STEAM HEATING

WARREN WEBSTER & CO.
CAMDEN, NEW JERSEY

STEAM HEATING

A Manual of Practical Data

Compiled by

THE GENERAL ENGINEERING COMMITTEE

OF

WARREN WEBSTER & COMPANY

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FOREWORD

THE subject of Heating and Ventilation has been covered broadly in many handbooks that are available for reference, but there has been a demand also for a book of information confined exclusively to Steam Heating and covering that field in all necessary detail.

Steam Heating is therefore the one topic of this volume and the editors have aimed to cover the subject with comprehensive data, arranged in such convenient and useful form as will best meet the needs of technical men in the engineering and contracting fields.

The information given is authentic, being based upon actual practice and largely upon the experience of Warren Webster & Company, who, as pioneers, have specialized for more than thirty years in the effective use of steam for all heating purposes. Many of the designs and methods originated by this firm are now the recognized service standards.

Special articles and helpful suggestions have been contributed by John A. Serrell, by the General Engineering Committee, and by John B. Dobson, Ralph T. Coe, William Roebuck, Russell C. Brown, William F. Bilyeu and other members of the Warren Webster & Company organization.

"Steam Heating" offers the best thought of this organization, and as part of Webster Service is intended to be of real value throughout the profession. The observance of good judgment and painstaking care in following its teachings will do much toward obtaining creditable and satisfactory results.

If further explanations, additional information or helpful co-operation are desired, the Engineers and Service Men in the branch offices of Warren Webster & Company throughout the country are always available for consultation and assistance.

General Engineering Committee of Warren Webster & Company

William M. Treadwell, *Chairman*

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CHAPTER I

Elements of Steam Heating

THE purpose of a heating system is to warm the interior of a structure to a desired degree of temperature and to maintain this condition against a lower exterior degree. It is usual to assume the exterior temperature to be the average minimum expected in the locality.

To warm the interior and to maintain a given temperature, heat is required to replace that which is absorbed by the contents and that transmitted through the structure to the exterior.

The unit measure of heat in English-speaking countries is the British thermal unit, which is the heat necessary to raise the temperature of one pound of water from 39 to 40 deg. fahr. This is commonly known as B.t.u., or heat unit.

The quantity of heat required to raise the temperature of a given weight of a substance through one deg. fahr. as compared with the quantity of heat required to raise the same weight of water from 62 deg. to 63 deg. fahr. is called the specific heat of that substance. (See Table 27-00, page 00.)

The heat content, or quantity of heat per degree of a given mass of a substance, is the product of its specific heat and its weight in pounds.

The rate at which initial heat is required to raise the temperature of a cold structure and its contents to the desired degree in a given time may be much greater than that necessary to maintain the required temperature after initial heating, or warming up, has been accomplished.

The greater the length of time permitted for initial warming, the less difference there will be between the heat requirement per unit of time during initial heating and that required during subsequent maintenance.

Heat losses by transmission through various forms of building structure have been ascertained with more or less accuracy, and much information on this subject has been published from time to time. These data are being constantly improved as new forms of construction appear.

The principal discrepancies between published data on transmission are probably due mainly to various allowances which have been included for infiltration. Infiltration, or air leakage, should be considered independently of structural transmission.

Local differences in workmanship and material of structure, as well as errors in observation, have further contributed to discrepancies, and in many instances the results of tests observed at one temperature difference have been reduced by direct proportion to a "per-degree-difference" basis.

Until recently it has not been generally recognized that this last-mentioned basis is in error, in that it is likely to give too high a rate of heat loss for smaller temperature difference and too low a rate for larger temperature difference than that existing during the test.

The heat loss factors in Chapter 3 are based upon experience with various substances used in construction under average conditions at a difference

of 70 deg. fahr. between interior and exterior temperatures. Factors for other temperature differences are stated as percentages of the 70 deg. normal. The effects of exposure and of varying wind velocities are separately considered as losses due to infiltration.

In order to determine the amount of heat required it is necessary to determine:

First: The lowest temperature to which the interior will fall, that is, the "initial" temperature; and the temperature which it is desired shall be maintained within the enclosure, or the "maintained" temperature;

Second: The time period in which it is required that the structure and its contents must be raised from initial to maintained temperature;

Third: The nature and the weight of the building and its contents (especially if large quantities of glass, metal or water are included);

Fourth: The minimum exterior temperature;

Fifth: The direction of the prevailing cold winds, and their anticipated velocities;

Sixth: The construction of the enclosure;

Seventh: The topography of the site, and other local peculiarities.

The heat transmitted hourly through the structure at a temperature difference between maintained interior and minimum exterior temperatures, plus the heat required to warm the infiltrated air through the same difference of temperature, gives the hourly maximum heat requirement during maintenance. In Chapters 3 and 4 these two causes for heat requirements are further discussed.

During initial heating or "warming up," heat units in addition to those required for maintenance must be supplied to raise the temperature of the structure and its contents of air and stored materials from their initial temperature to the desired temperature.

In practice the heat absorbed by the structure and its stored materials is usually neglected, as the error is small. However, if the interior walls or columns are massive, or if the contents of the building include large quantities of materials with high specific heats, such as iron, steel, water, glass, etc., heat absorption of these must be taken into account.

In almost all cases the heat required to raise the air contents of the enclosure from the initial to the maintained temperature must be considered.

After determining the amount of heat required to warm the various substances during initial heating, the hourly rate at which this additional heat must be supplied during initial heating is obtained by multiplying this heat quantity by the reciprocal of the warming-up period in hours.

Applications of the problem of determining the heat requirements will be found in Chapter 5.

Where the heating requirements for warming-up are large compared with those for maintenance, the radiation necessary for the warming-up requirements and consequently the heat emitted will be correspondingly

excessive during maintenance. It is often advisable to increase the length of the warming-up period first allowed in order to reduce this excess radiation.

Having estimated the total hourly heat requirement, the next consideration is the proper proportioning and distribution of radiating surfaces throughout the enclosure for obtaining the desired heating effect from the circulation of a fluid of higher temperature.

In the following chapters the fluid considered for conveying heat is steam at pressures slightly above that of the atmosphere. The high thermal value, or B.t.u. per pound, of steam and the convenience with which it can be utilized by means of commercial boilers, radiating surfaces, pipe and fittings and the special apparatus of the Webster System, has demonstrated the superiority of steam at low initial pressures for the great majority of installations.

The radiating surfaces, or radiation, normally used in low-pressure steam heating to transmit heat from steam to the enclosure to be warmed, are of two general classes—Direct and Indirect—each of which has many specific sub-divisions.

Direct radiation, properly classified, comprises only those arrangements of radiating surface which are located directly in the space to be heated.

Radiation which is not wholly exposed in the space to be heated is termed *indirect* radiation. Units which are concealed under window boxes, or in housings having an air inlet near the floor line and a heated air outlet above the radiation, or which are enclosed in casings outside of the space to be heated and which have a cool air inlet from any source and a warm air connection to convey by heated air the necessary heat units to the space to be heated are examples of this type of radiation.

Until recently the circulation of air for indirect heating by the method last mentioned was induced entirely by the difference in weight of the air columns before and after coming into contact with the enclosed radiating surface. Present usage designates such surfaces as *indirect*, distinguishing them from surfaces used in the later development, where additional circulating velocity is imparted mechanically by a fan or blower. Where mechanical means are used these surfaces are now designated as blower or fan-blast surface.

Certain forms of radiating surfaces exposed in a room and so arranged with dampers and ducts that air wholly from the room or partly from without may be used to convey heat from the surface of the radiator to the room, are called *direct-indirect* surfaces.

The rate of heat-flow through radiating surfaces from a given interior to a given exterior temperature varies not only with all classes of radiation but with all sub-divisions of those classes. This is due mainly to variation in convection, that is, in the facility for absorption of heat from the outer surface into the surrounding medium, and in a lesser degree, to the dispersion of radiant heat. So great is this variation that, under similar conditions of location and temperature difference, and even in the simplest form of direct radiation, a low, narrow radiator will give off 40 per cent. more heat per square foot of radiation than one that is extremely high and wide.

The term "square feet of radiation," therefore, means nothing specific and should not be used indiscriminately for sizing boilers, mains, or other apparatus in the heating system.

The type of radiating surface for the local conditions, heat requirements and architecture, having been selected and placed, the proper size of radiating units should be determined. For this purpose the information on heat emission of various types of radiation, Chapter 6, will be found useful.

The pipes which convey the heating fluid from its source to the radiating surfaces are termed supply mains. Those conveying the products of condensation from the radiating surfaces to the point of disposal are termed return mains. The vertical parts of these mains are usually called risers, to distinguish them from horizontal runs. Risers, in turn, are classified by direction of flow, as *up-feed* or *down-feed* risers. The small branches to individual units of radiation are known as run-outs; those supplying several units as branches, and those conveying all of the heating medium are usually termed trunk mains.

The flow of the heat-carrying medium is always toward a lower pressure, and if the medium is steam confined in pipes or ducts sealed from the atmosphere, the arbitrary dividing line conventionally drawn between "pressure" and "vacuum" does not enter. The problem involves only heat content, density, difference in pressure, condensation and friction.

If the lowest terminal pressure in the system is that of the atmosphere as in an open-return line or modulation system, the initial pressure must be somewhat above the atmosphere. The amount of pressure above the atmosphere depends largely upon the friction which must be overcome in the piping and upon the pressure necessary to give the steam its initial velocity. If, however, a terminal pressure lower than that of the atmosphere is mechanically maintained, as in vacuum systems, the initial pressure may be above, at or below that of the atmosphere as best meets the local conditions.

Vacuum system practice, with few exceptions, demands that a steam pressure equal to that of the atmosphere be maintained in the run-outs most distant from the source of steam supply, in order to avoid the air infiltration that would otherwise probably occur through minute leaks. This terminal pressure requires an initial pressure higher in some degree than that of the atmosphere. Local conditions, such as source of supply, length and character of pipe run, and use and permanency of the plant, make the selection of pressure difference one of good engineering judgment rather than the application of any fixed rule. The proper basis for proportioning the supply system is dealt with in Chapter 11.

The primary function of return mains is the removal and disposal of the products of condensation. These mains should provide gravity flow wherever possible. Pressure difference should be used to stimulate flow only where gravity alone is not practical.

The products to be removed consist of water, air, vapor, gases from impurities, and last, but not to be overlooked, dirt.

The last consists of initial impurities such as core-sand, gravel, chips, mill scale, grease, etc., left in the heating system when erected, together with rust particles and scale from impure feed water. Were it not for the

dirt which collects and the uncertainty as to its volume, return mains might be made much smaller.

Formulæ and tables of capacities of straight, smooth pipes laid on even grades for return of condensation, and tables of accepted capacities compensating for uncertainties of grade and dirt are given in Chapter 12.

The hot, distilled water should be returned to the boiler wherever possible. The saving due to the heat content of this water and its freedom from scale-forming and other impurities, warrants considerable initial outlay in return apparatus.

No specific type of return apparatus will best fit all conditions. The simple, low-pressure heating boiler may have its water line so located that the water of condensation will flow back into the boiler by gravity against the highest steam pressure carried. Between such a boiler and the modern high-pressure central generating plant, where part of the exhaust steam is used as a by-product for heating purposes in an extended group of buildings, there is a wide range of conditions. The selection of the best combination of return apparatus for the individual plant is, therefore, dependent upon comprehensive, practical experience.

Some of the possible combinations of return apparatus are described and shown in typical diagrams in Chapter 14, and basic rules are given for estimating proper sized apparatus. However, it is manifest that discussion in this volume cannot cover all requirements, and in this, as in the selection of all apparatus for special conditions, it is recommended that specific engineering advice be obtained from the Home Office of Warren Webster & Company or its nearest Branch Office, before a selection is made.

CHAPTER II

Basic Data Required for the Design of a Steam Heating System

INTELLIGENT design of any heating system in either new or existing buildings requires that certain basic data shall be available. For existing buildings the present use of which will continue, it is usually possible to obtain quite definite data to work upon. If only plans are available, much of the necessary information must be based upon assumptions of probable conditions.

In any event, good judgment, preferably founded upon ripe experience, must play its equal part with scientific knowledge in the final application of the data obtained.

Topography: The design of an efficient heating system especially where a group of buildings is being considered, requires that a careful study be made of the grade levels of the different buildings, each one to the other, so that, if possible, the condensation from the heating surfaces may flow by gravity to a central point from which it may be returned to the source of steam supply.

In cases where the conditions will not allow of the gravity flow of condensate to a central point, special methods for lifting the condensate to a higher level are necessary as described hereafter.

Location and Character of Source of Heat: It follows from the above that wherever possible the source of steam supply should be located at a lower level than that of the buildings to be heated.

In a plant consisting of a group of buildings there is usually a power generating plant, the by-product from which, in the form of exhaust steam, should be utilized to the fullest extent in the heating of the buildings. The economies incident to the use of this exhaust steam as a by-product frequently determine the adoption of an isolated power generating plant rather than the purchase of power from outside sources and the installation of a boiler plant for heating purposes only.

Exposure and Protective Conditions: By exposure is meant the relation of the outside surfaces of the building or buildings to the prevailing cold winds of winter, which by their pressure cause infiltration of excess quantities of cold air and the rapid removal of heat from the outside surfaces of the structure. To offset this, a greater amount of radiation must be provided on the sides having this exposure, than for the sides more favorably located with the protection of surrounding buildings or hills.

Consequently the designer should determine the direction of the prevailing winter winds, their probable velocities and duration as well as the topographic features which may afford protection.

Outside Temperatures: Although the records of the U. S. Weather Bureau (See Figure 2-1) may show an extreme minimum temperature much lower than that usually experienced in a given locality, it is not customary to estimate heating requirements with that extreme temperature as a basis.

Generally, the average minimum temperature, obtained from records

Fig. 2-1. Lowest Recorded Temperatures. U. S. Weather Bureau Statistics

over a period of years, is the fundamental consideration, or heating surface is provided for an estimated minimum outside temperature approximately 10 degrees above the lowest recorded minimum, unless this extreme temperature is likely to prevail throughout long periods.

It is possible to operate either the Webster Modulation System or the Webster Vacuum System with a slight increase in steam pressure which results in an increased rate of heat emission from the radiating surfaces. This flexibility may be used to advantage during short periods of unusually cold weather.

Floor Plans, Elevations and Cross Sections: To properly design the heating system for a building or buildings, it is necessary that complete floor plans and sufficient elevations and cross-sections, showing the details of construction, materials, etc., shall be available to assure an accurate calculation of the heating requirements.

In designing heating systems for existing buildings, accurate data may be obtained by survey, but with designs of new buildings it is often necessary to make certain assumptions which may or may not be justified when the construction is complete. A frequent element of error lies in deviations from the architect's original plans without proper consideration for their effect upon the heating system.

These possible discrepancies in construction and deviation in design from original plans make it quite necessary for the designer of the heating system to place himself on record as to the basic factors of his calculation.

Inside Temperature Requirements: The temperature which it is desired to maintain and the lowest degree to which the temperature will be allowed to fall, are usually governed by the use which will be made of the enclosure.

Inside temperatures are usually determined at the breathing line and not closer than five feet from the most exposed wall.

The important considerations for decision lie in the following questions:

Is the heat to be maintained continuously 24 hours per day or for stated portions of the total 24 hours?

If intermittent heating, how long a time may be allowed to raise the room temperature to the required maintained temperature?

Through how long a period will heat be shut off and how low may the room temperature become during this closed down period?

The following table indicates the usual range in maintained temperatures desired for various classes of occupancy, but it should be kept in mind that temperature is largely a matter of individual preference so that such a table can only be considered as a guide in the final selection.

Table 2-1. Temperature for Various Rooms in Degrees Fahrenheit

Bath rooms	75 to 85
Churches	60 to 70
Entrance halls to public buildings	50 to 60
Factories	60 to 70
Foundries	50 to 60
Gymnasiums	60 to 65

Homes for aged	80
Hospitals	72 to 75
Lecture halls	60 to 70
Living rooms	68 to 72
Machine shops	60 to 70
Offices	68 to 72
Operating room	70 to 90
Paint shop	80 to 90
Prisons—Day confinement	60 to 65
Prisons—Night confinement	50 to 55
Public buildings	68 to 72
Schools	70
Shops (stores)	50 to 65
Swimming halls	70 to 75
Vestibules for stores and office buildings	70 to 80

The relative humidity of the atmosphere which is likely to exist in any room or building has a bearing upon the desirable inside temperature.

For a living apartment, a normal temperature of 70 deg. fahr. and relative humidity of 50 per cent (about 4 grains of water vapor per cubic foot of content) is considered by most authorities to be a very satisfactory condition of the air. If the temperature is lower than 70 deg., the relative humidity should be higher than 50 per cent or if the temperature is higher, the relative humidity should be lower if the same effect of comfort to the occupant is to result.

It is usual, however, that the relative humidity is found to be much less than 50 per cent in living apartments heated to 70 deg. fahr. and has been observed to be as low as 28 per cent. With very low relative humidity the effect upon the occupant is a feeling of chilliness even though the temperature may be increased to 78 or 80 deg. fahr. This cooling effect is due to the rapid evaporation of moisture from the occupant's skin which is brought about by the low vapor pressure of the atmosphere. Conversely, where extremely high relative humidity exists, a temperature of 70 deg. fahr. might feel oppresively hot to the occupant.

Contents and Use of Enclosure: A very important consideration for the designer is that of the materials and machinery within the enclosure, and their capacities for absorbing heat. This has an important bearing upon the permissible time limit for warming up.

Large quantities of material or machinery having a high heat content will prolong the time for warming and will have an opposite effect of retarding the loss of temperature when the heat supply is cut off.

For consideration of this factor, the designer should have details of the weight and substance of each of the various items of machinery and materials. With this data and a table of specific heats of substances such as on page 000, the total heat contents or heat absorbing capacities which influence the warming up period can be determined.

Likewise, the designer should determine the total heat given off by the operation of the machinery, motors, lights, etc., although this is not of so much importance in buildings where the temperature requirements are those to be maintained during periods when machinery, etc., are not in operation.

In schools, theaters, auditoriums, churches, etc., where large numbers of persons may gather, it is necessary to allow for the heat given off by the human bodies if overheating is to be prevented. In such cases, ventilation is usually required to remove the bodily heat with its excessive humidity.

In manufacturing plants, portions of buildings often require unusual quantities of heat to warm the large amounts of air which replace that drawn from the rooms through exhausting fans on grinders, dryers, and similar apparatus. This condition requires a careful investigation of the factors involved in the unusual rate of air change.

Character and Location of Heating Surfaces: The selection of the radiation from a choice of direct, indirect, direct-indirect, or blast type depends largely upon the use for which the enclosure is intended, the ventilation requirements, the local building laws, school and labor codes, and other general considerations. (See Chapter 6.)

Whether pipe coils, cast-iron wall radiation or the usual cast-iron radiators are to be used for direct heating is usually a question of availability of materials, cost of installation and the esthetic effect required.

The selection of the type and location of the different radiating units may best be determined by a study of the plans and elevations of the building to be heated.

Location of Supply and Return Lines: In installations of the type of hotels, hospitals, office buildings or other public buildings with finished or decorated walls it is customary to conceal the steam and return risers, and their run-outs to radiators, in the wall and floor construction. In factory installations and other less expensive types of construction these lines are exposed and in many instances they are used as prime radiating surfaces.

In cases where the outlets from the risers are taken below the level of entrance to the radiators it is essential that the run-outs shall be so graded that the condensation will flow back by gravity into the risers regardless of the maximum velocity of steam which may flow in the opposite direction. It is therefore of prime importance that the maximum velocity shall be kept well below that at which the condensation will be swept along with the steam. This important feature of design is discussed in further detail in Chapter 12.

A down-feed system of supply is preferable wherever building conditions will permit since the condensation will then flow in the same direction and will be assisted by the flow of steam as well as by gravity. This permits the use of smaller supply risers and run-outs due to the higher velocities of steam flow which are permissible.

Return run-outs, risers and mains must grade in the direction of flow of condensation to some low point or points from which the condensation will be returned to the source of steam supply or other point of disposal.

CHAPTER III

Heat Transmission

THE same principle of flow of heat from a higher to a lower temperature which makes steam heating effective also functions in the transmission of heat through materials of construction to make such heating necessary.

Heat seeks equilibrium, and consequently flows from a higher to a lower temperature with greater or less rapidity, depending upon the difference in temperature and the character and thickness of the material through which it flows.

For the purpose of estimating the heat losses from enclosures, numerous tests and deductions from practice have been made to determine the rate of heat flow or transmission through the various types and materials of surfaces used for enclosing space. So many variables enter this problem that it is impossible to predict the heat flow exactly unless all of the peculiarities of any case under consideration have been previously determined.

Tables of Heat Transmission, therefore, attempt to provide for average conditions of construction of the enclosing substances. Due regard is given to the facility with which heat is absorbed and removed from the surfaces of the enclosing substances, and to the heat which is transmitted through them due to the difference between the temperatures existing at their surfaces which may be termed "heat head."

This heat head has been considered in many formulæ as a constant increase per degree of temperature difference. As the result of tests with the same substance under various temperature differences this deduction has been proved to be incorrect. Higher temperature differences cause a greater heat flow per degree difference than lower temperature differences.

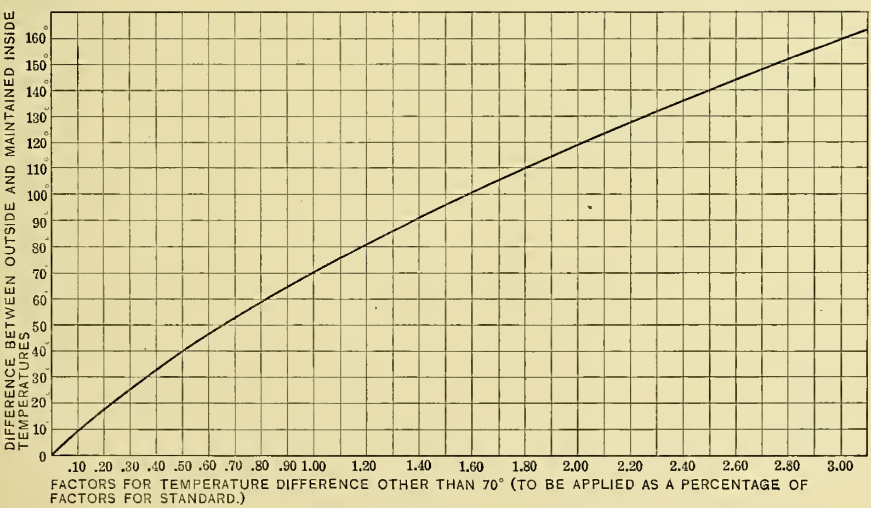


Fig. 3-1. Factors for temperature differences other than 70 deg.

