications. If the engineer proportions the boiler it is hardly fair to exact a guarantee from the maker as to the efficiency of the boiler. He should, however, guarantee that the boiler will operate at 50 per cent. over its rated capacity without showing signs of depreciation.

Chapter IV.—The Selection of the Types of Engines, Dimensions of Cylinders, Speed, Steam Pressure, Etc.

The method of buying a steam engine most frequently employed by engineers is to invite bids from a number of builders upon an engine which, with a certain number of revolutions, steam pressure and back pressure, will develop a given horse-power. The cylinder dimensions are usually supposed to be looked after sufficiently when the bidders are asked to state the steam consumption they will guarantee. On opening the bids thus obtained, it is sometimes found that some offer engines with smaller cylinders than others, to do the same work. The prices are naturally different, and it occasionally happens that the purchaser does not get the engine best suited to his needs, first cost and economy of operation both being considered. The non-technical factory manager, knowing nothing of engines, frequently makes a mistake in this matter of purchasing an engine, as he does not know the type of engine he wants and much less what its details should be. He, therefore, frequently buys something very different from what he needs. Such an individual should either engage a competent consulting engineer to invite bids for him on the type of engine best suited to the conditions, or go to one reputable engine builder, inform him of the conditions that exist, and have an engine built to suit them. There is no doubt but that there are builders fully competent to study the conditions involved and advise an owner what is required. Two or more builders should not, however, be placed in competition where the cheapest bid is likely to be the deciding factor in the selection of an engine, and an incompetent person is to be the judge of the propositions offered. It could hardly be possible to educate the non-technical purchaser of engines, but there are a number of engineers who are not steam engineering experts that have to buy engines without the aid of expert advice; and it is hoped to give some general information and data to assist such in determining the type of engine, approximate cylinder dimensions, etc., best suited for different situations, so that bids can be invited upon a specific machine and thus secure the benefit of a competition, where all bids are made upon the same basis. In doing this, the selection of the type of engine will first be discussed; afterward, the determination of those factors that fix the capacity and efficiency of an engine; and finally, the method of applying these factors in the general equation for determining the power of an engine so that the cylinder sizes or volumes may be calculated.

Selection of type.—To most people it is manifest that it would be bad engineering to install a costly triple-expansion engine that operates with the highest economy in a plant where fuel is of little or no value. Many situations, however, require the keenest judgment to select the type of engine best adapted for the service the engine is to perform. There is a saying that any expenditure for plant is warranted that results in savings which more than pay the interest on the expenditure, the depreciation and the cost of repairs on the plant. This is a simple proposition in itself, but it is not always easy to predict the saving that will follow the installation of a machine, or what the depreciation and repairs will be. As a basis for such calculations regarding steam engines, Table I is given, and in it is shown the steam consumption per

TABLE I.—STEAM CONSUMPTION OF DIFFERENT TYPES OF ENGINES.

Type of engine.	Pounds of steam per H.-P. per hr.	Steam pressure, lbs. gauge.
High-speed simple	32	80-100
High-speed compound non-condensing	24-26	150-110
High-speed compound condensing	19-21	150-110
Corliss simple non-condensing	26	80-100
Corliss simple condensing	21	80-100
Corliss compound non-condensing	20-22	150-110
Corliss compound condensing	14-15	150-125
Triple-expansion condensing	13	150-

horse-power per hour which might be expected from various types of engines with different steam pressures. The figures given are believed to be fairly accurate in a relative sense and are supposed to represent about what would be obtained with each engine running at its most economical load with valves and pistons in good but not the best condition. In the figures given for condensing engines, the steam used by the air pump is included. To arrive at the actual cost of power, the fuel cost and the cost of repairs, interest, depreciation, etc., should be taken into account. It can be safely assumed that with good anthracite or semi-bituminous coal a boiler will evaporate 8 pounds of water per pound of coal; hence the annual fuel cost of an engine will be:

Annual fuel cost $= P \times H.P. \times h \times c \div (8 \times 2{,}240)$,

in which,

$P =$ the pounds of steam used by the engine per horse-power per hour.

$H.P. =$ the average horse-power developed.

$h =$ the number of hours during the year the power is being used.

$c =$ the cost of coal in dollars per ton of 2,240 pounds.

To the annual fuel cost thus obtained, there should be added, say 4 per cent. of the cost of the engine to cover the interest on the investment and 8 per cent. for repairs, depreciation, etc. It is assumed that when the formula is used to compare condensing engines with simple engines that the boiler feed water in the case of the condensing plant is first warmed by the exhaust steam from the engine and afterward by the steam from the auxiliaries, so that there will not be enough difference in the temperature of the feed water in the two cases to warrant taking this difference into account.

No really reliable figures as to the cost of engines can be given, as this is constantly varying due to the condition of supply and demand and the cost of materials. If an engineer is in doubt as to the type of engine needed for a special situation, bids on the types that are to be considered should be obtained, and from the prices thus obtained a decision can be made. The increased cost of the most economical engines is partly offset by the fact that not so great a capacity in boilers is required to run them, hence the cost of the boilers should usually be considered. Generally speaking, it has been found that the greater expense of economically working engines is more than offset by the gain resulting from their use. For all steam pressure over 100 pounds, with engines over 150 horse-power, the compound engine in most situations, whether it is to be operated condensing or non-condensing, will save enough in fuel cost to pay for its increased first cost and cost of attendance, repairs, etc. There are situations where this statement would not be true; for instance when the fuel cost is exceptionally low or when the demand for exhaust steam for heating or for manufacturing purposes is in excess of the steam exhausted by the engines. It would then, it is manifest, be poor policy to buy an

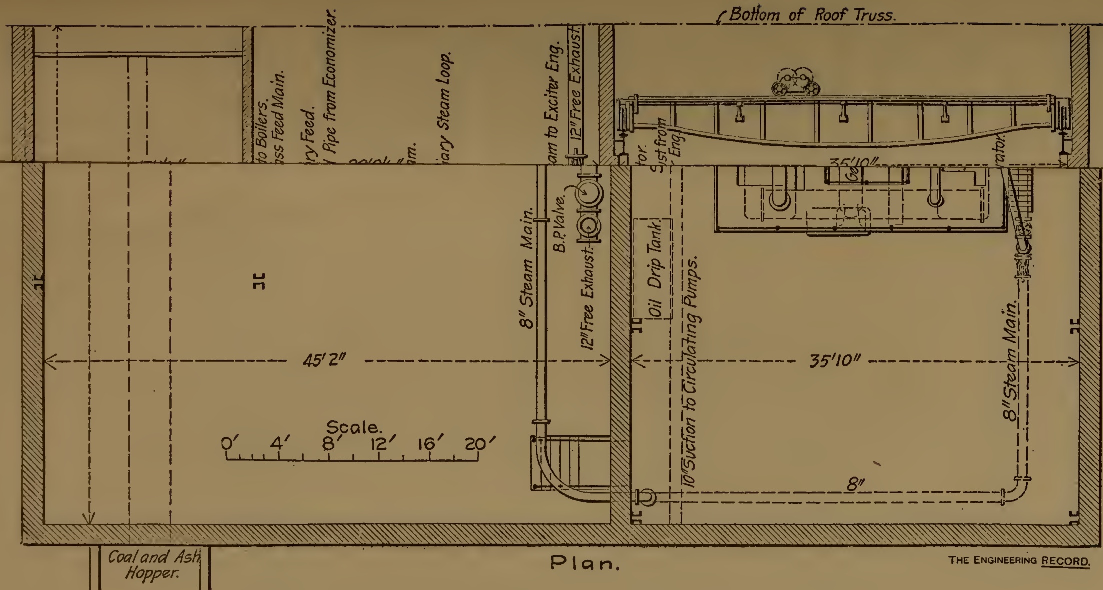


PLATE 5.—POWER STATION AT NAVY YARD, BROOKLYN, N. Y

REAR-ADMIRAL GEORGE W. MELVILLE, CHIEF ENGINEER.

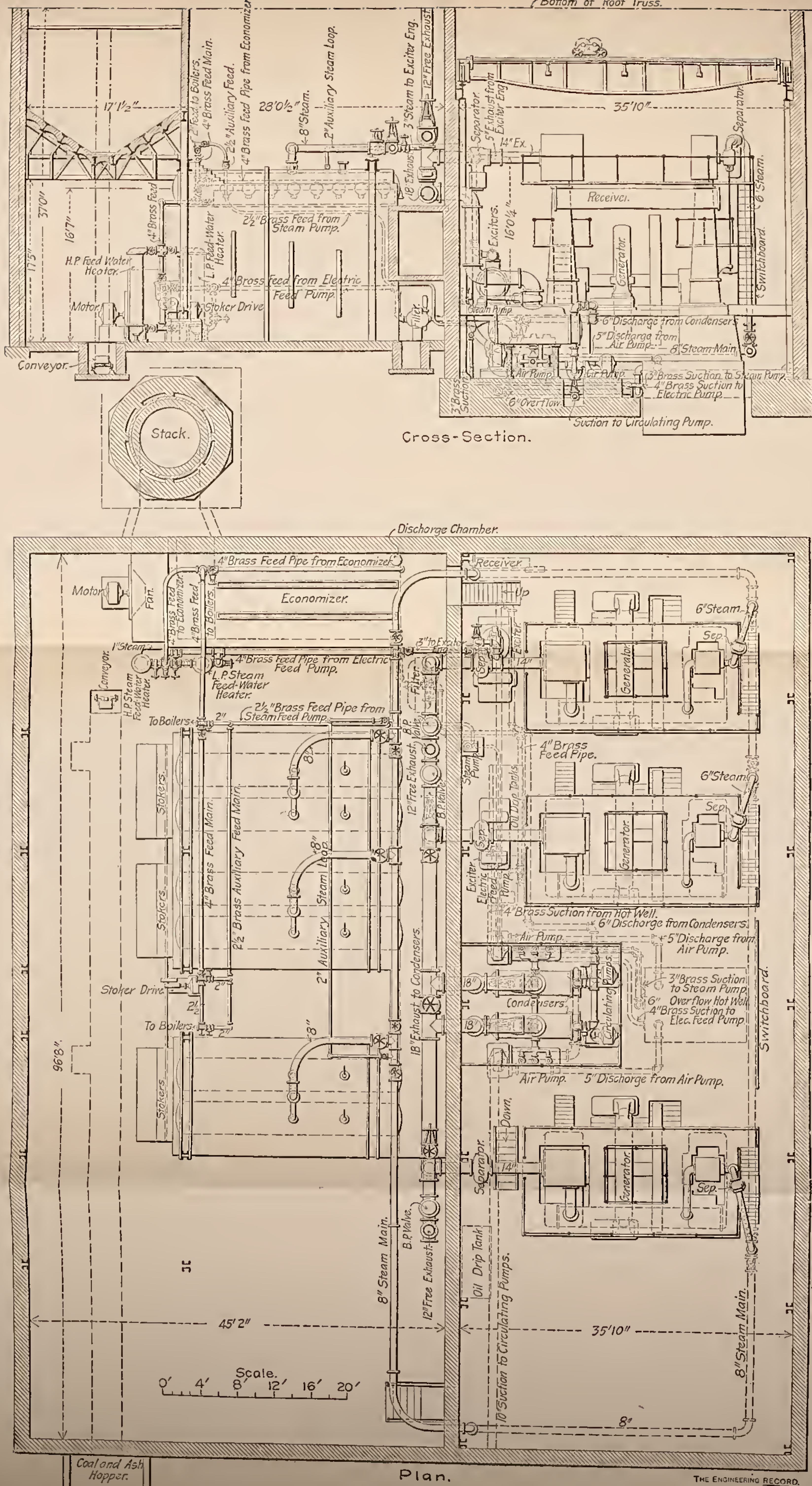


PLATE 5.—POWER STATION AT NAVY YARD, BROOKLYN, N. Y

REAR-ADMIRAL GEORGE W. MELVILLE, CHIEF ENGINEER.

expensive engine for the sake of getting an economical one, when the economical use of steam in the engine is no object. Again, if an engine is only to be run occasionally, as in the case of a relay engine to a water wheel, the engine should not be as costly a one as if it were to run more frequently. In the latter case, the interest on the increased cost of an economical engine over one using more fuel might be more than the difference in the fuel cost, owing to the short time that the engine is used. Generally speaking, triple-expansion engines for mill or electric work with steam pressures as low as 150 pounds have not shown that the saving due to their use is sufficient to pay for their increased cost over a compound engine. Engines used in the power plants of very large buildings, should, generally speaking, be of the compound type, if over 200 horse-power. They are naturally run non-condensing. Another point to be considered is the increased cost of attendants for the most economical engines over those with simpler parts and using more fuel.

Steam pressure.—The steam pressure employed in simple engines may be said, generally speaking, to vary from 80 to 120 pounds above the atmosphere and in compounds from 100 to 150 pounds, although the latter are sometimes run with lower pressures than 100 pounds. The tendency is toward the higher pressures. High-pressure steam especially in compound engines, is conducive to high economy, but it should not be forgotten that high steam pressures, and by that is meant pressures of 135 pounds and over, mean increased wear on the system, much trouble with steam piping if unusual care is not taken in constructing it, and an increased loss from leakage and condensation; but for all these objections the higher economy of engines with high steam pressures will more than compensate for the drawbacks if the plant is well designed and is placed in competent hands. In electric power stations and with mill and factory engines of large power, the pressure can well be from 135 to 150 pounds. In the latest large electric stations even higher steam pressures are used. High-pressure steam is of special importance where compound non-condensing engines are used. For plants for very large buildings employing competent engine attendants, a pressure of at least 120 pounds should be selected if this type of engine is to be used; and in large mills or large electric plants, where the steam plants will be in competent hands, the steam pressure with non-condensing compound engines should be as high as 150 pounds. Even with pressures as high as 125 pounds the steam piping should have unusual attention both in the design and erection.

Rotative speed.—The number of revolutions an engine is to run is frequently fixed by the fact that it is to be directly connected to a dynamo which has to be run at a certain speed, or by some other condition. The rotative speed is limited by the centrifugal force developed in fly wheels when in motion, by the maximum speed at which the piston should be run and, in the case of engines with a releasing valve gear, such as the Corliss, by the tendency of the valve gear to become noisy and give trouble at too high a speed. High-speed engines of 10-inch stroke with automatic cut-off controlled by a shaft governor, are usually run at about 325 revolutions per minute, and the rotary speed is decreased to about 150 revolutions in engines of 24 inches stroke. An engine may be run at much lower rotative speeds than those mentioned, but it would then, of course, require a proportionally larger cylinder to develop the same power, other conditions being equal. With releasing gear engines with ordinary air dash-pots, the highest rotative speed that should be employed seems to be about 90 revolutions per minute. Eighty revolutions is better practice. In electric work the rotative speed is sometimes increased above 90 revolutions per minute for engines directly connected to a dynamo, but unless some special valve gear is used, permitting high speeds, the life of the engine is shortened and it is apt to be noisy and require considerable attention. The rotative speed of an engine is frequently limited by the proper speed at which the piston should be run.

Piston speed.—Piston speed is usually defined as being the number of feet that the piston of an engine travels in a minute's time. It is obtained by multiplying the length of the stroke in feet by twice the number of revolutions per minute. Good practice with high-speed engines limits the piston speed of small engines with a length of stroke of 12 inches at about 550 feet per minute and with engines of the 2-feet stroke at about 600 feet. In larger engines 700 feet per minute is allowable, and in electric service this figure is sometimes exceeded in long stroke engines to obtain a high rotative speed. It is not well to go above 800 feet, however, for, when this piston speed is

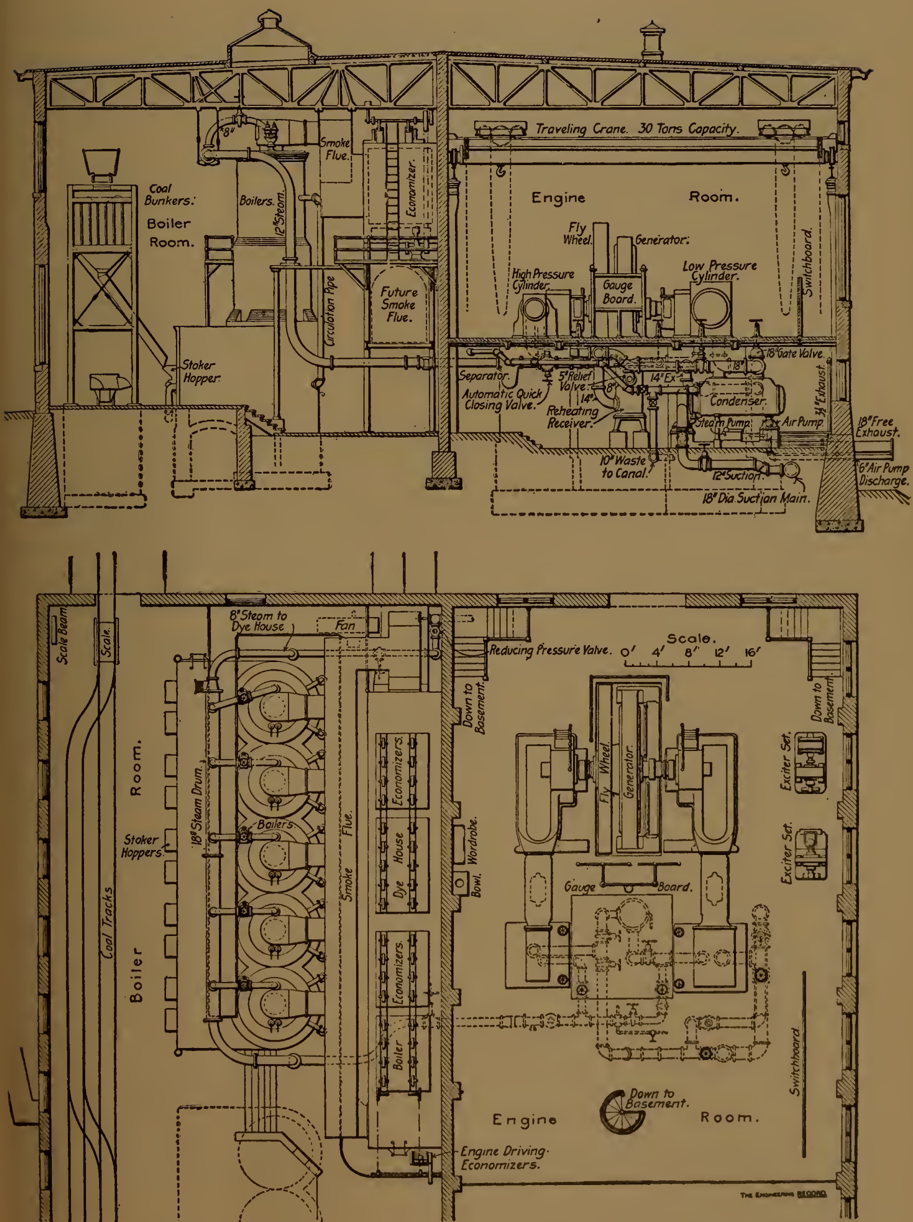


FIGURE 18.—POWER PLANT, LANCASTER MILLS, CLINTON, MASS.

LOCKWOOD, GREENE & CO., ENGINEERS.

exceeded, very large ports are necessary to admit the larger amount of steam required with high-piston speeds, and these large ports increase the clearance of the engine and make it less economical in the use of steam.

Mean effective pressure.—When steam is admitted to a cylinder during a portion of the stroke of an engine and then further supply is "cut-off," the steam, after "cut-off" occurs, begins to expand as the piston advances and consequently the pressure falls. The work done in the cylinder is proportional to the average effective pressure throughout the stroke, hence it is necessary in proportioning cylinders to know what this average or mean effective pressure should be with different steam pressures. The mean effective pressure is one of the factors in the general equation for determining the horse-power developed by an engine. Its selection is an important matter, for on it, more, perhaps, than anything else, the economy of the engine is dependent. Theoretically the greatest amount of work is obtained from steam when it is admitted to a cylinder up to such a point in the stroke that it will afterward expand to the pressure that the engine is exhausting against. Practically, it is not well to have so complete an expansion. The more complete the expansion is (that is, the number of times the volume of steam at cut-off is expanded), the less the mean forward pressure must be, hence engines operating with high economy work, up to a certain limit, with a lower mean effective pressure than a similar engine working with a poorer economy. The lower the mean effective pressure is, however, the larger the cylinder must be to accomplish the same work, hence the expansion may be so great and the mean effective pressure so small that the saving in fuel due to this complete expansion may not pay for the increased cost of the large cylinder. This is, in brief, the commercial problem involved in the selection of cylinder sizes.

Mean effective pressures for simple engines.—In Table II there are given the approximate mean effective pressures that are usually obtained with various steam pressures with Corliss and

TABLE II.—MEAN EFFECTIVE PRESSURES FOR DIFFERENT STEAM PRESSURES, IN POUNDS PER SQUARE INCH, FOR ENGINES OF THE SIMPLE TYPE.

Steam pressure, gauge	80	90	100
Corliss,* condensing	26	28	30
Corliss,* non-condensing	36	38	40
Single valve, non-condensing	42	46	50

*These data can be applied to four-valve, moderate or slow-speed engines.

other four-valve slow or medium-speed engines, condensing and non-condensing, when the minimum amount of steam per horse-power per hour is consumed. The table also shows the same data for simple single-valve engines of the high-speed type. The overload capacities of engines proportioned in accordance with the figures given in the table are described in a later paragraph.

Proportioning cylinders of single-cylinder Corliss engines.— The first thing to do in determining the size of cylinder for an engine to develop a given horse-power is to fix the steam pressure and afterward select a mean effective pressure proper for the conditions under which the engine is to work. Having the above data, the method of determining the cylinder size will be explained, in one case where the number of revolutions is fixed and in another case where it is not. It will first be supposed that the number of revolutions have been fixed.

The general equation by which the horse-power of an engine is determined is in the form:

$$\text{H. P.} = P \times l \times a \times n \div 16{,}500 \quad \text{(2)}$$

in which

P = the mean effective pressure in pounds per square inch.

l = the length of stroke in feet.

a = the mean area of the piston in square inches.

n = the number of revolutions per minute.

In equation (2) the terms H.-P. (the number of horse-power to be developed), P and n are assumed to be known, and transposing we have:

$$l \times a = \text{H.-P.} \times 16{,}500 \div (P \times n) \quad \text{(3)}$$

Substituting the value of P, n and H.-P., we can find the numerical value of $l \times a$. Both of these quantities are to be determined. Table III shows in columns 1 and 2 the diameter and length in inches of cylinders for Corliss engines for which practically all large engine builders carry patterns. The cylinder sizes given are taken from the catalogue of the Allis-Chalmers Company. Column 3 shows the number of revolutions at which it is recommended these engines should run. Column 4 shows the product of the cylinder area in square inches by the length of stroke in feet for each cylinder, and column 5 shows the quantities given in column 4 multiplied by the number of revolutions given in column 3. Going back to the calculations, the numerical value of $l \times a$, it will be remembered, has been found. We now look down column 4 until we find a number nearest to the numerical value of $1 \times a$. Opposite the number thus found are the dimensions of the cylinder that is seemingly required. One more operation remains. The length of stroke of this cylinder in feet should be multiplied by twice the number of revolutions

TABLE III.—DIMENSIONS OF CYLINDERS AND SPEEDS OF CORLISS ENGINES.

Dia. Cyl., in.	Length stroke, in.	Revs. per min.	Area cyl., sq. in. × stroke in ft. = 1 × a.	Area cyl., sq. in. × stroke, ft. × revs. = 1 × a × n.
12	30	90	282	25,447
12	36	85	339	28,840
14	36	85	461	39,252
14	42	82	538	44,177
16	36	82	603	49,461
16	42	78	703	54,889
18	36	80	763	61,070
18	42	78	890	69,467
18	48	75	1,017	76,338
20	42	75	1,099	82,467
20	48	72	1,256	90,478
20	60	75	1,570	117,810
22	42	75	1,330	99,784
22	48	72	1,520	109,477
22	60	65	1,900	123,542
24	48	70	1,809	126,669
24	60	65	2,261	147,026
26	48	70	2,123	148,660
26	60	65	2,654	172,552
28	48	68	2,463	167,484
28	60	65	3,078	200,118
30	48	68	2,827	192,265
30	60	62	3,534	219,126
30	72	55	4,241	233,263
32	48	65	3,217	209,105
32	60	62	4,021	249,317
32	72	55	4,825	265,402
34	48	65	3,631	236,058
34	60	62	4,539	281,455
34	72	55	5,447	299,613
36	48	72	4,071	293,146
36	60	62	5,089	315,539
36	72	55	6,107	335,897
38	60	60	5,670	340,233
40	48	70	5,026	351,859
40	60	62	6,283	389,558
40	72	55	7,539	414,691
40	84	50	8,796	439,824
42	48	70	5,541	387,923
42	60	62	6,927	429,486
42	72	55	8,312	457,195
44	48	70	6,082	425,748
44	60	62	7,602	471,364
44	72	55	9,123	501,774
46	60	62	8,309	515,189
46	72	55	9,971	548,427
48	60	62	9,047	560,963
48	72	55	10,857	597,154

that the engine is to run to obtain the piston speed; and if this exceeds good practice, as described in an earlier paragraph, a cylinder with a shorter stroke and larger diameter, but of approximately the same volume, can probably be found in the table, which will give the power without too great a piston speed.

If the revolutions are fixed in the first place, the problem is simpler. We then can transpose equation (2) to:

$1 \times a \times n = \text{H.-P.} \times 16{,}500 \div P$ . . . . (4)

and after obtaining the numerical value of $1 \times a \times n$, the proper diameter and stroke of cylinder and rotative speed of the engine required can be found opposite the number in column 5 of the table nearest to the numerical value of $1 \times a \times n$. If there is not a close agreement between one of the numbers in column 5 and the value of $1 \times a \times n$, the revolutions may be increased or decreased as the former is respectively greater or less than the latter.

It was explained that Table III gives the cylinder sizes of Corliss engines made by one company and that many builders carry patterns for these sizes. Most builders have additional

TABLE IV.—DIMENSIONS OF CYLINDERS AND SPEED FOR HIGH-SPEED AUTOMATIC CUT-OFF ENGINES.

Dia. cyl., in.	Length stroke, in.	Revs. per min.	Area cyl., sq. in. × stroke, ft. = 1 × a.	Area cyl., sq. in. × stroke, ft. × revs. = 1 × a × n.
9	10	325	53	17,225
10	10	325	65	21,125
11	10	325	79	25,675
11	12	300	95	28,500
12	12	300	113	33,900
13	12	300	132	39,600
14	12	300	153	45,900
14	14	275	178	48,950
15	14	275	205	56,375
16	16	250	268	67,000
17	16	250	302	75,500
18	16	250	338	84,500
18	18	225	381	85,725
20	18	225	471	105,975
20	20	200	523	104,600

patterns varying more or less in size from those given, and it may be for a particular case a builder might furnish a cylinder with slightly shorter stroke, and correspondingly greater area of piston, so that the cylinder volume would be the same. Such a cylinder would give the same power and, because of the shorter stroke, the engine would be less expensive to build, the frame, guides and all reciprocating parts being shorter. For this reason it is not well for the engineer to be too rigid in fixing the cylinder dimensions. Cylinders of different engines to do the same work should have the same volume; but the stroke should not be shortened too much.

Proportioning cylinders for single-cylinder high-speed engines.—Table IV contains the sizes of cylinders of a line of high-speed automatic engines. Although no one builder carries patterns for them all, most builders have patterns for cylinders of approximately the same volume as those given in the table. The engineer can determine the size cylinder required in the manner described for Corliss engines, using the mean effective pressure and rotative speed proper for this type of engine. A specification for this type of engine can name the length and diameter of cylinder wanted, but it should also state that bids upon engines of approximately the same cylinder volume will be considered. It is, of course, unfair to handicap a builder not possessing patterns of exactly the dimensions desired by demanding that he make a pattern to conform exactly to the engineer's specification.

Proportioning cylinders for medium-speed engines.—It is impossible to give a table that is representative of the cylinder sizes of medium-speed engines, as those four-valve engines which run at higher rotative speed than Corliss engines and at lower rotative speed than the so-called high-speed engines are called. The reason for this is that no two builders have patterns for the same sizes of cylinders. It is, perhaps, just as well to select the engine size from the catalogue of one builder, using the same piston speed and mean effective pressure as for Corliss engines, and invite bids upon an engine with a cylinder or cylinders of equivalent volume. The shorter stroke should belong to the cheaper engine, other things being equal, for reasons explained above. An engine with too short a stroke in proportion to its diameter is not as economical usually, on account of greater clearances.

Proportioning cylinders for compound engines.—The method of determining the size of cylinders of compound engines is somewhat complicated by the fact that steam has to be expanded through two cylinders. Theoretically, the size of the high-pressure cylinder for a given number of expansions in the engine has absolutely nothing to do with the capacity of the engine. In fact, a simple engine, with its single cylinder the same size as the low-pressure cylinder of a compound engine, would, theoretically, develop the same power as the compound if the cut-off in the single cylinder were adjusted so as to give the same number of expansions as exist in the compound engine. In practice, however, the large number of expansions common in compound engines would not do in a simple engine, because of the excessive cylinder- condensation that would occur. Dividing the expansion between two cylinders, as is done in the compound engine, reduces the condensation, and that is why compound engines are used. For the purpose of determining the cylinder sizes of a compound engine it can be assumed that all of the work is done in the low-pressure cylinder and then determine its size by the formulas given for the simple engines, assuming a mean effective pressure that is proper for a compound engine. After the size of the low-pressure cylinder is determined, the high-pressure cylinder dimensions can be found by selecting one whose volume is in the same ratio to the volume of the low-pressure cylinder, that practice has found to be proper.

Mean effective pressures for compound engines.—The mean effective pressures in compound engines are different from those in simple engines for the reason that the steam pressures and ratios of expansion are higher. In Table V there are given values that can be assumed for the mean effective pressures for

TABLE V.—MEAN EFFECTIVE PRESSURES FOR DIFFERENT STEAM PRESSURES, IN POUNDS PER SQUARE INCH, FOR COMPOUND ENGINES.

Steam pressure, gauge	100	125	150
Corliss,* condensing	18	20	22
Corliss,* non-condensing	29	31	33
Single valve, high-speed, condensing	22	24	26
Single valve, high-speed, non-condensing	32	34	36

*These data can be applied to four-valve, moderate or slow-speed engines.

substitution in equations (3) and (4) to find the size of the low-pressure cylinder of compound engines of various types. The mean effective pressures given are equivalent to the mean effective pressures actually found in the low-pressure cylinder added to the mean effective pressure in the high-pressure cylinder multiplied by the ratio of high-pressure to the low-pressure piston areas. The mean effective pressures given for compound engines are believed to be such as will ensure the lowest steam consumption per horse-power. Over load capacities are discussed later.

The data given in the tables have been gathered mainly from an examination of a number of tests made by engineers known to obtain trustworthy results, where the conditions have been such as to obtain the highest efficiency. In all cases the mean effective pressure is based upon a back pressure of 16 pounds absolute in the case of non-condensing engines, and a vacuum equivalent to 26 inches of mercury in the case of condensing engines. After the mean effective pressure is decided on, formula (3) should be used if the revolutions are not fixed, and formula (4) if they are determined upon, just as was done with simple engines, in calculating the area and length of stroke of the low-pressure cylinder of a compound engine.

After the size of the low-pressure cylinder is determined, the high pressure can be found by selecting one whose area is in the same ratio to the area of the low-pressure cylinder, as practice has found to be proper, as has been said. There is a difference of opinion among engineers as to the proper ratio of cylinder volumes with compound engines. Some engineers proportion the cylinders so that the amount of work done in each will be the same. This results, particularly with high pressures, in a considerably greater range of temperature in the high-pressure than in the low-pressure cylinder; and by many this is held to be bad practice, for it is well known that too great a temperature range results in excessive cylinder condensation, and it is to overcome this enemy to high economy in the steam engine that compounding is resorted to. The general practice with Corliss condensing engines, running with constant loads, has been to use a ratio of 1 : 3, 1 : 3½ and 1 : 4 with steam pressures of 125, 135 and 150 pounds respectively. Recently the tendency of a few engineers and builders has been toward a comparatively larger low-pressure cylinder than is given. Mr. George I. Rockwood, M. Am. Soc. M. E., has persistently advocated a cylinder ratio as high as 1 to 7 for a steam pressure of 150 pounds, and the most economical compound engine ever tested, that at the Grosvenordale Mills, Grosvenordale, Conn., which gave a steam consumption under 12 pounds per horse-power per hour, had a cylinder ratio of about that proportion. A number of other engines with similar cylinder ratios have shown almost as good an efficiency. With high-speed automatic single-valve engines the cylinder ratio varies from about 1 : 2½ with 100 pounds pressure to about 1 : 3 with a pressure of 150 pounds. An advantage in having the high-pressure cylinder fairly large in comparison with the low-pressure is the greater capacity for overloads that the engine will then have.

Engines with variable loads and overload capacities.—The mean effective pressures that are given for both simple and compound engines assume that the engines are to run with a steady load and that it is desirable that they should work with the highest economy. In other words, the mean effective pressures given are such as will secure the lowest steam consumption. The matter of maximum economy is often of less importance in an engine than its maximum capacity, to take care of overloads for short periods. All engines are able only to work at maximum economy at a certain load. As the load diminishes or increases, the steam consumed per horse-power developed becomes greater. With a variable load an engine has to be large enough to supply the maximum power required, but when the load varies it is, generally speaking, better to proportion the engine so that it will be operating at the highest economy when working against a load

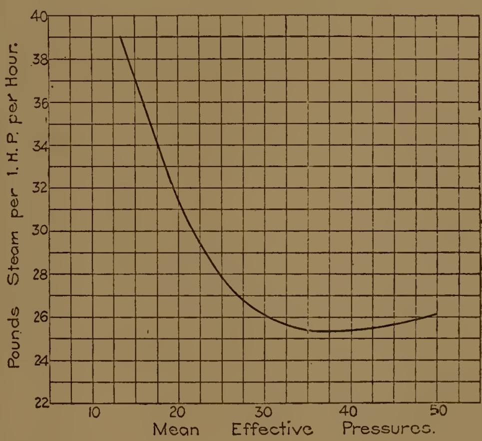


FIGURE 19.—ECONOMY CURVE SIMPLE CORLISS ENGINE.

that is something less than the maximum that it will be called upon to supply, for the reason that it will be working most of the time partly loaded, and it should be proportioned to meet this average load with the consumption of as little steam as possible. An engine should be of such a size that its most economical load, that is, the load under which it will operate with the least steam per horse-power, is equal to the average power required, provided the maximum power that it can develop is not less than the maximum power that will be required. Hence, if an engine is to furnish power where the maximum demand is 400 horse-power and the average demand 300 horse-power, the engine required would be one whose most economical load was 300 horse-power, and 300 horse-power would be the quantity substituted in the general equations (3) or (4) for determining the cylinder sizes.

For the purpose of showing the variation in steam consumption due to a variation in load, the results of a number of tests, believed to be reliable, made upon engines of various types, are shown in the accompanying diagrams. Figure 19 shows the result of a test by Mr. George H. Barrus upon a 16 x 42-inch simple non-condensing Harris-Corliss engine running 85 revolutions per minute with a steam pressure of 100 pounds. It will be noticed that the most economical rating on the basis of steam consumed per indicated horse-power per hour was when the mean effective pressure was about 40 pounds, increasing slightly at 47 pounds, which was probably very near the maximum load of the engine. If, therefore, an engine is rated in accordance with the mean effective pressures given in Table II it is probable that the simple non-condensing Corliss engines, with a single eccentric driving both the steam inlet and the exhaust valves, would stand an overload of at least 25 per cent. and simple condensing Corliss engines over 50 per cent.

A number of tests made upon simple high-speed engines, with single valves, running non-condensing, are reproduced in Figure 20, and the curves there shown representing the steam consumption with the variation in the mean effective pressure are numbered to correspond with the data in Table VI. Tests numbered 1 and 2 were made by Mr. J. M. Whitham and were printed in The Engineering Record of July 9, 1898. Test number 4 was

TABLE VI.—DATA OF SIMPLE NON-CONDENSING HIGH-SPEED ENGINE TESTS SHOWN IN FIGURE 20.