Table 3-1. Factors for Temperature Differences other than 70 degrees Fahrenheit

Where temperature difference is	40°	50°	60°	70°	80°	90°	100°	110°	120°	130°	140°	150°	160°
Multiply Transmission Losses for 70 deg. difference by.....	.50	.66	.82	1.00	1.19	1.39	1.59	1.80	2.00	2.25	2.50	2.75	3.00

The probable variation in heat flow under various conditions of heat head is shown in Table 3-1. The rate of flow for any difference between inside and outside temperatures other than 70 deg. is expressed as a percentage of that which will flow at 70 deg. difference. (See Figure 3-1, which is a curve constructed from the values in Table 3-1.)

The discussion of Rates of Heat Transmission in this book recognizes the following fundamental conditions:

(1) The maintained inside temperature is that normally existing at the breathing line (5 feet above the floor) and about 5 feet from a wall. The breathing line is more often mentioned hereafter as the datum line.

(2) The basic rate of transmission for any substance is the number of B.t.u. which will be transmitted in an hour through each square foot of surface of that substance when the outside temperature is zero and the maintained inside temperature is 70 deg. fahr.

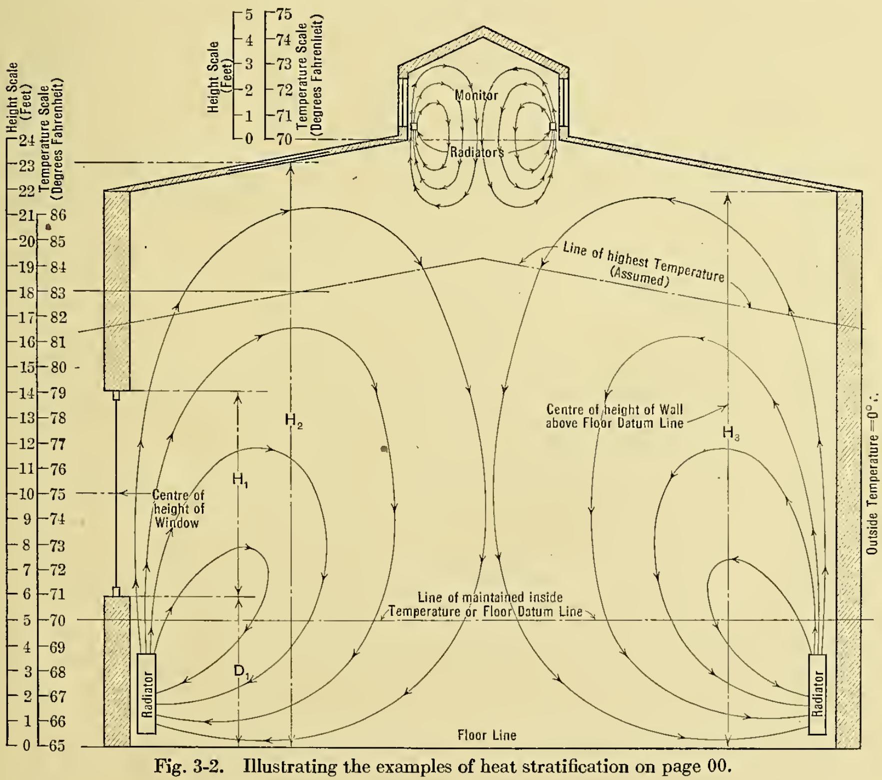


Fig. 3-2. Illustrating the examples of heat stratification on page 00.

(3) From the above it will be evident that the basic rate is that which is transmitted at the datum line.

(4) Tests in structures where the air is not agitated or mixed by moving fans, belts, pulleys, etc., indicate that an increase of temperature of about 1 deg. for each foot from the datum line may be expected up to the point where heat is lost by transmission through the roof or ceiling.

(5) This point where heat is lost through a roof or ceiling is assumed to be 5 feet below the surface of same.

(6) Because of this stratification of heat, it is necessary, in the case of high structures, to figure upon an increase in rate of transmission due to the greater heat head caused by the higher average inside temperature.

The application of this so-called stratification, or height factor, differs for roofs, for windows, and for such vertical surfaces as walls, doors, partitions, etc.

For instance, if the average height or center of a window is 10 feet above the floor line, as in Figure 3-2, the temperature T_1 which might be expected at that point is 70 deg. plus 1 deg. for each foot of height of the center of the window above the 5-foot datum line, or 75 deg.

Due to the higher temperature difference of zero to 75 deg. instead of the standard 0 to 70 deg. and the consequent increase of heat head, the transmission losses must be increased by a factor or per cent of the loss existing at the 5-foot line where 0 to 70 deg. is assumed to exist.

This factor F is obtained from Figure 3-1 and for 75 deg. difference the transmission losses are found to be 109 per cent of the normal.

For any other window the inside temperature T_1 assumed to exist at the average height is

$$T_1 = \left(\frac{H_1}{2} + D_1 - 5\right) \text{deg.} + 70 \text{ deg. or}$$

$$T_1 = \frac{H_1}{2} + D_1 + 65$$

where D_1 is the number of feet of height above floor of the lower edge of window opening and H_1 is the number of feet of height between upper and lower edges of window opening. In every case the factor F to be applied to normal transmission loss is obtained from Figure 3-1.

Because of the transmission of heat through roof construction, it is usual to consider in connection with roof factors that the inside temperature does not increase due to stratification beyond 5 feet below the under side of the roof. Therefore, the factor for stratification assumes the existence of two limits, one 5 feet above the floor and the other 5 feet below the roof.

For instance, in a building having a sloping roof which has an average height above floor of 23 feet, as in Figure 3-2, the average temperature under the roof is assumed to be that existing 5 feet below the roof or 18 feet above the floor. The temperature at that point is assumed to be 70 + 18-5 or 83 deg.

The temperature T_1 for the factor to be used for a roof of any other distance above the floor is $T_1 = (H_2 - 10)$ deg. + 70 deg. or $H_2 + 60$ where H_2 is the number of feet of average height of the roof above the floor.

With the temperature T_1 known, the factor F to be applied for heat head may be obtained from Figure 3-1.

For convenience, the revised heat transmission rates for different heights of roofs have been shown in the following tables.

Heat losses through monitors must be specially considered. In such cases it is usual to install heating surfaces within the monitor construction, and for that reason the entire monitor construction should be considered as an individual unit of enclosure with an imaginary floor across the space between its lower edges.

However, the factors for stratification for figuring heat losses from monitors should disregard the 5-foot roof datum line; that is, it is assumed that the temperature at this imaginary floor line is approximately 70 deg.

For walls, doors, partitions, or any other vertical surfaces the lower edge of which is at floor level, the method of obtaining the factor for stratification is the same as for windows, above described, except that the formula for obtaining the temperature (T_1) at the point of average height is

$$T_1 = \left(\frac{H_3}{2} - 5\right) \text{deg.} + 70 \text{ deg.}$$

where H_3 is the number of feet of height above floor of the upper edge of the surface.

In the cases where consideration must be given to the transmission of heat through floors, such as floors above cellars or other cold spaces, and floors laid upon the ground, it is unnecessary to provide factors for stratification, since the transmission rate at the established difference in temperature between underside of floor and datum line covers this loss with a margin of safety.

It will be understood that the foregoing factors have all been adjusted for the basic temperature difference between zero outside and 70 deg. maintained inside.

If the outside temperature for which any particular enclosure is figured is different from zero, or if the temperature to be maintained at the breathing line is more or less than 70 deg., or if both inside and outside temperatures are different from basic, the rates of transmission should again be adjusted for the new difference in temperature.

This new factor is obtained from Table 3-1 or Figure 3-1, and is applied to the total of all transmission losses through the structure which have previously been calculated for the standard 70 deg. difference.

Rates of Heat Transmission in B. t. u. per hour per sq. ft. of Surface for Temperature Difference, 0 deg. to 70 deg. Fahr.

Table 3-2. Walls, Clapboard

CONSTRUCTION	TOTAL HEIGHT OF WALL 10 Ft. F=1.00	15 Ft. F=1.04	20 Ft. F=1.09	25 Ft. F=1.14	35 Ft. F=1.23	45 Ft. F=1.33	55 Ft. F=1.43
Clapboard on studs—bare	50	52	55	57	62	67	72
Clapboard on studs—papered inside	45	47	49	51	55	60	64
Clapboard on studs—with lath and plaster	35	36	38	40	43	47	50
Clapboard on studs—paper with lath and plaster	30	31	33	34	37	40	43
Clapboard on studs—with 1-in. sheathing bare	40	42	44	46	49	53	57
Clapboard on studs—with 1-in. sheathing papered	35	36	38	40	43	47	50
Clapboard on studs—with 1-in. sheathing lath and plaster	32	33	35	37	39	43	46
Clapboard on studs—with 1-in. sheathing papered, lath and plaster	25	26	27	29	31	33	36
Clapboard on studs—with 1-in. sheathing wood insulation, papered	35	36	38	40	44	47	50
Clapboard on studs—with brick fill—bare	28	29	31	32	34	37	40
Clapboard on studs—with brick fill—papered	25	26	27	29	31	33	36
Clapboard on studs—with brick fill—lath and plaster	22	23	24	25	27	29	31
Clapboard on studs—with brick fill—papered, lath and plaster	20	21	22	23	25	27	29
Clapboard on studs—saw dust fill, inside sheathing, lath and plaster	15	16	16	17	18	20	21
Clapboard on studs—saw dust fill, inside sheathing, papered, lath and plaster	10	10	11	11	12	13	14
Clapboard on studs—sheathing, back plaster, lath and plaster	25	26	27	29	31	33	36

Table 3-3. Walls, Stucco on Studs

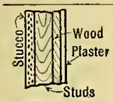
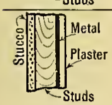
	CONSTRUCTION	TOTAL HEIGHT OF WALL 10 Ft. F=1.00	15 Ft. F=1.04	20 Ft. F=1.09	25 Ft. F=1.14	35 Ft. F=1.23	45 Ft. F=1.33	55 Ft. F=1.43
	With wood lath and plaster on the inside	40	42	44	46	49	53	57
	With metal lath and plaster on the inside	45	47	49	51	55	60	64

Table 3-4. Walls, Corrugated Iron

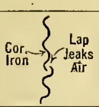
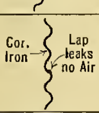
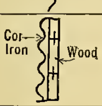
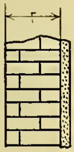
	CONSTRUCTION	TOTAL HEIGHT OF WALL 10 Ft. F=1.00	15 Ft. F=1.04	20 Ft. F=1.09	25 Ft. F=1.14	35 Ft. F=1.23	45 Ft. F=1.33	55 Ft. F=1.43
	Plain loose construction	125	130	136	143	154	166	179
	Plain tight construction	90	94	98	103	111	120	129
	Sheathed, tongued and grooved	45	47	49	51	55	60	64

Table 3-5. Walls, Brick

THICKNESS IN INCHES T	TOTAL HEIGHT OF WALL 10 Ft. F = 1.00	15 Ft. F = 1.04	20 Ft. F = 1.09	25 Ft. F = 1.14	35 Ft. F = 1.23	45 Ft. F = 1.33	55 Ft. F = 1.43
PLAIN							
4	50	52	55	57	62	67	72
8	30	31	33	34	37	40	43
12	22	23	24	25	27	29	31
16	18	19	20	21	22	24	26
20	16	17	17	18	20	21	23
24	14	15	15	16	17	19	20
28	12	13	13	14	15	16	17
32	10	10	11	11	12	13	14
36	8	8	9	9	10	11	11
PLASTERED							
4	48	50	52	55	59	64	69
8	28	29	31	32	34	37	40
12	20	21	22	23	25	26	29
16	15	16	16	17	18	20	21
20	14	15	15	16	17	19	20
24	12	13	13	14	15	16	17
28	11	12	12	13	14	15	16
32	10	10	11	11	12	13	14
36	8	8	9	9	10	11	11
FURRED AND PLASTERED							
4	32	33	35	37	39	43	46
8	20	21	22	23	25	27	29
12	15	16	16	17	18	20	21
16	12	13	13	14	15	16	17
20	11	12	12	13	14	15	16
24	9	9	10	10	11	12	13
28	8	8	9	9	10	11	11
32	7	7	8	8	9	9	10
36	6	6	7	7	7	8	9



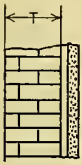


Table 3-6. Walls, Hollow Tile

THICKNESS IN INCHES T	TOTAL HEIGHT OF WALL 10 Ft. F = 1.00	15 Ft. F = 1.04	20 Ft. F = 1.09	25 Ft. F = 1.14	35 Ft. F = 1.23	45 Ft. F = 1.33	55 Ft. F = 1.43
PLAIN							
4	45	47	49	51	55	60	64
6	40	42	44	46	49	53	57
8	28	29	31	32	34	37	40
10	24	25	26	27	30	32	34
12	18	19	20	21	22	24	26

(Table 3-6 continued on next page)

Table 3-6. Walls, Hollow Tile *(Continued)*

Plastered on One Side

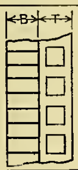
Thickness in Inches T	10 Ft. F=1.00	15 Ft. F=1.04	20 Ft. F=1.09	25 Ft. F=1.14	35 Ft. F=1.23	45 Ft. F=1.33	55 Ft. F=1.43
4	40	42	44	46	49	53	57
6	35	36	38	40	43	47	50
8	25	26	27	29	31	33	36
10	20	21	22	23	25	27	29
12	16	17	17	18	20	21	23

Stucco, Furred and Plastered

Thickness in Inches T	10 Ft. F=1.00	15 Ft. F=1.04	20 Ft. F=1.09	25 Ft. F=1.14	35 Ft. F=1.23	45 Ft. F=1.33	55 Ft. F=1.43
4	30	31	33	34	37	40	43
6	28	29	31	32	35	37	40
8	20	21	22	23	25	27	29
10	16	17	17	18	20	21	23
12	14	15	15	16	17	19	20

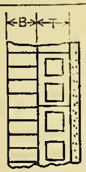
Table 3-7. Walls, Hollow Tile faced with Brick

Plain



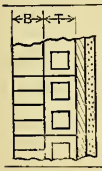
Thickness in Inches B	Thickness in Inches T	10 Ft. F=1.00	15 Ft. F=1.04	20 Ft. F=1.09	25 Ft. F=1.14	35 Ft. F=1.23	45 Ft. F=1.33	55 Ft. F=1.43
4	4	26	27	28	30	32	35	37
4	8	20	25	22	23	25	27	29
4	12	15	16	16	17	18	20	21
4	16	12	13	13	14	15	16	17

Plastered



Thickness in Inches B	Thickness in Inches T	10 Ft. F=1.00	15 Ft. F=1.04	20 Ft. F=1.09	25 Ft. F=1.14	35 Ft. F=1.23	45 Ft. F=1.33	55 Ft. F=1.43
4	4	24	25	26	27	30	32	34
4	8	18	19	20	21	22	24	26
4	12	12	13	13	14	15	16	17
4	16	10	10	11	11	12	13	14

Furred and Plastered



Thickness in Inches B	Thickness in Inches T	10 Ft. F=1.00	15 Ft. F=1.04	20 Ft. F=1.09	25 Ft. F=1.14	35 Ft. F=1.23	45 Ft. F=1.33	55 Ft. F=1.43
4	4	18	19	20	21	22	24	26
4	8	15	16	16	17	18	20	21
4	12	10	10	11	11	12	13	14
4	16	8	8	9	9	10	11	11

Table 3-8. Walls, Concrete faced with Brick 4 inches thick

Thickness in Inches B	T	Total Height of Wall 10 Ft. F = 1.00	15 Ft. F = 1.04	20 Ft. F = 1.09	25 Ft. F = 1.14	35 Ft. F = 1.23	45 Ft. F = 1.33	55 Ft. F = 1.43
Plain								
4	4	35	36	38	40	43	47	50
4	8	28	29	31	32	34	37	40
4	12	22	23	24	25	27	29	31
4	16	18	19	20	21	22	24	26
Plastered								
4	4	32	33	35	37	38	43	46
4	8	25	26	27	29	31	33	36
4	12	20	21	22	23	25	27	29
4	16	16	17	17	18	20	21	23
Furred and Plastered								
4	4	21	25	26	27	30	32	34
4	8	20	21	22	23	25	27	29
4	12	15	16	16	17	18	20	21
4	16	12	13	13	14	15	16	17

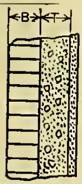


Table 3-9. Walls, Stone faced with Concrete 4 inches thick

Thickness in Inches C	T	Total Height of Wall 10 Ft. F = 1.00	15 Ft. F = 1.04	20 Ft. F = 1.09	25 Ft. F = 1.14	35 Ft. F = 1.23	45 Ft. F = 1.33	55 Ft. F = 1.43
Plain								
4	4	50	52	55	57	62	67	72
4		40	42	44	46	49	53	57
4	12	35	36	38	40	43	47	50
4	16	27	28	29	31	33	36	39

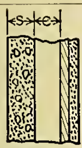
(Table 3-9 continued on next page)

Table 3-9- Walls, Stone faced with Concrete 4 inches thick (*Continued*)

Plastered

Thickness in Inches C	T	10 Ft. F=1.00	15 Ft. F=1.04	20 Ft. F=1.09	25 Ft. F=1.14	35 Ft. F=1.23	45 Ft. F=1.33	55 Ft. F=1.43
4	4	45	47	49	51	55	60	64
4	8	36	37	39	41	44	48	52
4	12	32	33	35	37	39	43	46
4	16	24	25	26	27	30	32	34

Furred and Plastered



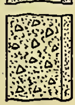
Thickness in Inches C	T	10 Ft. F=1.00	15 Ft. F=1.04	20 Ft. F=1.09	25 Ft. F=1.14	35 Ft. F=1.23	45 Ft. F=1.33	55 Ft. F=1.43
4	4	33	34	36	38	41	44	48
4	8	27	28	29	31	33	36	39
4	12	23	24	25	26	28	31	33
4	16	18	19	20	21	22	24	26

Table 3-10. Walls, Porous Stone or Porous Concrete

Plain

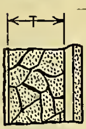
Thickness in Inches T	10 Ft. F=1.00	15 Ft. F=1.04	20 Ft. F=1.09	25 Ft. F=1.14	35 Ft. F=1.23	45 Ft. F=1.33	55 Ft. F=1.43
4	75	78	82	86	92	100	107
6	65	66	71	74	80	87	93
8	55	57	60	63	68	73	79
10	50	52	55	57	62	67	72
12	45	47	49	51	55	60	64
16	38	40	42	43	47	51	54
20	33	34	36	38	41	44	47
24	27	28	29	31	33	36	39

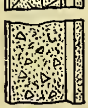
Plastered



Thickness in Inches T	10 Ft. F=1.00	15 Ft. F=1.04	20 Ft. F=1.09	25 Ft. F=1.14	35 Ft. F=1.23	45 Ft. F=1.33	55 Ft. F=1.43
4	67	70	73	76	83	89	96
6	58	60	63	66	71	77	83
8	49	51	53	56	60	65	70
10	45	47	49	51	55	60	64
12	41	43	45	47	50	55	59
16	34	35	37	39	42	45	49
20	29	30	32	33	36	39	42
24	24	25	26	27	30	32	34

Stucco, Furred and Plastered





Thickness in Inches T	10 Ft. F=1.00	15 Ft. F=1.04	20 Ft. F=1.09	25 Ft. F=1.14	35 Ft. F=1.23	45 Ft. F=1.33	55 Ft. F=1.43
4	50	52	55	57	62	67	72
6	43	45	47	49	53	57	62
8	37	39	40	42	46	49	53
10	33	34	36	38	41	44	47
12	30	31	33	34	37	40	43
16	25	26	27	29	31	33	36
20	22	23	24	25	27	29	31
24	18	19	20	21	22	24	26

Table 3-11. Walls, Hard Stone or Concrete

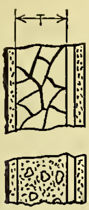
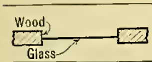
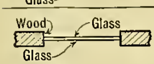
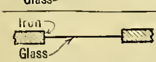
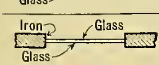
	THICKNESS IN INCHES T	TOTAL HEIGHT OF WALL 10 Ft. F = 1.00	15 Ft. F = 1.04	20 Ft. F = 1.09	25 Ft. F = 1.14	35 Ft. F = 1.23	45 Ft. F = 1.33	55 Ft. F = 1.43
				PLAIN				
	4	70	73	76	80	86	93	10
	6	60	62	65	68	74	80	86
	8	50	52	55	57	62	67	72
	10	45	47	49	51	55	60	64
	12	40	42	44	46	49	50	57
	16	35	36	38	40	43	47	50
	20	27	28	29	31	33	36	39
	24	20	21	22	23	25	27	29
				PLASTERED				
	4	63	66	69	72	78	84	90
	6	54	56	59	62	67	72	77
	8	45	47	49	51	55	60	64
	10	41	43	45	47	51	55	59
	12	36	37	39	41	44	48	52
	16	32	33	35	37	40	43	46
	20	24	25	26	27	30	32	34
	24	18	19	20	21	22	24	26
				STUCCO, FURRED AND PLASTERED				
	4	47	49	51	54	58	63	67
	6	40	42	44	46	49	53	57
	8	33	34	36	38	41	44	47
	10	30	31	33	34	37	40	43
	12	27	28	29	31	33	36	39
	16	23	24	25	26	28	31	33
	20	18	19	20	21	22	24	26
	24	13	14	14	15	16	17	19

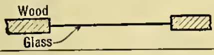
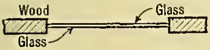
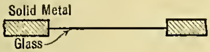
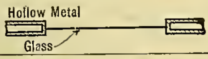
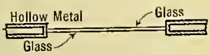
Table 3-12. Roof Glass and Skylights

The surface to be considered is the total surface of glass and frame.