No. test.	Diam. cylinder.	Stroke cylinder.	Revolutions per min.	Steam pressure.	Make of engine.
1	13	12	280	100	Ames
2	8	10	350	100	Ames
3	13	12	250	95	Ames
4	12	14	245	80	McEwen
5	13	12	280	95	Ames
6	13	12	250	95	Ames
7	13	12	250	95	Ames
8	17	16	225	100	Ames
9	17	16	270	100	Ames

made by Professor R. C. Carpenter and was taken from Paper DXXIII, Transactions American Society of Mechanical Engineers. The remaining tests were made by Mr. E. J. Armstrong and were taken with his permission from a paper read by him before the Engine Builders' Association. Test number 2 was made upon a very small engine, but the test was not carried far enough to give exact data as to the proper load for it. It is interesting as showing the influence of size upon the steam consumption. Curves 8 and 9 were obtained from an engine of the same make only considerably larger and show a better economy, due, doubtless, to the larger size of engine. Curve number 4 shows the steam consumption of an engine with a boiler pressure of only 80 pounds above the atmosphere. The most economical

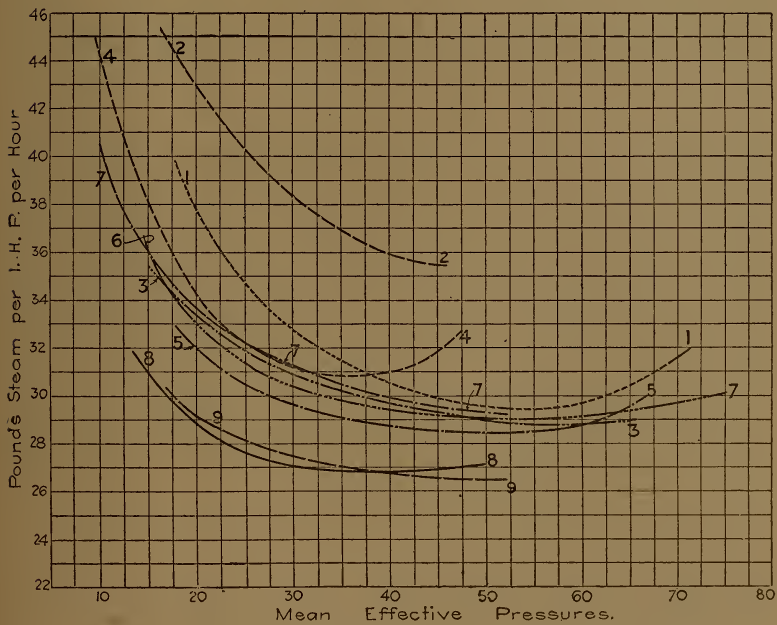


FIGURE 20.—ECONOMY CURVE SINGLE VALVE SIMPLE ENGINES.

mean effective pressure in the latter case appears to be a little less than half the boiler pressure. Most of the remaining curves show the steam consumption of engines operating under pressures of from 95 to 100 pounds. The most economical result with a steam pressure of 95 to 100 pounds seems to be with a mean effective pressure of from 45 to 55 pounds. Generally speaking, therefore, with the mean effective pressures recommended in Table II, the curves, with the exception of those numbered 2, 8 and 9, which were not carried out to the maximum capacity of the engine, seem to indicate that overloads of about $33^1/_3$ per cent. could be met without a reduction in the speed of the engine. Data upon single-valve high-speed engines running condensing were not obtainable, as they are not often used.

Curves 1 and 2, Figure 21, shows the variation in economy of a 9 and 16-inch by 15-inch McEwen compound single-valve engine operating under a steam pressure of 112 pounds, a vacuum of 22 inches and a rotative speed of 265 revolutions per minute. One test was made with the cylinders steam jacketed and one with the jackets shut off. The tests were made by Prof. R. C. Carpenter and have been taken from Volume DXXIII, Transactions of the American Society of Mechanical Engineers. It is seldom that engines of this type are provided with steam jackets, as the general impression seems to be that the saving does not warrant the expense. It will be noticed from the unjacketed test that the most economical rating, with a steam pressure of about 110 pounds, seems to be with a mean effective pressure, referred to the low-pressure cylinder, of 26 pounds. Upon this basis it would seem that a compound condensing high-speed engine could easily run with an overload of over 50 per cent., particularly if the automatic cut-off be applied to both cylinders.

Curve number 3 in Figure 21 shows the variation in economy in a 12 and 20-inch by 13-inch non-condensing engine, made by the Ball Engine Company and tested by Messrs. George H. Barrus and W. S. Monroe. The average boiler pressure was about 166 pounds, with practically no back pressure except that of the atmosphere. The rotative speed was about 175 revolutions per minute. The most economical mean effective pressure seems to be about 40 pounds compared with about 26 pounds for the unjacketed test of the McEwen condensing engine. While there is considerable difference in the steam pressure used by the two engines, it will be noticed that, if the loads were the same, the non-condensing engine would require a smaller cylinder than the condensing engine, provided both were operating with the lowest possible steam consumption per unit of power developed. Had the Ball engine been rated on the basis of a mean effective pressure of 40 pounds referred to the low-pressure cylinder, the overload capacity would have been about $33^1/_3$ per cent. The tests were not carried sufficiently far, however, to determine if the engine could be operated at greater load than that shown by the chart.

The compound non-condensing Corliss engine rated upon the mean effective pressures given in Table V can easily stand 25 per cent. overload if there is but a single eccentric driving the valve gear, and more than that if separate eccentrics are used to operate the steam and exhaust valves, for with two eccentrics a much later cut off in the cylinder can be secured than with one. With compound condensing Corliss engines rated upon the mean effective pressures given in Table V, overloads of at least 50 per cent. with one eccentric and probably 75 per cent. with two eccentrics could be withstood without very greatly affecting the engine's speed.

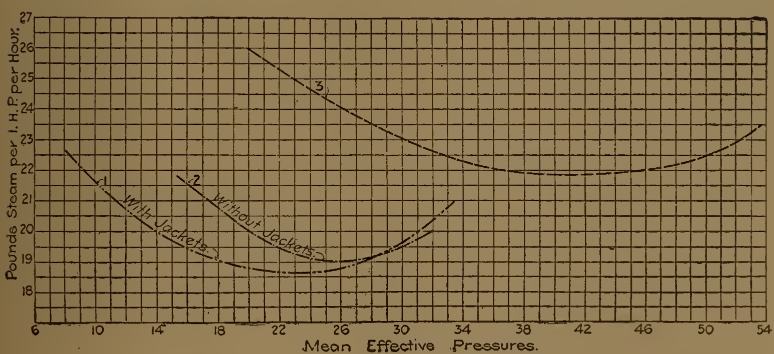


FIGURE 21.—ECONOMY CURVE SINGLE VALVE COMPOUND ENGINES.

Superheated steam, steam jackets and reheaters.—Superheated steam, steam jackets enclosing the walls of steam cylinders and reheating receivers placed between the cylinders of multiple expansion engines are used to reduce cylinder condensation. The almost universal use of superheated steam on the continent of Europe in steam plants of recent construction and the remarkably high economy secured, gains from 10 to 20 per cent. over the use of saturated steam having been reported, has drawn the attention of American engineers to this practice so that the use of superheated steam is increasing considerably in the United States. The practical objection to its use, heretofore, has been in the deterioration of the superheating devices used and in the difficulty of lubricating engines where the steam is highly superheated. Recently three or four makes of superheaters have been introduced in the United States, and their extensive use abroad has demonstrated their durability and efficiency. The use of poppet valves in steam engines has overcome the difficulty of lubrication, although for temperatures under 500 degrees Fahrenheit these are said to be unnecessary. Superheaters are of two types, those that are placed in the path of the gases in the boilers, or in the flue leading from the boiler to the chimney, and those that are heated by an independent furnace.

Steam jackets are seldom used upon any engines but those of the slow and medium rotative speed types, and their use becomes of less value as the speed is increased. Except for slow moving pumping engines their value is still a matter of doubt, although some engine builders provide them. Reheating receivers are in more common use, but their value is also a matter of dispute. The heating is done by a coil filled with steam of a higher pressure than that of the steam passing through the reheater. Compound and triple expansion engines of the slow and medium speed type are usually provided with them.

Steam turbines.—These devices for generating power by means of the impact of a current of steam upon the turbine buckets have, some time since, gotten beyond the experimental stage in the United States and they have been in commercial use in Europe for a considerable period. They possess a number of points of superiority over reciprocating engines and among these is the small variation in the steam consumption per unit of power developed that occurs with a change in the load, also, the fact that the bearings are the only rubbing surfaces hence the efficiency of the turbine is not impaired by use to the extent that it is with reciprocating engines. The economy of the steam turbine compares favorably with all but the most economical engines and the turbine is particularly adapted to use superheated steam, for with high superheating there can be none of the difficulty as regards lubrication that there is with some reciprocating engines. Dr. R. H. Thurston has called the steam turbine the engine of "maximum simplicity and highest thermal efficiency."

Chapter V.—Specifications for Steam Engines.

There is occasionally a tendency on the part of some to ridicule elaborate specifications for steam engines, yet as a specification is a description of what one party to a contract agrees to furnish to another, it should be sufficiently complete to define exactly what is to be supplied. A complete specification is unnecessary, perhaps, where an owner engages a builder, without competition, to construct and install an engine suited for existing conditions and agrees to pay for whatever the builder elects to supply. In competitive bidding, the builder is not legally bound to supply anything more than that which the specification calls for, and all bidders are not apt to figure on doing more than that in preparing their bids, knowing full well that other bidders will not do so. Hence, if it is found, after a contract is made, that there are some desirable details which have been overlooked by the engineer in his specification, they must be purchased and paid for as extras, at, naturally, a higher figure than what they would cost if specified in the beginning.

The amount of detail necessary in a specification naturally varies with the size of the engine to be purchased, and the extent to which the design departs from the standard types for the service. It is the author's intention to call attention to a number of details which must be considered in buying a large engine, although reference to them may be omitted in a specification for less important work. Many engineers believe in specifying the cylinder dimensions, at least in a general way, so that all bidders may bid upon engines of equal capacity. Methods of determining these dimensions for simple and compound engines of various types were given in the preceding chapter. While they may be fixed by the engineer, some latitude should be given bidders in order that standard patterns may be utilized.

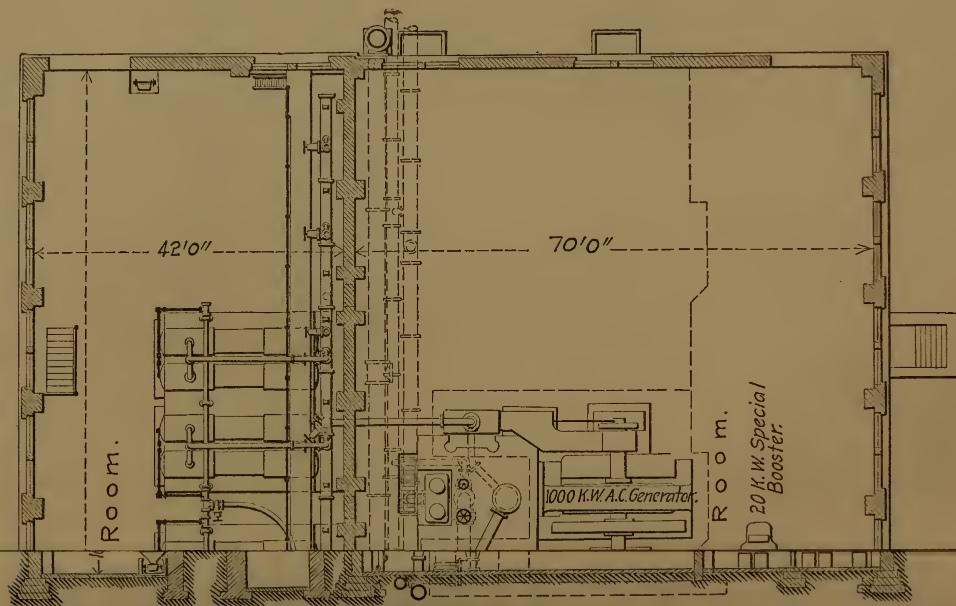
A specification usually begins by stating the type of engine or engines wanted, whether it is to be of the simple, compound, or triple-expansion type, whether it is to be run condensing or non-condensing, and where it is to be located. Even though the engineer fixes those dimensions that determine the capacity of the engine, the specification should state the load at which the engine is to operate with the highest economy, the maximum load that it is to be called upon to operate, and the kind of service to which it is to be subjected. If guarantees as to capacity and efficiency are to be required from the builder, these data are, of course, essential. Even if they are not, and the engineer assumes the responsibility for the fulfillment, by the engine, of the requirements imposed by existing conditions, as he does when he fixes the cylinder dimensions, steam pressure and rotative speed, the engine builder should be informed as to what will be required of the engine, as he is very apt to make calculations that will serve as a check on those of the engineer.

Sometimes a specification states that the price of the engine is to include its delivery and erection on a foundation supplied by the owner, and placing it in proper running condition. Again, however, the engine is sold free on board cars at the railway point nearest the locality where the engine is to be used. In the latter case the builder usually supplies a man to take charge of the erecting of the engine, which is done by labor employed by the owner under direction of the builder's erector.

It is assumed that the engineer has determined the cylinder dimensions, the rotative speed, steam pressure, and whether the engine is to be run condensing or non-condensing; hence the specification should give: Diameter and stroke of cylinders; number of revolutions per minute; horse-power to be developed when working at highest efficiency; horse-power at maximum load; steam pressure; back pressure if run non-condensing, or vacuum if run condensing.

As before stated, an engineer should not be too rigid in regard to cylinder dimensions, and it is well, particularly when purchasing high-speed or medium-speed engines, to state in the specifications that engines with cylinder volumes equivalent to those specified will be considered. There are objections to an engine with too short a stroke in proportion to the area of piston; these were stated in the preceding chapter. If the engine is to be run condensing and the builder is to supply the condenser, provision should be made for it.

The engineer should describe in a general way the kind of



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THE ENGINEERING RECORD.

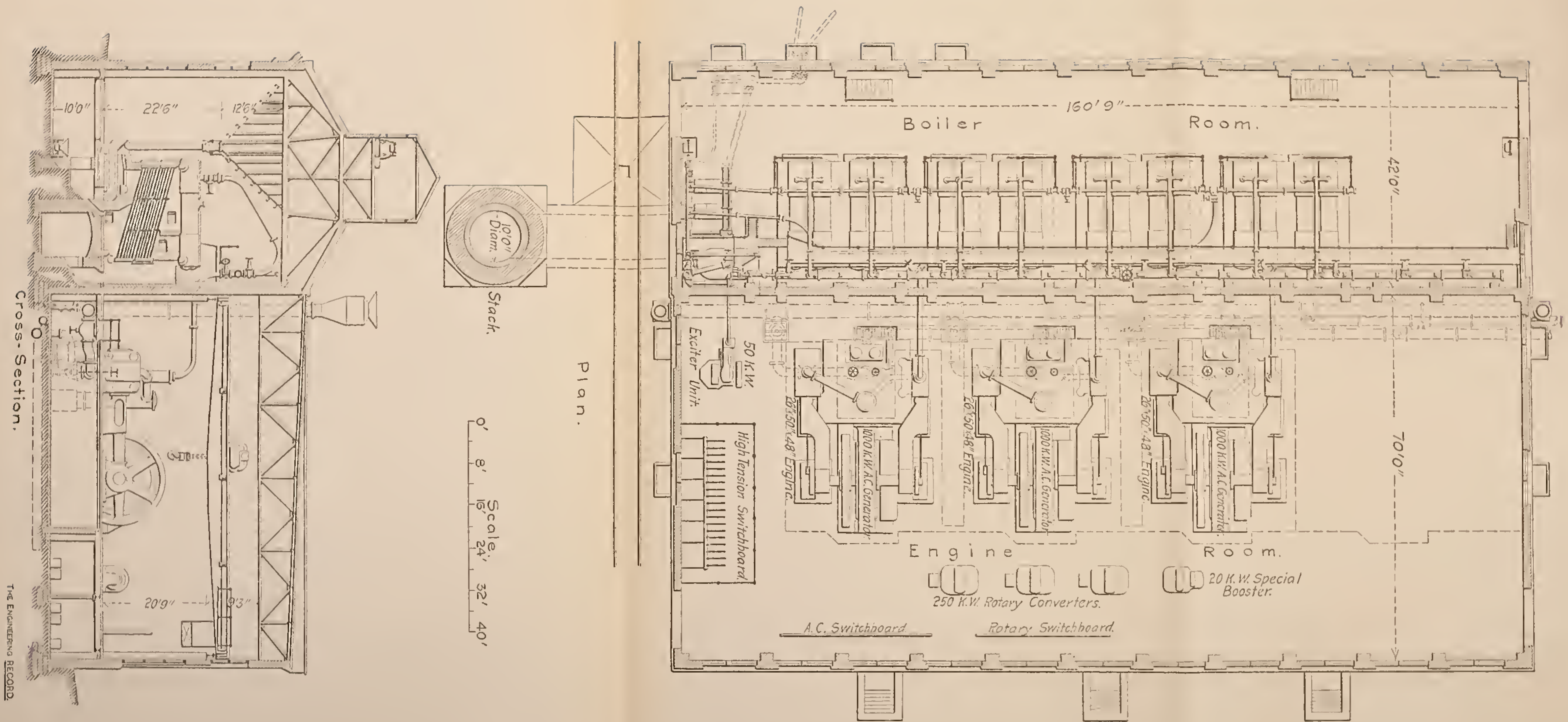


Plate 6.—Power Station, Union Traction Company, Anderson, Ind.

Sargent & Lundy, Engineers.

valve gear that is wanted. If the engine is to be of the compound Corliss type, the automatic cut-off might be applied to one or both cylinders. It is better to have it act on both cylinders if the load is variable, for the reason that a more equal distribution of the load between the cylinders will occur at light loads than there will if the cut-off is applied only to the high-pressure cylinder. Again, for Corliss engines, a separate wrist plate for the steam and exhaust valves, each driven by a separate eccentric, may be specified so that the engine may work with a later cut-off and consequently greater overload than is possible if both valves are driven from one eccentric. The cut-off may be applied to both cylinders in other types of compound engines.

If the cylinders are to be steam jacketed in barrels or heads, or both, it should be so stated. Consulting engineers seldom pay much attention to this question, as the value of steam jackets is still a disputed point. Generally, engines for electric service and factory work are not steam-jacketed, particularly if the piston speed is over 600 feet per minute. It is the custom in multi-cylinder engines of the medium-speed and slow-speed types, to place reheating receivers in the steam pipes between the cylinders, in which the steam is heated in transit from one cylinder to the other by live steam admitted to a coil of pipe. The value of these reheaters is also a matter of dispute. However, if they are wanted, they should be specified, also the manner in which the condensation that occurs in them is to be disposed of. The steam passing through the heater from one cylinder to the other is usually at a much lower pressure than the steam in the coils, hence the condensation must be drawn off by separate traps. The connection of these traps with the reheater and with a receptacle into which they can discharge, should be made either by the engine builder or the steam piping contractor.

If the engine is to be of the Corliss type, the kind of bed that is wanted should be mentioned. Two forms are made by most builders, one, known as the heavy-duty type, is adapted for heavy work and high pressures, and the other, the girder frame, is lighter and is of older design. The heavy-duty design is almost invariably used for pressures over 125 pounds and to an increasing extent in lower pressures; being much stronger, it is a little more expensive. If bids are called for a high rotative-speed automatic engine, the specifications should call for an iron sub-base with the engine, for all builders do not furnish them except at extra expense. Occasionally the iron sub-base is not used and the bed of the engine is bolted to a brick foundation which is built up above the floor the height of the sub-base.

There are a number of points concerning the design of the engine, chiefly relating to dimensions of wearing surfaces and the strength of parts, that engineers rarely fix. It is the custom of some, though, to ask each builder what he intends to furnish in respect to certain details, usually about as follows: Diameter and length of bearings, diameter and length of cross-head pins, diameter and length of crank pins, diameter of shaft in the body, dimensions of cross-head shoes, length of connecting rod, diameter and face of fly wheel (both may be fixed by the engineer), weight of fly wheel, weight of engine bed, weight of entire engine.

When these data are received from each builder the engineer can tabulate them and compare the proportions of the engines offered. If any part of an engine differs from that of others enough to warrant such a course, the engineer can ask the reason for this deviation, and if it is not a good one the builder can be requested to modify his design, or the bid can be rejected. The engine builder should be asked to furnish a blue print or drawing of some kind showing a plan and elevation and the principal dimensions of the engine he proposes to furnish.

Regarding details of construction of engines, most American engines are built after standard designs adopted by each builder and not much attention is usually paid in specifications to these details when builders of established reputation are bidding. Most builders furnish catalogues containing illustrations of the details of various parts of their engines, such as the valves, valve gear, governor, piston, main bearings, crosshead, etc. If an engineer is obtaining bids on an engine the details of which are unknown to him such illustrations can be asked for, When about to execute a contract with a builder, it can be stated in the contract that the details of construction are to correspond in design with drawings or blue prints or catalogue sketches furnished by the builder.

There are a great many minor details of an engine, such as oil cups, sight-feed lubricators, throttle valve, relief valves on the cylinders to prevent them from being damaged by water, cylinder lagging, steam and vacuum gauges, gauge board, provision for attaching the indicators to the cylinders, etc., that have to be looked after, and many builders furnish a printed specification which refers in more or less detail to some of these matters. This specification can be asked for and made part of the contract, but before so doing care should be taken that it does not conflict with the specification issued by the engineer.

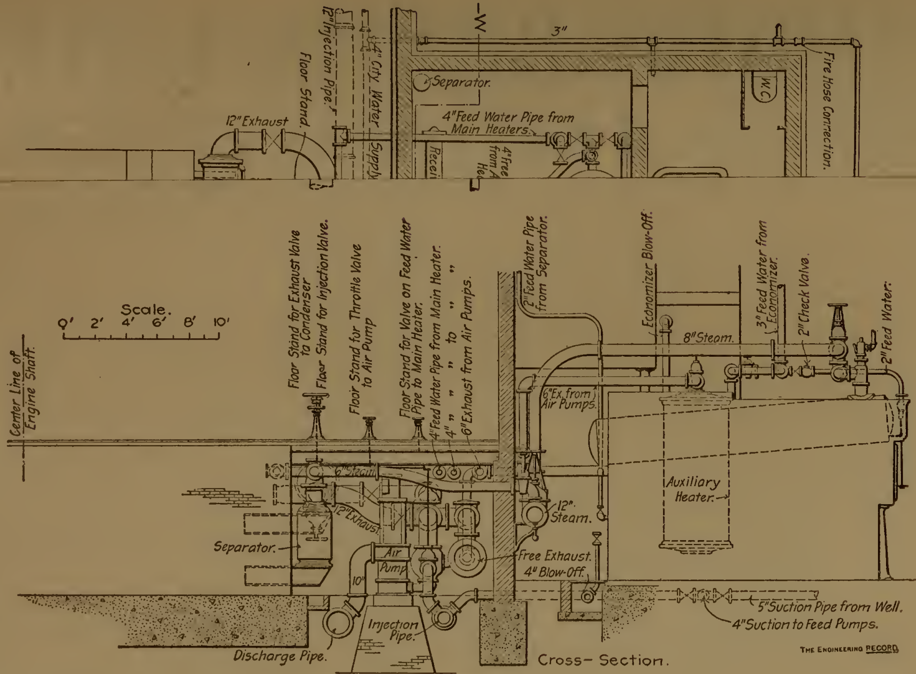
The character of materials used in engine construction is not usually given much attention in an engineer's specification, but recently the materials of which shafts of large engines are made is receiving more thought, and the kind of metal used, and its treatment, is frequently specified. The most advanced practice is found, probably, in the fluid-compressed hollow-forged treatment, perfected by the Bethlehem Steel Company. The molten steel is poured into a cylindrical mold and subjected to an enormous pressure while cooling, so as to diminish the blow holes that frequently occur in the ordinary method of casting. The ingot is cooled slowly and the impurities usually collect at its axis; after it has cooled, an axial hole is bored through it to remove these impurities. The ingot is then reheated and a mandrel is slipped through this hole, after which it is forged under a slow-moving hydraulic press, the action of which penetrates much more deeply into the metal than is the case with the blow of a steam hammer. After forging, the ingot is annealed to remove internal strains and improve the structure of the metal, which is then machined to size. Sometimes the shaft is oil-tempered after annealing. Open-hearth and nickel steels are used, the small percentage of nickel added tending to raise the physical properties of the mixture. A large number of recent important engines for mill and electric power house service have been equipped with shafts made in this way.

The time of delivery of an engine should be given in a specification, and with large engines the builder is sometimes required to furnish a man to operate the engine for a short period after it is erected. If the engine room is to be fitted with a traveling crane operated by electricity or by hand, it is well to notify the engine bidders to that effect, as the use of a crane reduces the cost of erecting engines and the owner should have the benefit of this.

In regard to steam piping, the engine builder usually supplies and connects the piping between the cylinders in a compound or triple-expansion engine. In large compound engines, particularly of the cross-compound type, that is with the cylinders placed side by side and driving separate cranks, it is frequently the custom to place a valve in the pipe carrying the exhaust steam from the high-pressure to the low-pressure cylinder, and to run a branch from the high-pressure steam pipe to this exhaust pipe connecting with it at a point between the low-pressure cylinder and the valve mentioned. With this arrangement the high-pressure cylinder can be cut out, and the engine driven by the low-pressure side alone. The pipe conveying high-pressure steam to the low-pressure cylinder is usually provided with a pressure reducing valve as well as an ordinary stop valve. By connecting the exhaust from the high-pressure cylinder at a point between that cylinder and the stop valve between the two cylinders, with the main exhaust pipe, and by placing a valve in the exhaust from the low-pressure cylinder, the low-pressure cylinder can be entirely cut out and the engine driven by the high-pressure cylinder. The advantage of such an arrangement in the event of the break-down of one side of the engine is obvious. An arrangement of this kind is shown very nicely in Figure 18.

For electric work, where revolving parts of dynamos are mounted on the engine shaft, as is the custom in directly connected work, the specifications should require the engine builder to cut a key-way in the shaft. The dynamo builder usually supplies a key for keying the armature of the dynamos on the engine shaft. The dynamo and engine are sometimes shipped by their respective builders to the site of the power plant and the engine or the dynamo builder fits the armature to the shaft. In other instances the armature of the dynamo is shipped to the engine builder and is put on the shaft by the latter. The specification should state what each builder is to do in this respect.

A specification often asks what steam consumption the builder of an engine will guarantee. If a guarantee as to steam consumption is to be made, it should be of the following general form: "The engine is guaranteed to consume not more than —— pounds of steam per indicated horse-power per hour when developing —— horse-power when running at a speed of —— revolutions per minute with a steam pressure of —— pounds above the atmosphere, and with a back pressure of —— pounds



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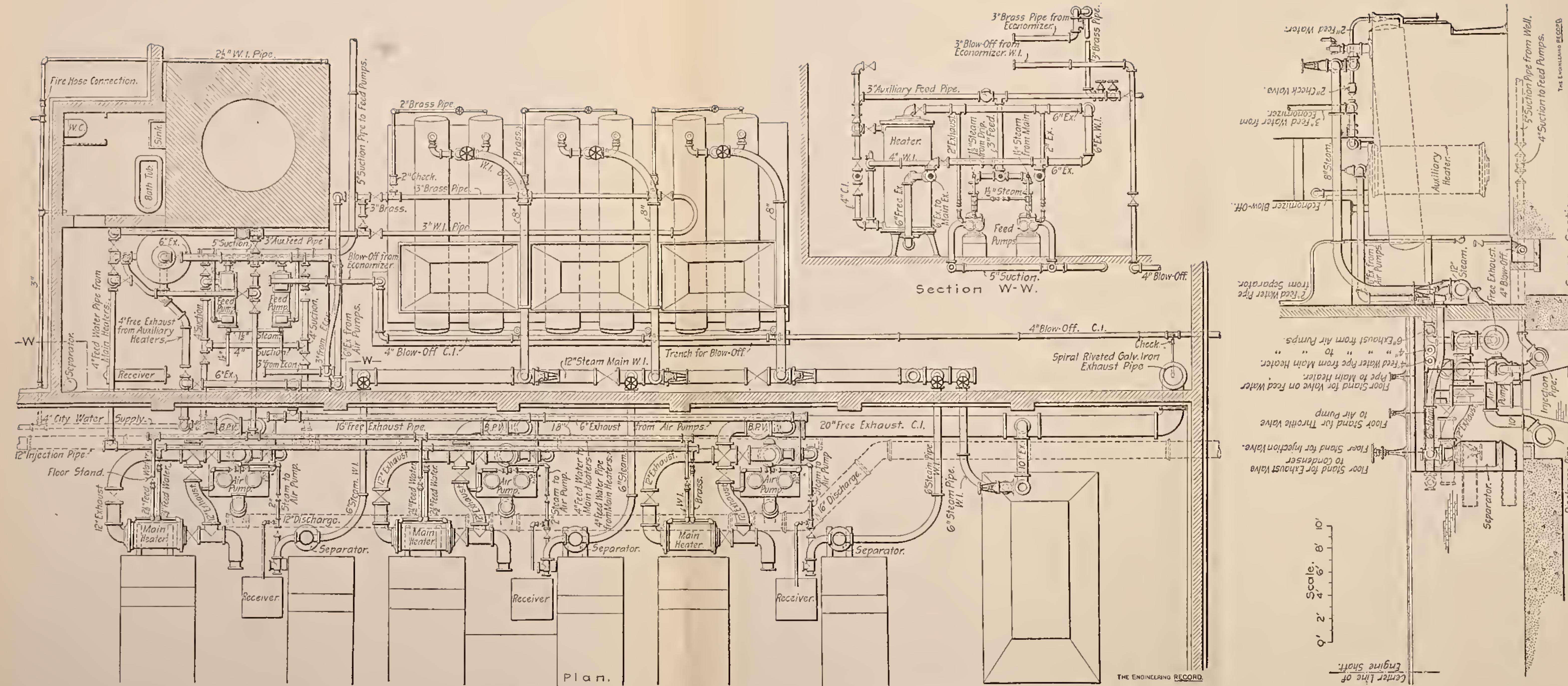


Plate 7.—Piping in Station, Woburn Heat, Light and Power Company, Woburn, Mass.

SHEAFF & JAASTED, ENGINEERS.

above the atmosphere." The specification should state that in case of dispute the steam pressure is to be the average pressure as obtained by a throttled steam gauge connected to the steam pipe close to the throttle valve, or by means of a steam pipe diagram obtained by attaching an indicator to the location named. If the engine is to run condensing instead of with a back pressure above the atmosphere the vacuum to be carried should be stated. The usual vacuum is 26 inches of mercury. If the engine is provided with steam jackets and reheating receivers, it should be stated that the steam used by them is to be included in the consumption of the engine. If the engine is to run condensing, the steam used by the air-pump should be included, provided the engine builder supplies and is thereby responsible for the efficiency of the condenser. Sometimes, when condensers are not supplied by the engine builder, the steam used by the air-pump is not considered as being part of the steam used by the engine. If the air-pump steam is or is not to be considered as part of the engine consumption, a statement to that effect should appear in the guarantee. It is usually the custom to measure the degree of vacuum obtained by a gauge attached to the exhaust pipe of the engine close to the engine and a specification should state that the vacuum is to be so measured.

A fair guarantee to ask concerning the regulation is that the speed of the engine shall not vary more than 1½ per cent. above or below the normal speed under any condition of load. The builder should guarantee the workmanship and materials to be of the best, and agree to make good at his expense any defects in the engine, not due to neglect, that develop during the first six months or year it is in operation.

The accompanying specification was prepared by Mr. Nicholas S. Hill, Jr., M. E., and it is printed here with his permission. It should be understood that it was drawn for the purpose of obtaining a small automatic high-speed engine for a private corporation that obtained bids from several reliable builders whose engines were well known. Perhaps in public work where competition is open to all and where the lowest bid has to be accepted further detail would have been necessary.

SPECIFICATION FOR A 300 I. H. P. NON-CONDENSING ENGINE.

In the following specification the noun "Company" is used to designate the purchaser, the (purchaser's name and address). The noun "Builder" is used to designate the seller, the contractor, the manufacturers of the engine.

Rejection.—The Company reserves the right to reject any or all bids.

Engineer.—The interpretation of the specifications hereinafter set forth shall be left to the Engineer appointed by the Company, and the inspection of all materials furnished and all tests for the determination of the fulfillment of the guarantee herein contained shall be made under the direction of the said Engineer.

Test.—The engine will be tested at such time, after the erection and completion of engine and generator, as the builders may select and after the Engineer shall have received at least one week's notice. The Company will furnish the necessary fuel, oil and supplies and the contractor will be required to furnish the indicator rig and prepare the engine for test. The Engineer will furnish the indicators. A run of ten hours will be made and the load will be maintained as nearly as possible at 300 I. H. P. The conditions of the test will be fixed at the time of the signing of the contract as agreed between the Engineer and the Builder.

Kind of Service.—The engine is intended to operate a generator driving motor driven tools located in the shops of the Company.

Location.—The engine is to be located on foundations in the power house of the (name) Company at (place).

Type of Engine.—The engine desired is of the single valve, tandem compound, fly wheel governor, automatic cut-off type, with extension sub-base, direct connected to a (name) generator, size No. —, 200 K. W., 250 volt generator.

Conditions of Erection.—The Builders will furnish and erect engine on foundation supplied by the Company. The engine may be unloaded directly from car at the door of the power station within 15 feet of the foundation. The station is equipped with a traveling crane of five tons capacity. The armature for the generator will be placed on shaft at the power house.

Conditions of Operation.—The engine is to run non-condensing at 200 revolutions per minute. Steam pressure 125 lbs. above the atmosphere. Back pressure 15 lbs. absolute. The maximum load for which engine is intended equals 400 I. H. P. The engine is to operate at highest efficiency with load equal 300 I. H. P. The average load will equal 175-200 I. H. P.

Size of Cylinders.—The cylinders shall be approximately 18 inches and 29½ in. x 18 in. or 18½ in. and 30 in. x 17 in.

The volumes of cylinders furnished shall be not less than the volumes herein specified and the ratio of the cylinder volumes shall be between 2.6 and 2.8.

Speed Regulation.—The speed regulation shall be within 1.5 per cent. above or below normal.

Piston Speed.—The piston speed shall be not less than 560 feet per minute nor more than 600 feet.

Clearance.—The clearance shall not exceed 8 per cent.

Indicator Attachment.—Brass piping for indicator with three way valve is to be furnished for both high and low pressure cylinders.

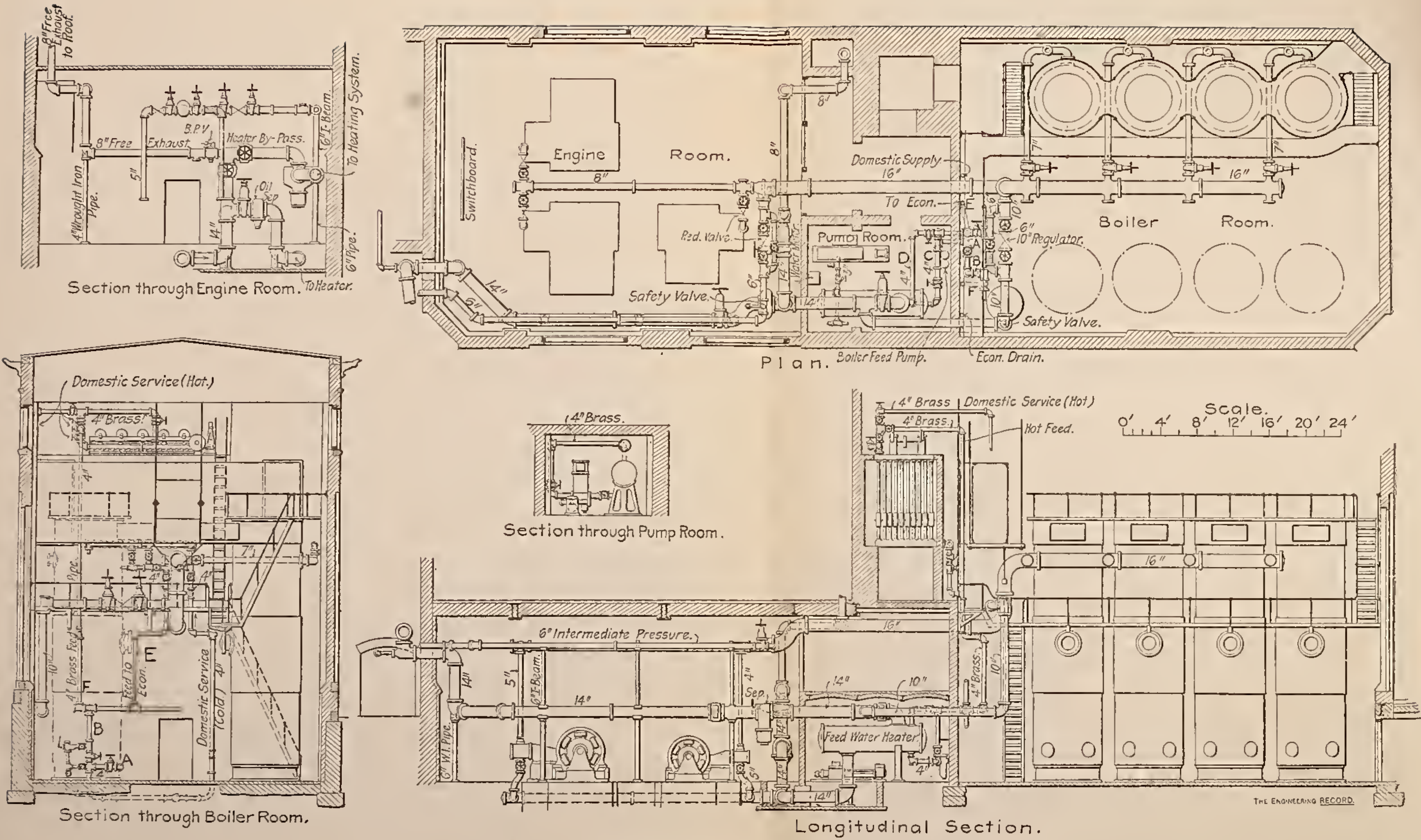


PLATE 8.—POWER PLANT, MASSACHUSETTS GENERAL HOSPITAL, BOSTON, MASS.

DEAN & MAIN, ENGINEERS.

Dimensions and Weights.—The Builder shall furnish the following data in regard to engine:

Total weight of engine.

Total weight of heaviest part.

Total weight of fly wheel.

Diameter and length of main bearings.

Diameter and length of cross-head pins.

Diameter and length of crank pins.

Diameter of main shaft in body.

Diameter and face of fly wheel.

Length of connecting rod.

Blue Prints, Template, Specifications.—The Builder shall furnish blue print of engine and foundations which shall form a part of these specifications after acceptance of proposal. A foundation template and foundation bolts shall be furnished by the Builder. The foundation bolts to be provided with washers at least 10 inches square and to be threaded for two nuts at top. The Builder shall also furnish specification of the lubricators, oil cups, tools, etc., to be supplied with engine.

Materials.—All materials used in the construction of the engine to be the best of their various kinds and to be in strict conformity with the latest modern practice.

Guarantee.—The engine shall be guaranteed to consume not more than 24 lbs. of steam per I. H. P. per hour, when developing 300 H. P., when running at a speed of 200 revolutions per minute with a steam pressure of 125 lbs. above the atmosphere and with a back pressure of 15 lbs. absolute.

The Builder shall also guarantee to make good any or all defects developed, within 150 days from the time of starting engine, which may be due to inherent defects in materials or faulty workmanship and design, provided such defects are developed when engine is running at less than 50 per cent. overload.

Painting.—All unfinished iron work about engine shall be filled, rubbed smooth and receive one coat of paint before leaving shop. A second coat is to be applied after erection and finally a finish coat highly enameled. The color is to conform with machinery already in place in the engine room. Quality of paint to be such as to insure against blistering, peeling off or fading and shall be satisfactory to the Engineer.

Covers.—The engine is to be supplied with a neatly fitting and well made oiled canvas cover to be approved by the Engineer.

Time of Delivery.—The Builder shall specify the earliest possible date of delivery.

Price.—The price submitted by the builder shall include delivery and erection in conformity with the preceding specification.

Chapter VI.—Steam and Water Piping.

Drawings.—After contracts have been let for the engines, boilers, pumps, feed-water heater and other auxiliaries of a plant, the engineer should obtain accurate drawings or blue prints of each, showing in plan and elevation the exact location of all steam and water inlets and outlets. When the engines, boilers and auxiliaries are located finally upon the plans, accurate drawings of the piping to connect them should be made. The piping should be shown in plan and in at least one elevation. The drawings should be to a scale of at least ⅜ inch to the foot, and should show the location of every fitting and valve in the system. It saves time to indicate a valve by drawing correctly the position of its flanges and joining them by two crossed lines. When a number of fittings are to be placed close together, however, they should be drawn accurately and in full, for if this is not done it may happen that the piping cannot be put together on account of too much being crowded into the space allotted. An accurate drawing would prevent an error of this kind from occurring. Complete drawings result in lower bids, and do away with extras that are usually the result of incomplete or inaccurate drawings.

Principles involved.—The fundamental object to be accomplished in steam piping is, of course, to carry steam without excessive loss of pressure; and next in importance is the requirement of safety, that the condensation loss shall be a minimum and that the piping shall not leak. The greatest enemy to safety is the liability of water entering the system, or collecting in it, due to condensation; and it is, therefore, of particular importance that piping should be so constructed that it does not contain pockets in which water can collect. Pipes are usually proportioned so that steam travels at the rate of about one mile a minute, hence if a "slug" of water, as a body of water is sometimes called, is picked up by the steam and carried along with it an accident is very apt to occur, either by the rupture of an elbow, at a change in direction of the pipe, or by the water entering the engine cylinder and wrecking it. In some instances pockets cannot be entirely done away with, but where they do occur they should be properly drained. Straightway globe valves should not be used in a steam pipe, as the valve seat acts like a dam in forming a water pocket. Valves of any kind should never be placed so as to form a water pocket whether they are closed or open, if it is possible to locate them any other way. For instance, one frequently sees the stop valve on a boiler placed immediately above the boiler nozzle and a vertical section of pipe above the valve leading to an elbow from which a horizontal pipe leads to a steam main. When a boiler so connected is out of service, water due to condensation accumulates in the vertical pipe over the stop valve, and although the pipe may be provided with a small drain pipe and valve, experience has shown that the latter are not always made use of to draw off the water, so that an accident may occur. It is just as effective to use an angle stop-valve at the top of the vertical pipe mentioned and to pitch the horizontal pipe that leads from it to the main so that condensation will flow toward the latter. With such an arrangement water cannot collect, and hence an opportunity for an accident does not exist.

Steam pipes should always be pitched so that the condensation that occurs in them will tend to flow in the direction in which the steam is moving, the reason for this being that if it is attempted to run condensation against the current of steam, water hammer is quite likely to occur, as the water accumulates into a slug, which is finally picked up by the steam and carried along until projected against a fitting. If one wants to carry steam a long distance, from one point to another at a higher level, the pipe should be laid at the proper inclination and at frequent intervals, say every 100 or 150 feet, the line should drop a few feet into a pocket that can be drained through a steam trap into a small return pipe running back to the boilers. A horizontal pipe should be inclined so that the condensation will tend to flow against the current of steam only when the pipe is excessively large, so that the velocity of steam flow will be much below the usual practice.

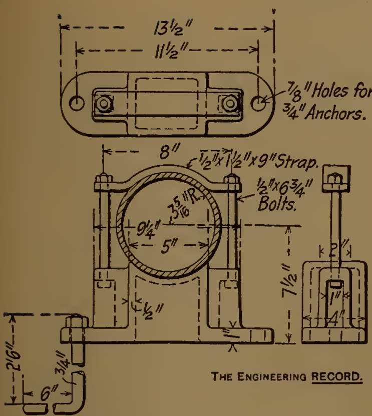
In systems of steam piping connecting several engines and the boilers supplying them it is usually the custom to connect each boiler with a steam main from which pipes lead to the engines. It is well to have this main of quite large size, so that the velocity of steam passing through it will be slow enough to allow any water that might be carried over from the boilers to accumulate in the main. The main, of course, should be drained by a pipe or pipes leading to a trap. If the main is divided into sections by valves, provision should be made for draining each section, for the reason that some parts may be shut off at times. Any branch to an engine should be connected to the top of the main to prevent, as far as possible, water from entering the branch.

It is impossible to give definite information as to the design of piping for all steam plants, for the conditions met with vary so much. With electric power-house work, however, this is not the case, for the reason that the construction of these plants, unless they be very large ones, almost invariably follow one of two types that are standard as far as the relative location of engines and boilers are concerned. These types are as follows: that in which the boilers and engines are placed back to back with a dividing wall between, and that with the boiler and engine rooms end to end, with the engines and boilers lying in the same direction. As far as the piping is concerned, the back to back type is the one most to be preferred, on account of the short and direct connection between the engines and boilers and the ease with which it can be enlarged. With the engine and boiler rooms placed end to end the condensation losses in the steam piping are greater, and this type of station, as far as the piping is concerned, is not easily enlarged.

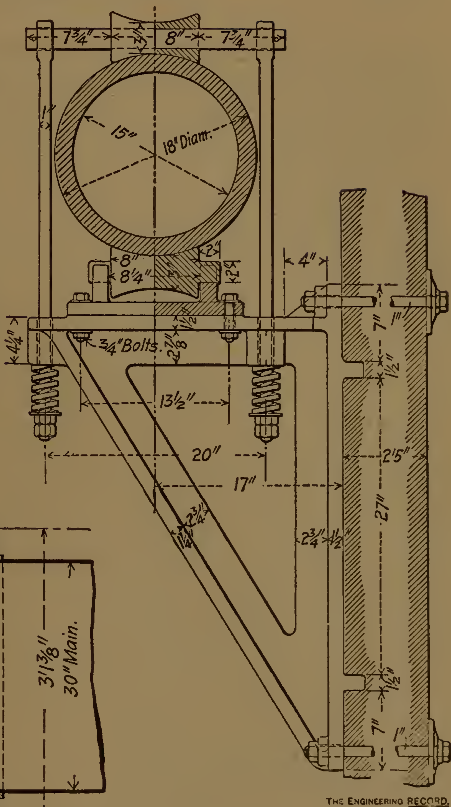
The steam piping for that type of station in which the engines and boilers are placed back to back and separated by a wall usually consists of a feeder from each boiler, connected with a main which is supported by the boiler room wall or suspended from the roof trusses and also connected to each engine.

The engine-room floor is usually a little higher than that of the boiler room. The main and as much of the piping as possible should be located in the boiler room, for the reason that if an explosion occurred in some section of the pipe in the boiler room, it would be possible after the steam-pressure fell to cut out the damaged section and operate the rest of the plant. If the engine room, on the other hand, was the scene of the explosion and became filled with steam, the electrical apparatus would, in all probability, not be fit for service without considerable overhauling.

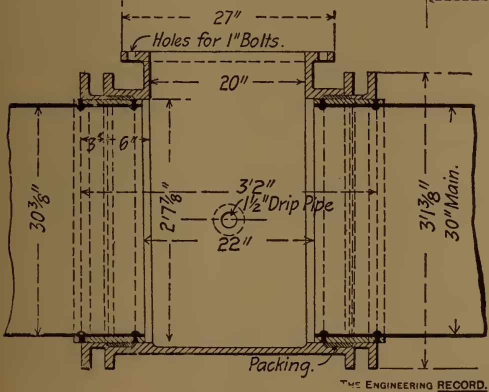
Two valves in a pipe leading from a boiler to a main are much more to be preferred than one and are usually provided in large



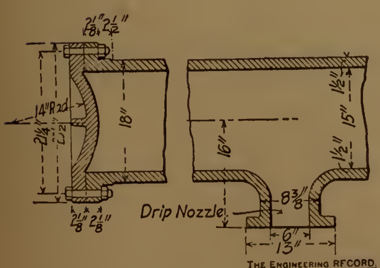
FEED MAIN ANCHOR.



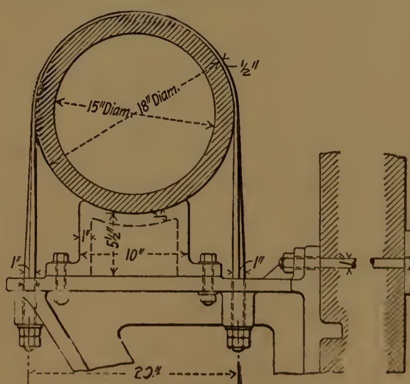
BRACKET FOR STEAM MAIN.



EXPANSION TEE FREE EXHAUST MAIN.



SECTION STEAM MAIN.



ANCHOR FOR STEAM MAIN.

FIGURE 22.—DETAILS OF STEAM PIPING, ANDERSON STATION (SEE PLATE 6).

SARGEANT & LUNDY, ENGINEERS.

work. One should be close to the boiler, the other at the main. If there is only one, however, and the steam main is supported on a bracket on the boiler room wall, it is a good arrangement to carry a pipe with a bend of long radius from the nozzle of each boiler to an angle stop valve bolted to a nozzle on the top of the main, as shown in Figure 23, and to run a connection to each engine from a stop valve similarly located. Instead of the angle valve shown in the figure, an elbow with a gate valve adjacent to it, with the axis of the valve in a horizontal position could be used. The valve in the pipe leading from the boiler to the main should not be placed over the boiler nozzle and thus form a pocket, for reasons previously explained. With both valves close to the main they are easily reached from a light walk, suspended from the roof trusses. Another advantage of placing the valve close to the main is that the condensation is less when the valve is closed. Some engineers prefer to place the angle stop valve immediately over the boiler nozzle and run a horizontal pipe from it to an elbow turning downward to a nozzle on the top of the main. The stop valve can then be reached by a person standing on top of the boiler setting. The engine connection should also be a bend of long radius. These connecting pipes will then have sufficient elasticity to permit expansion and contraction to occur without injury to the pipe. With the arrangement suggested there is no chance of water collecting in any part of the system, save in the main, and this should be large enough so that the water can settle there and pass out by the drain pipes. Frequently the pipe leading to each engine is provided with a separator close to the throttle valve. Besides intercepting moisture in the steam the separator performs another function of great value, in that it provides a reservoir of steam close to the cylinder, which insures a higher and more uniform pressure in the cylinder up to the point of cut-off than there would be if it were omitted; and it also reduces the vibrations in the steam piping due to the intermittent flow of steam as the valves of the engine open and close. For these reasons a separator, particularly if of large volume, is of much value.

When the steam piping in the engine room is run in the basement, the arrangement shown in Figure 24 can be resorted to. There the main is close to the floor of the boiler room. It can be supported on piers or wall brackets. The stop valve on the boilers is placed on the nozzle and a pipe with a long bend drops into the top of the main. The branch to each engine is run from an angle stop valve on the top of the main to a separator near each engine cylinder. This system is as unlikely to have accidents oc-

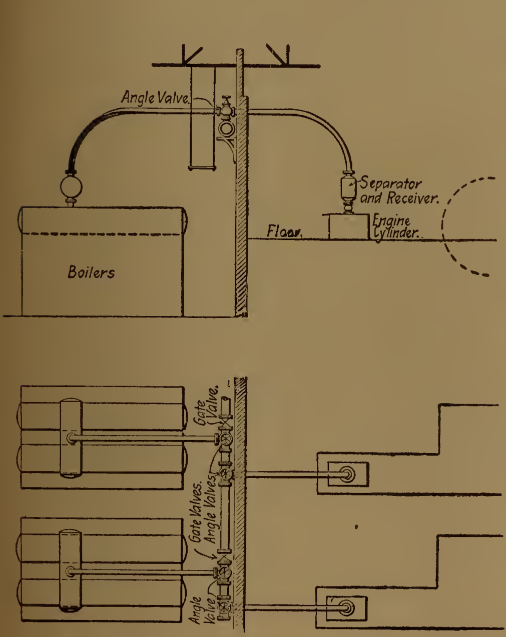


FIGURE 23.

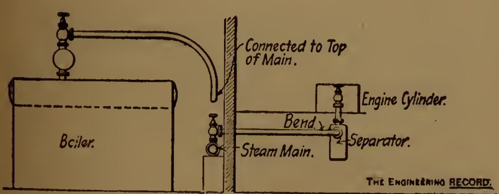


FIGURE 24.

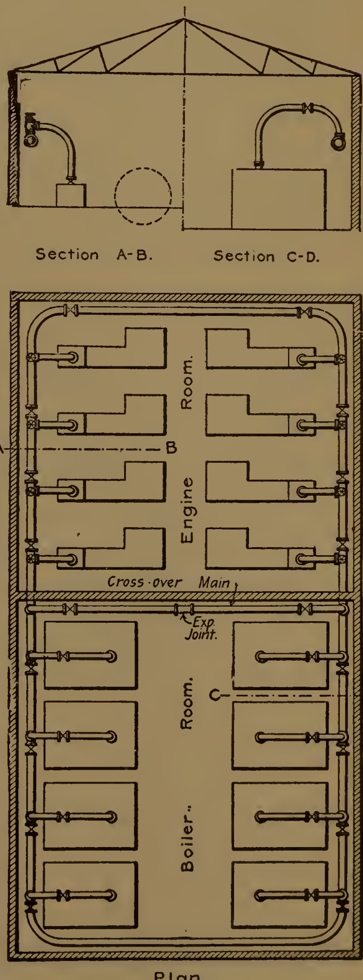


FIGURE 25.

cur to it as is the arrangement shown in Figure 23. An excellent example of this kind is shown in Plate 7.

Sometimes the steam piping is put in in duplicate, the two systems dividing at a double nozzle or Y on the boilers, and converging at a similar Y close to and connecting with the throttle valve of the engine. Duplicate systems were used much more frequently in the early days of electric power-house construction than they are at the present time. In fact, the opinion is fast becoming universal that a duplicate system is an expense that is unnecessary with the arrangement of boilers and engines and piping shown in Figures 23 or 24.

It is the custom in the latest work to subdivide the power house into complete and independent units. The plans of the Lincoln wharf power house of the Boston Elevated Railway in Plate 1 and Figure 3 shows how the piping is subdivided. By closing valves in the main each unit is entirely independent.

If the power station has the engine and boiler rooms placed end to end, as in Figure 25, the arrangement of piping shown there is probably the safest. It is arranged on the ring or loop system, and valves are so placed that if an accident occurs, the damaged section may be cut out and the steam carried around through the system in the opposite direction. In the station shown an expansion joint is placed in the cross connection. The cross-over pipe might, perhaps, be omitted. In a station of any size the expansion in the mains running lengthwise of the engine and boiler rooms could be taken care of by anchoring the mains at their middle points, so that the expansion would be divided equally between the two ends of the mains.

Exhaust piping for condensing plants.—A frequent method of running exhaust pipes in condensing plants is shown in Figure 26. Each engine is supposed to be provided with an independent condenser and air pump. Two exhaust pipe branches are shown, one branch dropping into the condenser and the other branch, which contains a relief valve, leading to the atmosphere. If advantageous to do so, the free exhaust pipes can be connected to a single pipe leading to the atmosphere. The purpose of the atmospheric connection is to provide means for allowing the steam to escape in case something happens to the condenser to prevent it from working. The relief valve is nothing more than a large check valve that is closed by the pressure of the atmosphere on one side, when a partial vacuum exists upon the other. Sometimes several engines exhaust into one condenser. The general arrangement can be the same. Exhaust piping for condensing plants should have flanged fittings most carefully put together. A leak through a very small hole will greatly affect the vacuum and the efficiency of the engine. The principal point to be looked after is to have the alignment of the pipe such that there is absolutely no chance for water to lodge in the piping system, for the reason that the water might be sucked back in the engine cylinder and destroy it as has frequently occurred. It is practically impossible to drain the exhaust pipe of a condensing engine, except toward the condenser, as the system is under a partial vacuum. One of the most important points is to have very generously proportioned exhaust pipes with as direct a run to the condenser and as few bends in the pipe as possible.

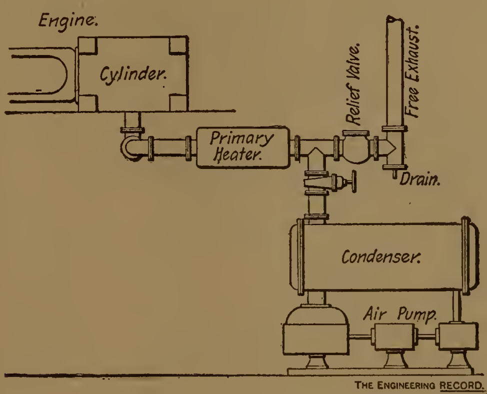


FIGURE 26.

Piping Between Cylinders of Compound Engines.—It is usually the custom in installing a cross-compound engine to arrange the piping between the cylinders so that high pressure steam may be admitted to the low pressure cylinder as well as the high pressure and to provide the necessary exhaust pipes so that both cylinders may be used at the same time as simple engines. If the engine is to be run condensing, it is sometimes so piped that the high pressure cylinder may run as a simple non-condensing engine while the low pressure cylinder may run as a simple condensing or non-condensing engine as may be desired. Designers have gone so far as to provide for running high pressure cylinders as a simple condensing engine and the low pressure cylinder a simple non-condensing engine although ordinarily intended to operate as an engine of the cross-compound condensing type. Various arrangements for accomplishing the purpose mentioned are shown in Plate 3 and in Figure 18.

Exhaust piping for non-condensing plants.—In non-condensing plants the exhaust steam can be carried long distances for heating or for manufacturing purposes, provided the pipes carrying it are large enough. When exhaust steam is thus utilized, the pipe carrying the exhaust steam from the engines to the outer air, which is known as the atmospheric exhaust, or the free-exhaust pipe, is provided with a back-pressure valve, the function of which is to preserve a sufficient pressure in the exhaust piping to cause the exhaust steam to flow through a system of pipes, also connected with this free-exhaust pipe, to the places where it

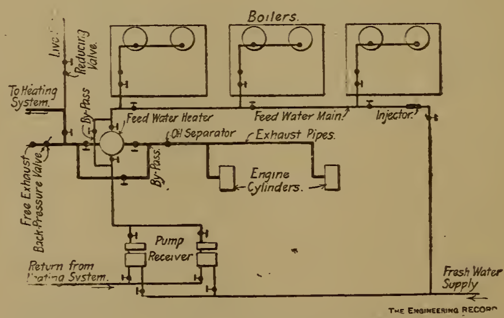


FIGURE 27.—EXHAUST AND FEED PIPING FOR NON-CONDENSING PLANT.

is to be used. These back-pressure valves are designed so as to open when the pressure in the exhaust system exceeds a certain amount, and thus allow sufficient steam to escape to reduce the pressure to that desired. It sometimes happens when exhaust steam is used for heating or for manufacturing purposes, that the supply is not sufficient to meet the demand, and if this is likely to occur, it is the custom to run a live-steam pipe from the high-pressure piping to the exhaust piping, and to place in this connecting pipe a reducing valve which automatically opens and allows live steam to enter the exhaust system when the pressure in the latter falls below that which it is desired to maintain.

When a system for utilizing exhaust steam is employed, it is frequently arranged in the manner shown in diagram by Figure

27. A grease separator should be provided to remove as much oil as possible from the exhaust steam, after which the steam is passed through the feed-water heater. The arrangement shown is designed for a feed-water heater of the closed type. From the heater the pipe branches, one branch supplying steam for heating or for a manufacturing purpose—this branch furnished with a high-pressure connection run from the boiler and containing a reducing valve. The other branch leads to the atmosphere, and it is provided with the back-pressure valve. The top of the free-exhaust pipe should have an exhaust head for intercepting the moisture that is blown out of the pipe by the exhaust steam. This head should have a drain pipe to a sewer or to waste.

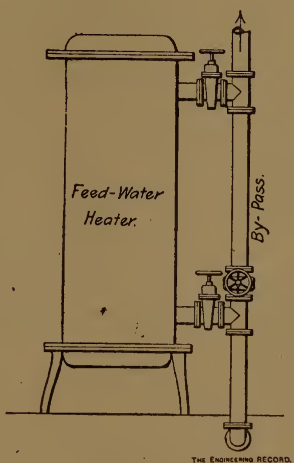


FIGURE 28.

Most power plants have a basement under the floor of the engine room, and in steam plants for buildings the exhaust piping is frequently run in covered trenches. When the exhaust piping is below the floor, a satisfactory method of connecting the feed-water heater is shown in Figure 28. The steam may pass through the heater or around it through the by-pass by properly adjusting the valves.

Care of drips.—The drainage from any part of the piping system is valuable on account of the value of the water, and the heat in the water that would be lost if the condensation was allowed to go to waste. By condensation is meant water due to condensation in the pipes. Sometimes the saving that is due to returning high-pressure condensation to the boilers is not sufficient to warrant the expenditure for the apparatus necessary to do this, but that is a point which must be determined for each case. Condensation from exhaust-steam pipes can sometimes well be allowed to go to waste, for the reason that it generally contains more or less oil; and the chance of this doing injury to the boilers is apt to be too great for the saving that would follow its utilization. If it is used, the water should be filtered.

There are various methods of returning high-pressure condensation to the boilers, and the most common are the Holly system and the "steam loop," both patented systems controlled by Westinghouse, Church, Kerr & Company, by means of the automatically-governed steam pump and receiver and by means of a return trap. With the pump and receiver all high-pressure drip pipes, from separators, steam jackets on the cylinders, from reheating receivers placed between the cylinders of compound and triple-expansion engines, and all other points from which water is drained can be connected to a main leading to an automatic pump and receiver. Each drain pipe should have a steam trap, if there is any chance of their being under different pressures. The automatically-governed pump can discharge the water into the boiler feed-pipe between the boiler feed-pump and the boilers. With very long steam pipes which have to be drained at several points, the drainage can be drawn off by a steam trap discharging into a return pipe leading to an automatic pump and receiver at the boiler-house. If the inclination of the steam pipe is such that the water will not run back by gravity in the return pipe to the boiler house, the drip can be carried to the lowest point in the system and an automatic pump and receiver, operated by steam from the main pipe, can be located there and used to pump the water up hill and back to the boiler house; that is, of course, provided the saving will warrant this arrangement. All high-pressure drip lines should be thoroughly covered with a non-conducting covering.

Low-pressure drips contain the condensation drawn from pipes carrying exhaust steam, steam condensed in feed-water heaters of the closed type, the drip from engine and pump cylinders and the drip from grease separators. All of this condensation should be thrown away on account of the grease that it contains, unless some form of oil filter is used. All low-pressure drips should be trapped independently into a drip main which can be led to any convenient place, such as the sewer, the pipe carrying off the discharge from the condenser, etc.

Feed-water piping.—Feed-water piping is sometimes put together with screwed and sometimes with flanged fittings, and these are either of brass or cast iron. The pipe is of brass or extra heavy wrought iron. The screwed fittings are said to be capable

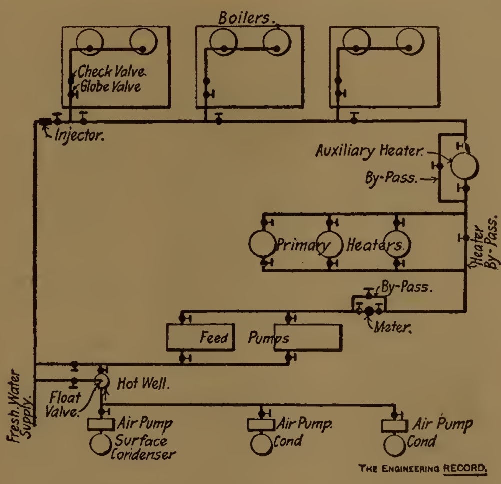


FIGURE 29.—FEED PIPING FOR CONDENSING PLANT.

of standing a pressure of 150 pounds, but for pressure over 125 pounds flanged fittings are frequently used. It is better to use elbows with a long radius to reduce friction. Gate valves should be used instead of globe valves, for the same reason.

An arrangement for feed-water piping for a typical plant equipped with surface condensers is shown in diagram by Figure 29. The plant is supposed to contain three engines, each with an independent air pump and surface condenser, two boiler feed-pumps, a primary heater in the exhaust pipe of each engine, between the engine and surface condenser, and a single auxiliary heater of the closed type receiving the exhaust steam of the air pumps and the boiler-feed pumps. The steam condensed in each condenser is drawn therefrom by the air pumps and forced into a hot well, from which the boiler-feed pumps draw their supply.

For occasionally replenishing the water from the condensers and supplying fresh water when needed, a fresh-water pipe is led to the hot well, and its supply can be controlled by a float valve or ball cock, so that fresh water will flow in the hot well in case the boiler-feed pumps draw water faster than it is discharged from the air pumps an occurrence that is unlikely to happen. The boiler-feed pumps are in duplicate. It is a good investment to put a water meter in the feed line, and one capable of measuring hot water should be used. The meter should have a by-pass and

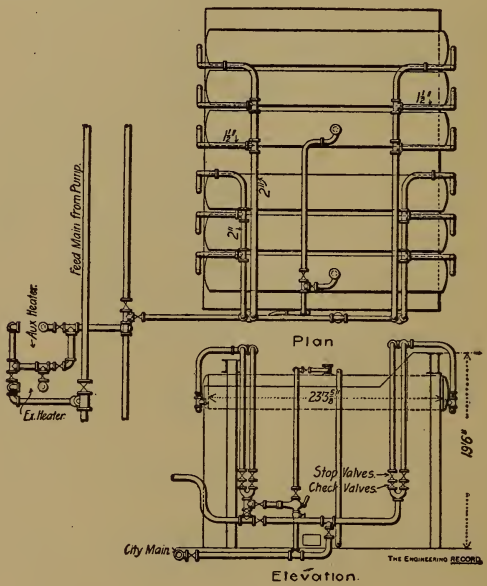


FIGURE 30.—FEED PIPING BOSTON ELECTRIC LIGHT COMPANY'S STATION.

be located at the pressure side of the pump, as shown. From the feed pumps the water is forced through the primary heaters and then through the auxiliary heater to the boilers. If the hot well is not provided with sponges or some material arranged to filter the oil from the air pump discharge, a cloth filter should be placed between the feed pumps and the boilers. Each heater should have a by-pass, and in the pipe leading to each heater two valves should be placed, one used as a stop valve and the other as a regulating valve, which should be adjusted when the plant is started, so that approximately equal amounts of water will pass through the heaters. It involves a little more complication to do this, but it insures each heater doing its full work.

The arrangement shown in Figure 29 provides for the use of a heater of the closed type between the engine and condenser for of course an open heater could not be used in such a situation. It is not often the custom to place a heater between the engine and condenser and where that is not done an open heater might be used to receive the exhaust steam of the auxiliaries, and if so,

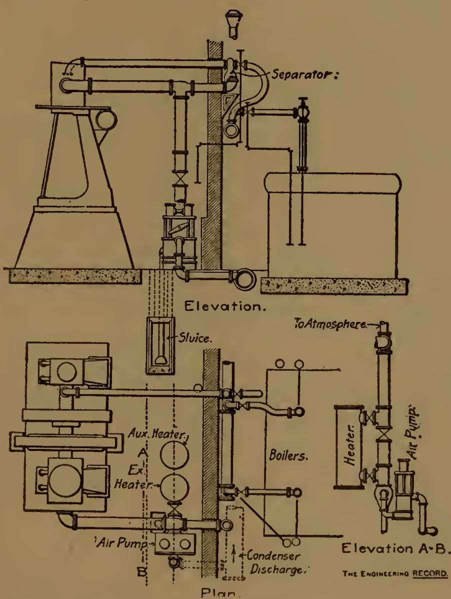


FIGURE 31.—ARRANGEMENT AUXILIARIES BOSTON ELECTRIC LIGHT COMPANY'S STATION.

the arrangement of the feed-water piping would have to be different from that shown. If the plant shown diagrammatically in Figure 29 was provided with jet condensers the boiler feed pumps might be arranged to draw their supply from the air pump discharge, as well as from the source of the fresh water supply and pump it through the heaters to the boilers. If ample feed water at low cost was available the air pump discharge would probably be allowed to go to waste because of the grease in it and the pumps connected to draw fresh water. If the latter was under pressure the water could be passed to an open heater first and the pump arranged to draw the water from it. This arrangement combined with an economizer is shown diagrammatically in Figure 32. If a surface condenser was used the air pump discharge might be led to an open hot well, arranged to filter the water by means of sponges or excelsior, etc., located at a higher elevation than the heater so the water would flow to the heater by gravity and from the heater to the feed pumps at a lower elevation than the heater so that the pump suction would be under pressure. If the filter was of the closed type containing cloth or similar material it would have to be placed between the pumps and the boilers. High pressure drips could be trapped into the heater, returns from the heating system could be connected with it and fresh cold water could be added to the heater by a float valve when the de-

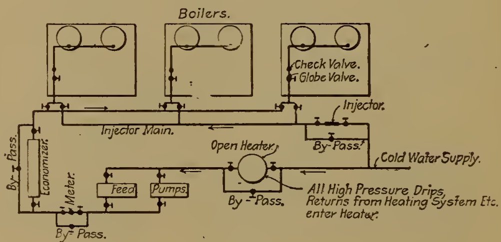


FIGURE 32.—FEED PIPING WITH OPEN HEATER.

mand of the pumps was in excess of the water entering the heater from the condensers, heating system, drips, etc. The boiler feed pumps would have to pump hot water with this arrangement, hence they should be constructed to do this. The pumps could deliver water to the boilers direct or first through an economizer where the water would be heated further by the waste gases from the boiler. An economizer is placed between the pumps and the boilers. Arrangements should be made to by-pass all heaters, economizers, meters, etc.

If a live steam purifier is used, which is a heater in which the feed water is exposed to the action of steam at full boiler pressure to precipitate scale forming salts in the purifier instead of having them precipitated in the boiler. The purifier is elevated above the boilers and receives the feed water after it has passed through all other heaters; and it is connected so that water will flow from it to the boilers by gravity.

In Figure 29 one end of the feed-water main in the boiler room is provided with an injector connected to the fresh-water supply. It would be a better arrangement, perhaps, to have an independent main from the injector, the main connecting with the feed pipe to each boiler through stop valves so that water from the in-

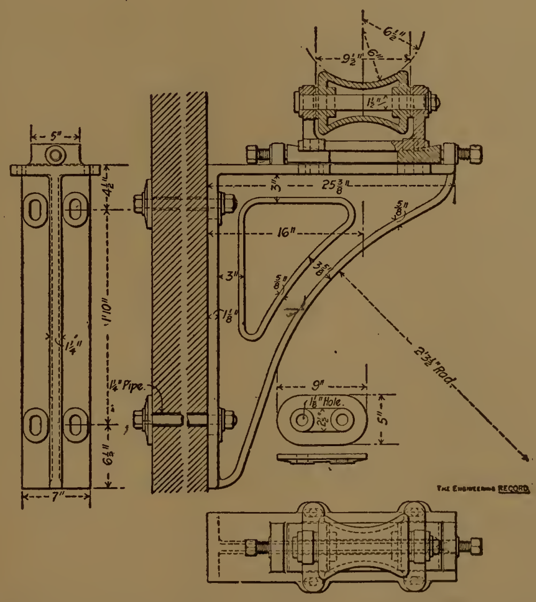


FIGURE 33.—PIPE BRACKET DESIGNED BY SHEAFF & JAASTED.

jector or the feed pump could be supplied to any boiler independent of the others as shown in Figure 32.

The method of running feed mains in a boiler room varies. Horizontal tubular boilers are frequently set with a number of boilers in one battery. Sometimes therefore, the feed main is run along the fronts of the boilers just above the fire doors. Water-tube boilers are generally set two in a battery, so that if a pipe extends across the front it blocks the passageway between the batteries. With this type of boiler the feed main is sometimes run in a covered trench in the floor along the fronts of the boilers. Again, the mains are run on top of the setting. In the latter event a branch from the main should extend to the boiler front, at an elevation low enough so that the regulating valve, which should be of the globe type, may easily be reached by the boiler attendant. A check valve should be placed in the pipe, as shown.

Feed-water piping for non-condensing plants with the closed type of feed water heater is shown in diagram in Figure 27. It is supposed that exhaust steam is used for heating and that the condensation is returned to the boilers. This may be brought back to one of the boiler-feed pumps, which can be connected to a return tank receiving the condensation in the heating system. The pump draws water from the tank and forces it through the feed-water heater to the boiler. A fresh-water pipe is led to the return tank and the supply is controlled by a float valve. If a feed-water heater of the open type is used, the condensation from a heating system is carried back to the heater, and usually enters it at the side or bottom and thus does away with the automatic pump and receiver. The boiler-feed pump draws its supply from the heater and forces the water to the boilers. Fresh water under pressure, controlled by a float valve or ball cock in the heater, is supplied to the heater at the top, and it falls to the bottom in direct contact with the exhaust steam and is thus heated. Cold water is only supplied when the boiler-feed pump draws the water from the heater so fast that the water surface falls below a certain level.

In electric stations it is frequently the practice, for safety's sake, to use a duplicate feed main from the pump to the boilers. With such an arrangement it is possible to test a boiler or group of boilers. If the steam piping is arranged so that one or more boilers can supply any one engine independently of the others, the duplicate boiler-feed main is of considerable value, for with such a duplicate system the steam used by an engine can be measured at any time and the condition of the engine determined. Boiler feed pumps should always be in duplicate.

Kind of pipe.—"Standard" sizes of steel pipe are used for steam piping. Table I. contains the various dimensions of pipe, those up to and including 10-inch pipe being the Briggs standard. The other sizes are in common use. As ordinary merchant pipe may vary in thickness from the standard, "full weight" pipe should be asked for. Full weight pipe may vary 5 per cent. from the standard thickness. In the connection between a boiler and a steam main, or the main and the engines, long bends made of pipe are an advantage, for several reasons. First, they reduce the friction very much; second, their use reduces the number of joints likely to leak; third, such a connection is very much more flexible than one composed of two straight pieces of pipe connected by an elbow. Their greater flexibility is of great advantage in taking care of expansion after the piping is in place, and furthermore, they are much easier to connect when erecting the piping.

TABLE I.—DIMENSIONS OF STANDARD WEIGHT PIPE.