	CONSTRUCTION	BASIC RATE F = 1.00	AVERAGE HEIGHT OF ROOF ABOVE FLOOR 25 Ft. F = 1.27	30 Ft. F = 1.38	35 Ft. F = 1.48	45 Ft. F = 1.70	55 Ft. F = 1.90
	Wood frame, single glazed....	75	95	104	111	128	143
	Wood frame, double glazed....	42	53	58	62	72	80
	Iron sash, single glazed.......	90	114	124	133	153	171
	Iron sash, double glazed......	60	76	83	89	102	114

Rates of Heat Transmission *(Continued)*

Table 3-13. Windows

CONSTRUCTION		Basic (or Rate) Factor
	Wood Frame Single Glazed	75
	Wood Frame Double Glazed	42
	Metal Frame Single Glazed	90
	Metal Frame Single Glazed	80
	Metal Frame Double Glazed	60
	Metal Frame Double Glazed	44

The factors in this table are for transmission rates at the datum line 5 feet from floor and a temperature of 70 deg. Fahr. The temperature T_1 at the centre of a window of any height above the floor will be

$$T_1=\left(\frac{H_1}{2}+D_1-5\right)^\circ+70^\circ$$

Where H_1 is the number of feet of height of the upper edge of window opening above lower edge.
D_1 is the number of feet of height of the lower edge of window opening above the floor.

With T_1 established, the factor for correcting the tabular values will be determined from Fig. 3-1. Apply this corrected factor to the entire area of window opening.

Monitors must be considered as separate problems as if they are structures of themselves with theoretical floors at the level of the base of the monitor. Their transmission losses and the sizing and placing of radiating surfaces should be figured accordingly. The factor should disregard the usual 5-foot datum line. That is, assume that the temperature at this imaginary floor line is 70 deg. Fahr.

Table 3-14. Roof Construction

CONSTRUCTION	BASIC RATE F=1.00	AVERAGE HEIGHT OF ROOF ABOVE FLOOR 25 Ft. F=1.27	30 Ft. F=1.38	35 Ft. F=1.48	45 Ft. F=1.70	55 Ft. F=1.90
Tile on strips	85	108	117	126	145	162
Tile on sheathing	45	57	62	67	77	86
Slate on strips	85	108	117	126	145	162
Slate on sheathing and paper	35	45	48	52	60	67
Corrugated iron on strips	125	159	173	185	213	238
Corrugated iron on sheathing	45	57	62	67	77	86
Tin on strips	110	140	152	163	187	209
Tin on sheathing	40	51	55	59	68	76
Tin on sheathing with paper	30	38	41	44	51	57
Shingles on strips	60	76	83	89	102	114
Shingles on sheathing	30	38	41	44	51	57
Shingles on strips over tar paper and tight boards	15	19	21	22	26	29
Cinder composition 2-in. paper, tar and gravel	25	32	35	37	43	48
Concrete composition 2-in. paper, tar and gravel	50	64	69	74	85	95
Concrete composition 3-in. paper, tar and gravel	45	57	62	67	77	86
Concrete composition 4-in. paper, tar and gravel	45	57	62	67	77	86
Hollow tile 4-in. paper, tar and gravel	20	25	28	30	34	38
Hollow tile 6-in. paper, tar and gravel	18	23	25	27	31	34
Metropolitan 3-in. paper, tar and gravel	20	25	28	30	34	38
Metropolitan 4-in. paper, tar and gravel	15	19	21	22	26	29
1-in. wood with 5 to 8-ply paper and gravel	20	25	28	30	34	38
1½-in. wood with 5 to 8-ply paper and gravel	18	23	25	27	31	34
2-in. wood with 5 to 8-ply paper and gravel	15	19	21	22	26	29
2½-in. wood with 5 to 8-ply paper and gravel	12	15	17	18	20	23
2-in. Federal cement tile, paper and tar and gravel	50	64	69	74	85	95

Rates of Heat Transmission *(Continued)*

Table 3-15. Floors

	Thickness in Inches	Basic (or Rate) Factor
Concrete	4	8
Tile or Metal	6	7
Laid on Cinder Fill on Ground without Air Space	8	6

NOTE—For figuring the opposite heat losses apply the factors only to that part of the floor area which lies within 5 feet of the outside walls.

The interior is to be considered for "warming up" only; that is the calculation for this part involves the use only of the specific heat of the material which is to be multiplied by its weight times the difference between initial and final temperature.

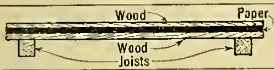
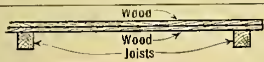
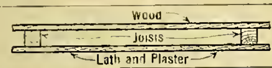
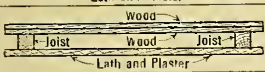
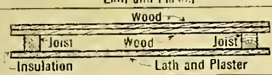
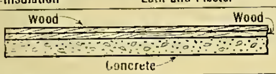
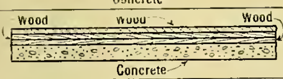
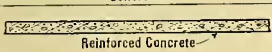
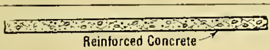
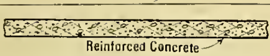
ABOVE COLD SPACE	DESCRIPTION	Basic Rate
	Mill construction—3-in. wood and paper plus ⅞-in. surface.	15
	Single wood on Joists..................................	40
	Double wood on Joists..................................	30
	Single wood on Joists with lath and plaster..............	30
	Double wood on Joists with lath and plater..............	20
	Double wood on Joists with insulation and lath and plaster.	10
	Double wood on fireproof concrete........................	10
	Wood flooring on double wood and fireproof concrete.....	5
	4-in. Concrete Slab, metal reinforced....................	70
	6-in. Concrete Slab, metal reinforced....................	60
	8-in. Concrete Slab, metal reinforced....................	50
	10-in. Concrete Slab, metal reinforced...................	45

Table 3-16. Doors and Wood Partitions

CONSTRUCTION	TOTAL HEIGHT OF DOOR OR WOOD PARTITION 10 Ft. F = 1.00	15 Ft. F = 1.04	20 Ft. F = 1.09	25 Ft. F = 1.14
¾" to 1" thick, tongued and grooved..........................	45	47	49	51
1" to 1¼" thick, tongued and grooved..........................	40	42	44	44
1¼" to 1½" thick, tongued and grooved..........................	35	36	38	40
1½" to 2" thick, tongued and grooved..........................	30	31	33	34
2" to 2½" thick, tongued and grooved..........................	25	26	27	29
2½" to 3" thick, tongued and grooved..........................	20	21	22	23

CHAPTER IV

Air Infiltration

WIND blowing against walls causes a leakage of air into the enclosure and an outward leakage from the enclosure through the opposite sides. Additional leakage is caused by temperature difference within and without regardless of wind velocity. These leakages are sometimes referred to as air change, but in this book are called *air infiltration.*

As the air enters and leaves the enclosure at different temperatures, sufficient B.t.u. or heat units must be provided to heat this air between the two temperatures. Air infiltration therefore becomes one of the important factors in the determination of heat losses in a room or an enclosure.

Some methods or formulae for determining the heat losses of an enclosure, either include the loss due to air infiltration in the heat transmission factors or base it upon the cubic contents of the space to be heated.

Examination of the air infiltration shows that most of the air leaks are around the doors, windows and other similar openings. The quantity that expresses the heat loss due to this infiltration of cold air should therefore be based upon the sum of the openings through which this leakage occurs, rather than upon the area of the doors, windows and similar openings of the structure.

Any determination of the quantity of air infiltrated must take into consideration the velocity and direction of the wind in relation to the openings of the enclosure. Where an enclosure has openings on more than one side, the infiltration for all openings must be determined and the radiation for this loss proportioned and located according to the maximum degree of infiltration that may occur on any side. This method will give an excess of radiation on the sides where leakage is outward, but there is no alternate without having some sides of the room feel cool at some wind direction.

The leakage in narrow monitors and rooms where cold drafts will not be objectionable may be considered only on the side where maximum wind velocities occur. A portion of the heat to care for this infiltration can then be applied to the other side. Where the wind strikes the surface at an angle, the resultant velocity at right angles to the surface must be considered. This is equal to the actual velocity times the sine of the angle of incidence.

Normally the same maximum wind velocity should be considered on the north and west sides, while on the south and east sides one-half of these velocities may be used except where special wind conditions exist.

A suggested extreme condition for New York and vicinity would be 20 miles per hr. wind velocity with a temperature of zero. Generally low wind velocities prevail at extreme low temperatures.

The many variables make reference to experiment easier than attempting to determine theoretically the perimeter air infiltration of windows, doors and similar openings. Little dependable experimental data is available at present, but this must be used as a basis until better is to be had.

Experiments on air infiltration of windows have been made by using a

fan to direct wind velocities against a test window set in the side of a tight enclosure and having an opening for Pitot tube readings on the opposite side. Further details regarding some of these experiments by Whitten will be found in the 1908 Transactions of the American Society of Heating and Ventilating Engineers, and others by Voorhees and Meyer in the 1916 Transactions.

Figure 4-1 gives the approximate leakage in cubic feet per minute per lineal foot of sash perimeter for a good frame, double-hung locked window, with and without metal weather strip.

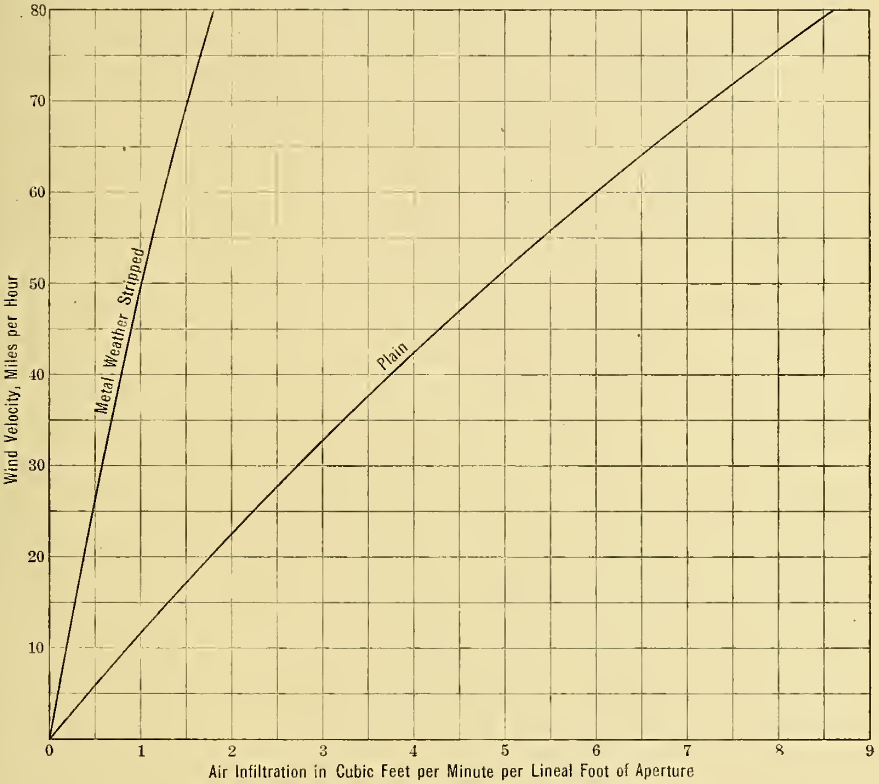


Fig. 4-1. Air infiltration for wood-sash window.

Due allowance must be made for loose-fitting sash, metal sash, pivoted sash, etc. In windows with steel section frames properly bedded, only the perimeter of that portion which opens need be considered. With standard double-hung sash, the meeting rail must be considered with the perimeter.

The leakage values as read from Figure 4-1 when multiplied by 60 times 0.087 (density of the air at zero), times 0.24 (specific heat of the air), will give the heat units per hour necessary to warm the infiltrated air one degree per foot of perimeter.

The following constants will be of value in calculating, as the product of the constant for the proper temperature difference and the air infiltration in cubic feet per minute gives the B.t.u. per hour required to heat this air through the temperature difference selected.

Example: Assume a double-hung frame window 3 ft. wide by 6 ft. high with perimeter of 21 ft., outside temperature 0, inside temperature 70 deg. fahr. with wind velocity of 20 miles per hr. Referring to Figure 4-1, the leakage per foot of perimeter is found to be 1.6. The conversion factor from the table is 87.696. Then $21 \times 1.6 \times 87.692 = 2946$ B.t.u. per hr. are required to heat the air infiltration from this window.

Temperature Difference between inside and outside air in deg. fahr.	Conversion factor Cubic ft. per min. to B.t.u. per hr.
1	1.253
40	50.112
50	62.640
60	75.168
80	100.224

CHAPTER V

Method of Calculating Heat Losses

CHAPTERS 1 and 2 give the general requirements that must be known in calculating the heat losses of any structure. Several rules and formulæ have been devised to determine the amount of heat that must be supplied to maintain a room or enclosure at a predetermined temperature with a known surrounding temperature.

Many of these formulæ are derived for an average size room and construction with standard size window openings, etc., and are not flexible enough to cover the problems of today.

If the air within an enclosure is maintained at a temperature higher than that surrounding, there must be a natural transfer of heat through the enclosing structure to the air of lower temperatures. This transfer may be to the air outside, to any adjoining rooms and to air above and below if these are at lower temperature than that in the room.

To heat the enclosure to and maintain it at a predetermined temperature, heat must be supplied equivalent to and at the rate at which it is lost. The most accurate method of determining this loss as generally agreed, is to determine the hourly rate of heat transfer from the heated enclosure to the surrounding air. This loss is usually calculated in British thermal units per hour; that is, on the B.t.u. basis.

The total losses are made up of four principal heat requirements.

First, is the heat required to warm to the desired inside temperature the air that leaks in through the various openings around the window and door perimeters, etc., from the outside. To calculate this loss, the width and lineal feet of the openings, and the wind velocity against the side of the enclosure where the openings are located, must be found, and with this data the air infiltration determined. The product of the air infiltrated in cubic feet per hour, the density of the air, its specific heat and the difference between the inside and outside temperatures is the heat required per hour for this loss. This subject is further discussed in Chapter 4 on Air Infiltration.

Second, is the loss by transmission of the heat through the various materials of which the enclosure is constructed. To calculate this loss, the area and kind of the various materials through which loss occurs, and the temperature difference between the air on the two sides of the material must be known.

The product of the area of any material in square feet, the transmission coefficient for that material in B.t.u. per hour, and the difference between the inside and outside temperatures will give the heat required per hour for the loss by transmission through that particular material. The sum of the losses so found for all materials of the structure is the total loss of heat from the enclosure by transmission.

A desired maintained interior temperature of 70 deg. fahr. and a minimum external temperature of zero have been adopted in this book as a standard. All transmission coefficients, therefore, are given in B.t.u. per hour per square foot of surface for this temperature difference, with correction factors for other differences.

A table of these factors for various materials used in building construction will be found on pages 00 to 00.

Third, a loss enters into the calculation where the heating is not continuous. This may be referred to as a warming-up loss, or the heat necessary to raise the air of the enclosure from its initial temperature to the desired maintained temperature. It is evident that if only sufficient heat is supplied to compensate for the air infiltration and transmission losses, the temperature of the enclosure would approach but not reach the predetermined temperature, unless additional heat units are supplied for heating an amount of air equivalent to the cubic contents of the space to be heated. To calculate this loss, the cubic contents of the enclosure, the initial and final temperatures of the internal air, and the time desired to raise the air through this temperature range must be determined.

The product of the quantity of air in cubic feet, the density of the air, its specific heat, and the temperature difference is the quantity of heat required for initial heating of the air. If this quantity be then multiplied by the reciprocal of the heating-up period in hours, the product will be the quantity of heat that must be supplied per hour during initial heating to supply the heat absorbed in heating the air.

Fourth, a loss or heat requirement should be included in calculations where the heating is not continuous, and where large quantities of materials such as iron, steel, water, glass, etc., are stored in the enclosure and must be heated like the air contents, from their initial to maintained inside temperature.

The product of the weight of such material in pounds, its specific heat and the desired temperature range is the heat absorbed by the material.

This quantity must also be multiplied by the reciprocal of the heating-up period in hours to obtain the hourly heat requirement during initial heating to compensate for this loss or absorption of heat. The longer the heating-up period selected the less will be the difference in the hourly requirements during initial and maintained heating.

The sum of these four losses gives the total hourly rate at which heat must be supplied to maintain the enclosure at a predetermined temperature, or to raise the temperature of the enclosure from its initial to predetermined temperature, as the case may be.

Applying this method of calculating the heat loss requirements to the house shown, Figure 5-1 represents the main floor of a residence with warm basement and second floor. Under these conditions, no ceiling or floor loss need be considered.

The quantities as taken from the plan are given in detail in the Heat-loss Computation Sheet, Table 5-1; also the basic requirements are given at the top of the sheet.

The losses are figured for each exposed side as in Room No. 1. The loss for the north side is 12618 B.t.u., for the east side 9601 B.t.u., for the west side 1635 B.t.u. and the B.t.u. required for initial heating of the air contents is 623, making a total maximum B.t.u. requirement of 24,477 per hour. The heat supply for this room should be placed under the north and east windows. The loss for the north and west sides, plus half of the heating-

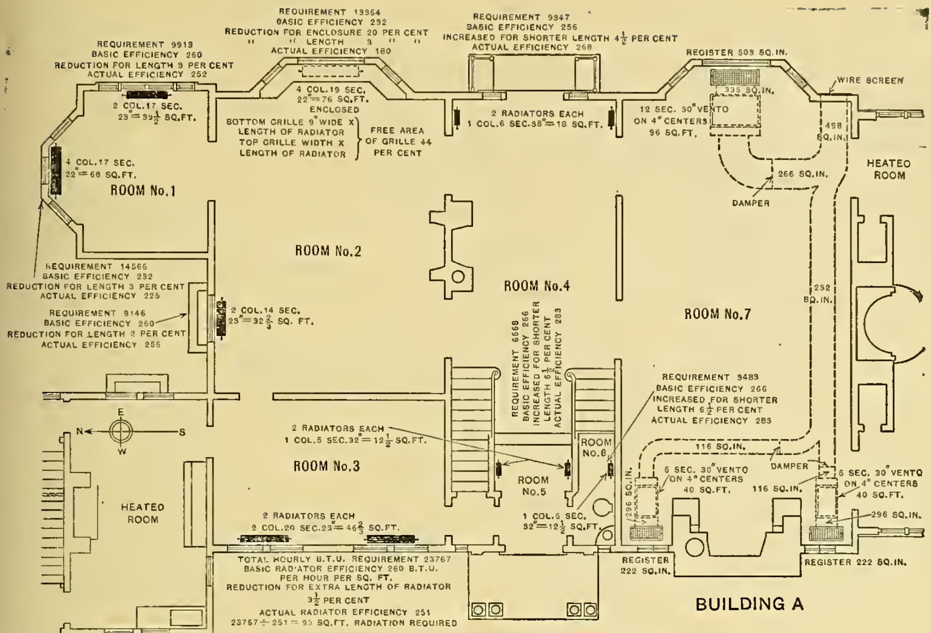


Fig. 5-1. Illustrating method of computing radiator for a residence

up loss for the air, can be taken care of by one unit placed at the west window. The loss for the east side, plus the balance of the heating-up loss for the air, can be taken care of by another unit located at the east window.

The losses in B.t.u. per hour as taken from the computation and divided in a similar manner are marked on the plan for each room.

Another illustration of the method of calculation is given in the Heat-loss Computation Sheets, Table 5-2, for the factory building shown in Figure 5-2.

The calculation has been separately made for the sections as marked in the figure, so that the losses may be proportioned to the exposures.

In the calculations for section "C," the north and south walls with their windows and doors from the floor to line a—b were made separately from the balance of the losses for this section.

As the air infiltration from the upper sash would not be felt directly by the operators in the building, the infiltration has been calculated for only the west or maximum-wind-velocity side.

The infiltration factor for the doors has been taken as double that of the windows, and in calculating the window infiltration losses only the perimeter of the ventilating portion of the window has been considered.

The requirements for the various walls and sections as taken from the calculations are marked on the drawing in their relative locations.