Inside diameter.	Actual outside diameter.	Thickness.	Actual inside diameter.	Length of pipe per square foot of outside surface.	Inside area.	Length of pipe containing one cubic foot.	Weight per foot.	Discharge per min., velocity 6,000 ft. per min., steam 100 lbs. gauge pressure.	Discharge per min., velocity 6,000 ft. per min., steam 150 lbs. gauge pressure.
Ins.	Ins.	Ins.	Ins.	Feet.	Ins.	Feet.	Lbs.	Lbs.	Lbs.
1	1.315	0.134	1.048	2.903	0.8627	166.9	1.670	.....	.....
1¼	1.66	0.140	1.380	2.301	1.496	96.25	2.258	.....	.....
1½	1.90	0.145	1.611	2.01	2.038	70.65	2.694	.....	.....
2	2.375	0.154	2.067	1.611	3.355	42.36	3.600	.....	.....
2½	2.875	0.204	2.468	1.328	4.783	30.11	5.773	.....	.....
3	3.50	0.217	3.067	1.091	7.388	19.49	7.547	80	113
3½	4.00	0.226	3.548	0.955	9.887	14.56	9.055	107	151
4	4.50	0.237	4.026	0.849	12.730	11.31	10.66	138	194
4½	5.00	0.247	4.508	0.765	15.939	9.03	12.34	173	243
5	5.563	0.259	5.045	0.629	19 990	7.20	14.[illegible]	218	305
6	6.625	0.280	6.065	0.577	28.889	4.98	18.767	315	442
7	7.625	0.301	7.023	0.595	38.737	3.72	23.27	422	592
8	8.625	0.322	7.982	0.444	50.039	2.88	28.177	545	764
9	9.625	0.344	9.001	0.394	63.633	2.26	33.70	694	974
10	10.75	0.366	10.019	0.355	78.838	1.80	40.06	872	1,223
11	12.00	0.375	11.25	0.318	98.942	1.455	45.95	1,079	1,514
12	12.75	0.375	12.000	0.293	116.535	1.235	48.98	1,275	1,788
13	14.00	0.375	13.25	0.273	134.582	1.069	53.92	1,468	2,060
14	15.00	0.375	14.25	0.254	155.968	.923	57.89	1,702	2,385
..	16.00	0.375	15.25	0.238	177.867	.809	61.77	1,940	2,722
..	18.00	0.375	17.25	0.212	225.907	.638	69.66	2,462	3,452
..	20.00	0.375	19.25	0.191	279.720	.515	77.57	3,050	4,275
..	22.00	0.375	21.25	0.174	354.66	.406	85.47	3,870	5,425

No matter how much care is taken in facing the flanges off square, it almost always happens that the flanges of the boiler nozzles are not in perfect alignment, or exactly horizontal, so that a considerable strain is introduced in the piping in forcing the abutting flanges to a seat. It is much better to make the bends in the piping of steel than copper, although the latter has been used to some extent. At the temperature the copper is subjected to in brazing the joint, the fibrous nature that copper acquires in rolling is destroyed and a serious reduction of its tensile strength and ductility results. Mr. James B. Berryman of the Crane Company states that unless the bends are of very short radius they are generally made of standard pipe for pressures of 125 pounds or less, full weight pipe up to 175 pounds, and extra heavy pipe for higher pressures.

Size of steam pipes.—It has been the custom to so proportion steam pipes for engines that the maximum velocity of steam flowing through them will be about 6,000 feet per minute. The pipe size can be obtained by assuming that the area of the steam pipe in square inches multiplied into the maximum velocity of steam in feet per minute is equal to the piston area in square inches multiplied into the piston speed in feet per minute. If the cut off is at one-third

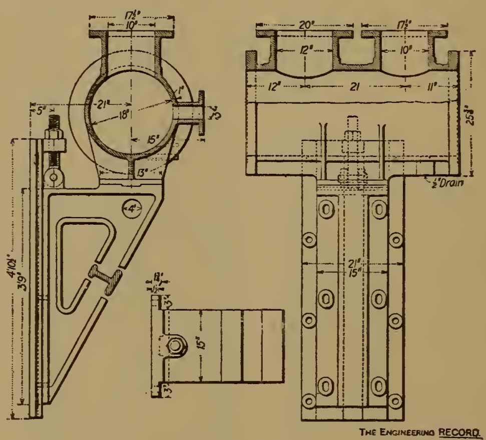


FIGURE 34.—PIPE BRACKET.

stroke the average velocity of steam would then be about 2,000 feet per minute. Friction would cause a considerable drop in pressure, with small sizes of pipe, when proportioned by this rule. Some experiments upon the flow of steam in pipes in a paper in volume XX. Transactions of The American Society of Mechanical Engineers, by Professor R. C. Carpenter, give some data that is of value. The tests were made upon 1, 1½, 2 and 3-inch pipes with lengths varying from 90 to 250 feet. The formula derived from the experiments checks very closely with the results obtained by M. Ledoux, who experimented with pipes varying from 1.85 inches to 3.94 inches in diameter and with lengths varying from 328 feet to 1,082 feet.

Professor Carpenter uses the formula

$$P=\frac{1}{20.663}K\left(1+\frac{3.6}{d}\right)\frac{w^2\ L}{Dd^5}$$

in which P equals the loss of pressure in pounds per square inch, d the diameter of the pipe in inches, W the flow of steam in pounds per minute, w the flow in pounds per second, D the density or weight per cubic foot, L the length of pipe in feet and K a constant taken, as a result of the experiment, at .0027. Table II., which was calculated by Mr. E. C. Sickles, is based upon the formula and was deduced by making every factor constant except the diameter, length of pipe, and discharge. The left-hand vertical column of the table contains the diameters (d) of the pipes, and the top horizontal column the length (L) in feet, while the body

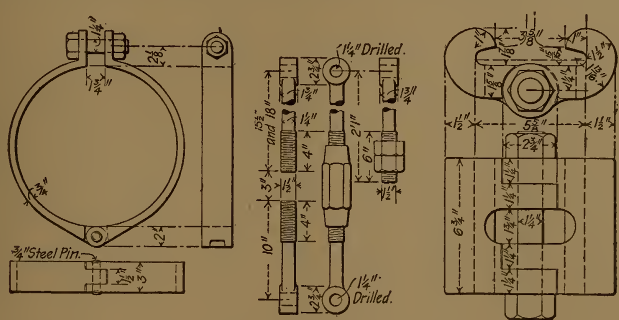


FIGURE 35.—PIPE SUPPORT DESIGNED BY DEAN & MAIN.

of the table gives values of (W) the pounds discharged per minute. Prof. Carpenter explains the table as follows:

"Thus, for instance, a 10-inch pipe 250 feet long will deliver 712 pounds of steam per minute with a drop of one pound in pressure, if there exists an average absolute pressure of 100 pounds; or, if all other conditions hold except the length of pipe, which varies, it may be seen that for 100 feet the discharge is 1,126 pounds, for 400 feet 563 pounds, and so for any number of feet given in the table.

"If any intermediate length of pipe is used other than those given in the tables, the discharges given by the tables may be corrected by consideration of the fact that the weight of discharge is inversely proportional to the square root of the length of the pipe.

"To meet the conditions where other average absolute pressures than 100 pounds exist, and higher drops than one pound are assumed, it is only necessary to use suitable factors which are calculated by means of the fundamental formula, and graphically represented by Curves 1 and 2, Figure 36.

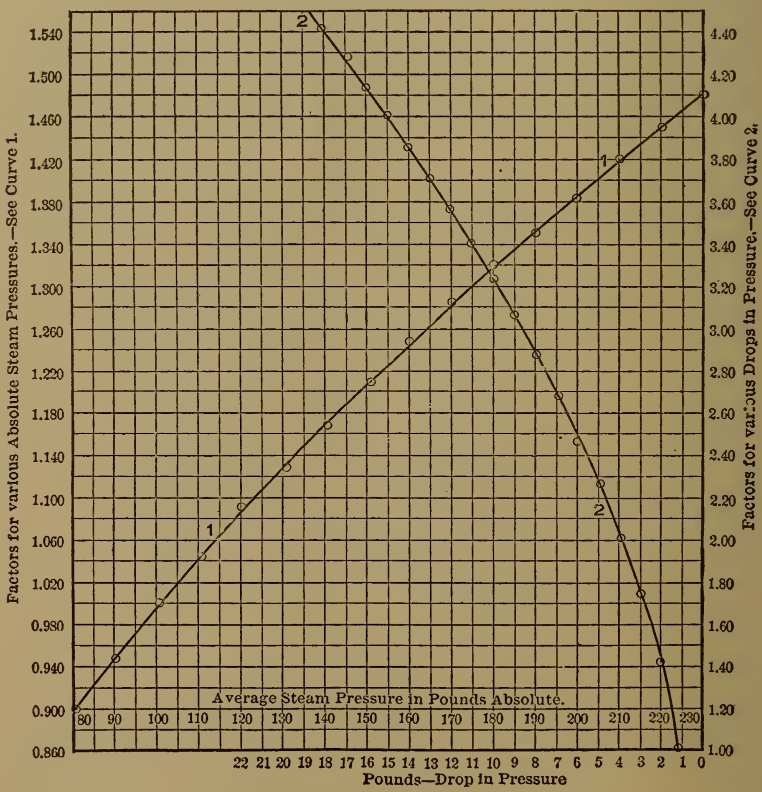


FIGURE 36.

"As an illustration of the use of the tables and curves, suppose it is desired to find what size pipe will be required to deliver 1,000 pounds of steam per minute a distance of 1,000 feet, the initial pressure of the steam being 157.5 pounds, and the final 152.5 pounds by gauge. Solution—It will be best to reduce all conditions to those of the tables and find the discharge, and from this the size of the required pipe. Looking at Curve 2, we find the factor of discharge for a 5-pound drop is about 2.23 times that for a 1-pound drop. Therefore, dividing the required discharge of 1,000 pounds by 2.23, we have about 450 pounds discharge for a 1-pound drop.

TABLE II.—FLOW OF STEAM IN PIPES. DELIVERY IN POUNDS PER MINUTE BY PIPES OF GIVEN DIAMETERS AND LENGTHS. (ABSOLUTE PRESSURE 100 POUNDS—DROP IN PRESSURE ONE POUND.)

Diam. in inches.	Length in Feet.									
	50	100	150	200	250	300	350	400	500	600
¾	1.55	1.10	.898	.700	.698	.633	.589	.550	.491	.450
1	3.10	2.20	1.79	1.55	1.39	1.26	1.17	1.10	.982	.900
1¼	6.90	4.88	3.93	3.44	3.08	2.81	2.62	2.44	2.18	1.96
1½	9.29	6.52	5.83	4.62	4.14	3.77	3.50	3.26	2.91	2.67
2	21.6	15.2	12.50	10.8	9.82	8.82	8.37	7.63	6.81	6.23
2½	35.9	25.4	20.8	17.9	16.2	14.62	13.6	12.7	11.3	10.4

Diam. in inches.	Length in Feet.									
	100	250	400	550	700	850	1,000	1,300	1,600	1,750
3	46.0	29.2	23.0	19.6	17.4	15.8	14.5	12.7	11.5	11.0
3½	69.5	44.6	34.7	29.6	26.3	23.8	22.0	19.2	17.4	16.6
4	97.6	61.8	48.8	41.5	36.7	33.5	30.8	27.1	24.4	23.3
4½	132.9	84.1	66.45	56.6	50.1	45.6	42.0	36.75	33.2	31.8
5	180.7	114.3	90.3	76.9	68.2	62.1	57.1	50.0	45.2	43.2
6	296.5	187.4	148.2	125.8	111.8	101.8	93.7	82.2	74.1	71.0
7	437	276	218	186	165	150	138	121	109	104
8	624	394	312	266	236	214	197	173	156	149
9	853	539	426	363	322	292	269	236	213	204
10	1,126	712	563	480	425	386	356	312	281	269
11	1,447	915	723	617	546	496	457	401	361	346
12	1,887	1,192	943	803	714	648	598	523	472	451
13	2,238	1,415	1,119	954	846	767	707	620	559	535

Diam. in inches.	Length in Feet.									
	100	200	300	500	800	1,000	1,400	1,800	2,000	2,200
14	2,714	1,920	1,567	1,213	959	858	725	639	606	578
15	3,250	2,300	1,873	1,453	1,149	1,028	868	766	726	693
16	4,000	2,830	2,315	1,785	1,413	1,268	1,072	945	895	854
17	4,500	3,210	2,635	2,021	1,591	1,424	1,200	1,061	1,006	962
18	5,211	3,685	3,008	2,330	1,843	1,648	1,393	1,228	1,165	1,111
19	5,992	4,237	3,459	2,680	2,119	1,895	1,602	1,412	1,340	1,278
20	6,839	4,835	3,948	3,059	2,418	2,163	1,828	1,612	1,529	1,458
22	8,743	6,183	5,048	3,910	3,093	2,765	2,337	2,061	1,955	1,864
24	113,08	7,990	6,535	5,065	3,995	3,580	3,023	2,665	2,535	2,415

"Again, the average pressure is 155 + 15, or 170 pounds absolute, and from Curve 1 it may be found the factor of discharge is 1.248 greater than for 100 pounds absolute. Therefore, dividing 450 pounds by 1.284 we have 350 pounds on the basis of the conditions given by the tables; and looking under 1,000 feet lengths for the discharge nearest to 350 pounds, we find a 10-inch pipe discharges 356 pounds per minute; therefore, it would be satisfactory."

Size of exhaust pipes.—Exhaust pipes should be proportioned for a much lower velocity of steam than high pressure pipes. The loss of a pound pressure between the engine and condenser means a very much greater increase in the amount of steam required to run the engine than would be occasioned by the loss of several pounds in the initial pressure, particularly with engines operating with low mean effective pressures. No general rule can be given for proportioning exhaust pipes. It is best to calculate the velocity that would occur in a pipe of a certain size for an assumed loss in pressure by means of formula for the flow of steam through pipes.

Kind of fittings.—There are two kinds of fittings used in steam piping, the screwed and the flanged fittings. The former, as the name implies, are put together with a screw thread, while the flanged fittings are bolted together. Screwed fittings are used to some extent in plants where the steam pressure is low, 80 pounds or under, and sometimes with higher pressures. Their use, however, seems to be going out for power plant work. The most used type of flange is shown in Figure 37. The pipes are screwed into the flanges as shown. On July 18, 1894, committees of the American Society of Mechanical Engineers, of the National Association of Steam and Hot-Water Fitters, and of manufacturers of fittings, met and adopted a schedule for the dimensions of flanges and this schedule is known as the A. S. M. E., or the "Master Steam Fitters" flange schedule. This schedule is printed in Table III. upon another page. Flanges dimensioned in accordance with this schedule are considered strong enough for a steam pressure of 100 pounds. For higher pressures, a schedule shown in Table IV. was adopted June 28, 1901, and this is suitable for pressures from 125 to 225 pounds per square inch. Flanged fittings, that is elbows, ties, crosses, etc., are made by a number of manufacturers and attention is drawn to the fact that the face to face and the face to axis dimensions vary with the different makes although the flange dimensions are standard. Specifications for flanges should require that the holes be drilled in the flange to straddle a vertical plane passing through the axis of the pipe. In the best work the pipe and flange are carefully threaded and the pipe screwed into the flange until it projects slightly beyond its face. The inner circumference of the flange at the face of the flange is then struck with a hammer, or peaned, as it is called. The pipe and flange are then put into a lathe where both are faced off. It is very essential that the pipe and flange be faced

TABLE III.—SCHEDULE OF STANDARD FLANGES ADOPTED JULY 18, 1894, BY A COMMITTEE OF THE MASTER STEAM AND HOT WATER FITTERS' ASSOCIATION, AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND VALVE AND FITTING MANUFACTURERS. SUITABLE FOR PRESSURES UNDER 100 POUNDS PER SQUARE INCH.

Size of flange, pipe size x diam.	Diameter of bolt circle.	Number of bolts.	Size of bolts, pressure under 80 lbs.	Size of bolts, pressure 80 lbs. and over.	Flange thickness at hub for iron pipe.	Flange thickness at edge.	Width of flange face
2 x 6	4¾	4	½ x 2	⅝ x 2	1 in.	⅝	2
2½ x 7	5½	4	½ x 2¼	⅝ x 2¼	1⅛	11/16	2¼
3 x 7½	6	4	½ x 2½	⅝ x 2½	1¼	¾	2¼
3½ x 8½	7	4	½ x 2½	⅝ x 2½	1¼	13/16	2½
4 x 9	7½	4	⅝ x 2¾	¾ x 2¾	1⅜	15/16	2½
4½ x 9¼	7¾	8	⅝ x 3	¾ x 3	1⅜	15/16	2⅜
5 x 10	8½	8	⅝ x 3	¾ x 3	1½	15/16	2 ½
6 x 11	9½	8	⅝ x 3	¾ x 3	1½	1	2½
7 x 12½	10¾	8	⅝ x 3¼	¾ x 3¼	1½	1 1/16	2¾
8 x 13½	11¾	8	⅝ x 3½	¾ x 3½	1¾	1⅛	2¾
9 x 15	13¼	12	⅝ x 3½	¾ x 3½	1¾	1⅛	3
10 x 16	14¼	12	¾ x 3⅝	⅞ x 3⅝	2	1 3/16	3
12 x 19	17	12	¾ x 3¾	⅞ x 3¾	2	1¼	3½
14 x 21	18¾	12	⅞ x 4¼	1 x 4¼	2	1⅜	3½
15 x 22¼	20	16	⅞ x 4¼	1 x 4¼	2	1⅜	3⅝
16 x 23½	21¼	16	⅞ x 4¼	1 x 4¼	2¼	1 7/16	3¾
18 x 25	22¾	16	1 x 4¾	1⅛ x 4¾	—	1 9/16	3½
20 x 27½	25	20	1 x 5	1⅛ x 5	—	1 11/16	3¾

TABLE IV.—SCHEDULE OF STANDARD FLANGES FOR EXTRA HEAVY STEEL PIPE, FITTINGS AND VALVES ADOPTED JUNE 28, 1901, BY VALVE AND FITTING MANUFACTURERS. SUITABLE FOR PRESSURE FROM 125 TO 250 POUNDS PER SQUARE INCH.

Size of pipe, inches.	Diameter of flange, inches.	Thickness of flange, inches.	Diameter of bolt circle, inches.	Number of bolts.	Diameter of bolts, inches.
2	6½	⅞	5	4	⅝
2½	7½	1	5⅞	4	¾
3	8¼	1⅛	6⅝	8	⅝
3½	9	1 3/16	7¼	8	⅝
4	10	1¼	7⅞	8	¾
4½	10½	1 5/16	8½	8	¾
5	11	1⅜	9¼	8	¾
6	12½	1 7/16	10⅝	12	¾
7	14	1½	11⅞	12	⅞
8	15	1⅝	13	12	⅞
9	16	1¾	14	12	⅞
10	17½	1⅞	15¼	16	⅞
12	20	2	17¾	16	⅞
14	22½	2⅛	20	20	⅞
15	23½	2 3/16	21	20	1
16	25	2¼	22½	20	1
18	27	2⅜	24½	24	1
20	29½	2½	26¾	24	1⅛
22	31½	2⅝	28¾	28	1⅛
24	34	2¾	31¼	28	1⅛

off in the lathe in order that the face of the flange will be perpendicular to the axis of the pipe. Pipe over 18 inches in size cannot be threaded so the flanges are riveted on pipes over that size, or else shrunk on. In the latter case an accurately bored flange is heated and forced on the end of the pipe. Sometimes rivets are used with smaller pipe than 18 inches but more often with larger sizes. Cast-iron flanges riveted to the pipe are apt to leak as it is very difficult to make a joint that will stay tight.

The difficulty with the flange shown in Figure 37 is the tendency to leak through the thread. This can be overcome by good workmanship, and some manufacturers have devoted a great deal

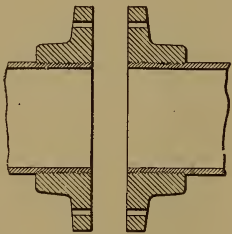


FIGURE 37.

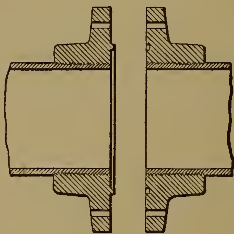


FIGURE 38.

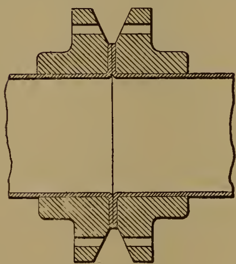


FIGURE 39.

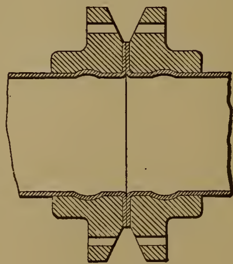


FIGURE 40.

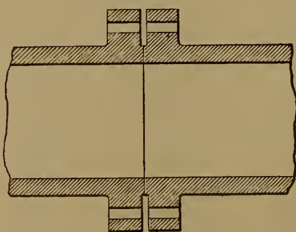


FIGURE 41.

THE ENGINEERING RECORD.

FIGURE 42.

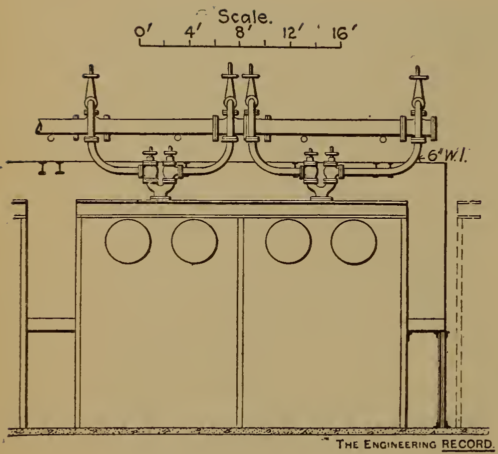
of attention to making flanges of this type for high steam pressures. This type of flange is frequently put together with a copper gasket, either in the form of a flat or a corrugated ring. There are various patented packings that have also given excellent satisfaction. In the flange shown by Figure 38, which has been used with a good deal of success, there is a circular tongue and groove, as shown, the groove containing a ring of copper as a gasket. One objection to this flange is that in places where the piping is concentrated and the connections short, it is difficult to spring the flanges apart a sufficient amount to take out a section of pipe for repairs.

The objection to any screwed flange is the tendency, unless the workmanship be of the best, for steam to leak through the joint between the flange and the pipe. A method of overcoming this leakage is shown in Figure 39. The sketch from which the cut was made was furnished by Mr. George I. Rockwood, Mem. Am. Soc. M. E., who designed the flange. The pipe is a steel boiler flue, and the flange is slipped over it, and the end of the pipe heated and flanged, as shown. The faces of the flange are cut away so as to caulk the joint. The "Walmanco" joint, made by the Walworth Manufacturing Company, has a recess in the flange in which the pipe is expanded as shown in Figure 40. This joint has been used successfully to a considerable extent.

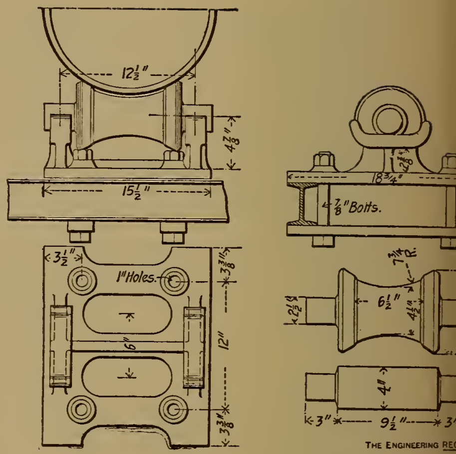
Cast iron pipe with flanges cast on the ends as shown in Figure 41 have been used by some engineers but the material is so treacherous in nature that its use is to be deprecated except for feed mains where the water is bad.

Wrought pipe with forged wrought flanges welded on the end of the pipe, shown in Figure 42, introduced within the last two or three years, seems destined to be used considerably in the future. The process of manufacture has been developed by the National Tube Company, which stated that in making the pipe a forged flange is bored out and forced on to the end of the pipe. It is then heated in a furnace and welded by means of a hammer. The flange is then faced and the bolt holes bored out. The piping is usually put together with a corrugated copper gasket, and is made in various sizes from 6 to 36 inches in diameter.

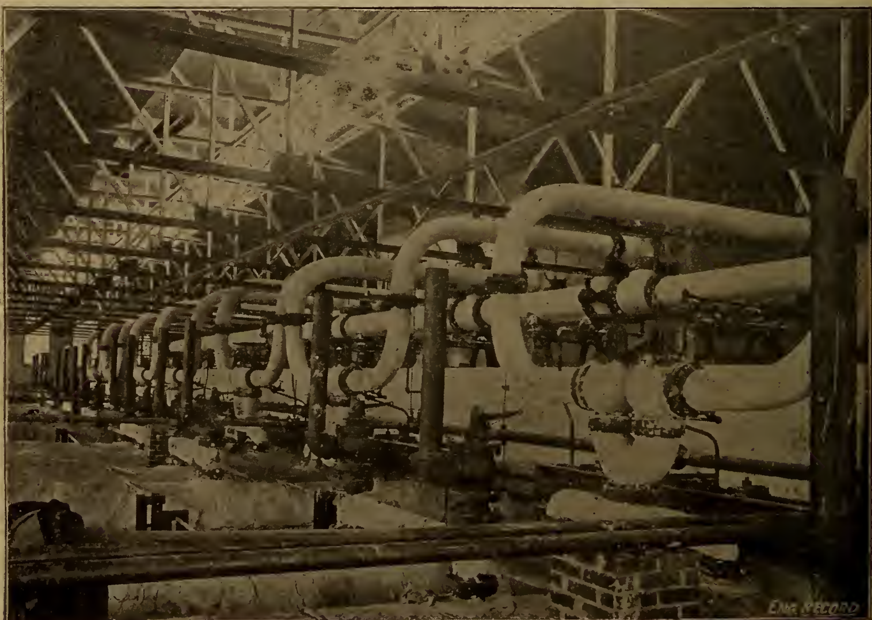
Covering pipes.—Steam pipes should be covered, for the double purpose of saving the latent heat in the steam that would otherwise be lost in condensation, and also to prevent the engines from being damaged by this water of condensation. With coal at $3 per ton, 10 feet of uncovered 6-inch pipe will cause an annual loss under average conditions of over $5 per year. A good covering that would reduce this loss to $1 per year would only cost about $5. From this it will be seen that it pays to buy the best



ELEVATION OF BOILER PIPING.



DETAIL OF PIPE SUPPORT.



METHOD OF CONNECTING DUPLICATE MAINS TO BOILERS.

FIGURE 43.—STEAM PIPING DETAILS AT GREAT NORTHERN PAPER CO.

SHEAFF AND JAASTED, ENGINEERS.

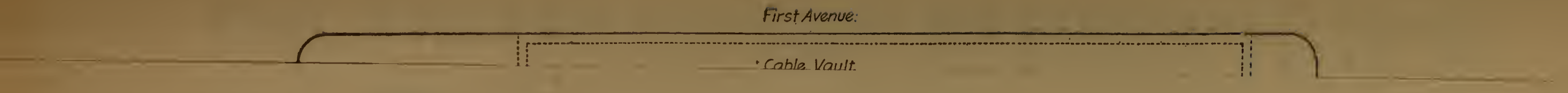
covering obtainable, and in making a selection the question of durability should be looked into fully as much as the efficiency of the covering when new. Sometimes the pipe covering is made part of the piping specifications, and again a contract is made for it with a manufacturer direct. The latter arrangement is much to be preferred.

Specifications for piping.—When an engineer prepares accurate and complete scale drawings of the entire system of pipe-fittings, etc., the written specifications can be quite brief. They should call for bids on the engineer's drawings, name the locality of the power house, give the time allowed to complete the work, and specify by name or make the following, if they are to be used, and are not marked on the drawings: Steam valves; water valves; reducing valves; back-pressure valves; steam traps; injectors; kind of packing used between the flanges; kind of pipe covering. The method of supporting pipes should be described, if it is not indicated in the drawings. The kind of fittings and pipe wanted should be clearly stated.

Chapter VII.—Condensers.

The purpose of attaching a condenser to a steam engine is to remove part of the pressure of the atmosphere, by creating a partial vacuum, from one side of the piston, and thus increasing the effective pressure acting upon the other side. If an engine is running without a condenser, with a mean effective pressure of 40 pounds, and a condenser is added removing 12 pounds from the back pressure caused by the pressure of the atmosphere, the mean effective pressure would be increased, theoretically, to 52 pounds, and the power would be correspondingly increased. If, however, the work done remains the same, the increased mean effective pressure, 52 pounds, can be reduced to 40 pounds by cutting off the steam earlier in the stroke and thus save steam nearly in the proportion that the cut-off is reduced. A condenser can, therefore, either increase the capacity of an engine, or it will reduce its steam consumption per horse-power if the load and other conditions remain the same. If an engine is operated non-condensing with the most economical load for it and a condenser be fitted to the engine, this same load will no longer be the most economical one, although the steam consumption per horse-power per hour will probably be considerably less with the condenser than without it. A condensing engine should have a larger cylinder than a non-condensing engine to do the same work, if both are to run with the lowest possible steam consumption per horse-power. The method of determining proper cylinder dimensions for engines running condensing and non-condensing was given in Chapter IV.

A condenser provides means for bringing cool water into contact with exhaust steam or passing it through thin tubes around which the exhaust steam passes, so that the latter is condensed. A vacuum is formed in the condenser by reason of this condensation, and thus part of the pressure of the atmosphere is removed. Because of the small volume occupied by the condensed steam relative to its volume as steam at the same temperature, condensed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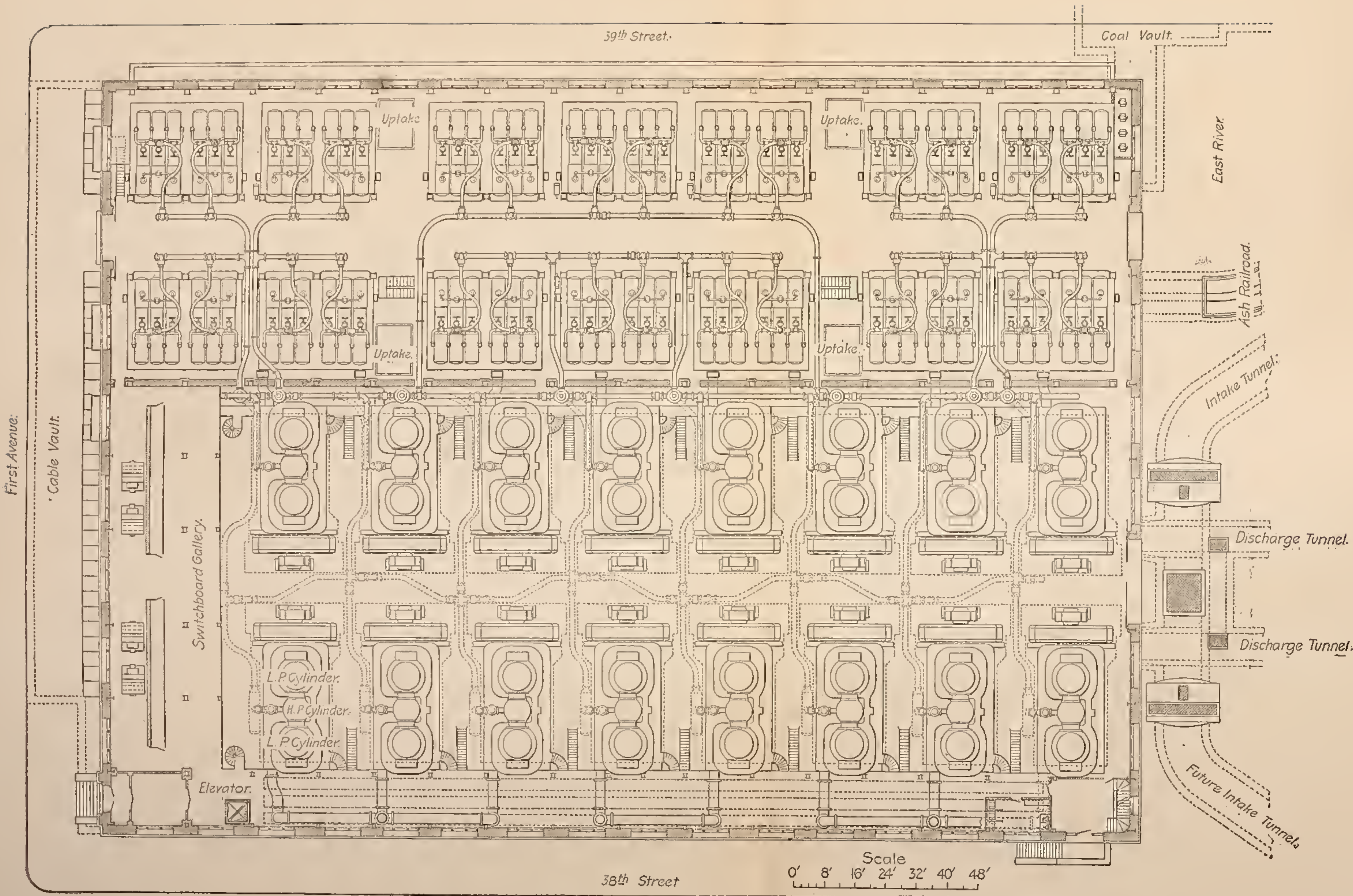


PLATE 9.—PLAN WATERSIDE STATION, NEW YORK EDISON COMPANY.
JOHN VAN VLECK, CONSTRUCTING ENGINEER.

steam or water may be removed from the condenser by a comparatively small pump using a small amount of power to operate it. This pump also removes such air as may be in the exhaust steam, hence it is usually called an air pump, although it serves to remove the water as well.

Saving due to condensers.—Generally speaking, a condensing engine will use from 75 to 80 per cent. of the steam required by a non-condensing engine. Power, however, is required to drive the air pump, and this saving is, therefore, not a net gain. The steam required to drive an independent steam-driven air pump is from one to four per cent. of that used by the main engine, depending on the size of the latter. This use of steam for driving the air pump need not necessitate a loss, if the exhaust steam from the pump is used to warm the feed water before the latter is delivered to the boilers. If the feed water is drawn from the air pump discharge or if it is fresh water heated in a feed water heater placed in the exhaust pipe between the engine and condenser, 110 degrees Fahr. is about as high a feed temperature as can be obtained on account of the vacuum in the condenser and the correspondingly low temperature of the exhaust steam. It is, therefore, advantageous to use the steam exhausted by the air pump, for feed-water heating in an auxiliary heater, so called because it receives the exhaust steam of the auxiliaries of the plant, such as the air and boiler feed pumps. This heater is sometimes called a secondary heater when a primary heater is placed in the exhaust pipe of the engine. By means of the auxiliary heater the feed water may be raised from 110 to about 170 degrees in a large well-proportioned plant, and to a greater extent in smaller plants, as the steam required for the air pump and boiler feed pump becomes a greater proportion of that used by the main engine. If the feed water is drawn from a city supply or reservoir and is not passed through a heater in the exhaust pipe of an engine, the exhaust from an air pump and boiler feed pump will raise it to almost the same temperature that it would if the feed water was at a temperature of 110 degrees.

With a non-condensing engine the feed water may be heated by the exhaust steam in a feed-water heater to within about 10 degrees of the temperature of the steam or to about 205 degrees with steam at atmospheric pressure; consequently, with condensing engines, the saving in coal used by the main engine due to a condenser over the use of the same engine running non-condensing is partly counterbalanced by the higher feed temperature that may be secured when running non-condensing. With feed water taken from the condenser discharge at a temperature of 110 degrees, there would be a saving, with a steam pressure of 100 pounds, of about 9 per cent. if the feed water could be further raised in temperature to about 205 degrees, as it probably could be if the engine was run non-condensing. However, by using the exhaust steam from the air and boiler feed pumps of a condensing plant to warm the feed water taken from the air pump discharge, there will be such a small difference between the final feed temperatures in the two types of plants that the slight difference can be neglected in considering the saving due to the use of a condenser. It is impossible, and perhaps unnecessary, to figure the exact saving a condenser will produce, but, generally speaking, it may be taken at about 20 per cent., hence plants are almost always fitted with these auxiliaries when condensing water is to be had, and the exhaust steam is not needed for heating or for some manufacturing purpose. When an abundant supply of condensing water is not available, steam plant owners often go to considerable expense for artificial cooling devices, so great is the economy in using condensers.

Types of condensers.—Condensers may be divided into three general types, known as the jet, the surface and the siphon condensers. The jet and surface condensers require an air pump, and each of these may be again divided into direct-driven or independent condensers, the former when their air pumps are driven by the main engine either by being directly connected to it or connected by a belt. The independent condensers are usually driven by steam cylinders forming part of their equipment. Condensers with air pumps driven directly from the main engine have been almost abandoned for stationary engines, the satisfactory manner in which the independent condensers have been perfected and their superiority over the direct-driven condenser having brought this about. The greatest advantage of the independent air pump is its greater flexibility, it being possible to change the speed of the separately driven type without regard to the speed of the main engine, so that it can be run to suit any condition of load.

The jet condenser usually consists of a pear-shaped chamber,

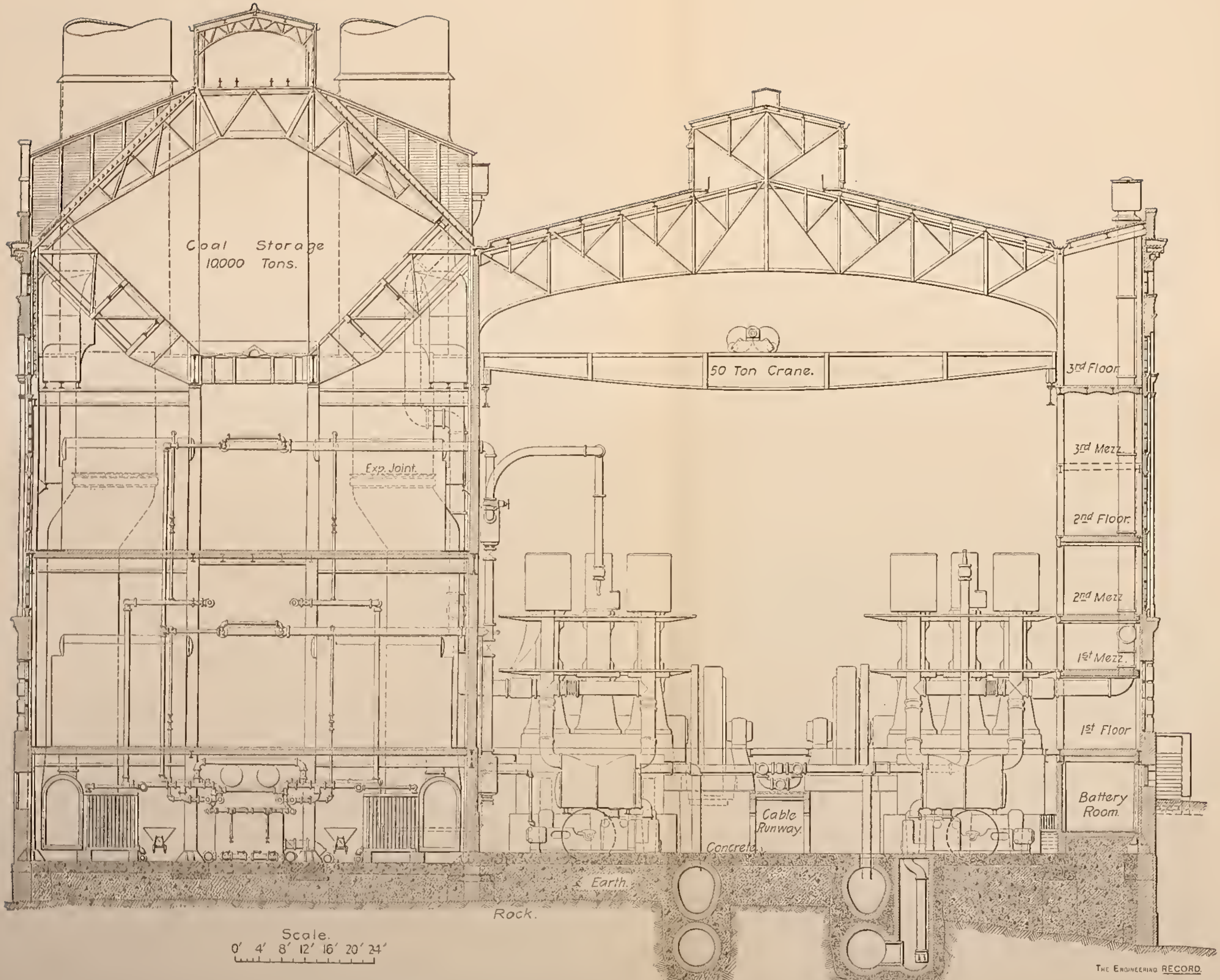


Plate 10.—Cross-Section Waterside Station, New York Edison Company.

at the top of which the exhaust steam enters, while at one side is a connection for the injection or condensing water. The bottom of the chamber has a contracted neck connecting with an air pump which is sometimes very similar to the ordinary direct-acting steam pump, or, it may be an air pump of special design. A cross-section of the Worthington jet condenser with horizontal double-acting air pump is shown in Figure 44.

The surface condenser consists of a shell, usually of cast iron, containing a large number of brass or copper pipes through which the condensing water is circulated, the exhaust steam being ad-

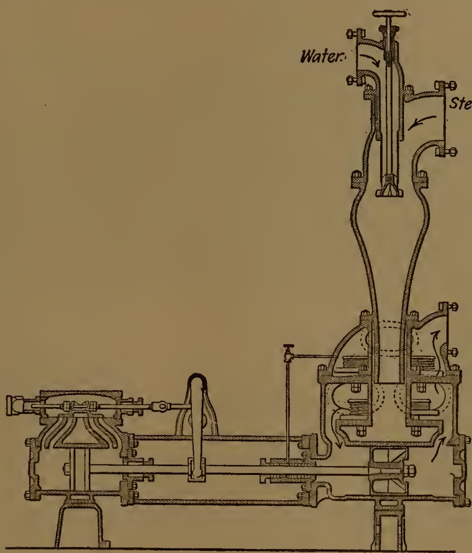


FIGURE 44.—JET CONDENSER.

mitted to the shell in the space surrounding the pipes. The steam coming in contact with the cooled pipes is condensed, and the condenser is so connected with the air pump that the condensed steam flows by gravity to the air pump, by which it is removed to maintain the vacuum. If the cooling water is under pressure no pump is necessary to circulate it through the condenser. If not, a circulating pump is required and this may be driven by the same steam cylinder that operates the air pump. It is usually the practice to place the air and circulating pump and the steam cylinder operating them in line, tandem, beneath the condenser and on a base supporting the whole. An arrangement of this kind showing the Wheeler method of mounting a surface condenser on the cylinders of a Knowles' air and circulating pump is shown in Figure 45. Circulating water is sometimes supplied by centrifugal pumps driven by an engine or electric motor. As the condensed steam does not come in contact with the circulating water in the surface condenser, this type can be used when the circulating water is of such a character that it should not be fed to the boilers. The condensed steam is used over again until it becomes so impregnated with cylinder oil from the engine that

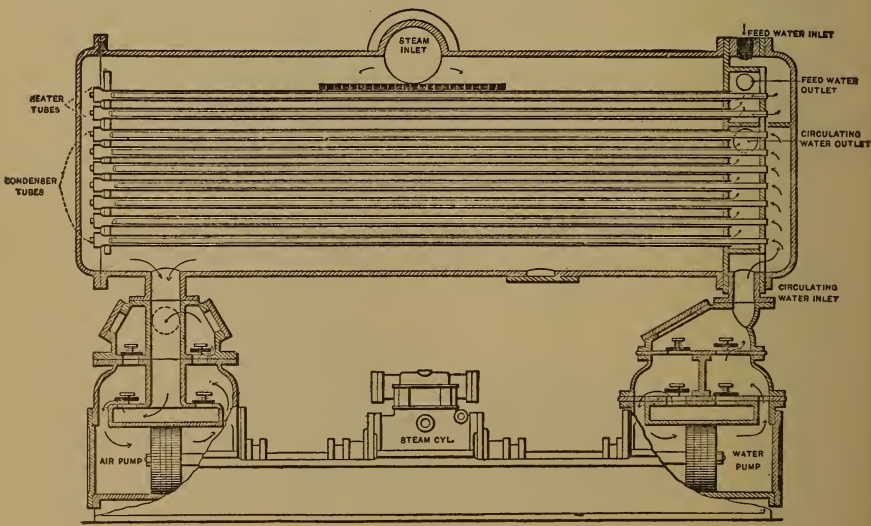


FIGURE 45.—SURFACE CONDENSER.

it has to be replaced by fresh water. The presence of cylinder oil in feed water causes trouble in steam boilers, and it is the fear of this trouble from oil that prevents the wider adoption of the surface condenser in power plants. However, by providing proper filters of sand, cloth, sponges, excelsior or similar material for the feed water and properly looking after them there is no reason why the surface condenser cannot be used with success.

The Bulkley siphon condenser, which in a general way is similar to others of this type, is shown in Figure 46. The steam from the engine is led in a pipe to the top of the condenser, which is elevated sufficiently to be placed about 34 feet above the surface of a hot well into which the condenser discharges. The injection water enters at the side and mingles with the steam at the lower edge of the cone shown. By contracting the neck of the condenser below the cone, sufficient velocity is given the water in falling to the hot well to maintain a siphon-like action that draws the steam from the exhaust pipe, and causes a vacuum to exist in it. The injection water can be supplied by a pump of either the steam-driven or the centrifugal type. If the injection water can be had under sufficient pressure, the pump is not necessary. This type of condenser will lift water from a source of supply, such as a tank or reservoir, through a height of 18 feet or less, but with this arrangement the siphon must be started. This can be done by running a horizontal pipe from the reservoir or tank across to a tee in the vertical discharge pipe of the condenser. Water flowing through this and down the discharge will gradually exhaust the air from the upper part of the discharge pipe until sufficient vacuum is formed to draw the water up to the condenser and start the water flowing through it. When this is done a valve in the cross-connection or starting pipe is closed.

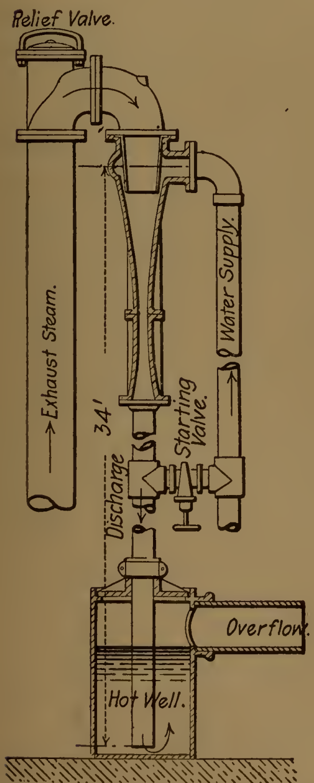


FIGURE 46.—SIPHON CONDENSER.

Figure 47 shows an elevation of the Worthington central jet condenser which is a modification of the siphon type. It is called the central system because it is often used to condense the steam of more than one engine. An enlarged cross-section of upper part is shown in Figure 48. The condensing cone is considerably larger than in other siphon condensers, the idea being that the steam and water are more thoroughly mixed, thus using less water. The spray is controlled by an adjustable nozzle. The neck of the condenser is not contracted, as it is in the ordinary siphon condenser, to give the high velocity to the descending water necessary to suck the air down into the discharge or tail pipe. The contracted neck was avoided to give the water an unrestricted fall so that it could not by any chance back up in the condenser and run over into the exhaust pipe. To assist in removing the air from the condenser cone an air pipe, shown in Figure 48, is led to a dry vacuum pump, which constantly removes air from this pipe and very considerably increases the vacuum over what would be obtained without it. The air cooler shown in the drawing is used to cool the air while it passes from the condenser cone to the vacuum pump, the air passing through tubes in the cooler around which the injection water is circulated. By cooling this air its volume is reduced, hence the vacuum pump has a little less work to do.

Location of condensers.—It is important to locate a condenser of the jet or surface type on a lower level than the engine, so that the pipe connecting the engine and condenser will be either perfectly horizontal or pitch slightly toward the condenser. This is necessary to prevent the existence of a pocket where water can collect in the pipe line. As this pipe is under a vacuum, water due to condensation that collects in it cannot be drawn off by a trap. If a pocket does exist, there is chance, in the event of broken vacuum, of this water getting back into the cylinder and wrecking it. One occasionally sees a condenser located on the floor with an engine. When this is done the exhaust pipe from the engine drops from the bottom of the cylinder, then runs horizontally, then rises to the condenser, thus forming a pocket, to which objection has been raised; this arrangement should be avoided.

There being a vacuum in the condenser, the injection water can be lifted from a reservoir or pond at a lower level. It is not advisable to lift water through a greater head than 20 feet, including the actual vertical distance the water is raised and also the friction of the water in the injection or suction pipe. A condenser is sometimes located in a pit considerably below the engine in order that the lift may not be too great. As stated in the first chapter of this series, the availability of a supply of condensing water often determines the location of the power house. If a river or reservoir is nearby and it is not desirable to locate the power house close to the river, water can be led to a well close to the condenser either in an open trench or in a tunnel or conduit

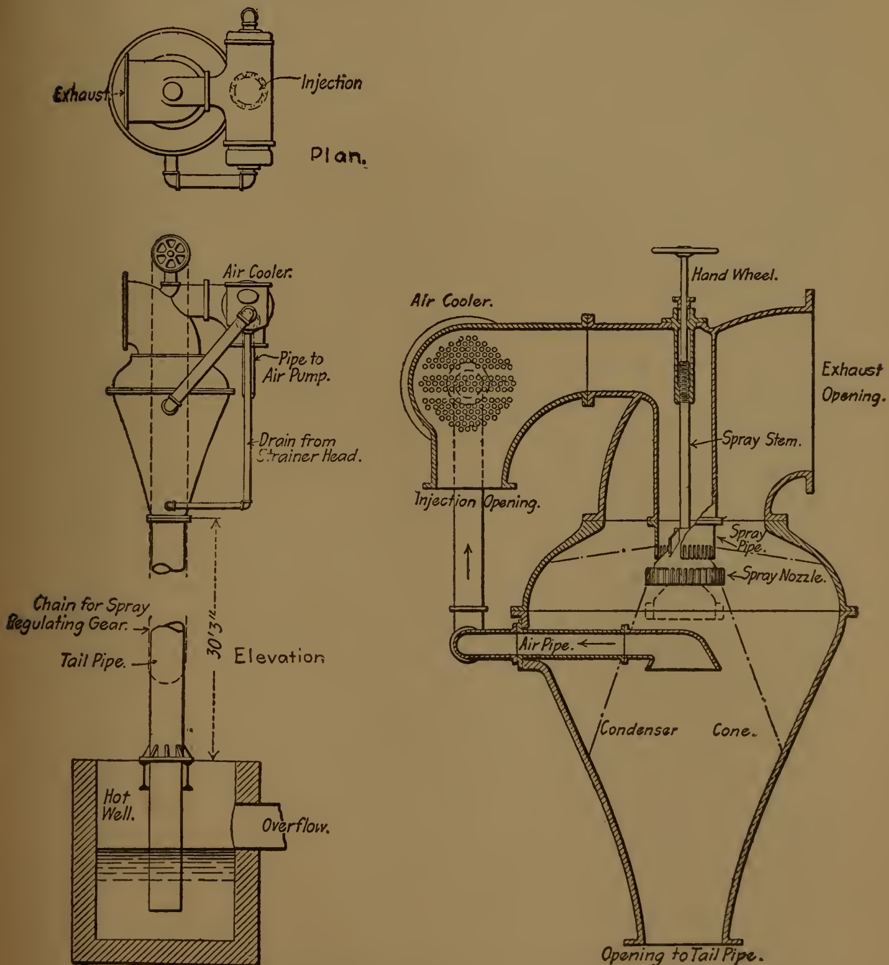


FIGURE 47.—CENTRAL JET CONDENSER.—FIGURE 48.

and the injection pipe run to the well. This would overcome the use of a long injection pipe. If the injection pipe is run underground it should not be covered over with earth until after the condenser is started and the pipe tested for air leaks. One of the most frequent causes of trouble with a condenser is a leaky suction pipe, hence the greatest care should be used to make sure that it is perfectly tight. Various methods of connecting condensers with engines were given in the chapter on "Steam Piping."

Water necessary for condensers.—A considerable amount of water must be available if a condenser be used, and to calculate this, one must first find the weight of water necessary to condense each pound of exhaust steam and then determine the amount necessary to condense all the steam exhausted by the engine. The former is found by dividing the rise in temperature that takes place in water used to condense the steam into the heat contained in one pound of steam at the pressure at which the exhaust valve in the engine begins to open less the heat in one pound of water at the temperature of the air pump discharge. Expressed algebraically, this equation is: $W = (H - h) \div (T - t)$; in which H is the total heat in steam at the terminal pressure, h is the heat in water at the temperature of the air pump discharge; T is the temperature of the discharge condensing water and t is the temperature of the entering condensing water. The value of H for different pressures and temperatures may be found in tables giving the properties of saturated steam, and may usually be taken at 1,150. Taking average values for a surface condenser with h at 120, T at 110, and t at 70, then W will be found to have a value of about 26 pounds per hour. If a jet condenser is used the condensed steam and air pump discharge would have the same temperature. When the same water is used over and over again to condense with, as is done with cooling towers or cooling reservoirs, the temperature of the condensing water is quite high when it enters the condenser, so that each pound can absorb a comparatively small amount of heat, hence a correspondingly greater volume of condensing water is required.

In estimating the quantity of injection water necessary, due consideration should be given the amount of steam exhausted by the engine during maximum load; also the increase in the steam consumption of the engine per horse-power that will occur when the engine has been in service for some time, due to the leakage of valves and piston on account of wear. The author believes in being very liberal in selecting condensers, for a good vacuum, particularly with engines operating with low mean effective pressures, is conducive to high economy. Some engineers hold that a high vacuum is a mistake, one of the reasons being that with it the temperature of the condenser discharge is lower and consequently the feed water will not be so hot. There is, however, a loss due to reducing the mean effective pressure acting on the engine piston that considerably more than offsets such a gain. By a decrease in the vacuum from 26 to 24 inches it would be impossible to increase the feed water temperature more than 15 degrees, and this would mean a saving of a little more than 1 per cent. This decrease in vacuum in a compound engine, with 125 pounds steam pressure, most economically loaded would effect, theoretically, an increase in the steam used of about 5 or 6 per cent., and with a simple engine about 2.5 per cent. It will be noticed on investigating duty trials of large pumping engines where high duty is desired that a very high vacuum is sought by the builder.

Sources of water supply for condensers.—Water for condensing purposes may be obtained from rivers, in which event the water goes to waste after it is discharged from the condensers, or from ponds or artificial reservoirs. When drawn from ponds or artificial reservoirs the source of supply has to have sufficient volume and surface so that it will be cooled naturally by the air. If the reservoir is too small for this it must be cooled artificially by some method that exposes the water sufficiently to the air to cool it, as in the cooling tower. Reservoirs where the cooling is done naturally require surface sufficiently large to allow enough water to come in contact with the air to remove the necessary amount of heat from it, and they should also have sufficient volume, if they are used continuously, so that the water can remain in the reservoir long enough to be cooled before being used again. Cooling reservoirs of this kind are rendered much more efficient by dividing them by partition walls so that the water is compelled to travel some distance in passing from the condenser discharge to the intake where it is drawn from the reservoir to return to the condenser. This is to prevent the discharge from the condenser from immediately entering the intake, or short-circuiting, before it has time to cool.

There is very little reliable information on the surface and volume required in cooling reservoirs. Thomas Box, in his work on "Heat," says that when an engine works day and night the depth of the reservoir is unimportant, and that 210 square feet of surface is required per horse-power. When an engine runs only 12 hours per day the surface may be reduced to 105 square feet per horse-power, but in the latter case the depth of the reservoir must be such as to give 300 cubic feet per horse-power. Box assumes the use of one cubic foot or 62½ pounds of water evaporated into steam per horse-power. These data are based on experiments when the water was reduced in temperature from 122 degrees Fahr. to 82 degrees, the air being at a temperature of 52 degrees and the humidity 85 per cent.

Cooling towers.—These appliances provide means for artificially cooling condensing water, and their successful development has made it possible to obtain the benefit of a condenser in many plants where condensers would be out of the question without them. Usually they consist of large cylinders of sheet steel open at the top and enclosing either mats or tiles or some similar substance presenting a large surface over which the hot water from the condensers is allowed to trickle downward, from distributing pipes at the top. The water falls into a reservoir at the bottom, from which it is returned to the condenser. An opening or openings in the side of the tower allow the air to enter and pass up through it, thus cooling the water. Usually a fan driven by an electric motor or small steam engine is used to stimulate this current of air. A cooling tower can be located almost anywhere outside of a power-house, on the ground or on the roof. There is a loss of circulating water attending their use of from 10 to 15 per cent. of that passing through them, owing to the evaporation that occurs. This, however, is more than made up, with jet condensers, by the discharge from the air pump of the condensed steam. Where cooling towers are located at a higher elevation than the engine, the condensing water must be pumped from the condenser to the tower, but the power required to do this is, with surface condensers, partly counterbalanced by the fall of water from the reservoir at the base of the tower to the condenser. The column of water in the condenser discharge pipe for the height of the tower itself is, of course, unbalanced.

Proportioning condensers.—In the jet type of condenser the air pump has to remove from the condenser the injection water, the steam that is condensed, and a certain amount of air that finds its way into the exhaust steam either from the boiler or through leaks in the pipe, condenser, etc. The air is a very uncertain quantity. With a surface condenser the cooling water does not come into contact with the steam, hence the air pump has to remove only the condensed steam and the air. To determine the size of the air pump for a jet condenser, the amount of injection water in pounds per minute necessary to condense the steam exhausted by the engine should be calculated in the manner previously described, and this reduced to cubic inches per minute. This quantity should then be doubled, if the air pump is to be of the horizontal double-acting type—to allow for the air, and the number of cubic inches thus obtained should be about equivalent to the volume in cubic inches that should be displaced by the air-pump piston in one minute. This latter quantity, the displacement, is equal to the product of the area of the piston in square inches, the number of double strokes per minute and the length of stroke in inches.

With vertical single-acting air pumps the product of the area of the cylinder in square inches and the length of stroke in inches and the number of double strokes per minute, should be 50 per cent. greater than the volume of injection water in cubic inches per minute. The size of air pump necessary to do a given amount of work can be determined with the data given, from the catalogues of condenser manufacturers. These give the area of cylinders, length of stroke and maximum number of strokes that the air pump should run per minute.

With surface condensers, the air-pump displacement should be 20 times the volume of steam condensed, with horizontal double-acting air pumps, and 15 times the volume of steam condensed with vertical single-acting air pumps. Surface condensers usually have one square foot of surface in brass or copper tubes for every 10 pounds of steam to be condensed per hour.

Specifications for condensers.—Generally in purchasing jet condensers, an engineer states the horse-power and type of engine for which a condenser is wanted, but it is better perhaps if the engineer is sure it will be correct, to give the number of pounds of steam that must be condensed in a given time, and the vacuum that is desired. This is usually 26 inches. If the engineer has calculated the volume of the air-pump cylinder, bids on condensers with air pumps of approximately this volume can be obtained. His specifications should also call for a blue print or drawing showing the condenser the builder intends to supply, and on this blue print should be given the outside dimensions of the condenser, the size of the steam cylinder, the diameter and location of steam and exhaust pipe openings, the injection or suction pipe and the discharge pipe. The delivery and erection of the condenser should be provided for, if the builder is to deliver and install it.

A specification for a surface condenser with combined air and circulating pumps should ask for a print of the apparatus, the size of the air, water and steam cylinders, the size of the steam and exhaust pipes, etc., also the square feet of heating surface in the condenser proper.

Guarantees.—Sometimes a condenser manufacturer is required to guarantee its efficiency, but this is not often done. The form of guarantee that is fairest to the manufacturer would be, perhaps, to ask that the apparatus with an air pump piston speed not exceeding a specified amount, should condense a given amount of steam delivered to it at a certain pressure, and maintain a given vacuum, usually 26 inches, with a given temperature of cooling water. The location at which the vacuum is to be obtained should be stated, for frequently there is considerable drop in pressure between the exhaust pipe next to the engine and the condenser owing to a too long or too small exhaust pipe. The vacuum ought to be measured close to the condenser, for the maker of the latter is not responsible for the selection of the exhaust pipe. He may not be responsible for the size and run of the injection pipe leading cold water to the condenser, or for the lift of water in it, and as these very materially affect the efficiency of the condenser, the manufacturer should have the opportunity, if a guarantee is made, to approve the details, size and arrangement of the injection pipes. These remarks apply also to jet and siphon condensers. With a surface condenser and combined air and circulating pumps, the only guarantee that should be asked is maintaining a specified vacuum with a piston speed of the air pump not exceeding a specified amount and a specified temperature of cooling water. The manufacturer would have to satisfy himself with the arrangement of the water piping that it was proposed to use. If a complete cooling tower and condensing outfit was to be supplied the guarantee should state the vacuum that is to be obtained with a given temperature of outside air, and when that air contains a certain percentage of moisture.

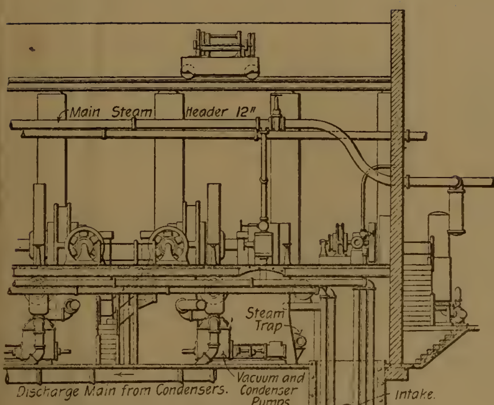
Chapter VIII.—Feed Water Heaters and Economizers.

Value of feed-water heaters.—Exhaust-steam feed-water heaters are used to heat the water fed to boilers with steam exhausted by engines and pumps, and the saving due to their use is so great that plants are seldom constructed without them. Generally speaking, for every 11 degrees that feed water is warmed there is a saving of 1 per cent. in the fuel burned. With sufficient exhaust steam available, cold feed water at 70 degrees Fahrenheit can be raised in temperature to 200 degrees, thus saving nearly 12 per cent. of the fuel. The heater will in many locations, therefore, reduce the fuel consumption enough to pay for itself in a few months, the exact time depending upon the cost of the fuel.

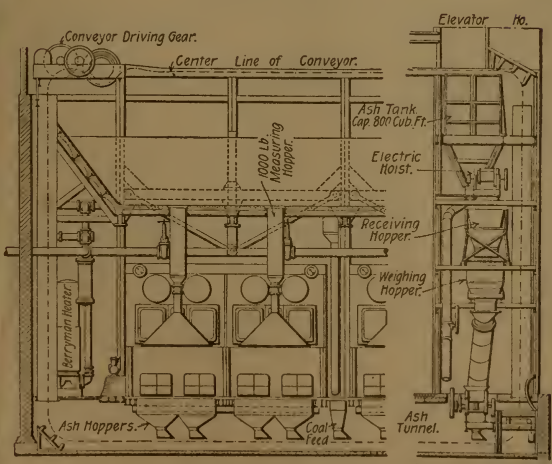
Types of heaters.—Exhaust-steam feed-water heaters are of two types, the open and the closed. In the former, the heater consists of a box-like receptacle of cast-iron or boiler steel to which the exhaust steam is led so as to fill its interior. The cold feed water is admitted at the top and in most designs trickles to the bottom over a series of trays and is thus brought in contact with the steam and is heated by it. If the water contains scale-forming salts which are precipitated at temperatures below 200 degrees, they collect on the trays, which are removable for cleaning. This type of heater sometimes has a filter in its base for further removing these precipitated salts and impurities that can be intercepted by filtration. The feed water is drawn from the bottom of the heater and pumped to the boilers. The pump must be located so as to receive the water under pressure, as it will not lift the water if it is under a high temperature. The condensation from heating systems, the discharge from traps connected to high-pressure drips, engine jackets, reheaters, etc., which are to be returned to the boilers, can be connected to a heater of this type. A certain water level is maintained in the heater, and to do this the supply of cold feed water is controlled by a valve operated by a float in the heater, so that cold water is admitted when the boiler feed pump draws water from it faster than it is supplied by the heating returns, drips, etc. It is absolutely essential that an efficient grease separator be placed in the steam pipe leading to an open heater, so that oil cannot enter it and pass on to the boilers mixed with the feed water.

Closed heaters usually consist of a cylindrical shell of cast-iron or boiler steel containing tubes extending from one head to the other, or coils of pipe. Sometimes the exhaust steam is admitted to the shell so as to surround the pipes or coils containing the feed water, in which event the heater is said to be of the water-tube type. If the steam is inside of the coils with the water surrounding them, it is a steam-tube heater. Closed feed-water heaters are usually supplied with a steam inlet and outlet, although they are sometimes arranged with only a single connection, reliance being placed on the vacuum that forms in the heater when steam is condensed to cause more steam to flow into it. As a certain amount of air exists in the exhaust steam, this will find its way into the heater and is apt to impair its efficiency. For that reason many engineers believe that there should be a double steam connection, unless some means of removing the air is provided, to prevent air from accumulating and thus to insure a thorough circulation of steam through the heater. Experience has shown in water-tube heaters that the best results are obtained when the water is compelled to pass through the tubes successively for the reason that the velocity of the water per unit of heating surface is greater. Closed heaters are made to rest in a horizontal or vertical position. The latter are to be preferred if it is convenient to use them as they take up much less floor area and the circulation of water in them is much more thorough unless the tubes are subdivided into groups.

Uses for heaters.—The manner in which feed-water heaters are used depends upon the type of plant in which they are installed. With non-condensing engines, where the exhaust steam is available for feed-water heating, the feed water may be raised to a temperature of about 205 degrees Fahrenheit; and when this is done, it is usually the custom to allow one square foot of heating surface in brass or copper pipes for every 90 pounds of water passed through the heater per hour. With condensing engines it is sometimes the practice to place a feed-water heater of the closed type in the exhaust pipe of the engine, unless two different



Longitudinal Section.



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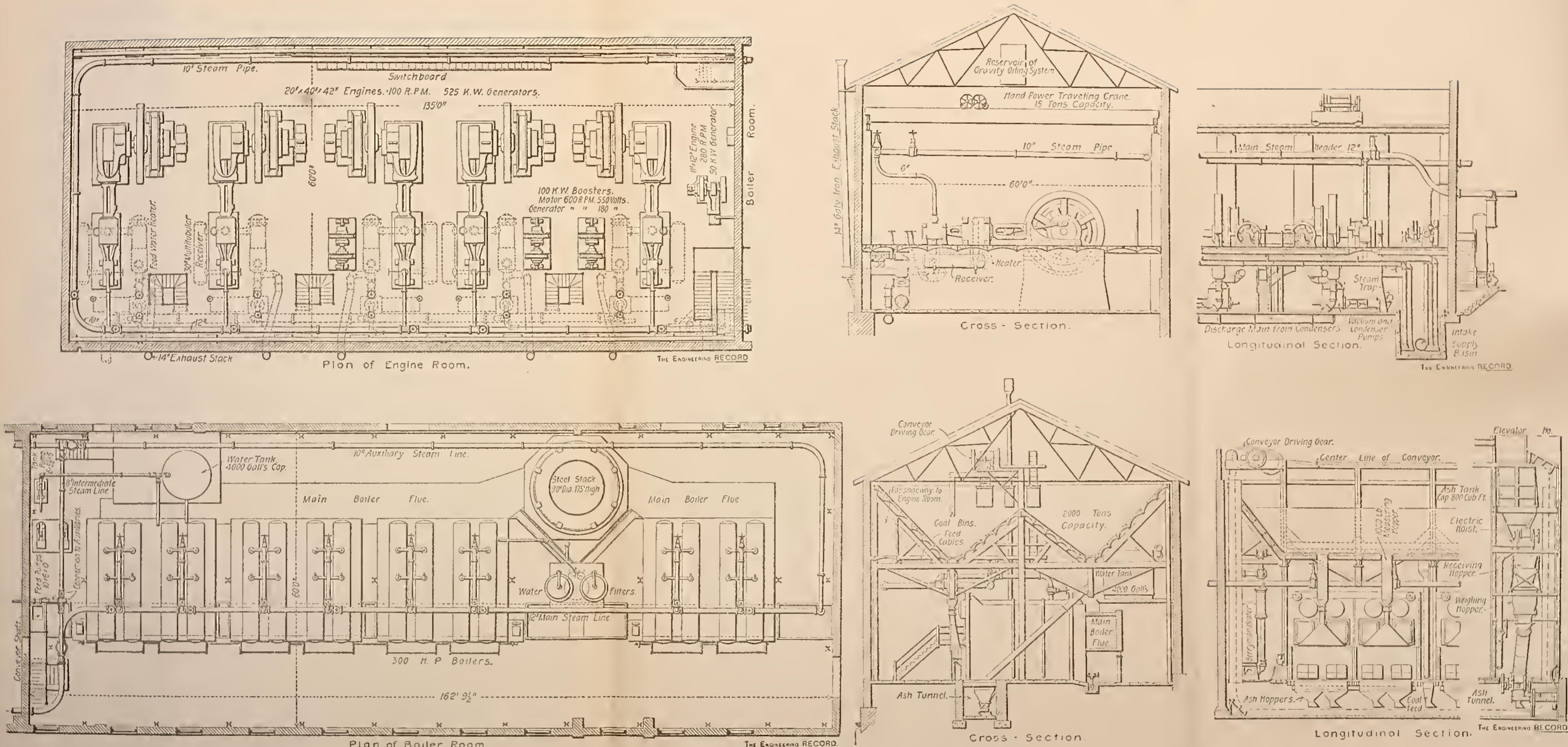


PLATE 11.—POWER STATION, CAPITAL TRACTION COMPANY, WASHINGTON, D. C.

D. S. CARLL, CHIEF ENGINEER.

sets of conditions exist. If the air pumps and boiler-feed pumps are steam driven, and the steam exhausted by them added to the steam exhausted by all other pumps or steam auxiliaries in the plant amounts to one-seventh or more of the steam generated in the boilers, this exhaust steam from the auxiliaries can be led to an auxiliary feed-water heater and the feed water warmed by it from about 75 to about 205 degrees. In this event, a heater in the exhaust pipe of the engine would be of no value; nor would it, if the feed water were drawn from the condenser discharge, for in that event it would be at nearly as high a temperature as the exhaust steam from the main engine, so that it would hardly pay to attempt to use this steam for heating. The exhaust steam from the main engine being under a partial vacuum, say 26 inches, would have a temperature of about 125 degrees, and the condenser discharge probably about 100 degrees. If the exhaust steam from the auxiliaries be not sufficient in quantity to raise the feed water from a temperature of 60 or 70 degrees to about 205 degrees, then a feed-water heater can be placed in the exhaust pipe of the engine and the water warmed from the lower temperature to about 115 degrees and finally passed through an auxiliary heater and there warmed to as high a temperature as possible with the auxiliary exhaust steam available. This auxiliary heater is almost invariably used in the latest plants while only occasionally is a heater placed in the main exhaust pipe of a condensing engine. Auxiliary heaters of the closed type should have the same amount of heating surface as heaters receiving steam at atmospheric pressure. This rating was previously given. Heaters placed in the exhaust pipes of condensing engines should have more surface, usually one square foot for every 60 pounds of water passed through the heater per hour, this being necessary because the difference between the temperature of the steam and the mean water temperature is less than it is with heaters using steam at atmospheric pressure.

Condensation in heaters.—The condensation that occurs in a feed-water heater amounts to about one-seventh of the weight of the feed water passing through the heater, when the water is raised from 70 to about 205 degrees. In an open heater this, of course, mingles with the feed water and passes on to the boilers with it. In a closed heater, the steam not being in contact with the water, the condensation occurring has to be removed from the heater, and this can be done by leading it to a steam trap. It should not be forgotten that the steam pressure in a heater is frequently at practically the same pressure as the atmosphere, hence the discharge from the trap should be carried downward, or on a level rather than upward, as there is no pressure to lift the water. It is not necessary to save the condensation in a closed heater unless the cost of this water which, as has been said, is about one-seventh of the total feed water, is sufficient to make the saving of it advisable; or unless there is not enough exhaust steam to warm the feed water to as high a temperature as may be possible. If there is plenty of exhaust steam, that would otherwise go to waste, available for heating, the heat in this condensation in the heater is of no value. If the condensation from a closed feed-water heater is to be returned to the boilers, it should be trapped, and the discharge from the trap led to a receiving tank combined with a pump controlled by the level of the water in the tank. Other drips from cylinder jackets, high-pressure steam pipes, reheating receivers, etc., can also be led to this tank, each discharging through a trap, or the condensation can be returned to the boilers direct by the steam loop or the Holly system.

Purchasing heaters.—In purchasing a feed-water heater for a power plant for a manufacturing or electric company, the heater is usually purchased by the owner direct, although in some plants, notably those in office buildings, the purchase, delivery and installation of the feed-water heater is frequently included in the specifications for the piping. The writer believes that it is much better to make the purchase of the heater a separate contract. The specifications for a closed heater should state the amount of heating surface required, the kind of metal that the tubes are to be made of, whether it is to be of the steam or water-tube type, and horizontal or vertical. If the supplying and installation of the heater is to be made part of the contract for the steam piping or some other part of the plant, the contractor undertaking it should be required to deliver and erect it. If the heater is to be purchased direct from its manufacturers, the specifications should ask that the bids cover the delivery of the heater free on board cars at the nearest railway point to the power plant. The specifications should ask each bidder to furnish a blue print or drawing showing the details of the construction of the heater it is pro-

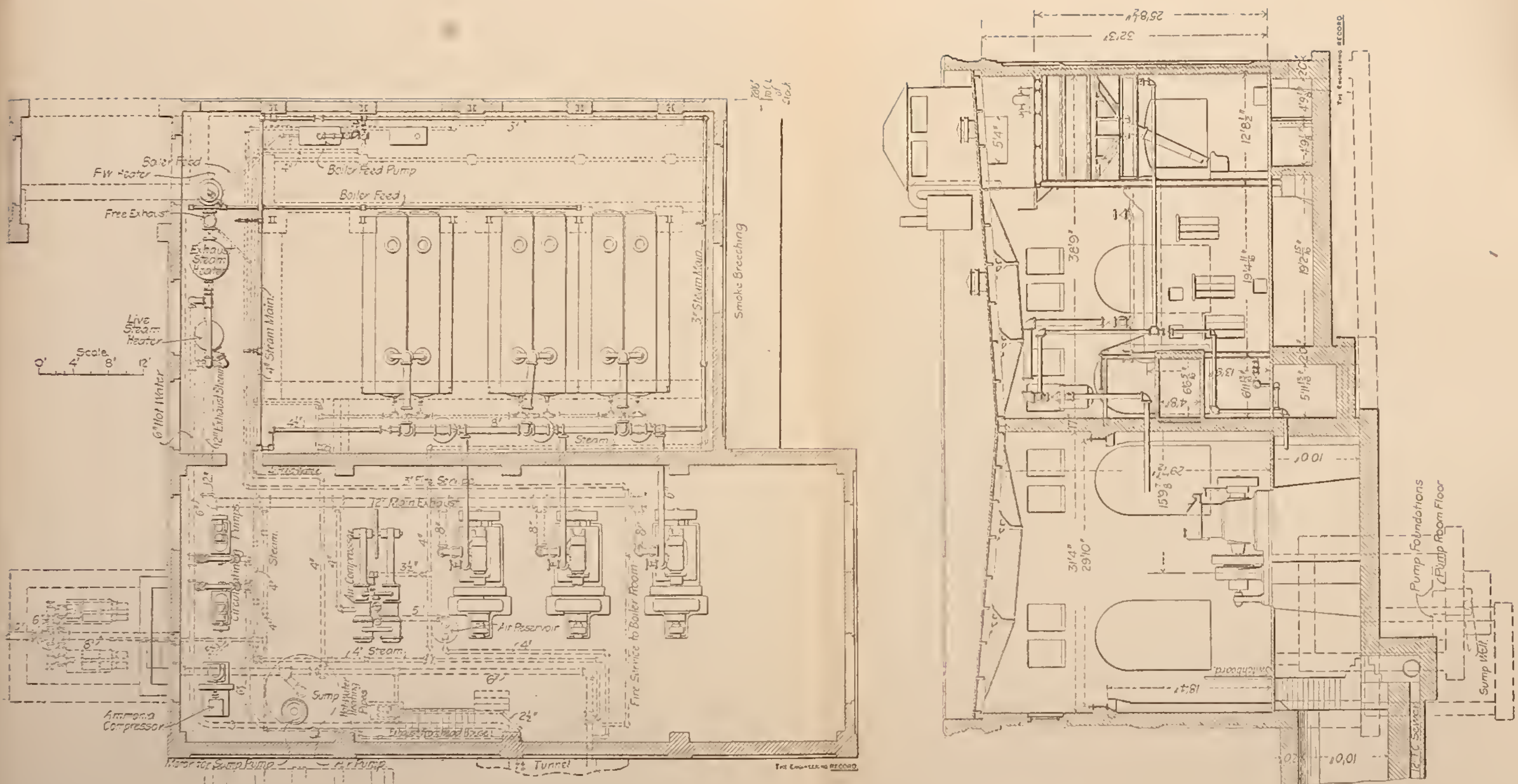


Plate 12.—Power Plant, Pittsburg and Lake Erie Railroad, Pittsburg, Pa.