Outside Temp.	0°	Wind Velocity Miles Per Hour	
Inside Temp.	70°	N.	20
Initial Temp.	50°	S.	10
Heating Period	1 Hr.	E.	10
		W.	20

Table 5-1. Heat Loss Computation Sheet

Name: Building A
Location:

Loss	Material	Compas Point	Number or Length	Width	Height	Volume, Area or Perimeter	Deductions	Total net Volume, Area or Perimeter	Temperature Difference	Basic Factor	Correction Factor		Total B.t.u. Per Hour Loss	Total B.t.u. Per Hour Loss
	Room No. 1													
Infiltration.....	Window......................	N	......	2′ 4″	6′ 6″	20	....	20	70	1.75	87.7	...	3070	......
Infiltration.....	Window......................	N45°	2	2′ 4″	6′ 6″	40	....	40	70	1.75	87.7	.7	4298	......
Window........	Single glass....................	N	3	2′ 4″	6′ 6″	46	....	46	70	75	1	...	3450	......
Wall...........	12″ Brick, furred and plastered...	N	17′ 6″	......	9′ 6″	166	46	120	70	15	1	...	1800	12618
Infiltration.....	Window......................	E	2	1′ 9″	6′ 6″	37	} ..	64	70	.85	87.7	...	4771	......
Infiltration.....	Window......................	E	......	4′ 6″	6′ 6″	27								
Window........	Single glass....................	E	2	1′ 9″	6′ 6″	23	} ..	52	70	75	I	...	3900	......
Window........	Single glass....................	E	......	4′ 6″	6′ 6″	29								
Wall...........	12″ Brick, furred and plastered..	E	......	12′	9′ 6″	114	52	62	70	15	1	...	930	9601
Wall...........	12″ Brick, furred and plastered..	W	......	11′ 6″	9′ 6″	109	....	109	70	15	I	...	1635	1635
Initial heating...	Air, sp. ht. .24, density .078.....	.....	14	11′	9′ 6″	1463	} ..	1663	20	.24	.078	...	623	623
Initial heating...	Air, sp. ht. .24, density .078.....	.....	7	3′	9′ 6″	200								
														24477
	Room No. 2													
Infiltration.....	Window......................	N	......	4′	6′ 6″	26	....	26	70	1.75	87.7	...	3990	......
Window........	Single glass....................	N	......	4′	6′ 6″	26	....	26	70	75	1	...	1950	......
Wall...........	12″ Brick, furred and plastered..	N	......	11′ 6″	9′ 6″	109	26	83	70	15	1	...	1245	7185
Infiltration.....	Window......................	E	......	6′	6′ 6″	32	....	32	70	.85	87.7	...	2385	......
Infiltration.....	Window......................	E45°	2	2′	6′ 6″	38	....	38	70	.85	87.7	.7	2833	......
Window........	Single glass....................	E	......	6′	6′ 6″	39	} ..	65	70	75	1	...	4875	......
Window........	Single glass....................	E	2	2′	6′ 6″	26								
Wall...........	12″ Brick, furred and plastered..	E	......	23′	9′ 6″	219	65	154	70	15	1	...	2310	12403
Initial heating...	Air, sp. ht. .24, density .078.....	.....	25′ 6″	20′	9′ 6″	4845	} ..	5130	20	.24	.078	...	1921	1921
Initial heating...	Air, sp. ht. .24, density .078.....	.....	10	3′	9′ 6″	285								
														21509
	Room No. 3													
Infiltration.....	Window......................	W	4	3′	6′ 6″	100	....	100	70	1.75	87.7	...	15348	......
Window........	Single glass....................	W	4	3′	6′ 6″	78	....	78	70	75	1	...	5850	......
Wall...........	12″ Brick, furred and plastered..	W	......	20′	9′ 6″	190	78	112	70	15	1	...	1680	......
Initial heating...	Air, sp. ht. .24, density .078.....	.....	20	12′ 6″	9′ 6″	2375	....	2375	20	.24	.078	...	889	......
														23767
	Room No. 4													
Infiltration.....	Window......................	E	2	1′ 6″	6′ 6″	35	} ..	60	70	.85	87.7	...	4473	......
Infiltration.....	Door..........................	E	......	5′	7′ 6″	25								
Window........	Single glass....................	E	2	1′ 6″	6′ 6″	20	....	20	70	75	1	...	1500	......
Door...........	1½″ Wood.....................	E	......	5′	7′ 6″	38	....	38	69	35	.98	...	1303	......
Wall...........	12″ Brick, furred and plastered..	E	......	14′	9′ 6″	133	58	75	70	15	1	...	1125	......
Initial heating...	Air, sp. ht. .24, density .078.....	.....	19	14′	9′ 6″	2527	....	2527	20	.24	.078	...	946	......
														9347
	Room No. 5													
Infiltration.....	Door..........................	W	......	5′	7′ 6″	25	....	25	70	1.75	87.7	...	3837	......
Door...........	1½″ Wood.....................	W	......	5′	7′ 6″	38	....	38	69	35	.98	...	1500	......
Wall...........	12″ Brick, furred and plastered..	W	......	6′	9′ 6″	57	38	19	70	15	1	...	285	......
Initial heating...	Air, sp. ht. .24, density .078.....	.....	19	14′	9′ 6″	2527	....	2527	20	.24	.078	...	946	......
														6568
	Room No. 6													
Infiltration.....	Window......................	W	......	1′	6′ 6″	16	....	16	70	1.75	87.7	...	2456	......
Window........	Single glass....................	W	......	1′	6′ 6″	7	....	7	70	75	1	...	525	......
Wall...........	12″ Brick, furred and plastered..	W	......	3′ 6″	9′ 6″	33	7	26	70	15	1	...	390	......
Initial heating...	Air, sp. ht. .24, density .078.....	.....	9	3′ 6″	9′ 6″	299	....	299	20	.24	.078	...	112	......
														3483
	Room No. 7													
Infiltration.....	Window......................	E	......	6′	6′ 6″	32	....	32	70	.85	87.7	...	2385	......
Infiltration.....	Window......................	E45°	2	2′	6′ 6″	38	....	38	70	.85	87.7	.7	2833	......
Window........	Single glass....................	E	......	6′	6′ 6″	39	} ..	65	70	75	1	...	4875	......
Window........	Single glass....................	E	2	2′	6′ 6″	26								
Wall...........	12″ Brick, furred and plastered..	E	......	24′	9′ 6″	228	65	163	70	15	1	...	2445	12538
Infiltration.....	Window......................	W	2	2′ 6″	6′ 6″	42	....	42	70	1.75	87.7	...	6446	......
Window........	Single glass....................	W	2	2′ 6″	6′ 6″	33	....	33	70	.75	1	...	2475	......
Wall...........	12″ Brick, furred and plastered..	W	......	11′	9′ 6″	105	33	72	70	15	1	...	1080	......
Wall...........	28″ Brick, furred and plastered..	W	......	11′	9′ 6″	105	....	105	70	8	1	...	840	10841
Initial heating...	Air, sp. ht. .24, density .078.....	.....	38	20′	9′ 6″	7220	} ..	7505	20	.24	.078	...	2810	2810
Initial heating...	Air, sp. ht. .24, density .078.....	.....	10	3′	9′ 6″	285								
														26189

NOTE—Where .7 is added as factor in last column of infiltration calculation, this is the sine of 45 deg., the angle at which the wind strikes the window.

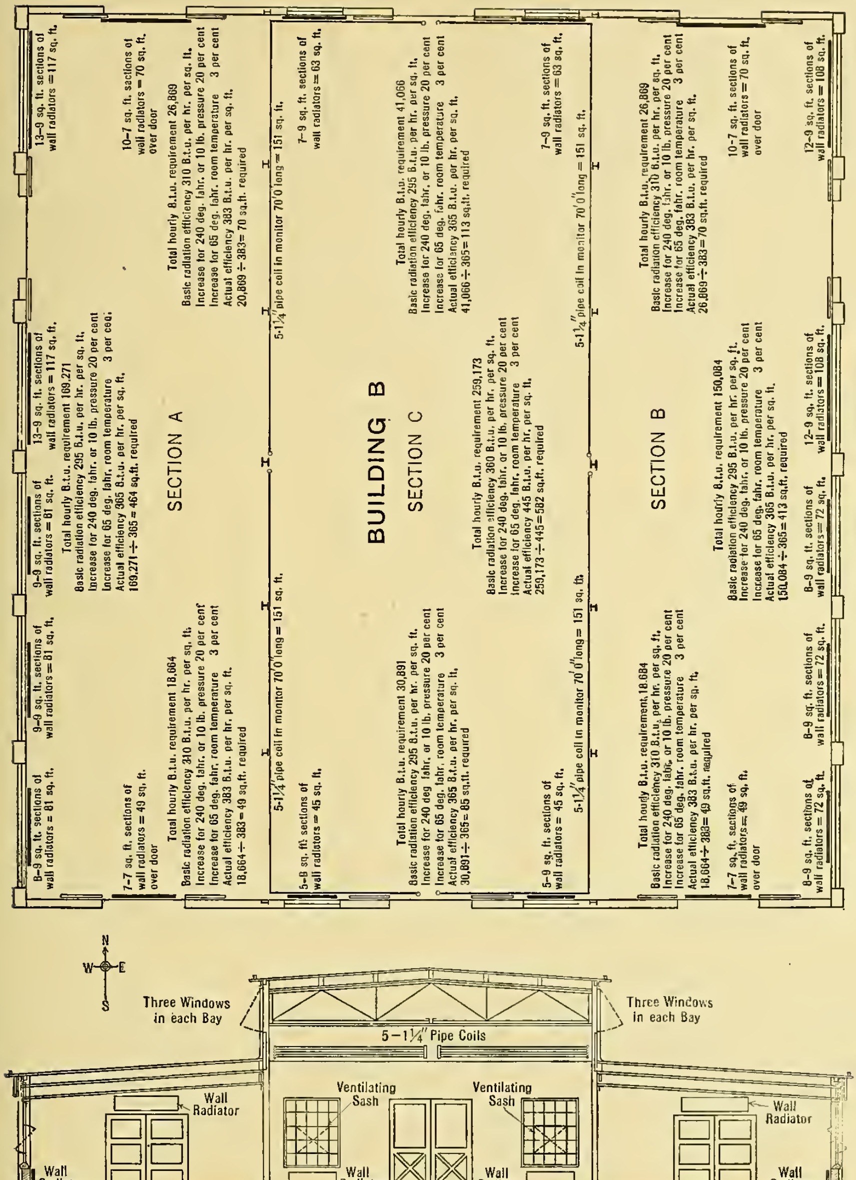


Fig. 5-2. Illustrating method of computing radiation in a factory building

		Wind Velocity Miles Per Hour	
Outside Temp.	0°	N.	20
Inside Temp.	65°	S.	10
Initial Temp.	40°	E.	10
Heating Period	2 Hr.	W.	20

Table 5-2. Heat Loss Computation Sheet

Name: Building B
Location:

Loss	Material	Compass Point	Number or Length	Width	Height	Volume, Area or Perimeter	Deduction	Total net Volume, Area or Perimeter	Temperature Difference	Basic Factor	Correction Factor		Total B.t.u. Per Hour Loss	Total B.t.u. Per Hour Loss
	Section "A"													
Infiltration	Window	W	10	3′ 6″	4′ 0″	150	....	150	65	1.75	81.4	...	21368	......
Infiltration	Door	W	......	10′	12′	56	....	56	65	3.5	81.4	...	15954	......
Window	Single glass	W	10	7′	8′	560	....	560	68	75	.96	...	40320	......
Door	2″ Wood	W	......	10′	12′	120	....	120	66	30	.92	...	3312	......
Wall	12″ Brick, plain	W	110′	......	15′	1650	943	607	68	22	.96	...	12820	......
Wall	16″ Concrete pier	W	7	2′ 6″	15′	263	....	263	68	35	.96	...	8837	102611
Infiltration	Door	N	......	10′	12′	56	....	56	65	3.5	81.4	...	15954	......
Door	2″ Wood	N	......	10′	12′	120	....	120	66	30	.92	...	3312	......
Wall	12″ Brick, plain	N	30′	......	16′	480	120	360	68	22	.96	...	7603	26869
Infiltration	Door	S	......	10′	12′	56	....	56	65	1.7	81.4	...	7749	......
Door	2″ Wood	S	......	10′	12′	120	....	120	66	30	.92	...	3312	......
Wall	12″ Brick, plain	S	30′	......	16′	480	120	360	68	22	.96	...	7603	18664
Roof	2″ Wood, paper and gravel	.....	110′	30′	......	3300	....	3300	71	15	1.01	...	49995	......
Floor	6″ concrete on cinder fill	.....	110′	30	......	3300	....	3300	15	7	.18	...	4158	......
Initial heating	Air, sp. ht. .24, density .079	.....	110′	30	16′	52800	....	52800	25	.24	.079	.5	12514	66667
														214811
	Section "B"													
Infiltration	Window	E	10	3′ 6″	4′ 0″	150	....	150	65	.85	81.4	...	10379	......
Infiltration	Door	E	......	10′	12′	56	....	56	65	1.7	81.4	...	7749	......
Window	Single glass	E	10	7′	8′	560	....	560	68	75	.96	...	40320	......
Door	2″ Wood	E	......	10′	12′	120	....	120	66	30	.92	...	3312	......
Wall	12″ Brick, plain	E	110	......	15′	1650	943	607	68	22	.96	...	12820	......
Wall	16″ Concrete	E	7	2′ 6″	15′	263	....	263	68	35	.96	...	8837	83417
Infiltration	Door	N	......	10	12′	56	....	56	65	3.5	81.4	...	15954	......
Door	2″ Wood	N	......	10	12′	120	....	120	66	30	.92	...	3312	......
Wall	12″ Brick, plain	N	30	......	16′	480	120	360	68	22	.96	...	7603	26869
Infiltration	Door	S	......	10	12′	56	....	56	65	1.7	81.4	...	7749	......
Door	2″ Wood	S	......	10	12′	120	....	120	66	30	.92	...	3312	......
Wall	12″ Brick, plain	S	30	......	16′	480	120	360	68	22	.96	...	7603	18664
Roof	2″ Wood, paper and gravel	.....	110	30	......	3300	....	3300	71	15	1.01	...	49995	......
Floor	6″ Concrete on cinder fill	.....	110	30	......	3300	....	3300	15	7	.18	...	4158	......
Initial heating	Air, ap. ht. .24, density .079	.....	110	30	16′	52800	....	52800	25	.24	.079	.5	12514	66667
														195617
	Section "C" lower													
Infiltration	Window	N	2	3′ 6″	4	30	....	30	65	1.75	81.4	...	4274	......
Infiltration	Door	N	......	10	12	56	....	56	65	3.5	81.4	...	15954	......
Window	Single glass	N	2	7	8	112	....	112	68	75	.96	...	8064	......
Door	2″ Wood	N	......	10	12	120	....	120	66	30	.92	...	3312	......
Wall	12″ Brick, plain	N	......	40	17	680	232	448	68	22	.96	...	9462	41066
Infiltration	Window	S	2	3′ 6″	4	30	....	30	65	.85	81.4	...	2076	......
Inﬁltration	Door	S	......	10	12	56	....	56	65	1.75	81.4	...	7977	......
Window	Single glass	S	2	7	8	112	....	112	68	75	.96	...	8064	......
Door	2″ Wood	S	......	10	12	120	....	120	66	30	.92	...	3312	......
Wall	12″ Brick, plain	S	......	40	17	680	232	448	68	22	.96	...	9462	30891
														71957
	Section "C" upper													
Infiltration	Window	W	18	5	5′ 6″	378	....	378	65	1.75	81.4	...	53846	......
Window	Single glasa	W	18	5	5′ 6″	495	....	495	67	75	.94	...	34898	......
Wall	12″ Brick, plain	W	110	......	10′	1100	495	605	65	22	.9	...	11979	......
Wall	12″ Brick, plain	N	40	......	10′ 6″	420	....	420	65	22	.9	...	8316	......
Window	Single glass	E	18	5	5′ 6″	495	....	495	67	75	.94	...	34898	......
Wall	12″ Brick, plain	E	110	......	10′	1100	495	605	65	22	.9	...	11979	......
Wall	12″ Brick, plain	S	40	......	10′ 6″	420	....	420	65	22	.9	...	8316	......
Roof	2″ Wood, paper and gravel	.....	110	40	......	4400	....	4400	66	15	.92	...	60720	......
Floor	6″ Concrete on cinder fill	.....	110	40	......	4400	....	4400	15	7	.18	...	5544	......
Initial heating	Air, sp. ht. .24, density .079	.....	110	40	27′ 6″	121000	....	121000	25	.24	.079	.5	28677	......
														259173

CHAPTER VI

Method of Computing and Selecting Radiation

DETERMINATION of the radiating surface depends first upon the total hourly heat losses, which are assumed to have been calculated as described in the preceding chapter. The radiating surface must supply enough heat units to compensate for the losses and should be of the form that best fits the conditions for the room or enclosure.

The method of heat supply must first be determined—that is, whether the radiation is to be direct, indirect or direct-indirect. The last two methods are used principally when ventilation must be considered in addition to the heating requirements, although the indirect method is considerably used when it is not desired to have the radiation located in the room to be heated.

Normally, the heat should be supplied at the locations where the greatest losses occur, and this is generally at the windows, where in addition to a high transmission loss, there is the air infiltration loss as well.

Rooms or enclosures where more than one unit of radiation is to be installed should have the radiating surface divided in proportion to the losses of the spaces served.

Radiation placed under the windows should not project above the sills, should be as wide as the window openings, and should also be installed with a 2½-inch space between the wall and the radiation, as this distance gives maximum efficiency of heat emission.

Direct radiation, inasmuch as it is used in a large majority of installations, should be considered first. Residences, office, school, library, hospital and similar buildings, usually have cast-iron column radiation together with some cast-iron wall radiation. Factory and manufacturing buildings are usually heated by means of wrought-iron or steel pipe coils or cast-iron wall radiation.

Fig. 6-1. Cast-iron wall radiation on side walls under window, for heating a factory building

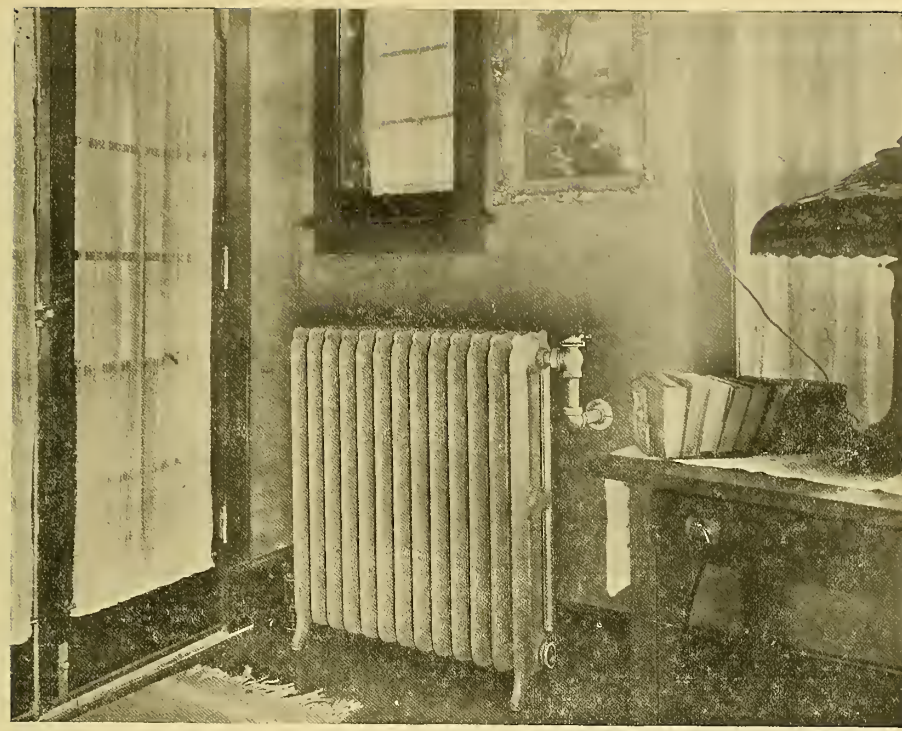


Fig. 6-2. Connection to a direct hot-water type radiator showing Webster Modulation Supply Valve and Webster Return Trap

Hot-water pattern radiation is preferable for those Webster systems in which Modulation Supply Valves are to be used. The supply valve should be placed at the upper inlet and the return trap at the lower opening diagonally opposite.

Good practice in the use of groups of wall radiation suggests that no individual group exceed 30 feet in length, as expansion and contraction become an important factor on longer runs. Where greater lengths of this type of radiation must be used, the supply connection should be made at top and bottom and expansion and contraction properly provided for.

Pipe coil practice demands a spring or mitre piece in the coil to provide for expansion and contraction, and the desirable length is limited to sixty feet not including the mitre piece. Coils should be securely anchored at the return header so as to throw the expansion toward the mitre end, the length of which should be not less than one-twelfth the coil length for 1-in. pipe and one-tenth for $1\frac{1}{4}$-in. or $1\frac{1}{2}$-in. pipe.

The amount of heat emitted from any given type of direct radiation is usually stated in B.t.u. per hour per square foot of radiator surface. This heat is given off in two ways, by convection directly to the air which passes

over the heated surface, and by radiation directly to surrounding materials independent of that carried off by the air. The heat given off by radiation does not heat the air through which it passes, but travels in straight lines and heats the materials on which it impinges.

After selecting the type of radiation best suited for the particular case, the number of square feet of radiating surface required should be determined next. The total number of heat units that must be supplied per hour divided by the heat units emitted per hour per square foot of radiation gives the required surface in square feet of radiation.

Table 6-1 will be of assistance in determining the heat emitted by different types of radiation.

Table 6-1. B. t. u. Emitted per Hour per Square Foot of Radiating Surface*
Radiators Ten Sections Long
Steam Temperature 215 deg. fahr. Room Temperature 70 deg. fahr.

Number of Columns	Height of Radiator	B.t.u. by Convection	B.t.u. by Radiation	Total B.t.u.	Ratio of Radiating to Total Surface	Percent Convected heat of Total heat
One	38 in.	150	106	256	0.53	58.6
"	32 in.	158	108	266	0.54	59.4
"	26 in.	162	111	273	0.555	59.4
"	23 in.	160	119	279	0.595	57.4
"	20 in.	166	117	283	0.584	58.7
Two	45 in.	148	86	234	0.43	63.
"	38 in.	148	92	240	0.458	62.
"	32 in.	154	94	248	0.47	62.
"	26 in.	149	106	255	0.53	58.
"	23 in.	151	109	260	0.544	58.
"	20 in.	153	112	265	0.56	58.
Three	45 in.	142	76	218	0.382	65.
"	38 in.	147	79	226	0.394	65.
"	32 in.	158	75	233	0.375	68.
"	26 in.	166	75	241	0.376	69.
"	22 in.	166	82	248	0.407	67.
"	18 in.	162	92	254	0.46	64.
Four	45 in.	149	56	205	0.28	73.
"	38 in.	150	60	210	0.30	71.5
"	32 in.	151	66	217	0.331	69.5
"	26 in.	155	70	225	0.35	69.
"	22 in.	156	76	232	0.382	67.
"	18 in.	151	87	238	0.435	63.5
Wall Radiation						
3 in. wide	14 in.	152	171	323	0.854	47.
" "	22 in.	154	156	310	0.78	49.7
" "	29 in.	138	157	295	0.784	48.
Pipe Coil	6–1¼ in. Pipes			360		
" "	8–1¼ in. "			343		
" "	10–1¼ in. "			330		
" "	12–1¼ in. "			319		

* John R. Allen A. S. H. & V. E. Journal—January 1920.

From Table 6-1 it will be noted that low, narrow radiation is most efficient and that the efficiency decreases as the height and width increase.

A number of factors other than variation of the height and width of section vary the amount of heat emitted from radiation. Some of these

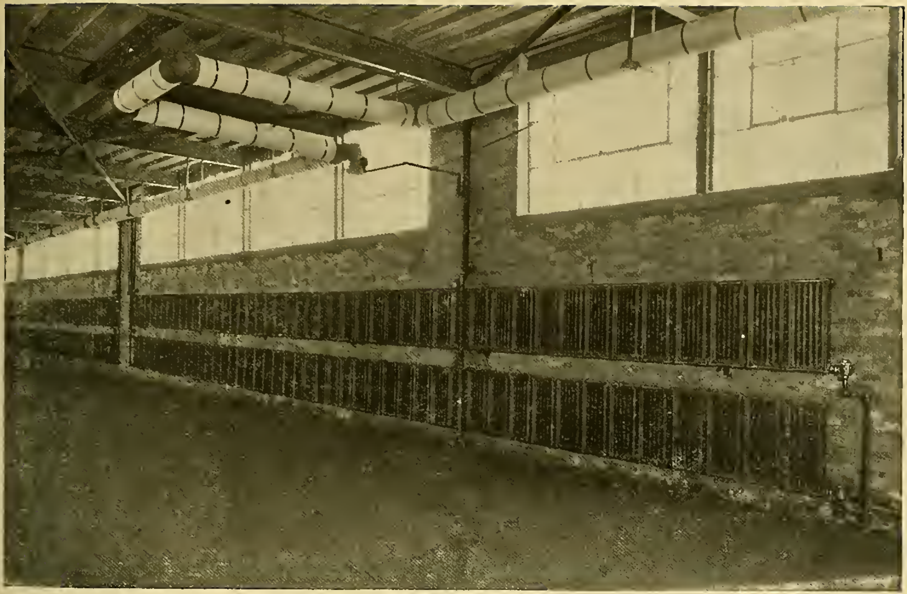


Fig. 6-3. Arrangement of cast-iron wall radiation on side wall of a factory building.