J. A. Atwood, Ch. Eng'r, Westinghouse, Church, Kerr & Co., Eng'rs and Contr's.

posed to furnish, its exact dimensions and the location of the steam and water outlets and their sizes.

There is no general rule that the writer is aware of for proportioning heaters of the open type. Therefore a specification for a heater of this kind should state the quantity of water to be passed through the heater in a given time and also the initial temperature of the water and the final temperature desired. A print or drawing of the heater a bidder is to provide should be obtained and the inside dimensions of the heater, the volume of the steam and water space, etc., should be investigated. The heater must not be too small, otherwise the water in passing through it cannot be broken up in sufficiently small particles to allow the water to mix thoroughly with the steam and thus be sufficiently heated by it. If a guarantee as to the efficiency of a feed-water heater is required, it should be in the following form: The maker guarantees the heater to be capable of warming —— pounds of water per hour from a temperature of —— degrees, Fahrenheit, to —— degrees Fahrenheit with sufficient steam at atmospheric pressure.

Economizers.—Economizers are used in connection with steam boilers to warm water either for boiler feeding or for some manufacturing purpose, by the heat in the gases from boilers that would otherwise go to waste. Economizers are also useful in increasing the capacity of a boiler plant already in operation, in providing means for storing a large quantity of water at high temperatures, which is of advantage in the event of a sudden increase in the demand for steam. They also deliver the water to boilers at high temperature and reduce strains in boilers due to the admission of cold water. One disadvantage of economizers is that they reduce the draft slightly owing to the friction of the gases passing through them and to the reduction in temperature of the gases. Provision should be made for this by the use of mechanical draft or of larger chimneys than would be necessary if economizers are not used. In plants already constructed the addition of economizers reduces the coal consumption, and this counterbalances in a measure the loss in draft, as less draft is necessary with them because less fuel is burned.

Economizers consist of vertical cast-iron pipes about 4 inches in diameter and 9 feet long placed in rows several inches apart, each row being connected at the top and bottom to cast-iron headers through which the water is supplied and withdrawn. They are provided with scrapers that encircle the pipes and that are continuously raised and lowered by a suitable mechanism, the power coming usually from a small steam engine. Economizers are placed in brick flues between the boilers and the chimney, and

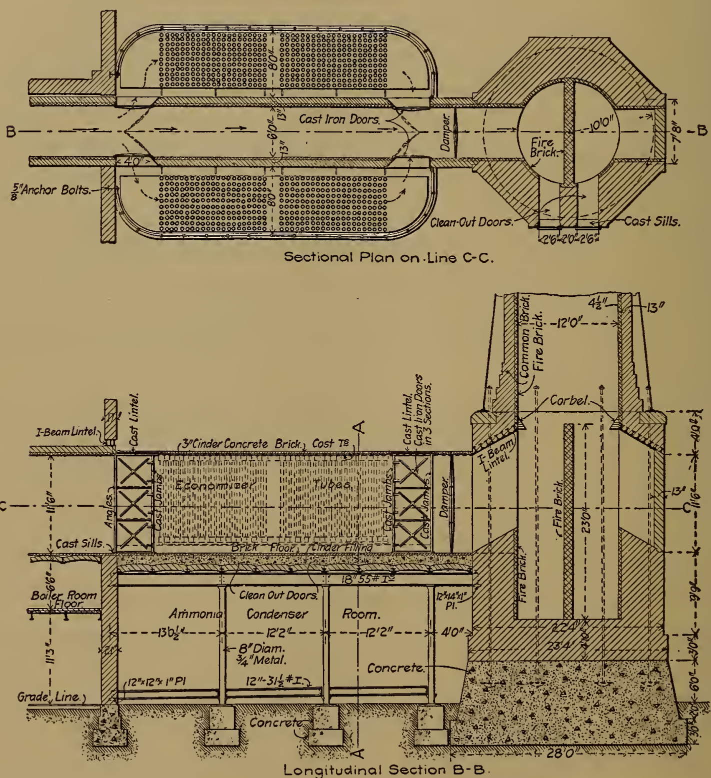


FIGURE 49.—ECONOMIZER ARRANGEMENT, PLANT OF SCHWARZCHILD & SULZBERGER, CHICAGO, ILL.

L. LEVY, CHIEF ENGINEER.

a by-pass flue must be provided for use when the economizer is out of service, either for cleaning, which is necessary at intervals, or for repairs. Economizers are used to a very considerable extent in Europe, more so in fact than in the United States, where engineers are now, however, rapidly appreciating their advantages. When first introduced, a number of failures occurred mainly due to improper design. They were then either constructed of poor materials improperly put together or there was a lack of sufficient heating surface necessary to make the saving that they should produce. Since the economizer has been perfected and made of durable materials, their advantages have become recognized.

The heating surface in economizers takes the place, in a measure, of additional boiler-heating surface. If a boiler is operating under certain conditions so that, with a certain rate of evaporation per square foot of heating surface, a certain temperature of waste gases follows, it would be possible to add more heating surface and abstract more heat from these gases; but the boiler heating surface would not be so efficient in doing this as an equivalent amount of economizer surface, for the reason that the average temperature of the water in the economizer is lower than that of the water in the boiler; this causes a more rapid transfer of heat from the gases to the water in the economizers than there would be from the gases to the water in the boiler. The saving that an economizer can produce increases as the flue temperature increases because there is more heat for it to absorb. An economizer can only heat the water to a given point, hence as the temperature of the entering water increases, due to previous heating, as in an exhaust steam heater, the saving possible with the economizer becomes less. If the ratio of boiler heating surface to the amount of coal burned is such that as much of the heat in the waste gases is absorbed as is possible, there is not as much heat left in the gases for the economizers to save as there would be if the ratio of boiler heating surface to the amount of coal burned were less. Barrus, in his book on "Boiler Tests," states in effect that where a boiler is operated most efficiently, the temperature of the waste gases should not exceed about 400 degrees Fahrenheit, and boilers are usually so proportioned that about this temperature exists when the highest efficiency is attained. Owing, however, to the accumulation of soot upon the boiler surface or to running at a higher rate of evaporation than the normal, the temperature of the waste gases usually exceeds that given so that an economizer is valuable in even a well proportioned boiler plant. Mr. Barrus gives some excellent data showing the saving made by economizers with low temperatures of flue gases, and they are reproduced in Table I, herewith:

TABLE I.—BARRUS' TESTS OF ECONOMIZERS.

Heating surface, boiler, sq. ft.	1,894	1,058	5,592	3,126
Heating surface, economizer, sq. ft.	1,600	1,920	1,280	1,600
Temperature of gases leaving boiler, deg.	376	361	403	435
Temperature of gases leaving economizer, deg.	231	254	299	279
Temperature of feed water entering economizer, deg.	95	79	111	84
Temperature of feed water entering boiler, deg.	175	145	169	196
Increased evaporation produced by economizer, per cent.	10.5	7	9.3	12.8

Mr. William R. Roney in a paper on "Mechanical Draft" (Transactions, Am. Soc. M. E., Vol. XV), gives some additional data on the saving due to economizers working under various conditions in nine different plants. They are given in Table II.

TABLE II.—RONEY'S TESTS OF ECONOMIZERS.

Plants tested	1	2	3	4	5	6	7	8	9
Gases entering economizer, deg.	610	505	550	522	505	465	490	495	595
Gases leaving economizer, deg.	340	212	205	320	320	250	290	190	299
Water entering economizer, deg.	110	84	185	155	190	180	165	155	130
Water leaving economizer, deg.	287	276	305	300	300	295	280	320	311
Gain in temperature of water, deg.	117	192	120	145	110	115	115	165	181
Fuel saving, per cent.	16.7	17.1	11.7	13.8	10.7	11.2	11.0	15.5	16.8

In considering the saving that an economizer produces it is necessary to take into account the interest on the investment and the cost of repairs, cleaning, etc. Economizers cost, it is said, about $5.40 per boiler horse-power for plants of 1,000 boiler horse-power or over, on the basis of 4.8 square feet of economizer surface per boiler horse-power. This includes the cost of the brick setting, delivering and erecting, etc. Three per cent. of the investment will probably do more than pay for the cost of the operation, cleaning and repairs. Assume a 1,000-horse-power boiler plant to operate 300 ten-hour days per year, that the coal consumption is 3½ pounds per boiler horse-power per hour and that coal costs $3 per ton of 2,000 pounds delivered. The annual fuel cost would then be $15,750, and if an economizer reduced this by 12 per cent., then the saving that an economizer would produce would be $1,890. The cost of the economizer at $5.40 per boiler horse-power would be $5,400 and 8 per cent. of this for interest, repairs, operating and cleaning is $432. Deducting $432 from $1,890 would leave a net saving of $1,458, which is sufficient to pay for the economizer in less than four years. If the plant was operated continuously, the annual fuel cost would be $45,990, and the net saving $5,085, sufficient to pay for the economizer in about one year. The 12 per cent. that was assumed in the previous calculations is a conservative estimate for the saving with a low temperature of feed water. In the four tests made by Mr. Barrus, previously noted, the average saving was 9.9 per cent. and this was obtained with unusually low temperatures of the escaping gases, due partly to the low-steam pressure under which the plants were operated, these varying in the different tests from 68 to 82 pounds. Economizer makers will guarantee that they can produce a saving of 6½ per cent. when the temperature of the water entering them is as high as 200 degrees, the economizers having 4.5 square feet of heating surface per boiler horsepower and the boilers working at their normal rating.

Chapter IX.—Mechanical Draft.

Theory of combustion.—As pointed out in a previous chapter, there are two independent factors that effect the efficiency of a steam boiler. One is the efficiency of the heating surface or its ability to transmit to the water the heat to which it is exposed, and the other is the efficiency of the furnace, by which is meant the amount of heat actually contained in each unit volume of furnace gas compared to that theoretically possible of attainment. A high surface efficiency demands that the temperature of the gases in contact with the boiler be as high as possible, for the rate of heat transfer is some function of the difference in temperature of the water and gases, and the greater the difference the more heat per unit of surface will be transmitted. A high furnace efficiency demands that just the proper amount of air be supplied to the furnace per pound of fuel burned. Too little air, due to a too thick bed of fuel on the grate in proportion to the available draft, results in loss due to the incomplete combustion of the fuel, part of the carbon being burned only to carbonic oxide instead of to carbonic acid, as it should if the combustion is complete. Too much air, on the other hand, due to too thin a bed of fuel, lowers the temperature of the furnace gases and thereby decreases the heat transferred per unit of heating surface, owing to the smaller difference between the gas and the water temperatures; it also increases the volume of gases and this in turn necessarily increases the velocity of their flow through the passages of the boiler, thus giving less time for their heat to be absorbed.

A thin fire allows an excess of air to enter the furnace, while a thick fire tends to reduce it on account of the greater resistance offered. Mr. Walter B. Snow writes as follows in explaining the higher efficiency of a high rate of combustion: "If the size of a grate is reduced but the same amount of fuel burned by increasing the rate of combustion, the diminished area of the grate and of the exposed interstices between the fuel necessitates a higher velocity to secure the admission of a given volume of air. This increased velocity in turn requires greater draft or air pressure. The volume at any given temperature passing through the coal is proportional to the velocity, but the pressure varies as the square of the velocity. Therefore, if a given grate be reduced one-half and the rate of combustion doubled, so as to maintain the same total consumption, the same volume of air would have to travel through the exposed interstices at twice the velocity. But the pressure or vacuum would be four times as great, and, as a consequence, the air would be forced or drawn into spaces between the fuel which it could not reach under lesser impelling force. Much more intimate contact and distribution are the results. Less free oxygen passes through the fuel bed unconsumed, and for a given supply of air a higher efficiency of the fuel is attained." Most leading authorities unite in the belief that a higher efficiency is secured in steam boilers when operating at comparatively high rates of combustion. It is advisable to provide for this in plants using coal that can be burned rapidly, care being taken at the same time to provide sufficient draft to run the boilers over their normal rating when occasion demands. Certain coals high in ash and in sulphur cannot be burned at more than ordinary rates for reasons explained in Chapter II in the section relating to coals.

Necessity for ample draft.—Means for providing a strong draft for steam boilers is one of the most essential features of a plant, for if sufficient coal can be burned a boiler will generate steam several times as rapidly as under normal conditions. This means that a smaller number of the boilers can be relied upon to furnish the steam required, while others are shut down for repairs or cleaning, than would be needed if less draft was available. The investment for boilers, therefore, need not be so great. Of course, when making steam at abnormally high rates of evaporation, the efficiency is not so good as under the conditions of normal working, but it is cheaper to allow some fuel to go to waste occasionally than to spend more money in the first place for boilers only used at long intervals. More complaints with steam boilers are probably traceable to insufficient draft than to any other cause. Draft is secured by a chimney, by mechanical draft produced by fans, or by steam jet blowers.

Mechanical draft.—Mechanical draft may be secured by two methods, known as induced draft and forced draft. In the former, a fan is connected with the smoke flue from a boiler or batteries of boilers, so as to suck the gases through the furnace, gas passages and flues to the fan, which discharges it usually through a short chimney, but sometimes through a high one. Forced draft is the term applied to that system where air is forced into the furnace beneath the grate bars, either by a fan or by a steam jet blower. If mechanical draft is to be used, induced draft is usually installed in new plants. Forced draft with a fan is sometimes provided for in new plants, but more often in old plants, as it is sometimes easier to install than the induced system. The objection to forced draft lies in the fact that a pressure greater than that of the atmosphere is created in the furnace and gas passages which may cause the gas to pass outward through cracks in the setting of boilers and through the fire doors when the fires are being replenished or cleaned. Dampers in the blast pipe and uptake, to be closed when the fire doors are opened, may overcome this latter objection. With induced draft the leakage through the doors and brickwork is of course inward.

Steam jet blowers can only produce a moderate draft. Their steam is believed to be useful in preventing clinker from forming on the grates with certain kinds of anthracite coals. Steam jets are used to a considerable, but decreasing, extent in the anthracite coal regions. Experiments made by a Board of Steam Engineers of the Navy Department with five different types of steam jets showed that they used from 8.3 to 21.2 per cent. of the steam generated in the boiler to which the jets were applied. Draft can be obtained from fans with only a fractional part of these amounts. Under ordinary conditions a fan for mechanical draft can be driven by from 1 to 5 per cent. of the steam evaporated in the boilers operating under ordinary conditions, depending upon the size of the plant.

Advantages of mechanical draft.—Draft produced by fans possesses many advantages over chimneys as ordinarily proportioned. Probably the greatest of these is its flexibility, it being possible to regulate the speed of the fan so that the proper rate of combustion for the amount of steam required is maintained entirely independent of the weather conditions; another important advantage is the ability of the fans to create a much greater draft than is possible with a chimney. Steam engines for driving fans are frequently fitted with valves arranged to govern the speed of the engine according as the boiler pressure varies, increasing it as the pressure falls and reducing it as it rises above the normal. Mechanical draft enables economizers to be placed in the flue and reduce the temperature of the escaping gases by heating the feed-water far below the temperature that is necessary in a chimney

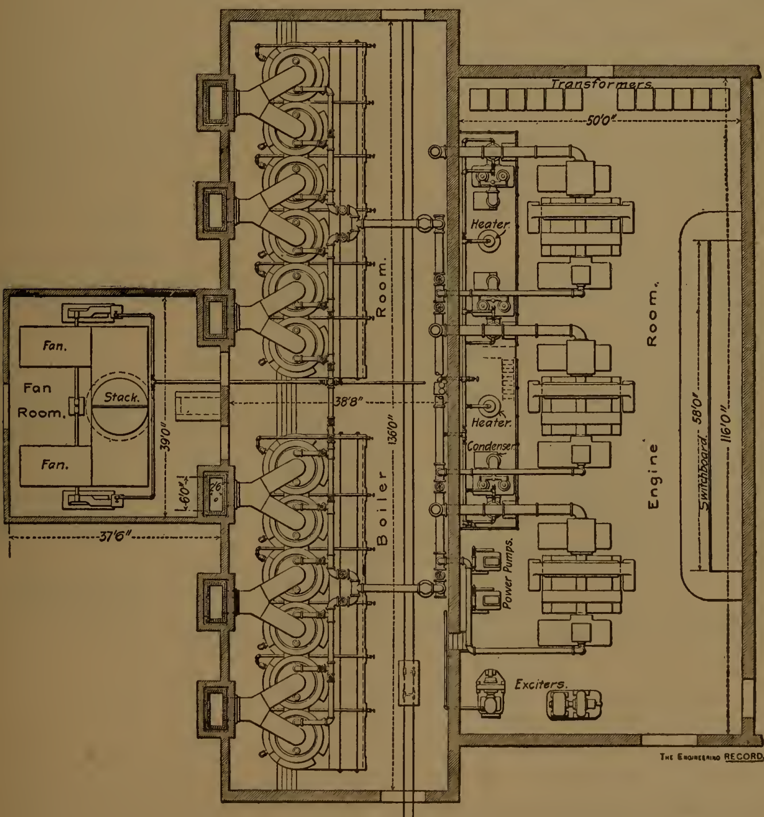


Figure 50.—Power House, Olympia Mills, Columbia, S. C.

W. B. SMITH-WHALEY, ENGINEER.

to create a draft. The reduction in draft due to the use of economizers is a much greater percentage of the available draft with a chimney than it is of the draft where fans are employed. Again, the greater draft of fans enables cheap low-grade fuels to be burned that could not easily be used with the chimney draft, and the saving that these fuels brings about in some localities is a very considerable sum of money. Still another point in favor of mechanical draft lies in the portability of the fans in case a change of location is desired.

Theory of fans.—Before explaining the method of selecting a fan for a given service it is necessary to consider briefly their theory of operation. Fans for mechanical draft are invariably of the centrifugal type and consist of a paddle wheel revolving in a sheet iron casing, the air entering an opening in the casing at the axis of the wheel and being propelled radially to the casing by the action of centrifugal force, finally escaping by an outlet provided. The velocity with which the air is moved is expressed by the well-known formula:

$$v = \sqrt{2gh}, \tag{1}$$

in which v equals the velocity and h equals the head due to the velocity and also to the pressure divided by the density. Therefore,

$$v = \sqrt{(2gp \div d)}. \tag{2}$$

The work done in moving this air is equal to the product of the velocity of the air in feet per second, the pressure in pounds per square foot and the effective area in square feet over which this pressure is exerted. If W represents the work done, p the pressure in pounds per square foot, a the area in square feet and v the velocity in feet per second, then:

$$W = pav. \tag{3}$$

From (2) we have, by squaring and transposing:

$$p = dv^2 \div 2g. \tag{4}$$

and substituting its value in (3) we have:

$$W = dav^3 \div 2g. \tag{5}$$

From this it will be seen that the power required to drive a fan varies as the cube of the velocity. In other words, if the velocity is doubled the power required will be eight-fold; if tripled, 27-fold. As the velocity of the air is practically the same as the peripheral speed of the fan it will be seen how essential it is to use a large fan at its proper speed rather than a small fan running at a higher speed than is necessary to obtain the desired pressure, and thus the desired volume. The pressure varies as the square or the speed, as shown in formula (4), hence the pressure is quadrupled by doubling the speed.

Design of fans.—From the work upon "Mechanical Draft" by Mr. Walter B. Snow, published by the B. F. Sturtevant Company, the following quotations relating to the design of fans have been taken:

"In the design of a wheel to meet given requirements, it is necessary to make its peripheral speed such as to create the desired pressure, and then so proportion its width as to provide for the required air volume. Evidently the velocity and corresponding pressure may be obtained either with a small wheel running at high speed or a large wheel running at low speed. But if the diameter of the wheel be taken too small, it may be impossible

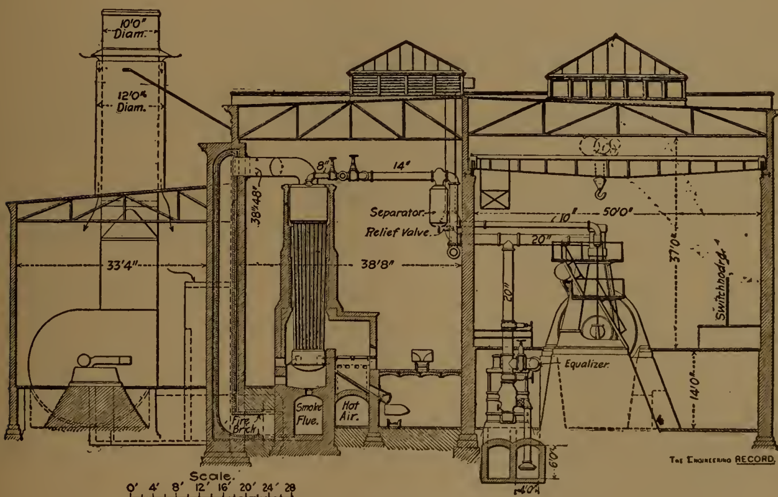


FIGURE 51.—CROSS-SECTION, POWER HOUSE, OLYMPIA MILLS.

to adopt a width, within reasonable limits, which will permit of the passage of the necessary amount of air under the desired pressure. Under this condition it will be necessary to run the fan at higher speed in order to obtain the desired volume. But this results in raising the pressure above that desired, and in unnecessarily increasing the power required. On the other hand, if the wheel be made of excessive diameter it will become almost impracticable on account of its narrowness. Between these two extremes a diameter must be intelligently adopted which will give the best proportions.

"It has been determined experimentally that a peripheral discharge fan, if enclosed in a case, has the ability, if driven to a certain speed, to maintain the pressure corresponding to its tip velocity over an effective area which is usually denominated the 'square inches of blast.' This area is the limit of its capacity to maintain the given pressure. If it be increased the pressure will be reduced, but if decreased the pressure will remain the same. As fan housings are usually constructed, this area is considerably less than that of either the regular inlet or outlet.

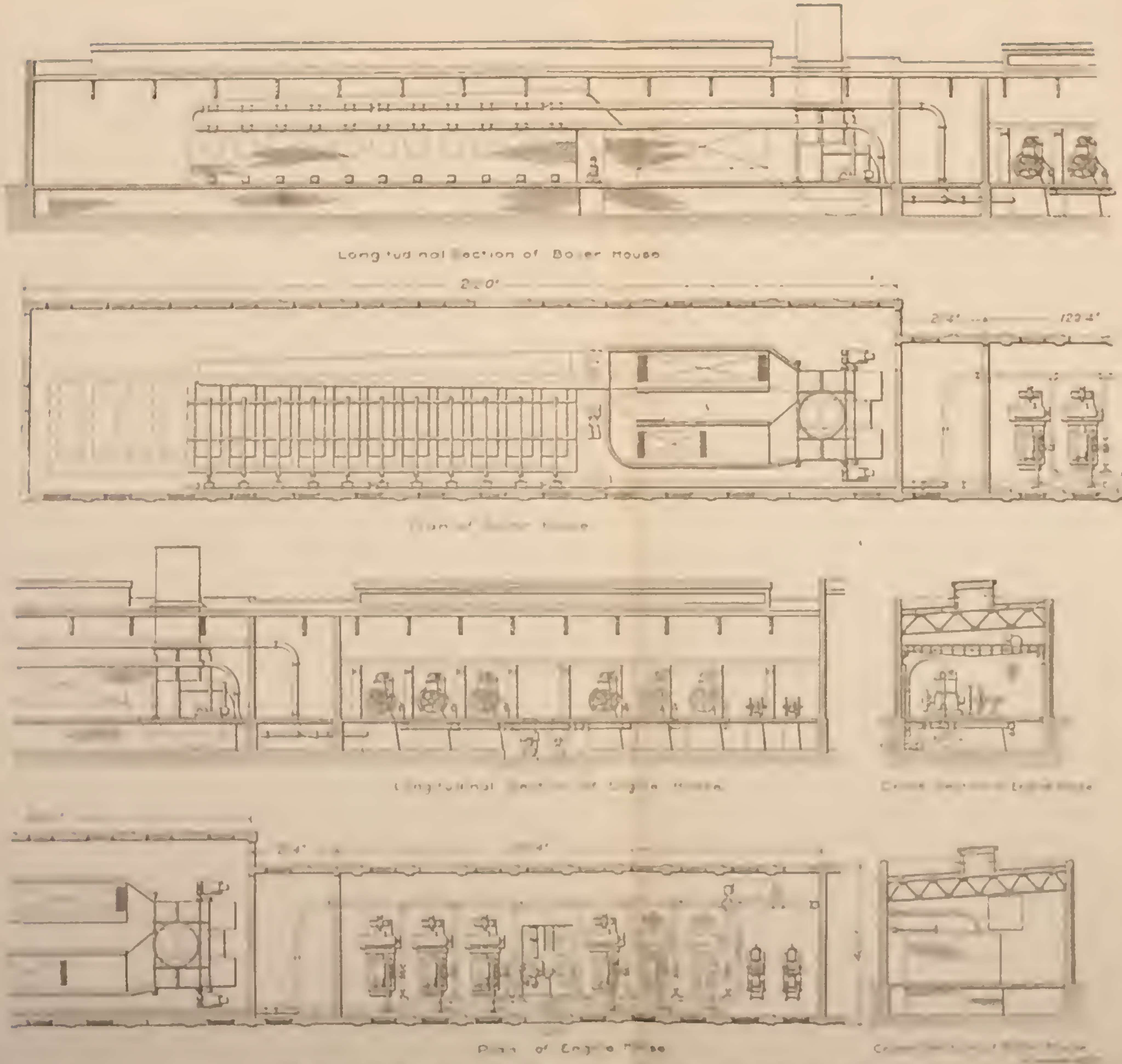
"The square inches of blast, or, as it may be termed, the capacity area of a cased fan, may be approximately expressed by the empirical formula:

$$\text{Capacity area} = DW \div X.$$

In which D = diameter of fan wheel in inches.
W = width of fan wheel at circumference in inches.
X = a constant dependent upon the type of fan and casing.

"An approximate value for X for Sturtevant fans for general practice is not far from 3, but this is to be used only to determine the capacity area over which the given pressure may be maintained. This is not a measure of the area of the casing outlet, which is always larger than the square inches of blast. As a consequence, the pressure is lower and the volume discharged is somewhat greater than would result through an outlet having the square inches of blast for its area. But the maximum pressure may be realized when the sum of resistances is equivalent to a reduction of effective outlet area to that of square inches of blast. The volume of air which, under the given pressure, will flow through the given capacity area, and hence the volumetric capacity of the fan under the given conditions, may be determined from Table I. In a similar manner the horse-power may be ascertained, the proper efficiency coefficient being applied.

"Both the volume and the power required will evidently increase with the area of the outlet, being greater with the normal outlet than with that representing the capacity area. But this increase will not be proportional to the area, for the pressure and consequently the velocity will be lower with the larger area. The greatest delivery of air and the largest consumption of power will occur when the casing is entirely removed and the fan left free to discharge entirely around its periphery.



Power House, Southern Terminal Station, Boston, Mass.

"If volume alone regardless of pressure is the requisite, the larger the fan the less the power required. There is a strong temptation, however, for a purchaser to buy a smaller fan and run it at a higher speed; for he sees only the first cost and does not realize the entailed expenditure for extra power. If possible a fan should never be made so small that it is necessary to run it above the required pressure in order to deliver the necessary volume. To double the volume under such circumstances requires eight times the power; three times the volume demands twenty-seven times the power.

"When a fan is employed for exhausting hot air or gases, the speed required to maintain a given pressure difference is evidently greater than that necessary when cold air is handled, the difference being due to, and inversely proportional to, the absolute temperature."

With forced draft it will be safe to assume that 300 cubic feet of air are supplied per pound of coal burned, and with induced draft 300 cubic feet supplied but 600 cubic feet exhausted by the fan if the gases are to be at a temperature of about 500 degrees Fahrenheit or 450 cubic feet exhausted if at 300 degrees temperature, as might be expected with economizers. With induced draft the fan has to handle hot gases of approximately double the volume that it would if the fan was supplying air to the grates. As the gases are lighter, however, the power required per cubic foot of gas removed is less than it is per cubic foot of air with forced draft. Knowing the amount of coal burned under ordinary conditions, say four pounds per boiler horse-power per hour, the amount of air required per minute can be determined. The pressures usually required in forced draft vary from ½ to 1 ounce per square inch and in induced draft from ½ to ¾ ounce, depending on the fineness of fuel, the readiness with which it burns, the length of flues and number of bends in them, etc.

Tables I and II are taken from "Mechanical Draft," before mentioned. In the former is given the volume of air in cubic feet for various pressures which may be discharged in one minute through an orifice having an effective area of discharge of one square inch, or per square inch of blast. The method of using these tables can best be explained, perhaps, by working out a typical case. Suppose we are to design an induced-draft fan to handle the gases from 1,000 horse-power of boilers against a pressure of ¾ ounce. At 4 pounds of coal per horse-power per hour and 600 cubic feet of gas per pound, 40,000 cubic feet of gas would have to be exhausted per minute. Opposite ¾ ounce pressure in Table I, it will be seen that a fan will supply 31.06 cubic feet, say 31 cubic feet, per square inch of blast. Dividing 40,000 by 31 would give 1,290 as the number of square inches of blast the fan would require. But the square inches of blast equal $DW \div 3$. In standard fans of the Sturtevant make W is approximately equal to $D \div 2.4$, therefore:

$$\text{Square inches of blast} = D^2 \div 7.2.$$

Substituting 1,290 for the square inches of blast we have: $1,290 = D^2 \div 7.2$, from which D equals 96 inches or 8 feet for the diameter of the fan.

To determine the velocity at which the fan should be run, reference should be made to Table II. For the column corresponding to ¾ ounce it will be seen that an 8-foot fan should run at 178 revolutions per minute to give the capacity required. By increasing the speed at times of overload the pressure and volume of gases can be increased.

TABLE I.—VOLUME OF AIR DISCHARGED AND HORSE-POWER REQUIRED WHEN AIR UNDER GIVEN PRESSURE IS ALLOWED TO ESCAPE INTO THE ATMOSPHERE.

Pressure in ounces per sq. in.	Volume of air discharged through an orifice of an effective area of discharge of 1 sq. in. (or per square inch of blast).	Horse-power required to move the given volumes of air under given conditions of discharge.
¼	17.95 cubic feet per minute.	0.00122
⅜	21.98 " " " "	0.00225
½	25.37 " " " "	0.00346
⅝	28.36 " " " "	0.00483
¾	31.06 " " " "	0.00635
⅞	33.54 " " " "	0.00800
1	35.85 " " " "	0.00978
1⅛	38.01 " " " "	0.01166
1¼	40.06 " " " "	0.01366
1⅜	42.0 " " " "	0.01577
1½	43.86 " " " "	0.01794

In the last column of Table I is given the horse-power required per square inch of blast to move given quantities of air, under different pressures and at a temperature of 50 degrees Fahr. For forced draft these quantities can be used directly, but it should be remembered that they refer to the theoretical power required to move the air and they should be at least doubled to allow for the fan and engine friction and the power required to run the fan considering it was moving no air and to overcome the friction of the air in the fan casing.

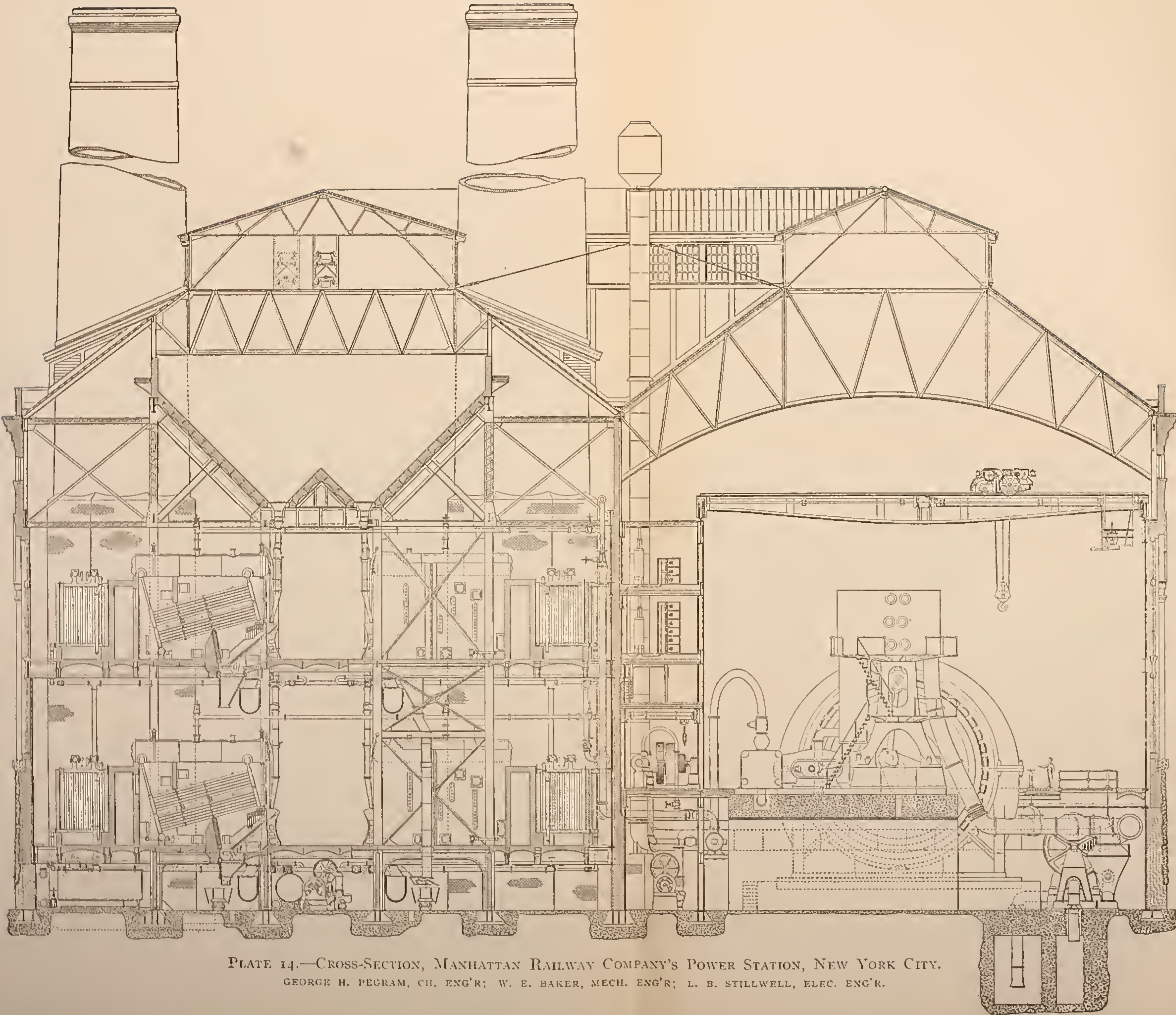


PLATE 14.—CROSS-SECTION, MANHATTAN RAILWAY COMPANY'S POWER STATION, NEW YORK CITY.

GEORGE H. PEGRAM, CH. ENG'R; W. E. BAKER, MECH. ENG'R; L. B. STILLWELL, ELEC. ENG'R.

With induced draft the air is of less density on account of its higher temperature, so that less power is required per cubic foot of gas moved. If the gases have a temperature of 600 degrees the quantities given in the last column of Table I can be multiplied by 0.5, and by 0.58 if the temperature is 450 degrees, to obtain the actual power required to move the air. The engine driving the fan should have its cylinders so proportioned that it can easily develop power 50 per cent. in excess of that calculated.

Table III shows the capacity of the American Blower Company's fans used for induced draft, according to its catalogue.

TABLE II.—REVOLUTIONS OF FAN OF GIVEN DIAMETER NECESSARY TO MAINTAIN A GIVEN PRESSURE OVER AN AREA WHICH IS WITHIN THE CAPACITY OF THE FAN.

Diameter of fan wheel, in feet.	Pressure, in Ounces per Square Inch.									
	⅛	¼	⅜	½	⅝	¾	⅞	1	1⅛	1¼
3	194	274	336	388	433	475	513	548	581	612
3½	166	235	288	332	372	407	439	469	498	525
4	146	206	252	291	325	356	384	411	436	459
4½	129	183	224	258	289	316	342	365	387	408
5	116	164	202	232	260	285	308	329	349	367
5½	106	149	183	211	236	259	280	299	317	334
6	97	137	168	194	217	238	256	274	290	306
6½	90	126	155	179	200	219	236	253	268	282
7	83	117	144	166	186	203	220	235	240	262
7½	78	110	135	155	173	190	204	219	232	245
8	73	103	126	146	163	178	192	205	218	230
8½	69	97	119	137	153	167	181	194	205	216
9	65	92	112	129	144	158	171	183	194	204
9½	61	87	106	123	137	149	162	173	183	193
10	58	82	101	116	130	142	154	164	174	184
11	53	75	92	106	118	129	140	150	158	167
12	49	69	84	97	108	119	128	137	145	153
13	45	63	78	90	100	110	116	126	130	141
14	42	59	72	83	93	102	110	117	124	131
15	39	55	67	78	87	95	102	110	116	122

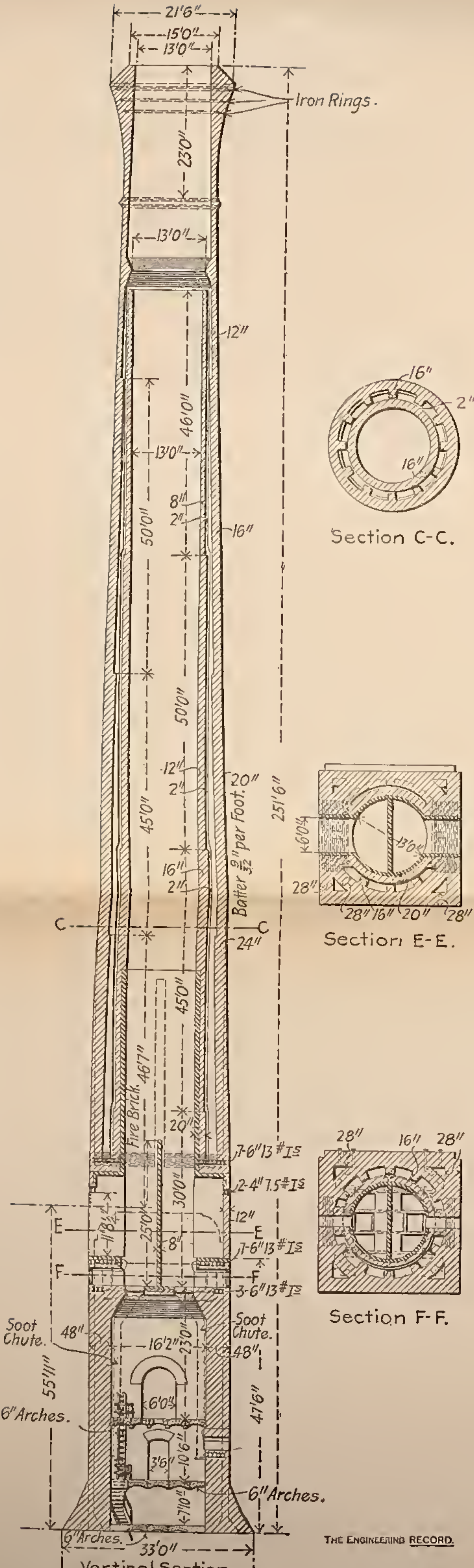
It is based on a temperature of 550 degrees Fahrenheit for the gases, and air temperature of 62 degrees, on 18 pounds or 234 cubic feet of air per pound of coal, and a consumption of five pounds of coal per boiler horse-power. Mr. F. R. Still, of the American Blower Company, recently wrote as follows explaining it: "This table gives the diameter of the fan wheel, width at periphery, diameter of the inlet, size of the outlet and the maximum speed necessary to produce a draft of one inch of water, which is about the strongest draft that is ordinarily required. The table also gives the capacity of the fan per inch of width (of blade) at periphery, and the horse-power per inch of width. If it is desired to run the fan at a slower speed than is given opposite any of our sizes, determine on the speed at which the fan is to run, find the capacity of same per inch width and divide

TABLE III.—INDUCED DRAFT CAPACITY TABLE FOR FANS.

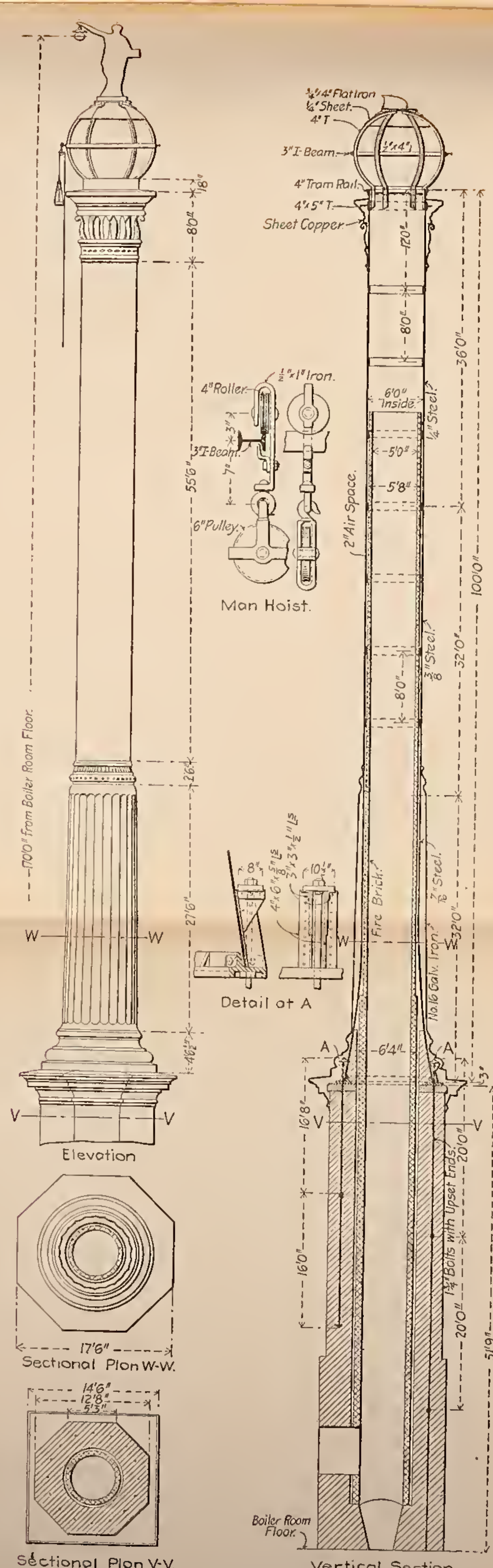
Size of fan.	Diam. of wheel, inches,	Width at periphery.	Diam. of inlet.	Size of outlet, square inches.	Speed R.P.M., for 1-inch draft.	Capacity of fan in cubic feet per minute, temp. gases 550 deg. F.	H.-P. boiler capacity from fan capacity.	Pounds coal per hour at 5 lbs. per H.-P. hour.	Brake H.-P. to drive fan at speed.	Capacity of fan per inch width at periphery.	H.-P. per inch width at periphery.	Cu. ft. air per minute for combustion. Temp. 62 deg. F.
50	30	12½	20	18	740	5,030	116	580	2.02	403	.162	2,260
60	36	14¼	23	21½	615	6,900	159	795	2.76	485	.194	3,100
70	42	16½	26	24½	530	9,325	215	1,075	3.72	563	.226	4,200
80	48	17½	30	27	460	11,300	261	1,305	4.50	645	.258	5,100
90	54	20¾	34	30½	410	15,100	349	1,745	6.03	728	.290	6,800
100	60	23¼	38	34½	370	18,750	433	2,165	7.50	844	.322	8,450
110	66	26	42	37½	335	23,000	532	2,660	9.20	885	.354	10,370
120	72	30¼	46	41½	310	29,250	677	3,385	11.72	970	.387	13,200
140	84	34⅜	53	48	265	38,800	896	4,480	15.50	1,132	.452	17,450
160	96	38	60	54	230	49,000	1,130	5,650	19.60	1,290	.516	22,000
180	108	41¼	68	60	205	59,900	1,385	6,925	24.00	1,453	.581	27,000
200	120	47	76	66	185	75,500	1,746	8,730	30.25	1,610	.644	34,000
220	132	50	84	72	170	88,600	2,050	10,250	35.50	1,775	.710	40,000
240	144	54	92	78	155	104,600	2,420	12,100	41.80	1,940	.775	47,000

the required capacity for a given horse-power in the plant by this capacity per inch and it will determine how wide the fan should be at the periphery, with the increased diameter and the slower speed. The horse-power can be determined by multiplying this width by the horse-power per inches in width."


2'6"



CHIMNEY LINCOLN WHARF STATION, BOSTON ELEVATED RAILWAY COMPANY.
GEORGE A. KIMBALL, CH. ENG'R.



CHIMNEY FOR U. S. GOVERNMENT, MANILA, P. I.
F. L. STRONG, CON. ENG'R; EDWARD BARROTH, ARCH.

PLATE 15.

Chapter X.—Chimneys.

Size of Chimneys.—There are two factors that affect the capacity of a chimney, its cross-sectional area and its height. The intensity or force of draft is proportional to the square root of the height of the chimney, and, for a given height, the capacity is directly proportional to its cross-sectional area within certain limits. Earlier in this book attention has been called to the importance of having a strong draft in order that cheap, low-grade fuels may be burned successfully, and also, that there may be sufficient reserve capacity in the boilers at times of abnormal demands. In this latter respect ample draft is just as good as additional boiler-heating surface, and, as a general proposition, it costs less. Although an engineer once said that high chimneys "are monuments to the folly of their builders," yet this opinion should not deter one from building chimneys from 150 to 200 feet in height, and for very large plants still higher. In certain localities where smoke or obnoxious gases are objectionable, tall chimneys are necessary. Chimneys of the heights mentioned are desirable where high rates of combustion are to be employed, and the height of the chimney is governed somewhat by the amount of coal that is to be burned per square foot of grate in a given time. As most chimney formulas contain factors representing the height and sectional area, it is convenient to first fix the height suitable for the kind of coal to be burned and then determine the area by established formulas.

Height of Chimneys.—Mr. J. J. De Kinder, an engineer of considerable experience in steam plant operation, recommends the following heights for chimneys with the coals mentioned: 75 feet for free-burning bituminous coal, 100 feet for slow-burning bituminous slack, 115 feet for slow-burning bituminous coal, 125 feet for anthracite pea coal, 150 feet for anthracite buckwheat coal. These recommendations are probably intended for boiler plants of moderate size only. The author believes that with western coals such as Collinsville or Mt. Olive a 150-foot stack is desirable. This will make it possible to burn as a maximum at times of overload from 30 to 35 pounds of coal per square foot of grate per hour, this coal being burned ordinarily at 20 pounds per square foot of grate per hour. With plants operating 800 or more horse-power of boilers, 150 feet is the minimum height of a chimney that should be selected, irrespective of the kind of coal that is to be burned. For large plants 200 feet is not excessive. No designer knows when the cost of coal may be such that it will be cheaper to burn low-grade fuels, which cannot be done without strong draft.

Capacity of Chimneys.—Most chimney formulas are based on coal consumption but do not take into account the fact that the amount of gases given off by different coals varies. Col. E. D. Meier has called attention to this, and by calculating the volume of gases from their composition finds that the relative areas for chimneys for certain much used coals should be as follows: Anthracite 100, New River (Va. semi-bit.) 93, Youghiogheny (Penn. bit.) 102, Mt. Olive (Ill. bit.) 128, Collinsville (Ill. bit.) 138.

Three well-known chimney formulas are those of Kent, Gale and Christie. In the former

$$\text{H, P.} = 3.33\ E \sqrt{H}$$

in which H is the height in feet and E is the effective area, the actual area being determined by increasing the diameter of the chimney if it be round or the side of the chimney if it be square by 4 inches to allow for the lining of the chimney by a layer of gas that is assumed to have no velocity. E is approximately equal to $(A - 0.6 \sqrt{A})$ for round chimneys and to $A - {}^2/_3 \sqrt{A}$ for square chimneys. As this formula is based upon the burning of 5 pounds of coal per horse power

$$C = 16.6\ E \sqrt{H}$$

where C equals the number of pounds of coal burned per hour. The Kent formula assumes that the height and area are interdependent, and it only holds within certain limits.

Gale's formula may be expressed in the form

$$A = 0.07\ C^{3/4} \text{ and } H = \frac{180}{t}\left(\frac{C}{G}\right)^2$$

in which t is the temperature of the chimney gases and G the grate area in square feet. Col. E. D. Meier, who has a great deal of ex-

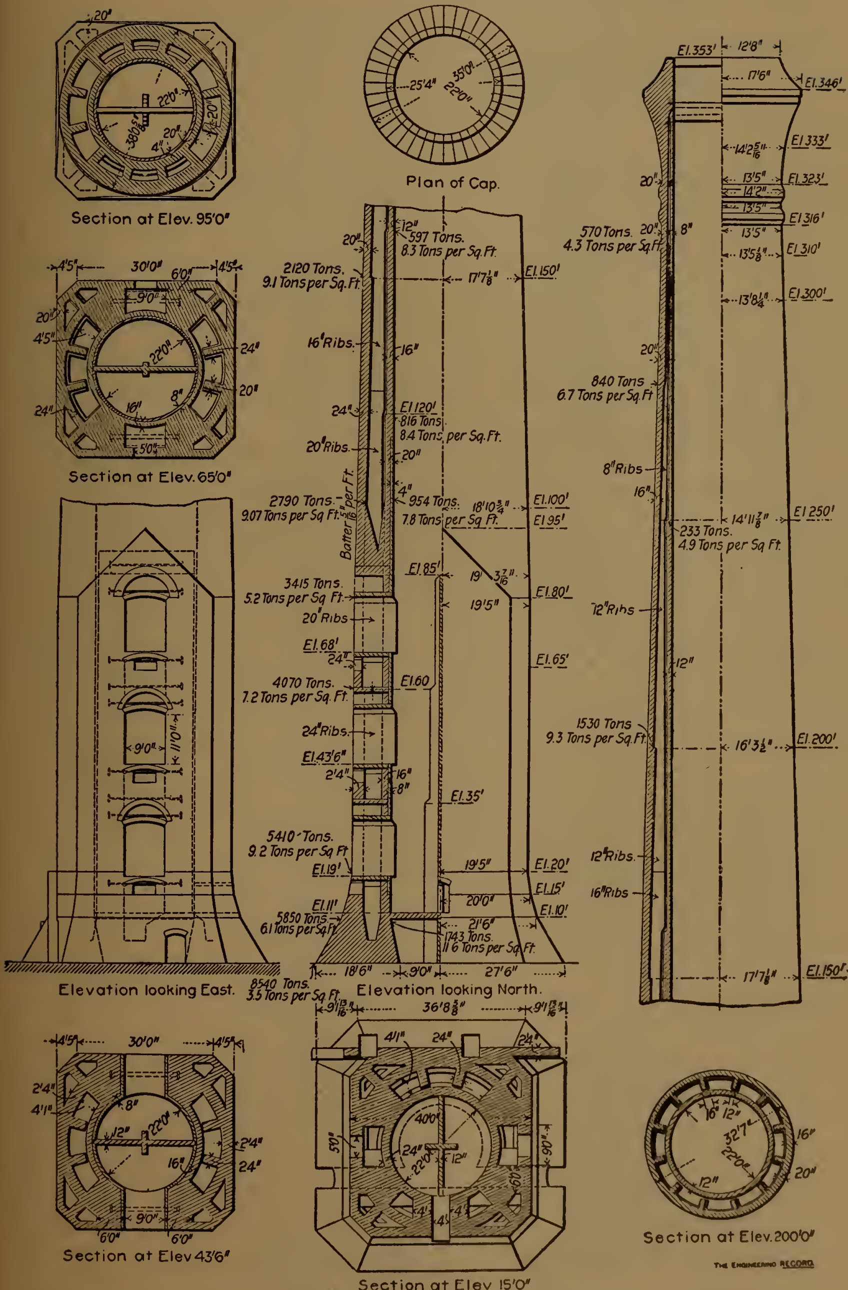


FIGURE 52.—CHIMNEY METROPOLITAN STREET RAILWAY CO., NEW YORK.

perience with western coals, states that this formula gives rather too large results and recommends that

$$H = \frac{120}{t}\left(\frac{C}{G}\right)^2$$

and that after the height is found by this formula the area be obtained by the one proposed by Mr. Kent.

Mr. George A. Orrok in a recent issue of "Power" states that the constant in the general chimney formula, for which Kent gives the value of 16.66, varies greatly in the formulas of different authorities, and Mr. Orrok recommends that a value of 12 be given it for brick-lined stacks, but that in case of an unlined steel stack the value of this constant may be increased to 14 or 15 and for small stacks 16 may be used.

Mr. W. W. Christie in his book on "Chimney Design" gives the chimney formula: Horse power $= 3.24\ A \sqrt{H}$, it being assumed that four pounds of coal are burned per horse power, A being the area of the flue and H the height, both in feet. If C is the coal burned per hour $C = 12.96\ A \sqrt{H}$.

Thickness of Chimney Walls.—In designing a chimney of a given height and inside diameter it is necessary to determine the thickness of the walls required to provide sufficient weight to prevent its being overthrown by the wind. It is customary to step out the inner walls at different levels so as to divide the shell into a series of sections, each of a uniform thickness which is less than that of the section immediately below it. As the walls have a slight batter inside and outside, the diameters at the top and bottom and the thickness and heights of the different sections of the shell have to be determined. To simplify the problem it is proposed to give a method of designing a chimney with straight inner and outer walls of the proper batter. From this a chimney of approximately equivalent weight but made up of several sections each of uniform thickness can be designed.

It is usually customary to make the weight of a chimney such that it will bear a certain relation to the wind pressure, a rule commonly used requiring that the prolongation of the resultant of the total wind pressure acting through the center pressure and the weight of the chimney intersect a horizontal plane through the chimney base at a point not more remote from the axis of the chimney than a distance equal to $D_b \div 6$ where D_b represents the outside

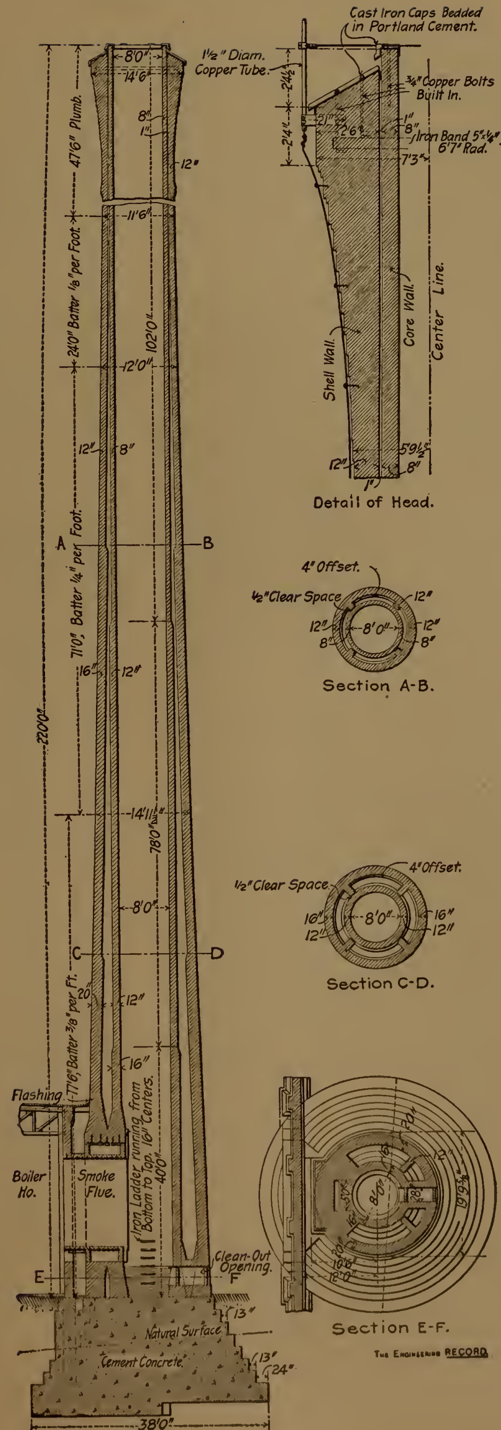


FIGURE 53.—BRICK CHIMNEY DESIGNED BY LOCKWOOD, GREENE & CO.

diameter at the base, or at the elevation at which the calculation for stability is made. If the chimney be round, square or octagonal, the diameters referred to are those of the inscribed circles. To illustrate, if, in Figure 54, CD represents the wind pressure considered to be acting through the center of pressure C and CE represents the weight of the chimney, and the prolongation of their resultant CF intersects the base AB at a point G distant from H by a distance less than $D_b \div 6$, then the conditions as to stability are fulfilled. In Figure 54, C is the center of pressure; CD represents the total wind pressure, P; CE the weight W, and CH equals h the height in feet of the center of pressure above the base AB. Expressed algebraically the formula is:

$$\frac{W}{P} = \frac{h}{D_b \div 6} \qquad (1)$$

The value of W can be found from the volume of the chimney, assuming that one cubic foot of brick masonry weighs 115 pounds. Then $W = 115$ V. Therefore

$$W = 115\ H\ \pi \left(\frac{(D_b^2 + D_t^2 + D_b D_t) - (d_b^2 + d_t^2 + d_b d_t)}{12}\right) \qquad (2)$$

Throughout this discussion the terms used have the following significance:

$H =$ height of chimney above base in feet.
$h =$ height of center of pressure above base in feet.
$D_t =$ outside diameter top in feet.
$d_t =$ inside diameter top in feet.
$D_b =$ outside diameter bottom in feet.
$d_b =$ inside diameter bottom in feet.
$P =$ equivalent total wind pressure.

Now if the value of W in equation (2) be substituted for W in equation (1) then:

$$115\ \pi\ H \left(\frac{(D_b^2 + D_t^2 + D_b D_t) - (d_b^2 + d_t^2 + d_b d_t)}{12}\right) = \frac{6\ h\ P}{D_b} \qquad (3)$$

There is but one factor in this equation, d_b, whose numerical value is not known either from data assumed at the outset or from its relation to known quantities in the equation. This being the case the equation can be solved for the value of d_b and its numerical value obtained. One would then have all the dimensions of a chimney with straight walls of a height and inside diameter necessary, and which would satisfy conditions of stability.

The inside diameter of the chimney at the top and the height must be assumed. From the former dimension the outside diam-

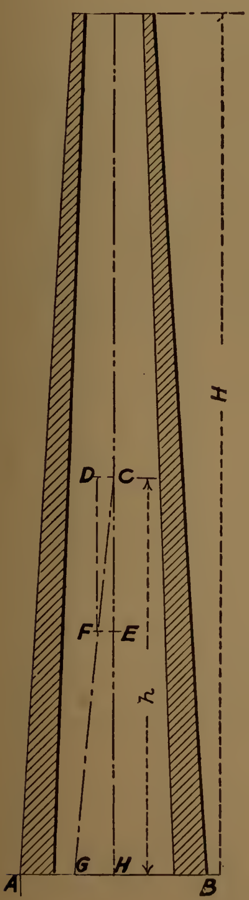


FIGURE 54.

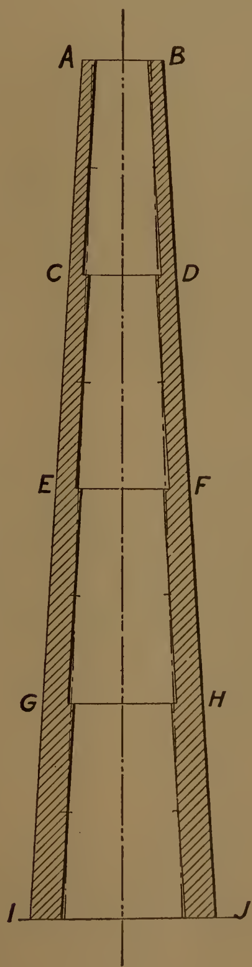


FIGURE 55.

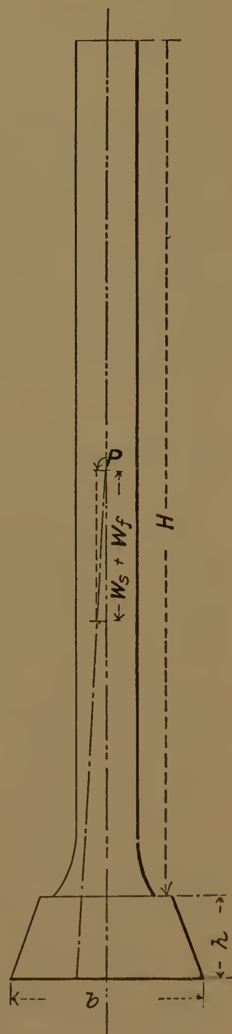


FIGURE 56.

eter at the top and the outside diameter at the bottom can be found readily by two empirical rules. The first of these is Prof. Lang's rule, given in his paper on the "Construction and Dimensions of Chimneys for Boiler Plants," of which a translation was printed in The Engineering Record of July 20 and 27, 1901. It is to the effect that the thickness in feet of a chimney at the top, t, neglecting the ornamentation, should be

$$t = 0.328 + 0.05\, d_t + 0.0005\, H$$

therefore

$$D_t = d_t + 2t = 0.656 + 1.1\, d_t + 0.001\, H \qquad (4)$$

The minimum value of t should be 0.58 feet for radial brick and 0.7 feet for common brick. As the thickness must be over a brick's length and as it cannot increase by less than half a brick's length, the value of t must be 0.7, 1.08 feet, etc. Few chimneys built of common brick have a value for t greater than 1.08 feet. The outside diameter at the bottom, or D_b, can be obtained from the outside diameter at the top by a rule which assumes a batter for the outside wall of 1 :30 to 1 :36 on a side. Assuming it to be 1 :32 then:

$$D_b = D_t + \frac{H}{16}$$

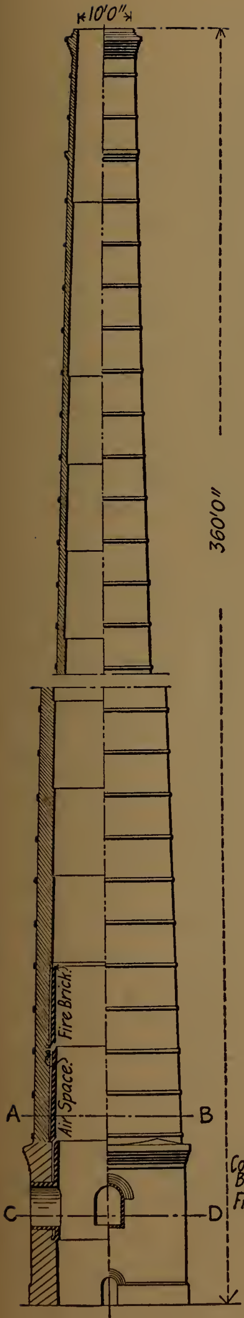
substituting the value of D_t from (4) then

$$D_b = 0.656 + 1.1\, d_t + 0.001\, H + \frac{H}{16} = 0.656 + 1.1\, d_t + 0.063\, H.$$

The value for P, the total wind pressure in equation (3), is found by multiplying the assumed wind pressure of 50 pounds per square foot by the equivalent of the vertical cross-section of the chimney through the axis, in square feet. With a square chimney this plane is taken parallel to two opposite sides. For square chimneys P has a value equal to their cross section in square feet; for round chimneys 0.50 times this area, and for octagonal ones 0.71 times this area.

The numerical values of H, D_b, D_t, d_t, P and H found in the manner indicated are substituted in the above equation, which then can be solved for the value of d_b. This determined we have all the dimensions of the straight walled chimney of the height and inside diameter required for the capacity, and which fulfills the conditions as to stability. As has been said, the interior of a chimney is seldom constructed with a straight batter, but in a series of steps each section from the top downward increasing in thickness by half a brick's length. Each step should then be of a thickness in inches that is divisible by 4½, this being the width of a brick and the necessary mortar. From the top down, therefore, the thicknesses of the different sections should be 8½, 13, 17½, 22, 26,½ etc., inches. Should the calculations for the thickness of the wall at the bottom call for a thickness intermediate between two of these thicknesses, the greater one can be selected, as the thickness of the bottom section. Should such calculation show that the thickness at the bottom should be 22 inches, the chimney can be divided into four sections of equal height, 8½, 13, 17½ and 22 inches thick. Radial brick for chimneys are made in several sizes so that the thickness, when they are used, increases by about 2 inches at the offsets.

After the thickness of walls of the different sections and their heights are determined, the calculation for stability must be made at the base of each section, paying no attention to that below. For instance in Figure 55, the stability of the part ABJI should be calculated considering IJ as



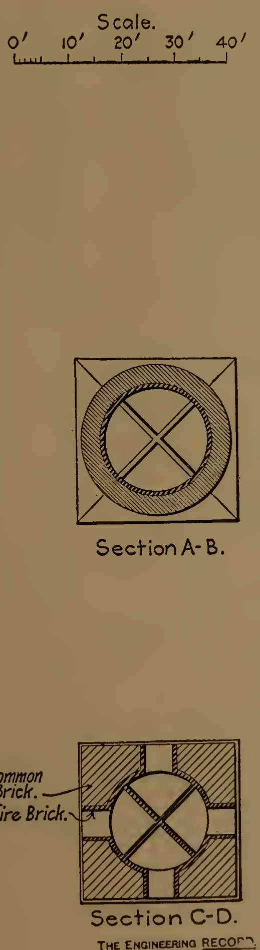


FIGURE 57.—CUSTODIS RADIAL BRICK CHIMNEY FOR ORFORD COPPER CO.

the base, also ABHG considering GH the base, etc., each time locating the center of pressure for the part under consideration.

Another operation yet remains, and that is to find the pressure imposed upon the brickwork by its own weight. Calculation must be made as to the pressure per square inch at the base, and at each level where the shell changes in thickness, that is, at the bottom of each section. According to Prof. Lang, the pressure in pounds per square inch should nowhere exceed that given by the formula $P = 71 + 0.65\ L$ where L denotes the distance in feet from the top of the chimney to the point in question. Should the pressure exceed that given by the formula the walls of the chimney should be made thicker. The function of the chimney height is introduced in the formula to allow a greater pressure in high chimneys, which are erected less quickly than shorter ones, the mortar therefore having more time to harden.

Chimney Linings, Etc.—Chimneys of common brick built in the United States are usually provided with an inner core or lining to protect the outer shell from the heat. Sometimes this core is carried up for half the height of the chimney, but more usually to the top. Care should be taken that it is built independently of the outer shell as the greater expansion of the core would injure the shell. With radial brick chimneys the inner core is not so common as the bricks are more carefully burned and selected than is the custom with common brick and are not so easily affected by heat. Any chimney likely to contain gases at a higher temperature than 600 degrees Fahrenheit should be lined with fire-brick set in fire clay or lime mortar, preferably the former. Sometimes the fire-brick lining extends from the bottom to one-third or one-half of the height. The core can be divided into sections, each about 40 to 50 feet high, and 4, 8½, 13, 17½ and 21 inches in thickness from the top down. If the chimney is of fairly large diameter the 4-inch section should not be over 25 or 30 feet. In the largest chimneys 8 inches is the minimum thickness of the core. It usually has a uniform inside diameter so that changes in thickness are secured by offsets on the outside. With the batter for the outer surface of the outside shell recommended in an earlier paragraph there is likely to be sufficient distance between the offsets of the core and those of the outer shell to provide proper clearance. When the chimney is drawn on paper this can be determined. Steel chimneys should also be lined to prevent loss of

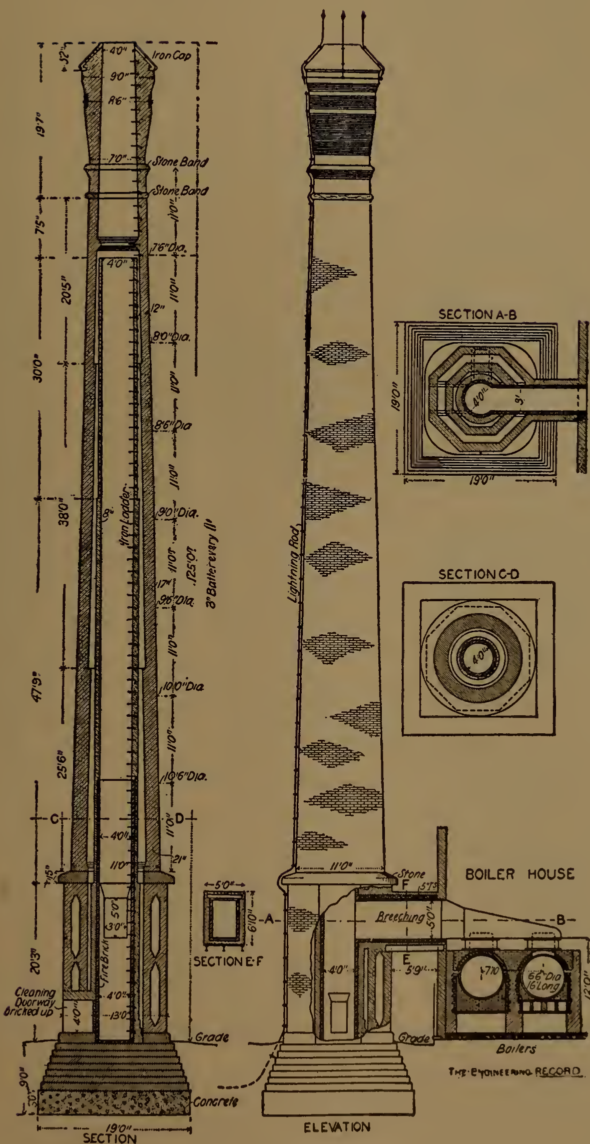


FIGURE 58.—CHIMNEY, LAIDLAW-DUNN-GORDON CO., CINCINNATI, O.

heat and also air leakage, which will occur unless the joints are carefully calked, a provision that is frequently overlooked.

In constructing the opening for smoke flues at the base of the chimney care should be taken that it is not weakened at that point. The top of the opening should be arched over or spanned with heavy steel beams built in the masonry. Should the chimney have an inner core the smoke flue should, of course, be continuous through both shells. Ladders for reaching the top of a chimney are usually located on the inside of brick chimneys and more frequently on the outside of steel ones. If the latter, the continuous hand rails at the sides should be so designed as to allow for the possibility of a difference in expansion between the ladder and the chimney. The tops of brick chimneys should be covered with a cast-iron cap held in place by anchor bolts, and lightning rods metallically connected to the ground should also be provided.

Materials.—Masonry chimneys are usually built of brick, but recently concrete chimneys reinforced with steel, as in the Ransome chimney, shown in Figure 59, have been used with success. For brick chimneys hard burned brick of high specific gravity should be used. A special brick for chimney building used to a large extent in Europe and in a rapidly increasing extent in the United States is the radial brick for round chimneys, with its inner and outer surface curved to conform to the curvature of the chimney. Several holes running through the brick aid in the burning and also serve to secure it in place more strongly as the mortar works into them when the brick is laid. The radial brick that have been used to a considerable extent by the Custodis and Heinicke companies in Germany are much stronger and more durable than common brick. Another feature in their favor is that they are considerably larger than common brick and less labor is required in laying them.