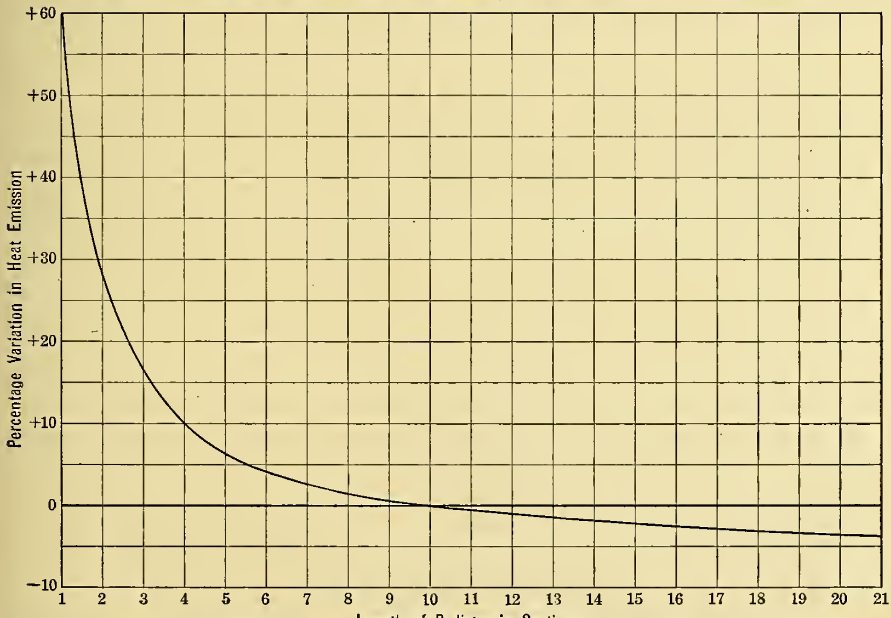


Fig. 6-4. Percentage of variation in heat emitted by cast-iron radiation per square foot due to variation in the number of sections from a standard 10-section radiator

factors and their effect upon the efficiency of the radiation are worthy of further explanation.

The preceding table is based upon a radiator ten sections wide. As the number of sections decrease, the efficiency increases, due to increase of the more efficient end section surface in proportion to total radiator surface; also a short radiator emits proportionally more radiant heat than a longer one. Figure 6-4 shows the effect of varying the number of sections, and that increasing the number of sections above ten has not as much effect as decreasing the number below ten. It will also be noted that a four-section radiator will give off about ten per cent more heat per square foot of surface than one ten sections long.

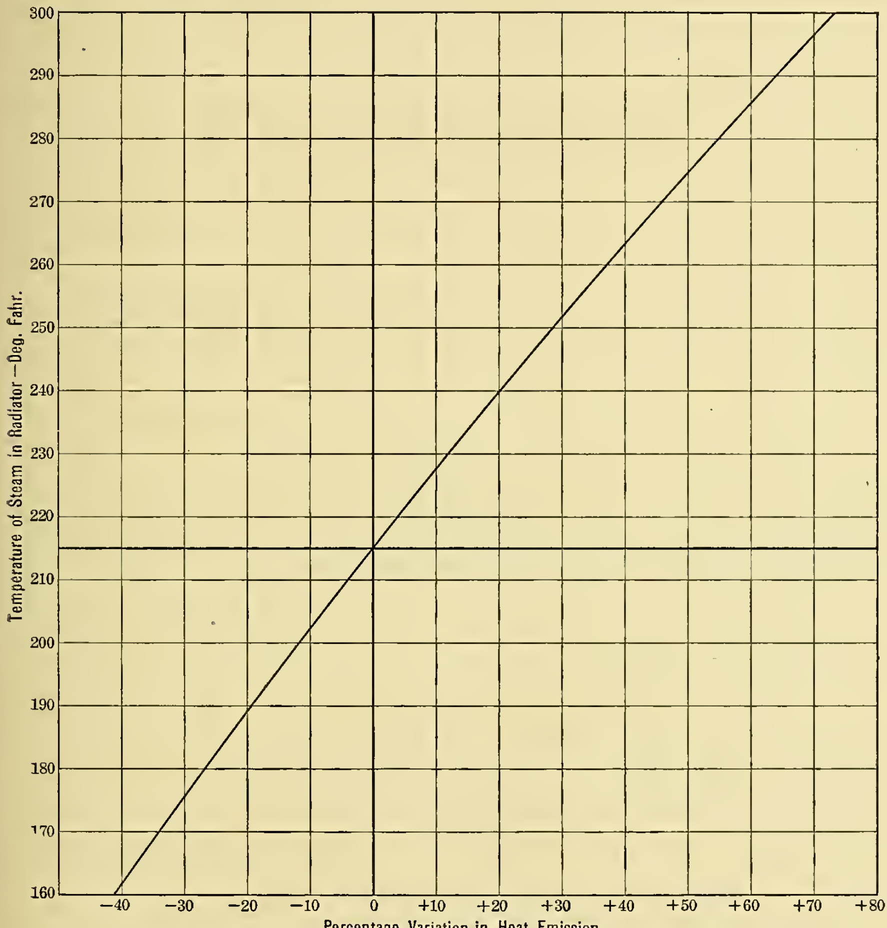


Fig. 6-5. Percentage variation in heat emitted by radiation by varying the steam temperature from 215 deg. fahr.

When the temperature of steam in the radiation is considered at 215 deg. fahr. as standard, the effect upon the heat emission of radiation due to varying this temperature is shown in Figure 6-5. The percentage variation can be read directly from the curve.

Example: If steam at a temperature of 230 deg. fahr. is supplied to the radiator, the heat emission will be increased twelve per cent over one supplied with steam at 215 deg. fahr.

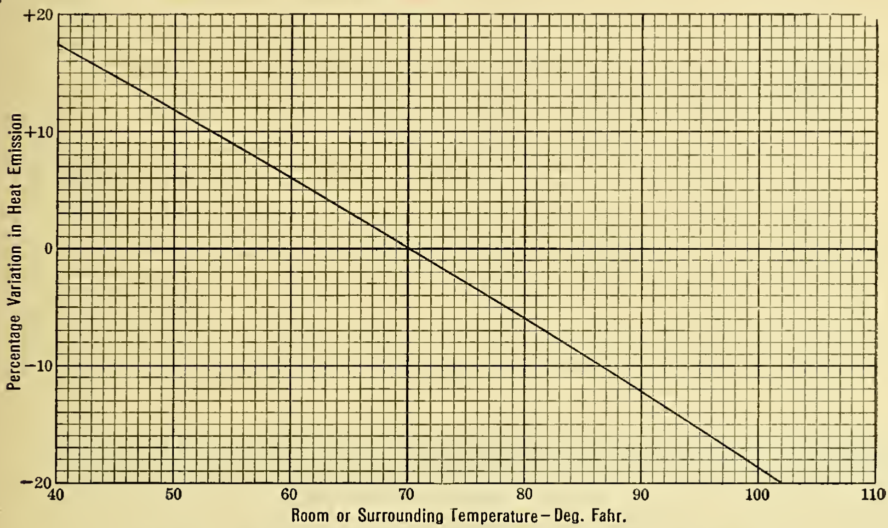


Fig. 6-6. Percentage variation in heat emitted by radiation by varying the room temperature from 70 deg. fahr.

The surrounding or room temperature is taken at 70 deg. fahr. as a standard—the effect upon the heat emitted by radiation, due to varying this temperature is shown graphically in Figure 6-6. From the curve it will be observed that, for instance, a radiator in a room temperature of 60 deg. fahr. will emit 6 per cent more heat than will same radiator in a room temperature of 70 deg. fahr.

The effect on heat emission due to variation in steam temperature is much greater than an equal temperature variation in the surrounding or room temperature.

The following example will illustrate the use of the curves in Figures 6-4, 6-5 and 6-6 for determining the heat emission under given conditions. It is desired to know the B.t.u. emitted per hour per square foot of radiating surface of a standard cast-iron radiator, two columns wide, 38 inches high, and six sections long when supplied with steam at 240 deg. fahr. and located in a room heated to 80 deg. fahr.

Referring to Table 6-1, a radiator as above except that it is ten sections long, with steam at 215 deg. fahr. in room temperature 70 deg. fahr. gives off 240 B.t.u. per hour per square foot. A radiator six sections long is 4.5 per cent. more efficient (Figure 6-4) when supplied with steam at 240 deg.

fahr., the efficiency is increased 20 per cent (Figure 6-5), and if located in a room heated to 80 deg. fahr. there is a decrease in efficiency of 6 per cent (Figure 6-6). The heat emission of the radiator required would be 240 x 1.045 x 1.20 x 0.94, or 283 B.t.u. per hr. per sq. ft. of radiation surface.

Painting the radiator affects only the heat emitted by radiation, and has practically no effect upon the convected heat. As painting affects only the surface, the final coat is the only one that must be considered. Paints generally produce only a very slight effect. That of flake metals is more marked. (See Table 6-2.)

Table 6-2. Approximate Effect of Painting on the Total Heat Emission of Radiators. Test made on a 2-column Radiator 33 Inches High and Ten Sections Long, Supplied with Steam at 215 deg. fahr. in Room Temperature of 70 deg. fahr.

Cast Iron Base	100%	Painted Maroon Japan	100%
Painted with Aluminum Bronze	83%	Painted with White Zinc Paint	101%
Painted with Gold Bronze	85%	Painted with No-lustre Green Enamel	96%
Painted White Enamel	101%		

Direct radiators are sometimes set behind grilles or screens, in window enclosures or wall recesses, all of which greatly decrease the efficiency of the radiation.

Tests by Professor Brabbee, as reported by George Stumpf, Heating and Ventilating Magazine, May, 1914, show that a radiator in an enclosure is most efficient when located with $2\frac{1}{2}$ inches between the wall and radiator

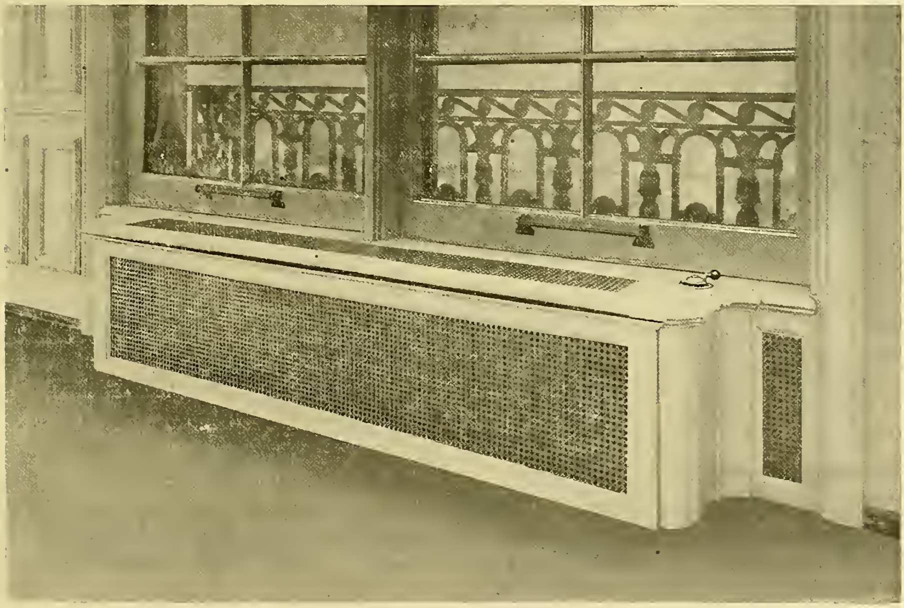


Fig. 6-7. An enclosed radiator having grilles or screens on front and top of enclosure. The Modulation Supply Valve Control is shown located on top of enclosure

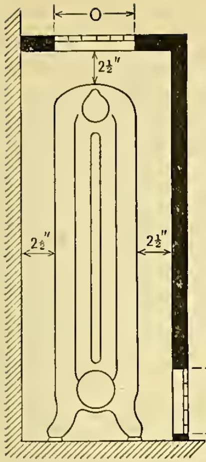


Fig. 6-8

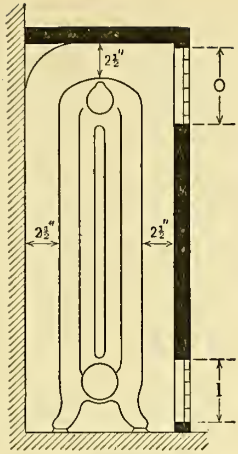


Fig. 6-9

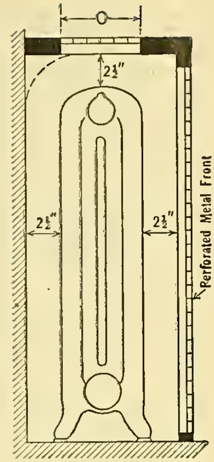


Fig. 6-10

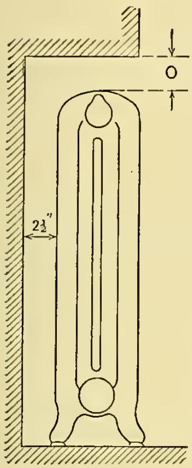


Fig. 6-11

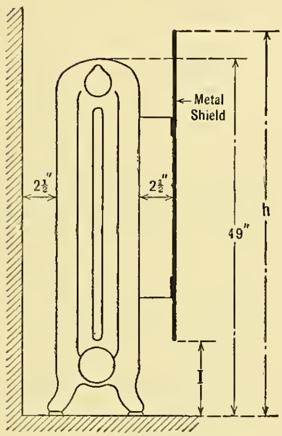


Fig. 6-12

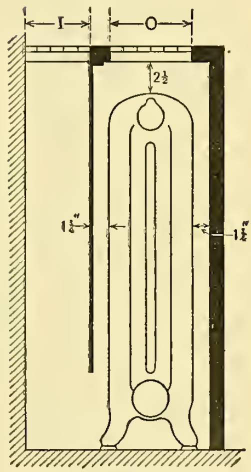
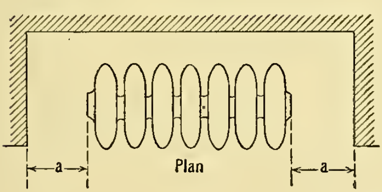


Fig. 6-13



Length of all outlets "O" = length of radiator.

Length of all inlets "I" = length of radiator.

Width of all outlets "O" = width of radiator or as given in table.

Screens or grilles have 44 per cent. free area.

Enclosures for radiators

and between the inside of the enclosure and the radiator. Abstracts from these tests follow.

The inlet and outlet openings of any form of enclosure should extend at least the entire length of the radiator. The width of the outlet is usually made that of the radiator. Tests show little gain in efficiency for wider outlets, but a decrease of about five per cent for each inch narrower than that of the radiator.

The outlets and inlets in Tables 6-3 to 6-8 are the full length of the radiators. The width of outlet "O" is the width of the radiator except in Table 6-4, where it is as given. The width of inlet "I" is as stated in the Tables. Both openings are covered with screen of 44 per cent free area.

The design of the screen or grille has no effect provided the free area is not changed.

Figure 6-8 shows a form of enclosure frequently used.

Table 6-3. Decrease in Radiator Efficiency with Form of Enclosure Shown in Fig. 6-8.

Radiator Width	Radiator Height	Width of I	Decrease in Efficiency
Two-column	42 in. and over	9 in.	15%
" "	Under 42 in.	9 in.	20%
" "	Under 42 in.	5 in.	25%
Three-column	42 in. and over	9 in.	15%
" "	32 in. to 38 in.	9 in.	15%
" "	32 in. to 38 in.	7 in.	20%
" "	26 in. and under	9 in.	20%
" "	26 in. and under	5 in.	25%

If the width of inlet is made proportional to the free area and not screened, the efficiency reduction will remain as above.

Another form of enclosure, Figure 6-9, gives the effect upon the radiation efficiency as shown in Table 6-4.

Table 6-3. Decrease in Radiator Efficiency with Form of Enclosure Shown in Fig. 6-9.

Radiation Width	Radiation Height	Width of O	Width of I	Decrease in Efficiency
Two-column	42 in. and over	8 in.	8 in.	20%
" "	32 in. to 38 in.	9 in.	9 in.	20%
" "	32 in. to 38 in.	7 in.	7 in.	25%
" "	26 in. and under	6 in.	6 in.	33%
Three-column	26 in. and over	9 in.	9 in.	20%
" "	26 in. and over	6 in.	6 in.	25%

Enclosure of the form shown in Figure 6-10 is sometimes used and by test gives the following effect:

Table 6-5. Decrease in Radiator Efficiency with Form of Enclosure Shown in Fig. 6-10.

Perforated screen full front of enclosure—decrease in efficiency.......... 20%
Same screen with deflector— " " " 15%

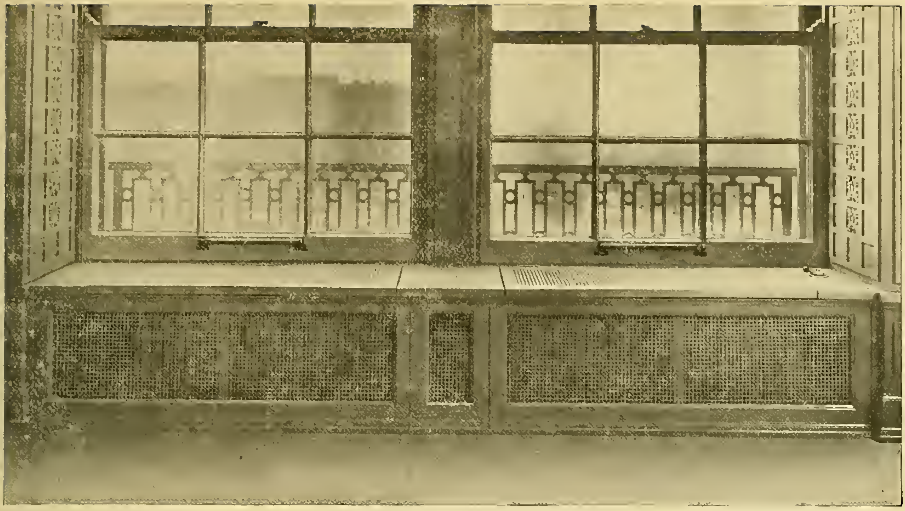


Fig. 6-15. An enclosed radiator in a window seat, with grilles of rattan cane. The Modulation Supply Valve control is placed on the window seat.

If an outlet "O" is provided and made equal to the width and length of the radiator, the efficiency decreases 10%.

Sometimes it is desirable to set the radiators in wall recesses, as shown in Figure 6-11, which causes a decrease in efficiency as follows:

Table 6-6 Decrease in Radiator Efficiency Due to Wall Recess (Fig. 6-11)

When 0 = 1½ inches—decrease in efficiency 11%
" 0 = 3 " " " " 7.3%
" 0 = 4 " " " " 6%

The distance "a" has little or no effect, and, therefore, need only be sufficient for connections to the radiator.

A shield in front of a radiator as shown in Figure 6-,12 increases the radiator efficiency as follows:

Table 6-. Increase in Radiator Efficiency by Use of a Shield (Fig. 6-12)

Height of shield, H	52 in.	52 in.	52 in.	72 in.
Width of open slot, I	6½ in.	9 in.	12 in.	12 in.
Increase in efficiency	2.2%	6.3%	12.5%	13%

Another form of enclosure, shown in Figure 6-13, by test gives the following effect upon the radiator efficiency:

Table 6-8. Decrease in Radiator Efficiency with Form of Enclosure Shown in Fig. 6-13

Width D	Percent Decrease in Efficiency
8 in.	10
6 in.	15
5 in.	20
4 in.	25
3 in.	33

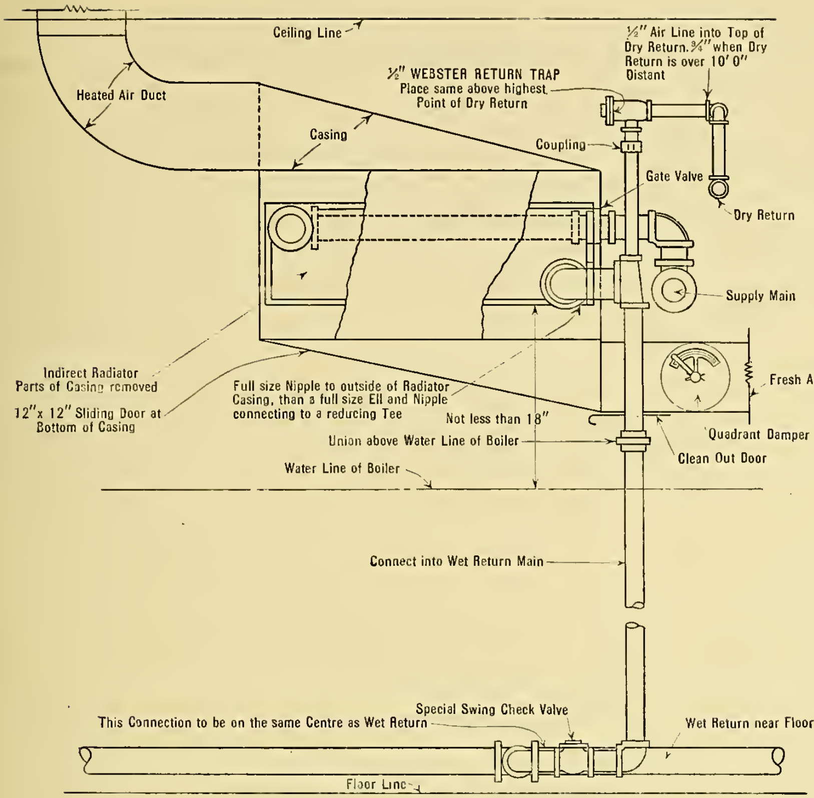


Fig. 6-16. Connections to an indirect radiator

Indirect radiation generally refers to that located below and outside of the room to be heated. (See Figure 6-16.) The heat is delivered to the room by a system of ducts that convey fresh air from outside. The air passes over the radiation, is heated and then discharged into the room through register faces located in the room floor or wall. This method of heating is called *fresh air indirect*, as a constant supply of fresh heated air is delivered into the room.

Where the air supply is taken from the room, passed over the heating surface and then discharged into the room again, the method is referred to as *recirculating indirect.*

In either system no radiation is located in the room to be heated.

The indirect method of heating is most used in the principal rooms of residences, clubs, churches and similar types of buildings, and is much more expensive to install and to operate than is the direct system.

All rooms heated by the fresh air indirect system must be provided with vents for the escape of the air replaced by that delivered by the "indirect stack," as this radiation is often called.

Many variable factors, each of prime importance, enter into an accurate calculation of the proper proportions of a system of this type. These variables include velocity and direction of the wind, frictional resistance to the air flow in the ducts, and the loss of heat due to transmission through the walls of hot air ducts.

Each manufacturer of radiation for this system has his own special design, which is usually sold by catalogue ratings in square feet of surface. Reliable data as to the free area between sections and the heating effect under the variable conditions of steam and air temperatures at various air velocities are unfortunately not available for each make of radiation used in this method of heating. Proper values are very difficult to assign to the variable factors, and the several rules for determining the proper proportions of such a system are all based upon some standard conditions and assumptions.

The general principle of this system is that the air be delivered to the room at a temperature higher than that of the room, and in such volume that in cooling to room temperature, sufficient heat units are given up to replace those lost by transmission, infiltration and otherwise.

The requirements for this method of heating are usually computed in the following way:

First: Calculate the total heat losses in B.t.u. per hour for the room to be heated as described in Chapter 5.

Second: Determine the height of the column of heated air, that is, the distance from center of indirect stack to center of the room register.

Third: Assume the temperature of the air entering the room. This is usually taken about 120 deg. fahr. when air enters the radiator at zero and the radiator is supplied with steam at atmospheric pressure or slightly above.

Fourth: Determine the velocity of air due to difference in densities between the heated and outside air for a column equal in height to that found.

Fifth: Ascertain from the manufacturer of the type of radiation selected the velocity at which air must pass through the radiator to produce the final temperature selected, when the radiator is supplied with steam at a predetermined temperature and air enters the heating stack at the minimum outside temperature. Ascertain also the temperature of the air on which this performance is based, the free area between the sections, and the number of square feet of heating surface per section.

With this data the amount of radiation may be determined as follows:

H = total B.t.u. losses per hour for the room.
t_f = temperature of air entering the room.
t_r = temperature of air in room (room temperature).
t_p = temperature of air on which radiation performance is based.
d = density of air at temperature t_p.
v = performance velocity of air in feet per minute.
a = free area per section of radiation.

$$\frac{H}{.24\ (t_f - t_r)\ 60} = \text{pounds of air required per minute} = P$$

where .24 is the specific heat of the air.

$\frac{P}{d}$ = Cubic feet of air per minute at t_p.

$\frac{P}{d}$ divided by av = number of sections of radiation required from which the square feet of radiation can be determined.

The sizes of the ducts or flues for conveying the air to and from the heating surface are dependent upon the velocity of the air due to the unbalanced air column. This velocity may be determined theoretically from the formula:

$$V = 480\sqrt{\frac{h\ (t - t_o.)}{460 \text{ plus } t}}$$

in which

V = velocity in feet per minute.

h = height of warm air column in feet or distance from center of heating surface to center of register.

t = average temperature of air in column.

t_o = average temperature of outside air.

To allow for friction in ducts, through heating surface, register face and elsewhere, velocities of one-third of the theoretical may be assumed.

The area of the hot-air duct may be determined as follows:

$$\text{Area in square inches} = \frac{144\ P}{d\ v}$$

in which

P = pounds of air required per minute.

d = density of air at average temperature in hot air duct.

V = velocity in feet per minute in duct.

The register should have a free area equal to the area of the hot-air duct. The area of the cold-air duct can be determined in a manner similar to the hot-air duct area, using density of the air at the cold inlet temperature.

Direct-indirect radiation, as the name implies, consists of radiators arranged so that a portion of each serves on the indirect principle and the remainder as a direct radiator; the entire surface, however, is located in the room to be heated. This combination is accomplished by providing a direct radiator and installing a metal box base under some of the sections. Cold fresh air is taken from the outside of the building directly through the wall and connected to this box base. The fresh air passes up through its portion of the radiator into the room. The balance of the section acts as plain direct radiation.

This method of heating has come into quite general use in recent years in some localities where the state ventilation laws for public buildings specify either the quantity of air to be supplied per minute per person, or the number of square inches of fresh-air inlet duct per person. The latter requirement can be met by this type of radiation.

The size of the opening in the wall or the wall box determines the size of the box base, and the number of sections of the radiator enclosed by the box base are to be considered as available only for heating the incoming air.

Sufficient additional direct heating surface must be provided, either by adding sections to the radiator, extending same outside of the box base on either end, or by installing separate units for supplying the heat required to compensate for the losses through wall, glass, and through infiltration, as already mentioned.

Vent flues must be extended from all rooms heated and ventilated by this method.

In order to obtain the desired air movement and prevent back draft in the flues, they must be provided with aspirating radiation or rotary type ventilators.

The radiation best suited for direct-indirect surface is that with high and wide sections. One manufacturer of the most modern devices for this type of system gives the size of the ventilating base, together with its capacity, fresh air inlet area and amount of radiating surface to be enclosed as given in Table 6-9.

Table 6-9. Dimensions of Direct-indirect Radiation Surface as given by one manufacturer. *Not standard with other builders*

Size of Wall Box	Capacity in Cu. Ft. per Min.	Area of Fresh Air Opening	Radiating Surface
8 in. x 20 in.	180	120	50
8 in. x 24 in.	240	144	50
8 in. x 30 in.	300	180	60
10½ in. x 20 in.	270	160	50
10½ in. x 24 in.	330	192	60
10½ in. x 30 in.	420	240	60

As an example of the application of Table 6-9, select and compute the radiation to supply the heat requirements as shown for the various rooms in Figure 5-1, page 5-3, based on steam at 215 deg. fahr., or 1 lb. per sq. in. pressure.

Room No. 3 requires a total of 23767 B.t.u. per hour and is to be heated by means of direct radiation. The window sills are 24 inches high. Therefore, 23-inch high radiators should be installed. For a room of this size, it appears that two-column radiation should give sufficient surface. The B.t.u. emitted by two-column, 23-inch high radiation is given in Table 6-1 as 260 B.t.u. per hr. per sq. ft. of surface. As these radiators will be twenty sections long instead of the standard ten, on which the above efficiency was based, the efficiency, or B.t.u. emitted will be reduced by 3.5 per cent, making an actual efficiency of 251. This divided into the total heat requirements gives 93 square feet of radiation required, which is supplied by two units of 46⅔ square feet each as marked on the plan.

Data as above for determination of the other units are marked on the plan. Room No. 7, which is to be heated by indirect surface, is calculated as follows: The total requirements for the west side are 13943 B.t.u. per hr.,

and assuming that the air enters the room at 120 deg. fahr., the pounds of air required in accordance with formula on page 6-12 would be 19.4 per minute.

Vento radiation 30 inches long on 4-inch centers gives a temperature rise of air from zero to 120 deg. fahr. at 100 ft. per min. velocity measured at 70 deg. fahr. volume. The free area per section is .225 sq. ft.

The pounds of air as found above divided by the density at 70 deg. fahr., or .0749, gives 259 cu. ft. of air per minute.

This volume divided by the velocity, then by the free area per section, gives 12 sections required.

The distance from the center of the radiation to the floor above is 27 inches, which head with 120 deg. fahr. temperature difference gives a theoretical velocity of 367 feet per minute by the formulæ on page 6-13. For determining the size of the ducts, one-third of this value, or 122 ft. per min. velocity may be used.

Using formula on page 6-13 with a density for air at 120 deg. fahr., the area of the hot-air duct is 335 sq. in. The register if of 66⅔ per cent free area should contain 503 square inches.

The cold-air duct by the above formula, using air density at zero, should have a sectional area of 266 sq. in.

The indirect surface for the requirement of the east side of this room was calculated similarly.

As another example, determine the necessary radiation to supply the heat required for the factory building as calculated in the previous chapter and shown in Figure 5-2, page 5-5.

Assume that steam at 10 lb. per sq. in. pressure or at a temperature of 240 deg. fahr. is available for heating this building under maximum load conditions. The increase in B.t.u. emission of the heating surfaces for this increased temperature above the standard or basic temperature is 20 per cent, and there would be a further increase in efficiency of 3 per cent due to a 65-deg. fahr. instead of 70-deg. fahr. room temperature.

This would make a total increase of 23.6 per cent in the B.t.u. emitted per hour per sq. ft. of radiation for this installation, over the basic value.

The monitor portion of the building is provided with 1¼-inch pipe coils under the windows, with expansion springs at the ends, as shown. For the lower portion of the building cast-iron wall radiation is to be installed, as shown. The efficiency and method of determining the amount of surface are shown on the plan.

Table 6-10. Surface in square feet of one to twelve $1\frac{1}{4}$-inch pipe coils, 1 to 100 feet long

Length of Coil	NUMBER OF $1\frac{1}{4}$" PIPES											
	1	2	3	4	5	6	7	8	9	10	11	12
1	.43	.86	1.29	1.72	2.15	2.58	3.01	3.44	3.87	4.30	4.73	5.16
2	1	2	3	3	4	5	6	7	8	9	9	10
3	1	3	4	5	6	8	9	10	12	13	14	15
4	2	3	5	7	9	10	12	14	15	17	19	21
5	2	4	6	9	11	13	15	17	19	22	24	26
6	3	5	8	10	13	15	18	21	23	26	28	31
7	3	6	9	12	14	18	21	24	27	30	33	36
8	3	7	10	14	17	21	24	28	31	34	38	41
9	4	8	12	15	19	23	27	31	35	39	43	46
10	4	9	13	17	22	26	30	34	39	43	47	52
11	5	9	14	19	24	28	33	38	43	47	52	57
12	5	10	15	21	26	31	36	41	46	52	57	62
13	6	11	17	22	28	34	39	45	50	56	61	67
14	6	12	18	24	30	36	42	48	54	60	66	72
15	6	13	19	26	32	39	45	52	58	65	71	77
16	7	14	21	28	34	41	48	55	62	69	76	83
17	7	15	22	29	37	44	51	58	66	73	80	88
18	8	15	23	31	39	46	54	62	70	77	85	93
19	8	16	25	33	41	49	57	65	74	82	90	98
20	9	17	26	34	43	52	60	69	77	86	95	103
21	9	18	27	36	45	54	63	72	81	90	99	108
22	9	19	28	38	47	57	66	76	85	95	104	114
23	10	20	30	40	49	59	69	79	89	99	109	119
24	10	21	31	41	52	62	72	83	93	103	114	124
25	11	22	32	43	54	65	75	86	97	108	118	129
26	11	22	34	45	56	67	78	89	101	112	123	134
27	12	23	35	46	58	70	81	93	104	116	128	139
28	12	24	36	48	60	72	84	96	108	120	132	144
29	12	25	37	50	62	75	87	100	112	125	137	150
30	13	26	39	52	65	77	90	103	116	129	142	155
31	13	27	40	53	67	80	93	107	120	133	147	160
32	14	28	41	55	69	83	96	110	124	138	151	165
33	14	28	43	57	71	85	99	114	128	142	156	170
34	15	29	44	58	73	88	102	117	132	146	161	175
35	15	30	45	60	75	90	105	120	135	151	166	181
36	15	31	46	62	77	93	108	124	139	155	170	186
37	16	32	48	64	80	95	111	127	143	159	175	191
38	16	33	49	65	82	98	114	131	147	163	180	196
39	17	34	50	67	84	101	117	134	151	168	184	201
40	17	34	52	69	86	103	120	138	155	172	189	206
41	18	35	53	71	88	106	123	141	159	176	194	212
42	18	36	54	72	90	108	126	144	163	181	199	217
43	18	37	55	74	92	111	129	148	166	185	203	222
44	19	38	57	76	95	114	132	151	170	189	208	227
45	19	39	58	77	97	116	135	155	174	194	213	232
46	20	40	59	79	99	119	138	158	178	198	218	237
47	20	40	61	81	101	121	141	162	182	202	222	243
48	21	41	62	83	103	124	144	165	186	206	227	248
49	21	42	63	84	105	126	147	169	190	211	232	253
50	22	43	65	86	108	129	151	172	194	215	237	258

Table 6-10. Surface in square feet of one to twelve 1¼-inch pipe coils, 1 to 100 feet long. *(Continued)*

Length of Coil	NUMBER OF 1¼" PIPES											
	1	2	3	4	5	6	7	8	9	10	11	12
51	22	44	66	88	110	132	154	175	197	219	241	263
52	22	45	67	89	112	134	157	179	201	224	246	268
53	23	46	68	91	114	137	160	182	205	228	251	273
54	23	46	70	93	116	139	163	186	209	232	255	279
55	24	47	71	95	118	142	166	189	213	237	260	284
56	24	48	72	96	120	144	169	193	217	241	265	289
57	25	49	74	98	123	147	172	196	221	245	270	294
58	25	50	75	100	125	150	175	200	224	249	274	299
59	25	51	76	101	127	152	178	203	228	254	279	304
60	26	52	77	103	129	155	181	206	232	258	284	310
61	26	52	79	105	131	157	184	210	236	262	289	315
62	27	53	80	107	133	160	187	213	240	267	293	320
63	27	54	81	108	135	163	190	217	244	271	298	325
64	28	55	83	110	138	165	193	220	248	275	303	330
65	28	56	84	112	140	168	196	224	252	280	307	335
66	28	57	85	114	142	170	199	227	255	284	312	341
67	29	58	86	115	144	173	202	230	259	288	317	346
68	29	58	88	117	146	175	205	234	263	292	322	351
69	30	59	89	119	148	178	208	237	267	297	326	356
70	30	60	90	120	151	181	211	241	271	301	331	361
71	31	61	92	122	153	183	214	244	275	305	336	366
72	31	62	93	124	155	186	217	248	279	310	341	372
73	31	63	94	126	157	188	220	251	283	314	345	377
74	32	64	95	127	159	191	223	255	286	318	350	382
75	32	65	97	129	161	194	226	258	290	323	355	387
76	33	65	98	131	163	196	229	261	294	327	359	392
77	33	66	99	132	166	199	232	265	298	331	364	397
78	34	67	101	134	168	201	235	268	302	335	369	402
79	34	68	102	136	170	204	238	272	306	340	374	408
80	34	69	103	138	172	206	241	275	310	344	378	413
81	35	70	104	139	174	209	244	279	313	348	383	418
82	35	71	106	141	176	212	247	282	317	353	388	423
83	36	71	107	143	178	214	250	286	321	357	393	428
84	36	72	108	144	181	217	253	289	325	361	397	433
85	37	73	110	146	183	219	256	292	329	366	402	439
86	37	74	111	148	185	222	259	296	333	370	407	444
87	37	75	112	150	187	224	262	299	337	374	412	449
88	38	76	114	151	189	227	265	303	341	378	416	454
89	38	77	115	153	191	230	268	306	344	383	421	459
90	39	77	116	155	194	232	271	310	348	387	426	464
91	39	78	117	157	196	235	274	313	352	391	430	470
92	40	79	119	158	198	237	277	316	356	396	435	475
93	40	80	120	160	200	240	280	320	360	400	440	480
94	40	81	121	162	202	243	283	323	364	404	445	485
95	41	82	123	163	204	245	286	327	368	409	449	490
96	41	83	124	165	206	248	289	330	372	413	454	495
97	42	83	125	167	209	250	292	334	375	417	459	501
98	42	84	126	169	211	253	295	337	379	421	464	506
99	43	85	128	170	213	255	298	341	383	426	468	511
100	43	86	129	172	215	258	301	344	387	430	473	516

CHAPTER VII

Ventilation Problems as They Affect the Design of Webster Heating Systems

VENTILATION in the past was based on more or less traditional and unscientific standards, but is now receiving more of the consideration warranted by its importance.

The necessity of providing adequate ventilating facilities for public buildings and buildings for various classes of industrial operations has been recognized by the legislative bodies of numerous states and cities, which have passed laws and ordinances governing the quantity of air to be supplied per person, and in some instances also the locations from which the air supply is to be brought into the room and the vitiated air removed.

Ventilation is classed, and rightly so, as a branch of applied science, and it is the duty of the ventilating engineer to apply the principles of this science to the problems with which he is dealing in such a manner that the results obtained will produce the most healthful and comfortable conditions in the ventilated rooms.

A ventilating system may be very satisfactory in regard to the quantity and means of distribution of the air but still fail to produce healthful and comfortable conditions. A good ventilating system should produce immediate physical comfort. The human body is the best indicator as to whether or not these conditions are realized.

Temperature and relative humidity are important factors in producing comfort; the human body is to a great extent influenced by the temperature of the surrounding air, and by the rate at which perspiration is evaporated from the body into the air, which again is influenced by the relative humidity of the air.

It is generally considered that the dry-bulb temperature to produce a sense of comfort to a person at rest is 68 to 70 deg. fahr., provided a proper relation between the dry and wet-bulb temperatures is maintained.

The human organism is very susceptible to abrupt changes such as might be experienced when passing from outdoors on a cold day into a heated room in which the relative humidity is below normal or vice versa.

A ventilating system, to produce conditions of comfort and health, should therefore provide for maintaining a satisfactory relation between temperature and humidity. This relation, with a room temperature of 68 to 70 deg. fahr., generally assumes a relative humidity not below 40 per cent, nor over 60 per cent. Although this assumption is entirely traditional, a relation of humidity to temperature may be found between the limits of which true comfort will result.

Investigations from time to time by various engineering organizations and civic bodies regarding ventilating methods employed in public buildings, and particularly in schools, have disclosed the fact that systems of complete hot-blast heating and ventilation have inherent defects. Many former advocates of this type of equipment now favor the more modern types of "split system."

It has been proven improper from the standpoint of health and comfort to employ a small quantity of highly heated air to replace the heat lost by transmission. The air supply should be large in volume and comparatively low in temperature in order to obtain the best ventilating effect. The nearer the temperature of the incoming air corresponds to the room temperature to be maintained, the more nearly is the ideal condition obtained.

To compensate for the heat losses through wall and glass and other exposures, direct radiating surface should be installed. This direct radiating surface, if placed under the windows, will also overcome the difficulties due to "outside wall and window chill" which, in the hot-blast system of heating, has been a source of considerable discomfort.

The close relation of ventilation and heating makes necessary a discussion as to the effect of various methods of ventilation upon the design of the heating plant. To illustrate these effects, some of the commonest applications of ventilation may be classified as follows:

The fireplace.
Direct-indirect system of heating and ventilation.
Indirect system of gravity ventilation.
Ventilating systems for school buildings.
Ventilating systems of large theatres and auditoriums.
Ventilation of churches.
Ventilation of banquet halls, dining rooms, kitchens, etc.
Exhaust ventilation of industrial plants.
Hot-blast systems of heating for industrial plants.

The fireplace: The purpose of fireplaces is twofold, *first*, ornamental effect, and *second*, utility for warming at times when the heating plant is not in operation. Incidentally, also, the flue or chimney of the fireplace acts as a vent, the chimney effect or flue draft causing continuous outflow of air from the room into the atmosphere.

This outflow of air from the room through the chimney of the fireplace has the tendency of lowering the air temperature and pressure in the room, causing a greater infiltration of air from outdoors than would take place without the fireplace. The additional air finding its way into the room tends to lower the temperature, unless compensation is provided in the form of sufficient additional radiating surface.

Direct-indirect system of heating and ventilation: This method of heating and ventilation, as described in Chapter 6, has come into quite general use in certain sections of the country for ventilating school buildings, public libraries and courthouses.

Indirect system of gravity ventilation: Heating by the indirect system, in which the heat is conveyed entirely by air to the space to be heated, also provides a fair means of ventilation, but is open to the objection of highly heated incoming air.

The amount of air to be circulated is generally stipulated, which requires knowing the temperature to which the incoming air is to be heated so that in cooling from incoming to maintained room temperature enough heat units will be provided to offset the heat losses through windows, wall, and other exposures.

In designing heating plants of the indirect type, the total air to be circulated must be known within a fair degree of accuracy in order to determine the quantity of steam required.

The indirect method of heating requires from three to four times the quantity of steam that would be needed with direct radiation for the same warming effect. This indicates the importance of carefully considering ventilating problems in connection with heating systems, in order to determine proper proportions for boilers, pipes, radiator supply valves, return traps, and any other heating system apparatus which would be affected by the increased steam requirement due to the ventilating equipment.

With the indirect system it is also necessary to provide aspirating radiators in the vent flues.

The method of computing indirect radiating surface for given heating effects and requirements is discussed in Chapter 6.

Ventilating systems for school buildings: The direct-indirect and the indirect systems of heating previously mentioned are frequently used for ventilating school houses of the smaller type, but for buildings of larger proportions mechanical systems of ventilation are generally installed.

The necessity for healthful and comfortable conditions in school buildings has been the main stimulus for enacting ventilating laws by various states and cities.

Great progress has been made in late years in the design of ventilating plants for school buildings. The antiquated hot-blast system of heating and ventilation without provision for humidification has been almost entirely abandoned and superseded by the modern split-system method of ventilating with tempered air, washed and humidified before being delivered into the rooms. Direct radiation is installed for taking care of the heat lost through direct exposures of walls, windows, doors, etc.

Air is generally supplied into the class rooms through registers or diffusers placed at a level of seven to eight feet above the floor with the vent registers near the floor. The most satisfactory arrangement is generally obtained when the heat and vent flues are placed in the corridor walls and the air is blown towards the windows. The vitiated air is discharged through the vent flues into the attic space and passes through ventilators in the roof into the atmosphere.

The cold air intake should preferably be at a point above the roof. The intake openings are dampered, and additional air intake openings are provided in the attic space, making the re-circulation of air from the building possible during the heating-up period in the morning. Delivering the air into the rooms at nearly the temperature to be maintained and with automatic temperature control or modulation supply valves on the direct radiators, gives ideal conditions as near as obtainable.

In computing the requirements for direct heating in the ventilated spaces, it is only necessary to take into account the heat losses due to exposures. Exceptions, however, must be made of rooms which are to be in use after the ventilating system is shut down, such as libraries, reading rooms and offices.