Masonry Chimney Foundations.—These should be of such an area that the load per square foot does not exceed one ton per square foot on soft clay; two tons per square foot on stiff clay, compact sand, loam, etc. These loads are exceeded in buildings but they should not be in chimneys unless solid rock underlies the foundation. The rock should be dressed off into steps with vertical sides so there will be no tendency to slide. With very large masonry chimneys and in fact with chimneys of moderate size in soil of low bearing power, pile foundations are frequently re-

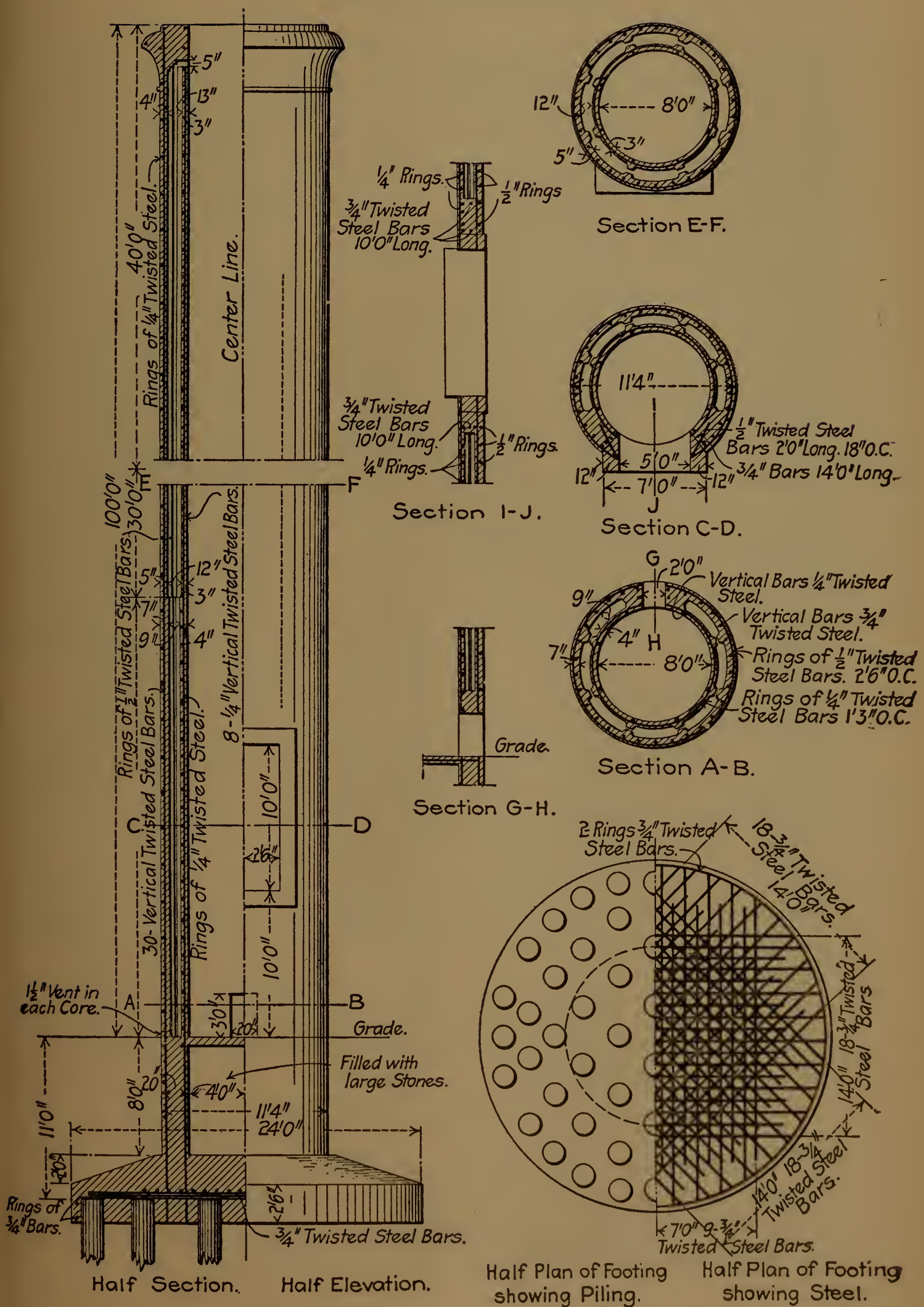


FIGURE 59.—RANSOME CONCRETE-STEEL CHIMNEY, CENTRAL LARD CO., HOBOKEN, N. J.

sorted to, the piles being driven on about 2½ foot centers and cut off below the level of surface water; they support a concrete bed two or more feet in thickness into which their tops extend. Concrete or brick foundations laid in cement mortar should be laid several weeks before the chimney is constructed in order that the cement should set properly. The sides of the foundations which are usually in the form of a truncated pyramid should have an inclination of at least 60 degrees to the horizontal. The depth of the foundation should be such as to properly distribute the load.

Steel Chimneys.—Chimneys built of sheet steel are common, particularly in Pennsylvania and the middle West. They may be either self-supporting or held in position by means of guy ropes. To save expense, it is not unusual for a considerable number of small guyed chimneys to be used in place of one large chimney. Steel chimneys are said to cost less than those built of brick but the market price of steel affects their relative cost considerably. Any steel chimney should be carefully calked at the joints and the vertical lap joints scarfed at the girth seams. The leakage of air that would otherwise occur and which unfortunately does occur in many cheaply built steel chimneys very greatly affects the draft. Self-supporting steel chimneys rest usually on cast iron base plates sometimes cast in sections and bolted together if the chimney be large enough to make this desirable. The base plate is held down by foundation bolts built in a brick or concrete foundation sufficiently heavy to prevent the chimney from overturning. The lower courses of a self-supporting stack are usually flared out at the base of the chimney in a conical or bell shape, to give stiffness to it, the height of this cone or bell being from 1½ to 2 times the diameter of the chimney above the bell and of a diameter at the base equal to the height of the bell.

A formula for determining the thickness of shell that is used by a firm which has built a large number of large self-supporting steel chimneys is as follows:

$$\frac{\text{Moment in inch pounds}}{0.7845\ D^2} = \text{stress per lineal inch.}$$

This assumes that the moment of the total wind pressure in pounds multiplied into the distance in inches of the section under consideration from the center of pressure, divided by the diameter of the chimney in inches squared multiplied by 0.7854, is

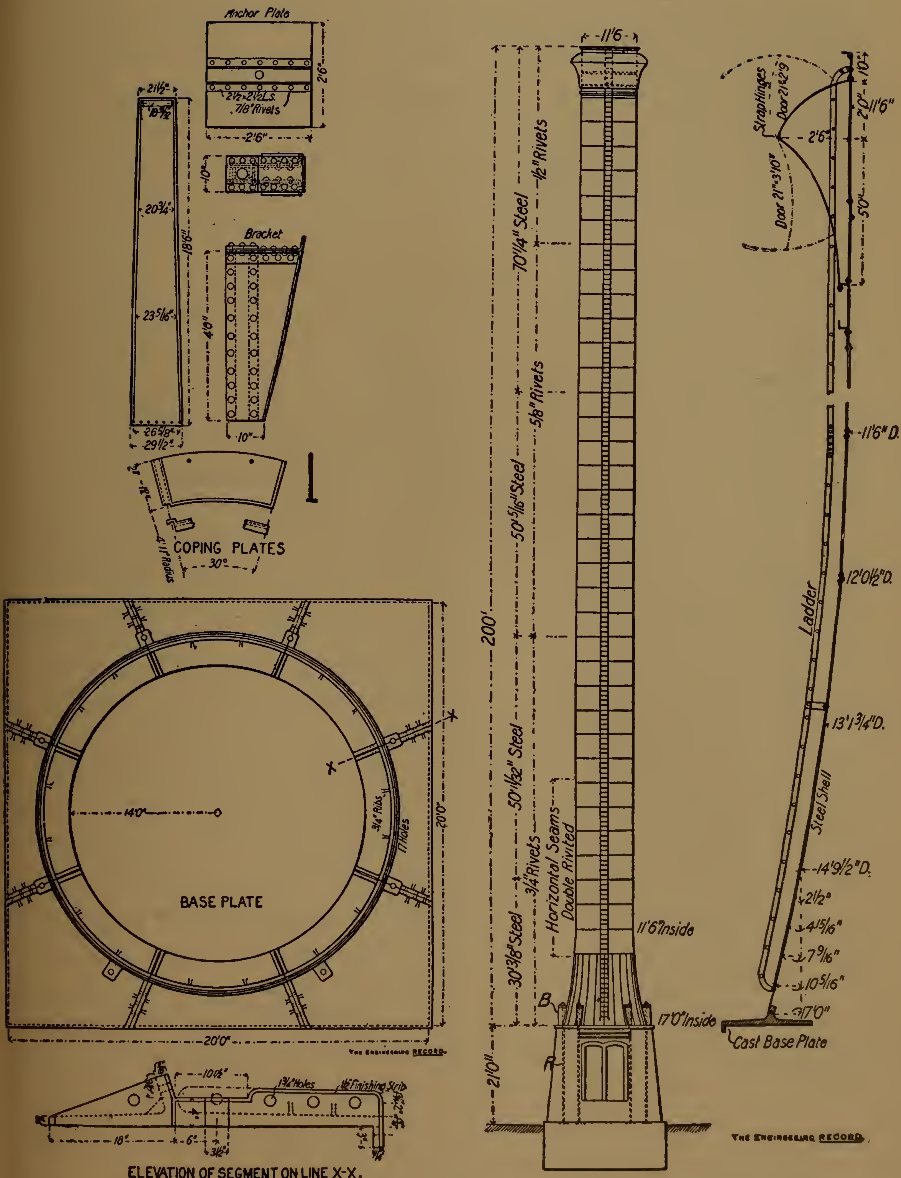


FIGURE 60.—STEEL CHIMNEY AT WILMERDING, PA.
BUILT BY THE RITER-CONLEY MANUFACTURING CO.

equal to the maximum stress per lineal inch in the shell. The total wind pressure is based on an assumed pressure of 25 pounds per square foot of projected area. A safe working stress is 10,000 pounds per square inch and this should be reduced by the efficiency of the riveted joint. If the efficiency of the riveted joint is 60 per cent., then 6,000 pounds per lineal inch would be a safe working strength and the ratio of the stress per lineal inch, as found by the equation, to 6,000 would be the thickness in inches of the shell required at the section under consideration. Calculation for the thickness of the shell should be made at the base of the stack, the top of the bell and several points between it and the top. The greatest strain occurs at the top of the bell. On account of deterioration that is apt to occur in the steel, it is undesirable to use shells less than 3/8 to 1/4 inch in thickness, depending upon the size of the chimney. This is particularly true near the top where the greatest corrosion is apt to occur, owing to the effect of the smoke that usually clings to the lee-side of a chimney.

For about one-fifth or one-quarter the height of a self-supporting chimney it is desirable that the girth seams be double-riveted. The lower edge of any sheet usually overlaps the upper edge of the sheet beneath it. For the sake of the greater stiffness, the vertical seams at the bell should also be double-riveted.

Foundations for Self-Supporting Steel Chimneys.—The foundations for self-supporting steel chimneys should have such a base that the load caused by the weight of the chimney and base and by the wind pressure on the leeward half of the base does not exceed the requirements of safety. The weight of the lining, if there be any, is to be neglected. The foundation must have sufficient mass so that the moment of the wind pressure into the height of the foundation plus half the height of the steel shell shall be equal to the weight of the shell plus the weight of the foundation into one-third the length of the base of the foundation.

If P is the total wind pressure, W_s the weight of the stack, W_f the weight of the foundation, all in pounds, then in Figure 56 the conditions as to stability are fulfilled when

$$P\left(\frac{H}{2}+h\right)=\left(W_s+W_f\right)\frac{b}{3}$$

The wind pressure can be obtained by calculation and the height of the foundation can be taken at from one-eighth to one-tenth the total height of the chimney. With these data and the weight of the stack being known, the weight of the foundation can be calculated. As concrete weighs about 140 pounds per cubic foot, the volume of the foundation can be determined and from this the area of the top and bottom. The slope of the sides can be to 1 to 3. Should the wind pressure be found to cause too great a load along the leeward edge of the foundations, a foundation of less depth which would insure one of greater area, can be assumed.

Chapter XI.—Coal Handling, Water Supply and Purification.

Coal Handling Machinery.—One of the most important factors governing the selection of a location for a steam power plant is the cost at which coal, assuming that fuel is to be used, can be transported to the boiler plant and the cost of the disposal of the ashes. Because coal can frequently be conveyed more cheaply by water than by rail large power plants are located upon navigable waterways, if it is convenient to do so, and if not, the site should be near a railroad if possible, so that coal can be delivered at the minimum expense. Of course the expenditure to which one should go in providing means for handling coal and ashes depends entirely upon the amount to be burned, as the cost of handling coal may be so small compared to other expenses as to be of secondary consideration. Even in the smallest plants, however, thought should be given the matter of coal delivery. If coal-handling machinery is not used it is a convenient arrangement to have coal pockets or coal bunkers located next to the wall of a boiler room so that coal will fall from them by gravity to the floor in front of the furnace doors. If possible, arrangements should be made for delivering the coal to the bunkers directly from the coal cars, whether they be steam railroad coal cars or small dumping cars of an automatic railway run from some nearby wharf. As plants increase in size greater expense for coal handling apparatus is warranted. Means should be provided for storing coal and the amount of storage space required depends, in a measure, upon the effect of an enforced shut down. For an electric lighting or railway power station, or for power and heat for hospital buildings or public institutions, large storage capacity is imperative, particularly in localities where there is likely to be an interruption in the supply of fuel.

With plants as small as 500 or 600 horse power in boilers, coal-handling machinery is frequently installed. It usually consists of a receiving hopper, into which the coal is delivered, feeding to an endless chain of buckets which elevates and conveys the coal to bunkers placed over the boilers; spouts lead from the bunkers to the floor near the furnace doors of the boilers if they are hand fired or to mechanical stokers if the latter are provided. Frequently the conveyor is arranged to pass beneath ash hoppers under the boiler grates so that when not handling coal it can be used to elevate the ashes into a storage bin, from which they can be removed by carts or otherwise. When coal is delivered by boat to a plant, particularly in large stations, it is frequently elevated by a self-filling bucket and raised by a boom to an elevated coal hopper in a tower on the wharf; the coal falls through a

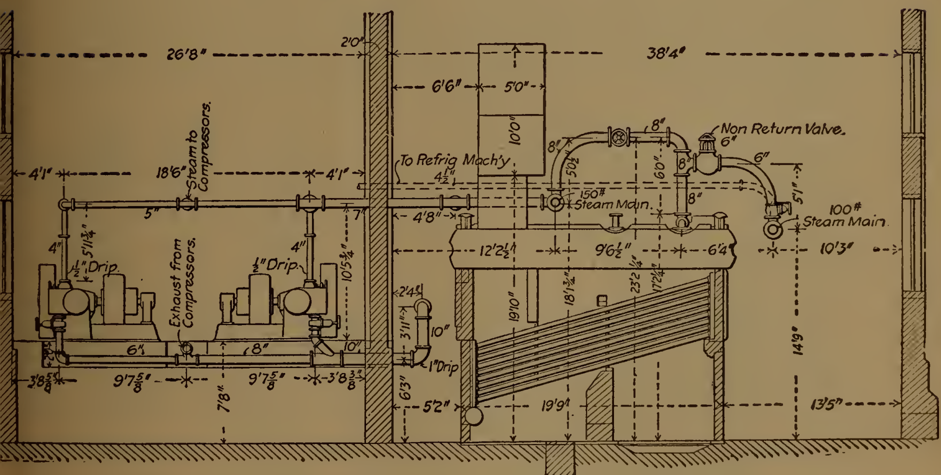


FIGURE 61.—CROSS-SECTION POWER HOUSE CENTRAL LARD CO.

crusher to reduce it in size and then passes through automatic weighing scales to the conveyors which deliver it into bunkers over the boilers. Sometimes large storage bunkers are built outside of a power house, these being filled by conveyors, while a second conveyor extends from the storage bunker to a smaller bunker over the boilers. Some idea of the varied manner in which coal handling machinery for power houses may be applied can be had by examining the illustrated catalogues of makers of this class of machinery.

Cost of Coal Handling by Machinery.—For the purpose of showing the relative cost of hand firing and a modern coal-handling equipment combined with mechanical stokers in large plants, some pertinent figures have been obtained through the courtesy of a well-known electrical supply company owning a plant, containing about 7,500 horse-power in boilers, which was operated for some time after construction without any kind of coal-handling machinery other than small hand cars which were loaded by hand from railway cars outside of the building and then hauled up a slight incline to the boiler house, so that the fuel could be dumped in front of the furnaces. The cost is given for two months, a year apart; during the first month, the plant was run without and during the second month with coal-handling machinery and mechanical stokers. The coal-handling plant consisted of a McCaslin conveyor so arranged that the coal was only handled by hand in shoveling it out of railway cars onto the conveying system:

May, 1900.	Wages.	Tons of coal burned.	Cost per ton.
16 firemen and one helper	$981.80	4,292	$0.229
11 coal and ash men. Ash removing by contract	634.66	.....	0.1478
May, 1901.			
3 firemen and 2 helpers	287.75	6,975	0.041
11 coal and ash men, 2 conveyor men	654.50	.....	0.0938

The saving in wages of firemen and helpers amounts to 18.8 cents per ton, which is 82.1 per cent. or $1,311.30 per month. The saving on coal and ash handling is 5.4 cents per ton, which is 41.4 per cent. or $376.55 per month, or a total saving of $1,687.85 per month or over $20,000 per year. Were it not for the fact that, owing to peculiar local conditions the coal has to be shoveled from the coal cars onto the conveyor system the cost of labor might be still further reduced.

Cost of Boiler-Room Labor.—This matter is of importance in deciding upon methods of handling coal in steam power plants and some valuable information in this connection was contained in a report by Mr. R. S. Hale to the Steam Users' Association, whose members represented nearly 400 mill owners, largely in New England and the Middle States. Judging from the replies received from members owning a total of about 600 boilers, it costs to move coal by hand (wheel barrow) about 1.6 cents per ton per yard up to distances of five yards, then about 0.1 cent per ton per yard for each additional yard. Mr. Hale found that one man, besides a night man, can run an engine and fire up to about

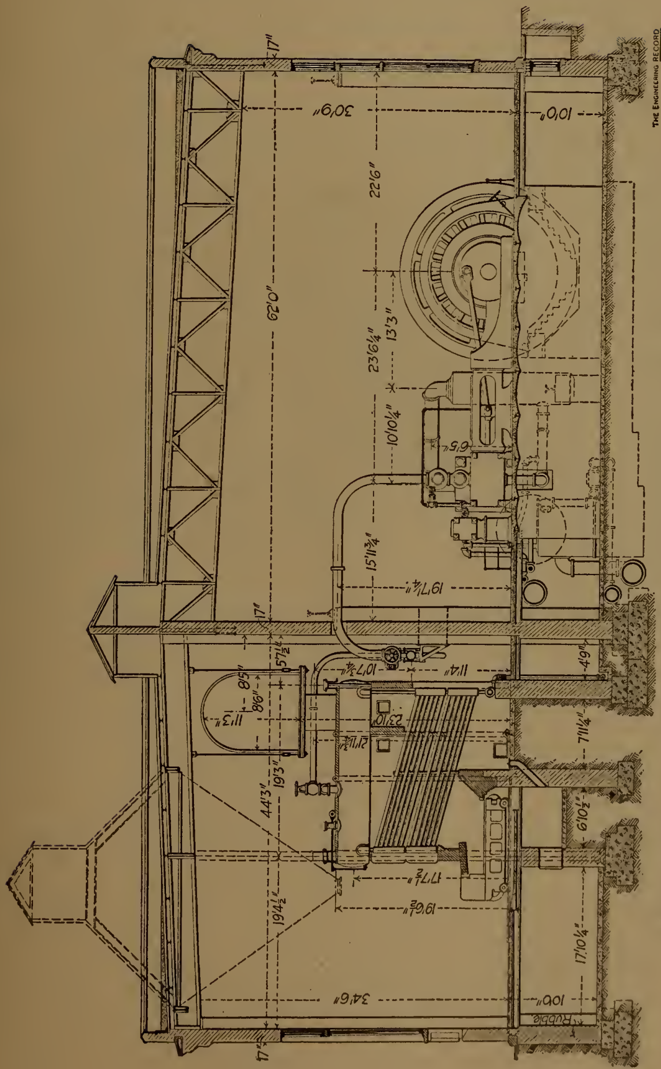


FIGURE 62.—CROSS-SECTION POWER HOUSE DES MOINES CITY RAILWAY CO.
SARGENT & LUNDY, CONS. ENG'RS.

10 tons of coal per week. One man, besides an engineer and night man, can fire up to about 35 tons per week. Two men, besides an engineer and night man, can fire up to about 80 tons per week. These figures assume that the night man does all he can of the banking, cleaning and starting. The figures are for average conditions. If the conditions are exceptional, as, for instance, where there is a very long wheeling distance or a very variable load, proper allowance should be made. Mr. Hale states in the report that mechanical stokers save from 30 to 40 per cent. of the labor in plants burning from 50 to 150 tons per week and save no labor in small plants. Boiler attendants are paid about $1.50 per day, working from 10 to 12 hours. The average cost of firing coal, according to the report, was 48 cents per ton, the maximum 71 cents and the minimum 26 cents.

Mechanical Stokers.—In the chapter on mechanical draft it was explained that perfect combustion required just the proper amount of air to be supplied the furnace of a steam boiler. That requirement is difficult to fulfill where fuel is fired by hand. If the bed of fuel is too thin there will be an excess of air, and if too thick too little air will enter the furnace, both of which will cause a loss for reasons previously explained. This is particularly true with bituminous coals. Theoretically, therefore, the best result would be obtained by firing small quantities at frequent intervals rather than larger quantities of fuel less often. Frequent firing requires that the fire doors be opened often, and this means a loss due to too much air entering while the doors are open. The practical objection to frequent firing lies in the fact that it is difficult to get a fireman who will fire a boiler properly, most of men who perform this kind of labor being of the class that prefers shoveling a large amount of fuel into the furnace and then sitting down for half an hour than to shoveling a lesser amount at shorter intervals. The kind of men who take an interest in stoking properly, and do it, are not likely to remain firemen. For the reasons given, the only way in which coal can be supplied to a furnace in such a manner as to produce the best results is by mechanical means or mechanical stokers. There is every reason to believe that a plant of boilers using bituminous coal mechanically fired will operate with a sufficient economy of fuel over hand-fired boilers to pay a good return on the increased investment required for the stoking apparatus. This result can

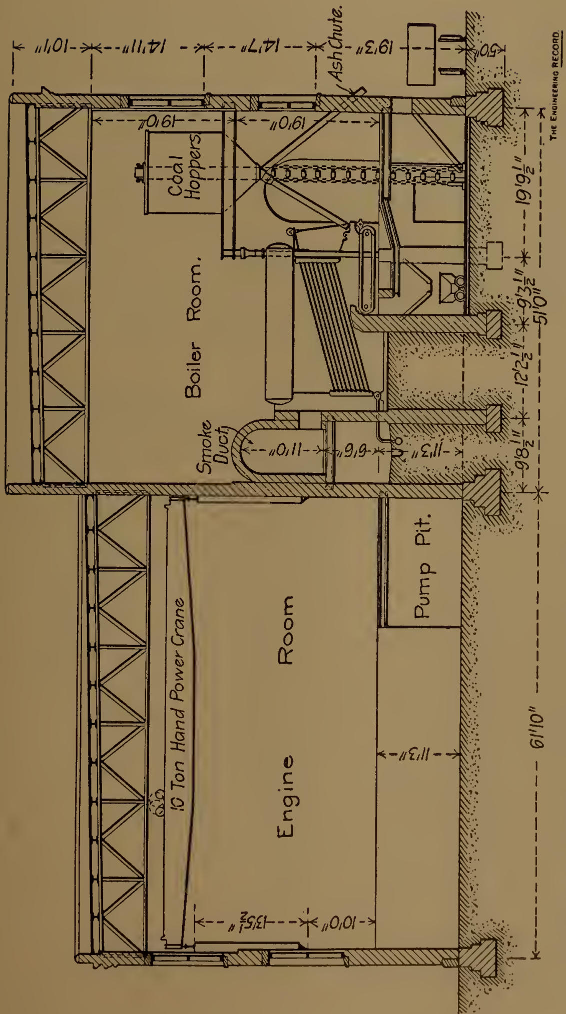
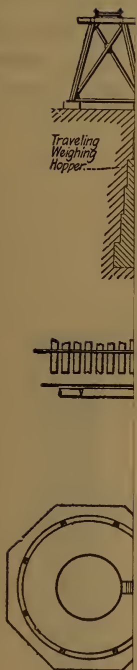


FIGURE 63.—CROSS-SECTION POWER HOUSE SCHWARZCHILD & SULZBERGER, CHICAGO, ILL.

L. LEVY, CHIEF ENGINEER.

be effected by any fireman who is sufficiently intelligent to do ordinary firing but not, however, unless the mechanical stoker is given some attention to see that holes in the fire do not occur or that clinker does not form to impede the uniform movement of the coal. The stoker is not wholly automatic, but, given a fair degree of attention, it is an indispensable aid to the firemen in the attainment of the perfect combustion of bituminous coal. Mechanical stokers are also of value in reducing smoke with bituminous coal and reducing the number of men in the boiler plants. They are of particular value in making it possible to burn low-grade smoking coals in situations where the emission of smoke would be objectionable. As an improper air supply affects the amount of smoke emitted by coal, mechanical stokers properly designed very much reduce the smoke and oftentimes entirely do away with it.

Burning Pulverized Fuel.—There is one method of mechanically supplying coal to boilers that gives promise of much success and that is after pulverization. The principal difficulty that has been met in the past has been the expense of pulverizing, but recently methods have come into use that make this method commercially possible. Various types of mills for this purpose have been in use in the cement industry, ever since the development of the rotary kiln, that have given excellent satisfaction. Bituminous coal is ground and stored in hoppers from which it is delivered usually by a small screw conveyor to a pipe about four inches in diameter and terminating in the mouth of the kiln in which the cement is burned. A blast of air passing through the pipe delivers the coal to the kiln where it burns. The jet of flame is six or eight feet long and of such an intensity that it could not be introduced directly under a steam boiler, hence a boiler would have to be equipped with a detached or separate furnace to use fuel in this way. One difficulty with this method of using pulverized fuel lies in the danger of storing it in quantities and to overcome this machines have been devised to pulverize the coal and deliver it directly to the furnace, each boiler being equipped with a pulverizer. By pulverizing the fuel the combustion of all the coal is complete as there is no loss due to the falling of fine particles of coal through the grate bars as there is in other methods of firing. The relative amounts of air and coal supplied can be adjusted to a nicety hence there is no loss through too little or


Traveling Weighing Hopper

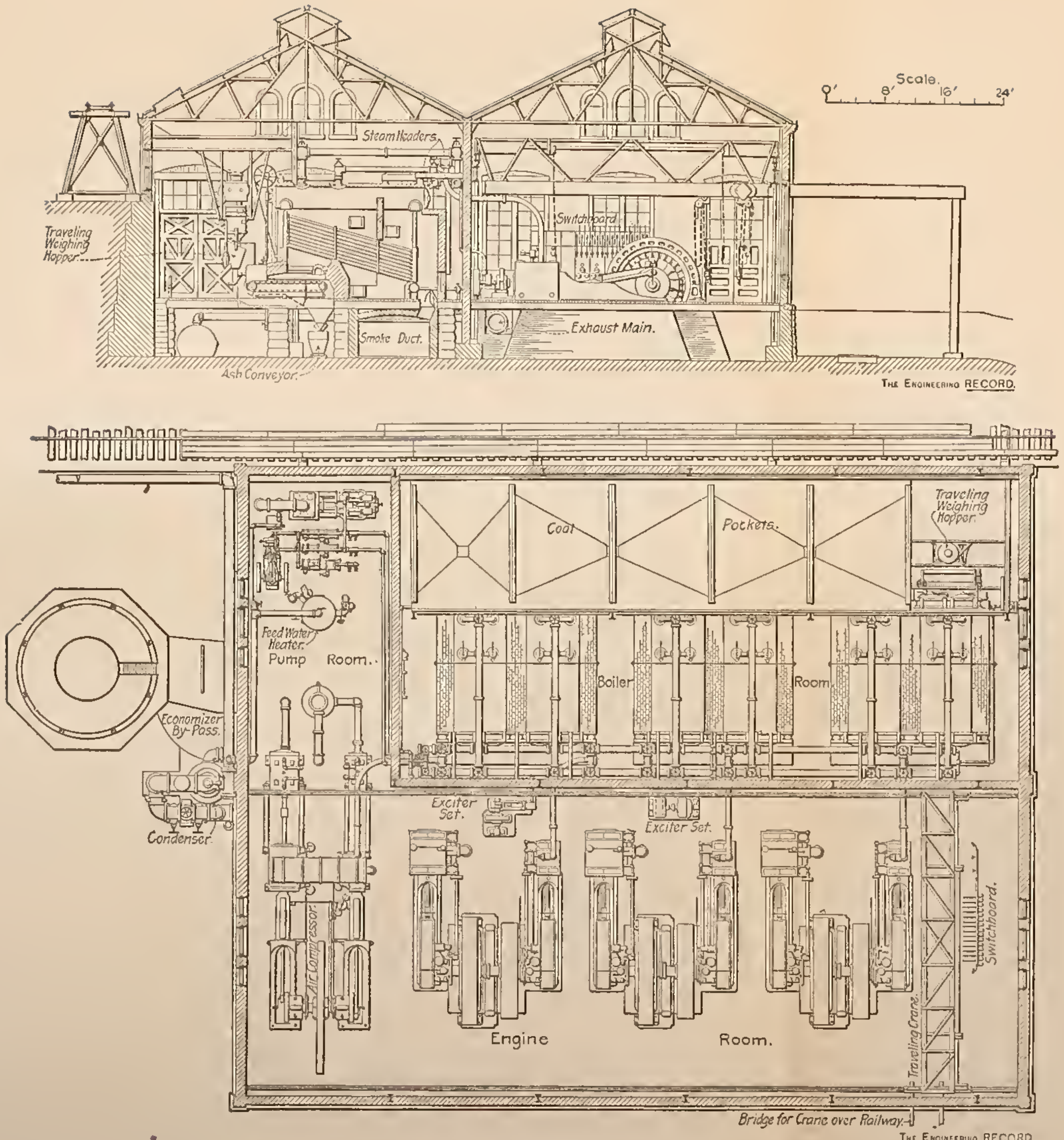


Plate 16.—La Bella Power Plant, Goldfield, Cal.

L. L. Summers, Engineer.

too much air. Furthermore, the coal can be burned with absolutely no smoke.

Supply of Boiler Water.—The amount of water used for steam boilers in large plants is of such a quantity that an abundant supply of good water at low cost is a deciding factor in the selection of a site for a power house. The cost of water from city mains is usually such that it is desirable for large plants to go to some other source, as a river if one be available, artesian wells, etc. Before determining upon a supply, investigation should be made

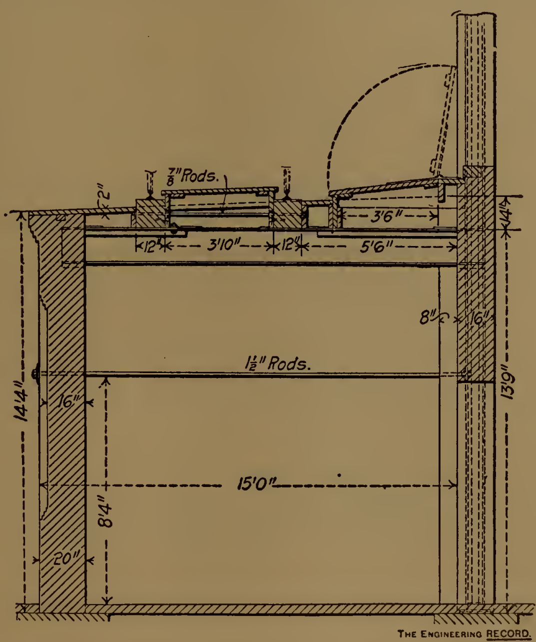


FIGURE 64.—COAL BUNKERS DESIGNED BY SHEAFF & JAASTED.

as to whether or not the water is suitable for boiler purposes, the opinion of a chemist, not a boiler-compound quack, being obtained upon this point. Water may be unfit for boiler purposes without treatment on account of the presence of sewage which will cause foaming, or of certain salts which will form hard scale on the heating surface of the boilers and not only impair their efficiency but also entail large expense for cleaning them and be a source of probable danger besides. Deep well waters frequently contain scale-forming salts and occasionally rivers receiving the rainfall from certain watersheds. River water is frequently contaminated with sewage. Sewage can probably be best removed for boiler purposes by mechanical filtration; silt and mud by sedimentation and the same process.

Water Softening.—The salts which give the most trouble in steam boiler waters by the formation of scale are the carbonates and sulphates of lime and magnesia. The presence in water of these salts, except the sulphate of magnesia, cause it to be hard, and their removal is known as water softening. Sulphate of magnesia does not cause scale to form, but it is precipitated by concentration. Its presence prevents the removal of other salts by chemical treatment due to the fact that when lime water is added to water containing it, it breaks up and sulphate of lime results, which does form scale. A small quantity of carbonate of lime is apt to remain in solution in almost pure water, but if the water be saturated with carbonic acid the amount of carbonate of lime in solution can be very much greater; most of it being precipitated when the carbonic acid is driven off, as it is when the water is heated. By adding lime water to water containing carbonate of lime and carbonic acid, the lime combines with the carbonic acid to form carbonate of lime which, while the carbonate of lime previously in the water, is precipitated because of the disappearance of the carbonic acid. Carbonate of magnesia is very similar in its properties to carbonate of lime. It is also acted upon by the lime water so as to form hydrate of magnesia, an almost insoluble salt, and carbonate of lime which is precipitated. The removal of the carbonates by the addition of lime water was proposed by an English chemist named Clark and this process generally bears his name.

Sulphate of lime and sulphate of magnesia may be broken up by adding sodium carbonate, the former resulting in the formation of calcium carbonate, which is precipitated, and sodium sulphate, a non scale forming salt. The magnesium sulphate is split up into sodium sulphate and carbonate of magnesia and the latter can be further broken up by adding lime to cause an additional reaction in the manner described. The amount of sulphate of lime that can be dissolved in water depends upon the temperature, but above a temperature of about 100 degrees Fahrenheit the solubility of this salt diminishes. At 300 degrees Fahrenheit it is said to be insoluble. By heating with live steam, therefore, it is possible to precipitate nearly all of the sulphate of lime in water containing it. Quite a little time, however, is required to complete the reaction. Heating water carrying sulphate of lime previous to its entering a boiler will only remove the excess of sulphate of lime over that possible to retain in solution at the temperature to which the water is heated, so that unless it is heated to a high temperature chemical treatment is necessary to precipitate that not precipitated by heating.

It will be noticed from what has been said that the carbonates

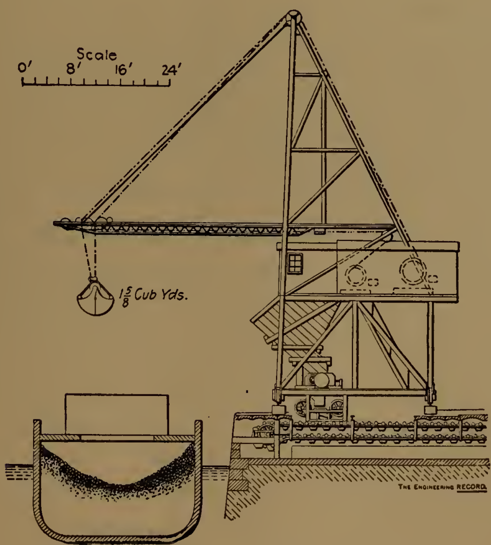


FIGURE 65.—TYPICAL COAL ELEVATING TOWER.

of lime and magnesia can be mostly removed by heating particularly if sufficient boiling occurs to drive off the carbonic acid, and some of the sulphate of lime can also be precipitated by heating. All of the four salts mentioned can be practically removed by chemical treatment. The heating method is more limited in its application but it is cheaper when it can be used. In certain cases chemical treatment or a combination of chemical treatment with the application of heat is alone possible. The best method can only be determined by a competent chemist after a proper investigation of the water is made, as it depends entirely upon the kind and quantity of salts present in the water.

Purifying with Exhaust Steam.—Almost all of the carbonates of lime and magnesia can be precipitated by moderate warming such as could be accomplished by exhaust steam in a feed-water heater of the open type. These are usually provided with trays over which the water trickles and filtering material to intercept the precipitated salts that do not lodge on the trays. Care should be taken that the heater is large enough to insure a low velocity of water passing through it to give time for the precipitation to occur in the heater and not in the boiler. Very often boiler waters are only objectionable because of the presence of salts which could be entirely removed practically in an exhaust-steam heater.

Purifying by Live Steam.—A live-steam purifier is a device similar in a general way to most open-type feed-water heaters in that it consists of a shell containing a number of trays that can be withdrawn by removing the head of the heater, the trays being arranged one over the other so that the water trickles slowly downward over them so as to become thoroughly heated by the live steam under full boiler pressure that is admitted to the heater. The heater is usually located over the boilers so that water will run from it to the boilers by gravity. As live steam is used in these heaters and the temperature is higher, much more of the sulphate of lime present in water will be precipitated than there will in an exhaust steam heater. This type of heater will eliminate the carbonates of lime and magnesia.

Purifying by Chemical Treatment.—The chemical treatment might be divided into two methods, the intermittent and continuous systems, and perhaps a third, with either combined with the application of heat to the water while being treated for the purpose of hastening the action of the reagents. The original Clark process was the intermittent method and it has been modified more or less by others. Usually this system consists of two large settling tanks in which the water to be treated is run, the proper amount of chemicals added, the mixture agitated and then allowed to stand for some hours while the precipitate settles, the clear treated water on top being used and the sludge that settles being blown off when necessary. This system usually requires two tanks so that the treatment can go on in one while purified water is being drawn from the other. An objection to it is the first cost. With the continuous method the untreated or raw water and the necessary chemicals are mixed in the desired proportions, the supply of both being relatively constant, both being fed into a tank of generous size so arranged that the current is sufficiently slow for the precipitate to settle in the bottom of the tank, the clear water passing on to the outlet. The continuous treatment combined with heat has been used with no little success. The Sorge-Cochrane system of water purification consists in heating water almost to the boiling point, purifying by a simple chemical treatment and at the same time neutralizing such acids as are present in the water. The supply of water to the boilers is made up of all the pure condensation that can be saved and utilized, and of just enough fresh cold water supplementing this condensation to supply the demand. The conversion of the sulphates and carbonates of lime and other soluble salts into insoluble and neutral salts is accomplished in a specially designed feed water heater of the open type by introducing into the water to be tested, before it enters the heater, a suitable chemical such as soda ash and simultaneously heating the water. Means are provided for varying the amount of the reagent, and also by means of a filter bed in the heater, of intercepting the precipitated salts and other insoluble matter.

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