Ventilating systems of school buildings are usually shut down after the close of the afternoon session. Any rooms that may be in use after

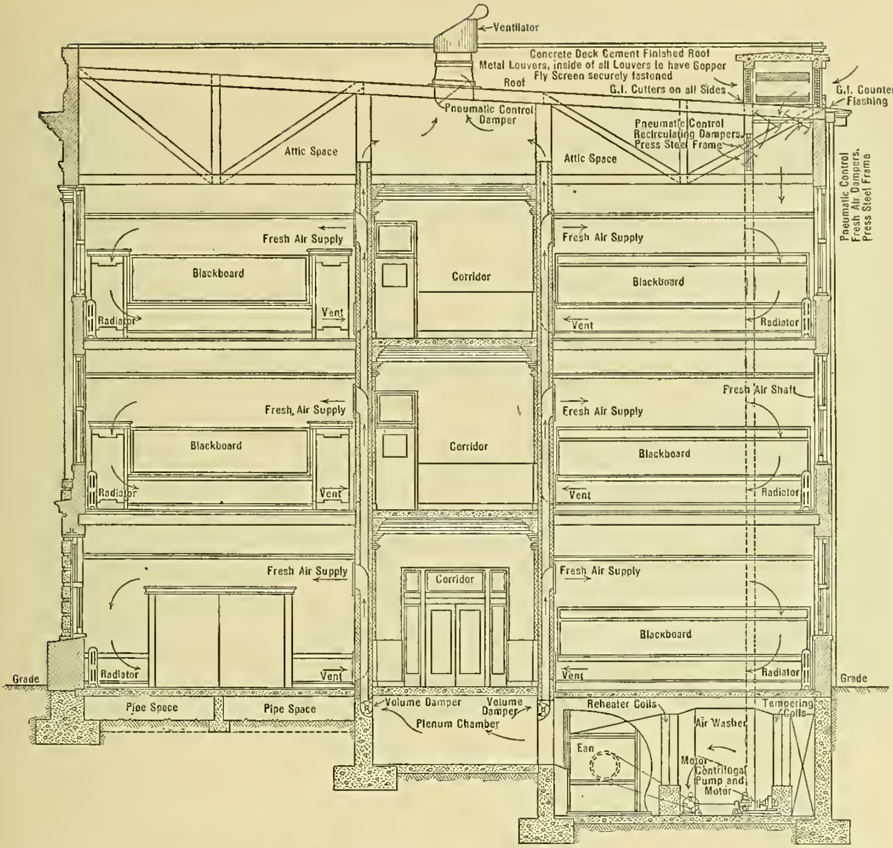


Fig. 7-1. Sectional elevation through class-rooms of a typical school, showing mechanical ventilating equipment and standard method of air supply and venting

that period should have sufficient direct radiation to take care of the maximum requirements without the assistance of the ventilating system.

The steam required to temper the air needed for the ventilating system is generally greatly in excess of that required for the direct system of heating.

Where air washers and humidity control systems are installed, additional steam is required to add to the heat in the air, compensating for the drop in temperature in passing through the air washer and to supply the humidity control apparatus.

Masonry ducts under floors, if used for the main trunk supply system for air distribution, should be so constructed that they can be kept dry at all times. The cooling effect of these masonry ducts must be considered in the design of heating and ventilating plants and during the heating-up period sufficient time should be allowed for heating the ducts thoroughly.

The entire heating plant, including boilers, vacuum pumps, piping system and direct radiation, is effected by the method of ventilation. In the design of the plant all phases of the application and operation of the ventilating system must therefore be known and analyzed to make possible a well balanced system.

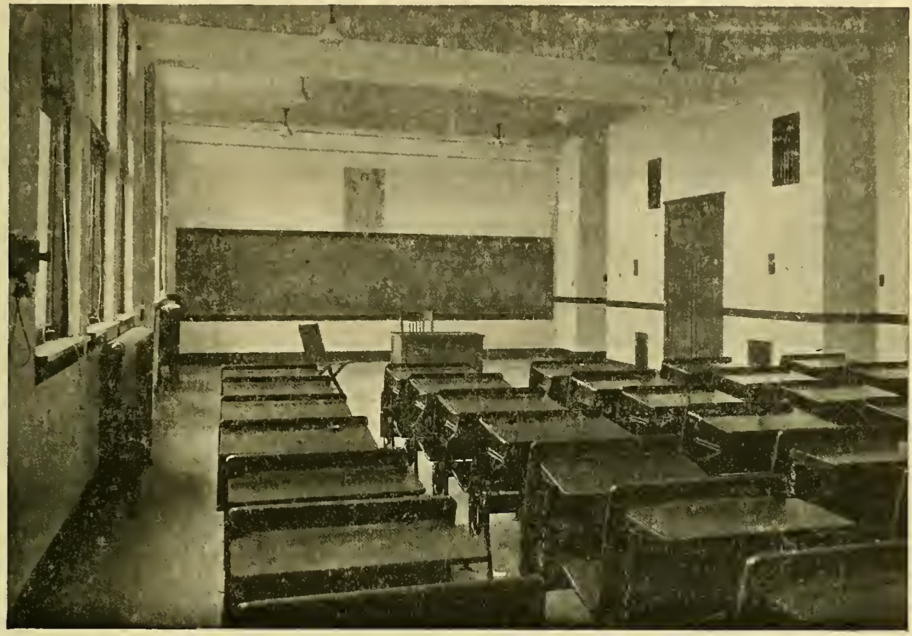


Fig. 7-2. Arrangement of fresh air diffusers, ventilation registers and direct radiators in a modern school room

Ventilation of theatres and auditoriums: The ventilation of theatres and auditoriums presents an entirely different problem from that encountered in the ventilation of a building subdivided into a number of comparatively small rooms.

The problem of proper air distribution in large spaces with seating capacities numbering into thousands requires special study to provide the required quota of fresh air for each occupant.

The down-flow system of air distribution for large rooms has been found to have serious shortcomings because of the difficulty of inducing the air flow toward the center portions. The exhaust flues are necessarily in the walls and the air currents flow close to them, causing a zone of practically no air movement in the center portion.

The up-flow system of air distribution, admitting the air through openings in the floor, under the seats, provides a means of distributing the air uniformly over the entire area. The spaces under the main floor and under floors of balconies are used as plenum chambers into which the fans dis-

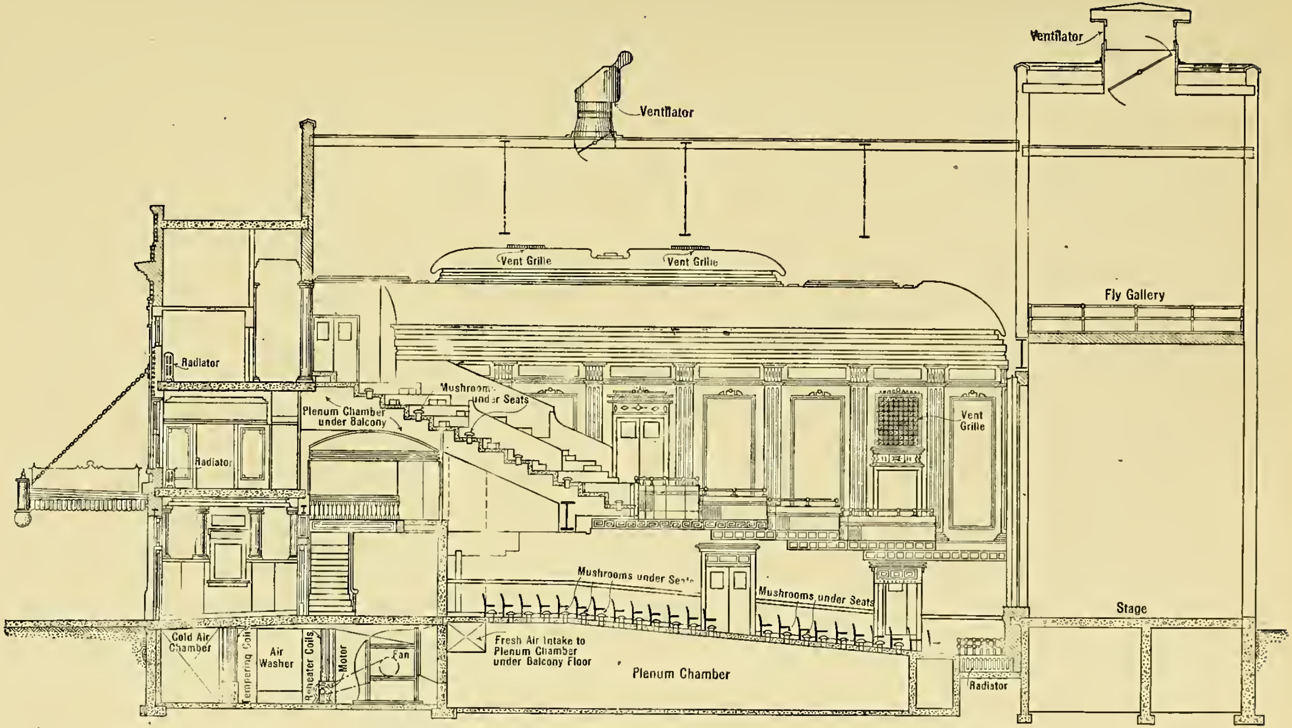


Fig. 7-3. Sectional elevation of auditorium, showing mushroom system of air supply

charge, building up in these chambers a static air pressure and producing a uniform outflow of air through each opening.

It is important that the air outlets shall be proportioned for an air velocity through them not exceeding 200 feet per minute. Higher outlet velocities are apt to produce objectionable drafts near the floor level.

With the up-flow system, the openings for the removal of air are placed in the ceiling or in the walls near the ceiling. Owing to the difficulty of supplying a sufficient quantity of warm air in the orchestra pit for proper heating, liberal amounts of direct radiation in the pit are usually necessary.

Ventilating systems for theatres and auditoriums are usually operated only during the performances, so that portions of the structure which are in use at other times should be heated by direct radiation.

The quantity of air supplied to theatre auditoriums, on the basis of 30 cubic feet per minute per occupant, is usually so large that sufficient heat is supplied by delivering the air into the space at a temperature a few degrees higher than that to be maintained. The temperature regulating system should be flexibile enough to automatically reduce the incoming air temperature when a large percentage of the seats are occupied and in this way prevent excessive temperature rise in the room.

The modern theatre would not be complete without the installation of air washers, humidity-control system, and, for summer use, a refrigerating system for cooling the air.

The design of heating and ventilating systems for large auditoriums presents an interesting problem in engineering. One is so closely affected by the other that both should be worked out together so that the results obtained will harmonize.

Ventilation of churches: Ventilation for churches is usually applied only to the main auditorium and the Sunday-school room, the balance of the building being heated by direct radiation. Most churches are not continuously heated, and the warming-up period should on that account receive careful consideration by the designer. The ventilating system is generally operated during the Sunday services only.

Whether to use the up-flow system of air distribution or to discharge the air into the room through registers in the wall will greatly depend on the size of the room to be ventilated. In large churches, a combination of both, blowing in the air partly through openings in the floors in the aisles, and partly through registers in the walls, will give good results. Vent openings are usually placed in the walls near the floor and in the ceiling.

The ventilating system for a church should supply air for ventilation only and no attempt should be made to use the fan system for heating. For satisfactory results, sufficient direct radiation should be provided to compensate for all heat losses due to direct exposures and infiltration. Arrangement for re-circulating the air before the building is occupied will be found a convenience, both from the standpoint of shortening the warming-up period and also of effecting a considerable economy in the fuel consumption.

It is considered good practice to have a separate boiler and piping system for that part of the heating and ventilating plant which will be in use Sundays only, having another boiler to heat the portions of the church in use during week days.

Ventilation of banquet halls, dining rooms, meeting rooms, etc.: In no other class of ventilated rooms is the efficiency or inefficiency of the ventilating system so noticeable as in banquet halls, dining rooms and meeting rooms. Smoke-laden air indicates that the ventilating system is not functioning properly, while if the air is clear and fresh in spite of smoking by the guests, a satisfactory diffusion of air in the room is shown.

As already pointed out in connection with other ventilating problems, the air should be brought in as nearly at room temperature as possible, and if heating of the room involves consideration of outside exposures, direct radiation should be used. The location and distribution of the exhaust openings is of prime importance and the exhaust should be accomplished by mechanical means. Vent openings should be placed near both floor and ceiling, and, if the structural conditions permit, additional vents should be provided in the ceiling toward the center of the room.

Kitchens require a very large air change, which should be accomplished by means of exhaust fans. Ordinances of some cities specify a three-minute air change for hotel kitchens, requiring a separate steel vent stack to be extended through the roof for this purpose. An exhaust fan, with inlet connected to this vent shaft, is usually placed in the penthouse. Above the

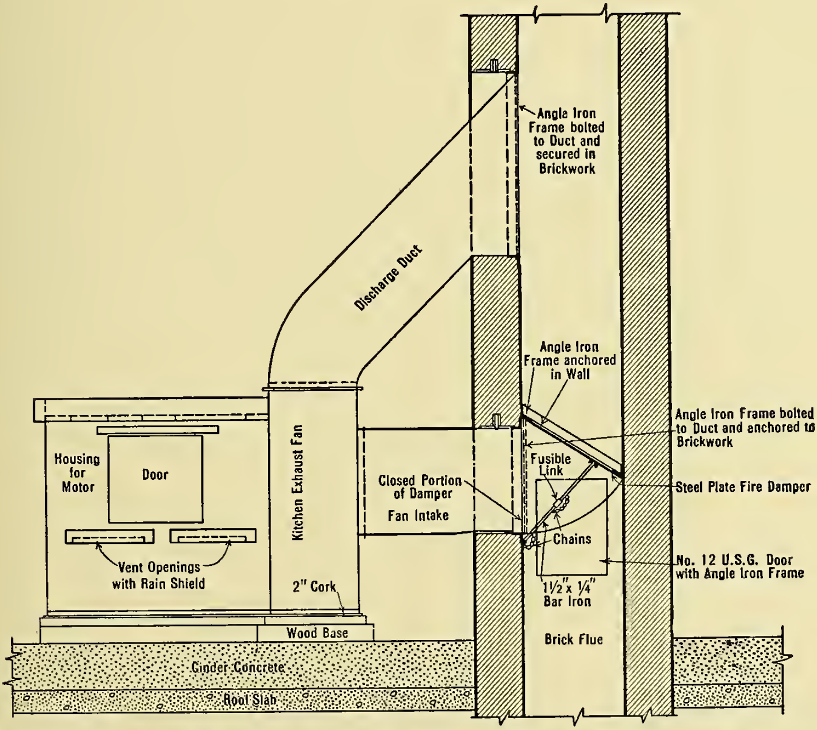


Fig. 7-4. Arrangement of fan, vent stack and safety damper of ventilating equipment for a kitchen

point where the fan inlet connection is made, a tight-fitting damper with chain connection is placed in the vent shaft, and the fan discharge is reconnected to the vent shaft above this damper. In case the chain connection is broken, the damper in the fan intake is closed by gravity, the fan inlet is closed and the stack is opened to the atmosphere. The chain connecting the two dampers is provided with a few links of a material having low fusing point, so that in case of fire in the stack, the links will melt. The damper then disengages, closes the fan intake and permits the stack to burn out without damaging the exhaust fan.

Where kitchens adjoin the dining rooms, the latter can conveniently be exhausted through the kitchen. This greatly reduces the inflow of air from outdoors into the kitchen and at the same time prevents odors from kitchen flowing to dining room.

Where existing conditions do not permit induction of air from warmed spaces to replace that exhaust, the air must necessarily find its way into the kitchen from outdoors and provision must be made to prevent a drop below the desired temperature. This is best accomplished by installing direct or indirect radiation for heating to the temperature needed.

Considerable heat is produced by the ranges and steam cooking utensils, so that the kitchen may be overloaded with radiation unless complete information is available as to the kitchen equipment to be used.

Exhaust ventilation of industrial plants: Industries, which in their operations produce dust, acid fumes, or in any other way contaminate the air, require positive means for removing the dust or fume-laden air from the premises. Mechanical systems of exhaust ventilation are used to maintain a continuous air change by exhausting the dust-laden air.

Various types of machines, such as grinders, buffers, and wood-working machines, are provided with sheet metal ducts running to the exhaust fans, which are usually centrally located, and discharge either into dust-collecting chambers or into the atmosphere, depending upon the nature of the dust or refuse to be handled.

The continuous exhausting of air from any space will cause a corresponding inflow of outdoor air which must be heated to avoid lowering the inside temperature.

If the ventilated spaces have outside exposures, the air is drawn directly from outdoors, and infiltration takes place uniformly over the entire exposed area. A sufficient amount of direct heating surface to heat this air to the temperature to be maintained must be added to the heating surface required for heating the space without the exhaust system.

If, however, the ventilated space has no direct exposure and connects with other rooms so that the air will be drawn from these, the additional radiation must be placed in the rooms from which the air is drawn or indirect inlets must be provided.

Chemical plants requiring the removal of acid fumes must usually exhaust large volumes of air from the rooms, and an equivalent quantity of air must be admitted directly from outdoors. This air is generally admitted through special openings in the walls and is drawn through tempering coils, so that it enters the room at the temperature to be maintained. In

such cases the content losses can be eliminated from the heat loss calculations, and the direct radiation should be sufficient only to compensate for the losses through direct exposures and infiltration. However, where the exhaust system is in use only at intervals, allowances for heating up the contents of the room should be made in figuring the warming-up period. Sufficient direct radiation should be added to supply the heat units required for this purpose.

Hot-blast systems of heating for industrial plants: In industrial structures, such as large foundries, machine shops, erecting shops and round-houses, the hot-blast system of heating, instead of the direct method, is often selected, owing to its lower first cost. From the operating standpoint, however, the hot-blast system is considerably more expensive than the direct, because of the greater amount of steam required for heating by any indirect method. This condition is particularly apparent in cases where all the air is taken

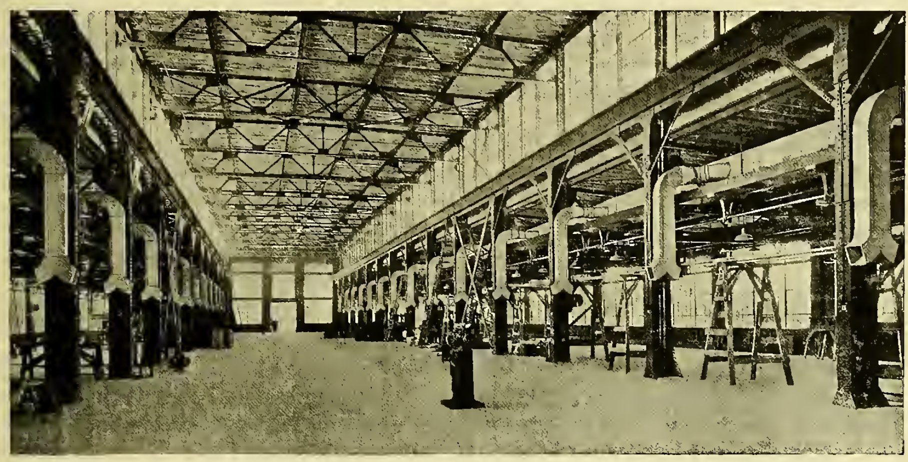


Fig. 7-5. Arrangement of hot-air ducts of hot-blast system in an industrial plant

directly from outdoors and after being circulated through the space is discharged into the atmosphere.

Where air can be taken from the space to be heated and re-circulated, instead of taking it from outdoors, the steam requirements are considerably reduced. In either case, the air must be heated at the fan to such a temperature that in cooling from the air-outlet temperature to that maintained inside all heat losses are offset under maximum conditions.

Only a few general ventilating problems and their direct effect upon heating plant design have been mentioned in this chapter, but these show the importance of analyzing each ventilating problem thoroughly and making all necessary provisions for the ventilating system in heating system design.

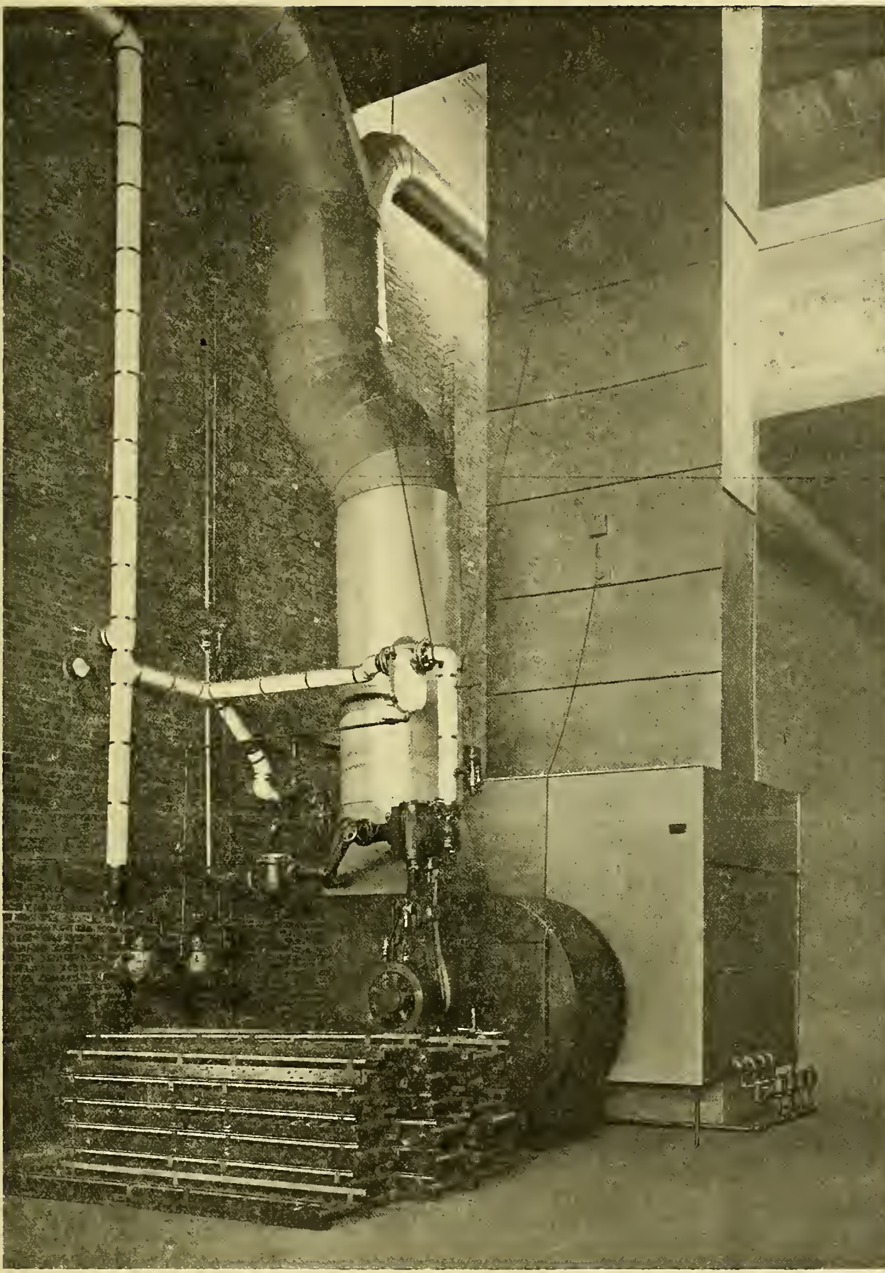


Fig. 7-6. Steam-engine-driven fan unit used for heating a storage warehouse

Air Quantities Required for Ventilation: Air quantities in many states and municipalities are fixed by legal restrictions which must be carefully followed. However, some of the generally accepted standards are mentioned in the following paragraphs.

The type of building and the purpose for which it is to be used are the main factors entering into the design of any ventilating system, not only as to the type of ventilation which is best adapted to each particular problem, but also as to the volume of air required.

Tables 7-1 and 7-2, list kinds of buildings, together with their air requirements. These quantities, with slight variation, have been universally adopted.

Table 7-1. Air Requirements of Various Buildings

Type of Building	Air Supply Cu. Ft. per Occupant per Hr.
School buildings	1800
Theatre and assembly halls	1500
Churches	1500
Prisons	2100
Hospitals: Ordinary	2600
Hospitals: Wounded	3500
Hospitals: Contagion	6000
Residence	1600 to 2000
Factories	2000 to 3000

Table 7-2. Allowable Air Velocities in Various Buildings in Feet per Minute

	Horizontal Ducts	Vertical Risers	Outlets
Factories	1500 to 2800	900 to 1500	600 to 1200
Schools	1000 to 1800	500 to 750	300 to 500
Hospitals	1000 to 1800	500 to 750	300 to 600
Theatres	1000 to 1800	500 to 750	300 to 600
Churches	1000 to 1800	500 to 750	300 to 600

Sizing of the Ducts: Two methods of estimating duct sizes are in common use:

First, the velocity method, in which the velocity is fixed in the various portions of the system, and decreases from the fan outlet to the various points of discharge. This method is applicable in single-duct systems and also in public buildings layouts, where certain velocity standards are required by law.

Referring to the duct design in Fig. 7-7, certain volumes and velocities are given. To determine the size of ducts at any particular point, the volume in cubic feet of air passing that point is divided by the velocity in feet at that point, which gives the required area in square feet.

Determination of the friction in any part of the duct is made by reference to the friction chart, Figure 7-8.

In a single-duct system, the longest duct, or the duct requiring greatest pressure, should be designed for certain velocities and the total pressure

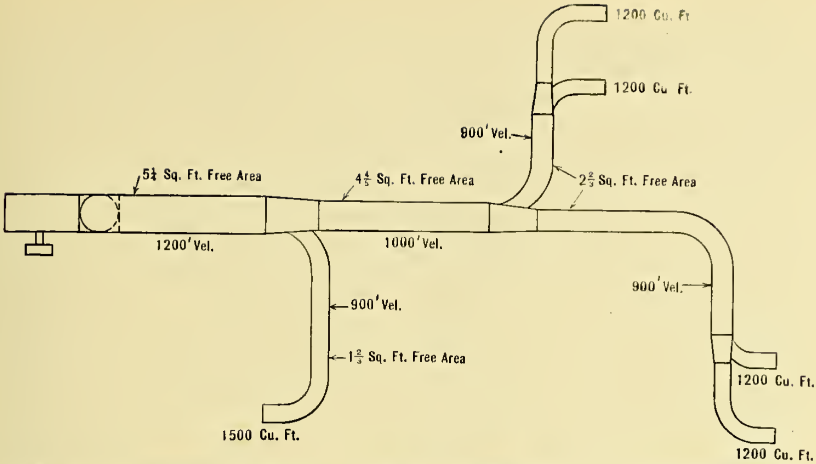


Fig. 7-7. Arrangement of ducts in a trunk-line system. Sized by the velocity method

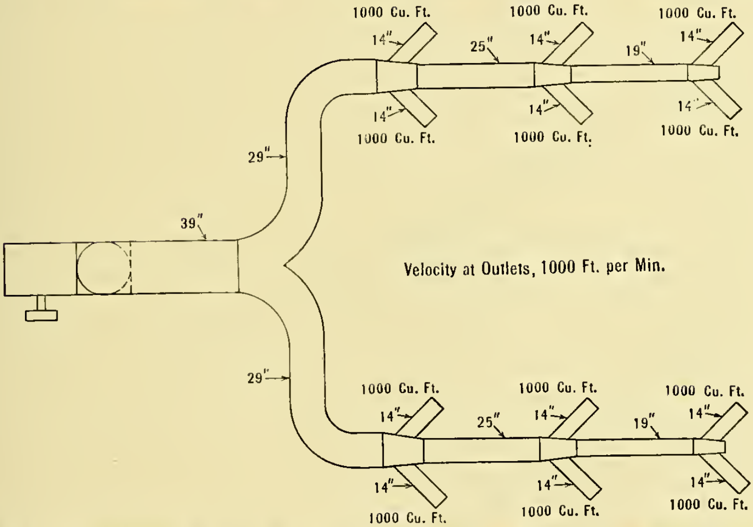


Fig. 7-8. Arrangement of ducts in a trunk-line system. Sized by the pressure-drop method

required at plenum chamber determined from the friction chart, Figure 7-8. All other ducts should then be designed for the same pressure.

Second—The friction-loss method, in which the duct is proportioned for equal friction pressure loss in every foot of run.

This method of duct sizing necessitates assumption of the velocity and volume at the outlets, and is adaptable to trunk-line duct systems such as are common in factories.

Table 7-3 gives an easy and accurate method for sizing ducts. An example of its application follows (See Figure 7-9):

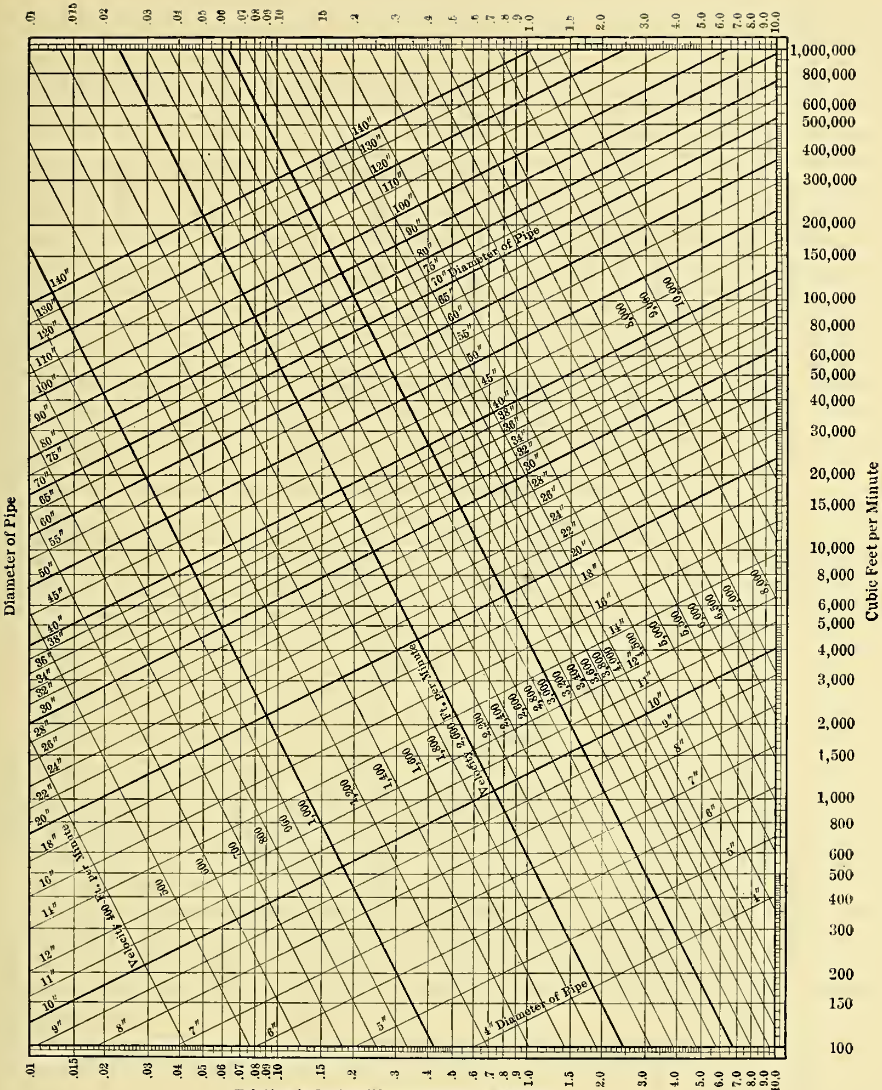


Fig. 7-9. Chart for determining pressure loss in ducts

Assuming a 1000 cu. ft. discharge from each outlet at 1000 ft. velocity per min. the area of the outlet is one (1) sq. ft. or say 14 inches in diameter.

Referring to Table 7-3, a 14-in. pipe is equivalent to 737 1-inch pipes and two 14-in. pipes are equivalent to 1474 1-in. pipes. Also, 1474 1-inch pipes are equivalent to approximately a 19-in. pipe, and so on. To deter-

Table 7-3. Comparison of the Air-carrying Capacity of Various Sizes of Pipes with That of a 1-in. Pipe of Same Length and Equal Friction Pressure Loss

For Example—With an equal pressure loss and equal length, a 4-in. diameter pipe will carry the same volume of air as thirty-two 1-in. pipes.

Diam.	1″ Pipes	Diam.	1″ Pipes	Diam.	1″ Pipes	Diam.	1″ Pipes	Diam.	1″ Pipes
1	1	21	1985	41	10565	61	28850	81	59122
2	5	22	2250	42	11300	62	30200	82	60831
3	16	23	2525	43	12030	63	31350	83	62540
4	32	24	2800	44	12621	64	32500	84	64249
5	56	25	3060	45	13400	65	33975	85	66396
6	88	26	3425	46	14100	66	35300	86	68542
7	129	27	3738	47	15000	67	36600	87	70687
8	180	28	4100	48	15850	68	38000	88	72833
9	244	29	4440	49	16610	69	39275	89	74979
10	317	30	4898	50	17600	70	40250	90	77125
11	402	31	5312	51	18275	71	41995	91	79271
12	501	32	5631	52	19335	72	43740	92	81416
13	613	33	6154	53	20000	73	45449	93	83562
14	737	34	6675	54	21500	74	47158	94	85708
15	876	35	7075	55	22300	75	48887	95	87854
16	1026	36	7735	56	23450	76	50576	96	89999
17	1197	37	8265	57	24500	77	52285		
18	1375	38	8715	58	25600	78	53995		
19	1580	39	9350	59	26700	79	55704		
20	1775	40	10060	60	27700	80	57413		

Table 7-4. Resistance of 90-degree Elbows

Radius of throat of elbow in diameters of pipe	Number of diameters of straight pipe offering equivalent resistance	Radius of throat of elbow in diameters of pipe	Number of diameters of straight pipe offering equivalent resistance
¼	67.0	2½	4.5
½	30.0	3	4.8
¾	16.0	3¼	5.0
1	10.0	4	5.2
1¼	7.5	4½	5.5
1½	6.0	5	5.8
1¾	5.0	5½	6.0
2	4.3		

mine the velocity at any point, the volume at that point is divided by the area in sq. ft. To determine the friction in any portion of the duct reference is made to friction chart Fig. 7-9.

Calculation of Resistance or Pressure: It is not the intention to go into the many complex formula entering into the loss of pressure in ducts but rather to arrange some easily workable method.

The friction chart, Figure 7-9, which is worked out from accepted pressure loss formula, provides a quick, accurate method for determining pressure loss.

Example: Assume that 30,000 cu. ft. of air per minute is passed through a duct 40 inches in diameter and 50 feet long. From the 30,000 cu. ft. division at the right of chart, trace horizontally to intersection with the line representing 40 dia. pipe. Perpendicularly down from this point the friction in inches of water per hundred feet of pipe is given—in this case .54 inches.

For 50 ft. the friction will be 50% of .54 or .27 in. of water. Friction in inches of water multiplied by 0.58 gives friction in ounces.

The resistance is expressed as that of the number of diameters of straight

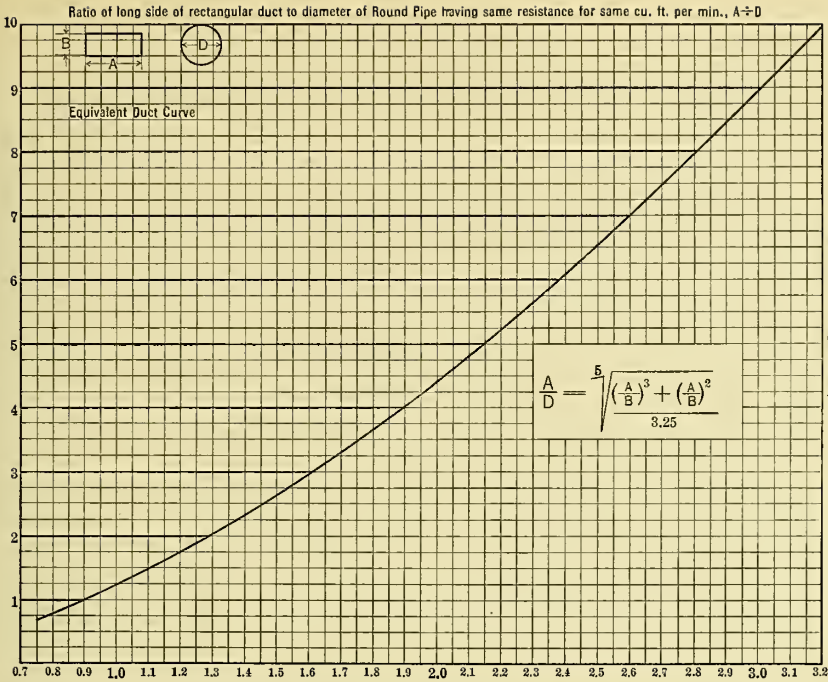


Fig. 7-10. Curve for determining the diameters of round pipes having the same friction loss for same capacity as rectangular ducts of various dimensions

pipe of same diameter as the elbow, and is given for elbows having different radii of throat, also expressed in diameters of pipe. For instance, a 90-deg. elbow of 24-inch pipe, having a radius of throat equal to 1 diameter, that is 24 inches, offers the same resistance to the flow of air as 10 diameters of straight pipe or 20 feet of straight pipe.

To the resistance of the duct system should be added the resistance through tempering and reheating coils, also air washers, plus a small factor of safety, thereby determining the total pressure against which the fan must deliver the specified volume of air.

Where each branch duct leaves the trunk line, there should be provided a volume damper with trunnion, quadrant and locking device, for balancing up the system.

Figure 7-10 is a curve for determining diameter of round pipe having same friction for same capacity as rectangular ducts of varying dimensions.

Selecting the Apparatus

Sizes and arrangement of fans: For fan performances and capacities, reference should be made to the tables issued by the manufacturers of such equipment.

Table 7-5. Cubic Feet of Air One B.t.u. Will Raise One Degree Fahr. at Different Temperatures

Specific heat of air .2375. At zero one cubic foot of air weighs .0864 lb. and $\frac{1 \text{ lb.}}{.0864} = 11.574$ cu. ft. $\frac{11.574}{.2375} = 48.77$ cu. ft. raised one degree by 1 B.t.u.

Table constructed from this formula, small fractional decimals omitted

Temp. air deg. fahr.	Weight of 1 cu. ft.	Cu. ft. in 1 lb.	Cu. ft. 1 B.t.u. will raise 1 deg. fahr.	Temp. air deg. fahr.	Weight of 1 cu. ft.	Cu. ft. in 1 lb.	Cu. ft. 1 B.t.u. will raise 1 deg. fahr.	Temp. air deg. fahr.	Weight of 1 cu. ft.	Cu. ft. in 1 lb.	Cu. ft. 1 B.t.u. will raise 1 deg. fahr.
0	.0864	11.58	48.77	72	.0747	13.39	56.40	152	.0649	15.40	64.90
12	.0842	11.87	50.00	82	.0733	13.64	57.40	162	.0638	15.65	66.00
22	.0824	12.14	51.00	92	.0720	13.90	58.60	172	.0628	15.90	67.00
32	.0807	12.40	52.20	102	.0707	14.14	59.20	182	.0618	16.17	68.00
42	.0791	12.64	53.10	112	.0694	14.40	60.60	192	.0609	16.42	69.10
52	.0776	12.88	54.10	122	.0682	14.65	61.60	202	.0600	16.67	70.10
62	.0761	13.13	55.20	132	.0671	14.90	62.80	212	.0591	16.92	71.30
70	.0750	13.31	56.30	142	.0660	15.15	63.80				

Heaters: To select a heater for any set of conditions it is necessary to know the volume of air to be handled, its initial temperature, and the temperature to which it must be raised.

Two methods for determining the above quantities are available where the building is heated as well as ventilated by the air. One applies where a definite air change is desired or where ventilation must be provided for a given number of people.

Example: Assume a building requiring 18,000 cu. ft. per min. measured at 70 deg. fahr. with a total of 860,000 B. t. u. loss through exposed glass, walls, etc. Then

$$\frac{\text{B.t.u. Loss}}{\text{Cu. ft. per min.} \times .2375 \times .075 \times 60} = \text{diffusion}$$

$$= \frac{860{,}000}{18000 \times .2375 \times .075 \times 60} = 45 \text{ deg. fahr.}$$

45 deg. diffusion + 10 deg. duct loss + 70 deg. desired room temperature = 125 deg. final temperature at coils. In this calculation .2375 is the specific heat of air and is constant and .075 is the weight of one cubic foot of air at the room temperature of 70 deg. (See Table 7-5).

The other method is to decide on the final temperature to be used with some fixed entering temperature.

Example: Suppose the heat loss through exposed walls, glass, etc., is 1,204,500 B.t.u. Assume a final temperature at the heater of 135 deg. fahr. and a loss of 10 deg. in the ducts. The temperature at the duct outlets will then be 125 deg. fahr. The room temperature desired is 65 deg. and the outside temperature is 0 deg.

The difference in the temperature between the duct outlets and the room temperature is available for heating.

$$\text{Cu. ft. per min.} = \frac{\text{B.t.u. per hr.}}{60 \times 60 \times .2375 \times .068} = \frac{1{,}204{,}500}{60 \times 60 \times .2375 \times .068} =$$

20,720 cu. ft. per min. required, in which .2375 is specific heat of air and is constant and .068 is weight of one cu. ft. of air at 65 deg. (See Table 7-5).

Either of the above formulas can be used on split systems where a portion of the losses through walls, glass, etc., are taken care of by direct radiation, and the balance by the incoming air. In the split system where all heat loss through walls, glass, etc., is taken care of by direct radiation, the final temperature of the air is of course, the same as the room temperature desired. However, in choosing the heater, allowance should be made for some temperature drop in the ducts (usually 10 to 20 degrees).

After we have determined the volume and final temperature of the air the size of heater can readily be chosen from tables furnished by manufacturers.

Table 7-6. B.t.u. Required for Heating Air

This table specifies the quantity of heat in British thermal units required to raise one cubic foot of air through any given temperature interval.

Temperature of Air in Room, deg. fahr.

External Temp.	40°	50°	60°	70°	80°	90°	100°	110°	120°	130°
−40°	1.802	2.027	2.252	2.479	2.703	2.928	3.154	3.379	3.604	3.829
−30°	1.540	1.760	1.980	2.200	2.420	2.640	2.860	3.080	3.300	3.520
−20°	1.290	1.505	1.720	1.935	2.150	2.365	2.580	2.795	3.010	3.225
−10°	1.051	1.262	1.473	1.684	1.892	2.102	2.311	2.522	2.732	2.943
0°	0.822	1.028	1.234	1.439	1.645	1.851	2.056	2.262	2.467	2.673
10°	0.601	0.805	1.007	1.208	1.409	1.611	1.812	2.013	2.215	2.416
20°	0.393	0.590	0.737	0.984	1.181	1.378	1.575	1.771	1.968	2.165
30°	0.192	0.385	0.578	0.770	0.963	1.155	1.345	1.540	1.733	1.925
40°	0.000	0.188	0.376	0.564	0.752	0.940	1.128	1.316	1.504	1.692
50°	0.000	0.000	0.184	0.367	0.551	0.735	0.918	1.102	1.286	1.470
60°	0.000	0.000	0.000	0.179	0.359	0.538	0.718	0.897	1.077	1.256
70°	0.000	0.000	0.000	0.000	0.175	0.350	0.525	0.700	0.875	1.049

Above table from F. Schumann's Manual of Heating and Ventilation.

Boiler Horsepower required: To determine the boiler horsepower required for air heating, the following formula can be used:

$$\frac{\text{Cu. ft. per min.} \times 60 \times A}{B} = \text{Lb. steam per hour.}$$

in which A = B.t.u. required for heating one cu. ft. of air from initial to final temperature (See Table 7-6).

B = latent heat of steam —

$$\frac{\text{Lb. steam per hr.}}{34.5} = \text{Boiler horsepower}$$

From the manufacturers' tables the condensation rates per square foot of surface are given for various velocities and temperatures, and it is well to check up the above formula from these given factors.

CHAPTER VIII

Proportioning of Chimneys

NO problem in the heating of buildings presents greater elements of uncertainty than that of properly proportioning the chimney.

In larger installations, such as isolated plants for the production of power, light and heat, the conditions may usually be very accurately determined in advance. By use of the formula given hereafter, proper results follow in almost every case.

A. Chimneys for House-Heating Boilers

In small plants, and particularly residence heating, it is not practicable to make such accurate advance determinations of all the conditions. Usually the chimney is built into the wall, thereby requiring that its cross-section must be proportioned to the width of brick. Chimneys so built are usually either smoothly mortared on the inside or lined with thin tile of rectangular or circular cross-section. The latter gives such freedom from friction and eddy currents and lessened surface for loss of heat in the gases that a round chimney lining will frequently give fully as good results as would be obtained in the square of brick-work in which it is enclosed.

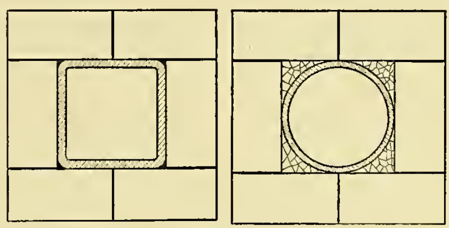


Fig. 8-1. Cross-sections through typical house chimneys.

The inclination to cut down cross-sectional area to save cost and space in the portion of building through which the chimney passes should be discouraged as false economy. Once the chimney is built into the structure, increase of area is practically impossible, and a chimney that is too small remains a source of discomfort and waste during the entire life of the structure. Little is saved in building an 8½ inch by 13 inch flue as compared with a 13 inch by 18 inch flue, the latter having more than twice the area and more than twice the capacity, while the bricks per course are as 9 is to 7. See Figure 8-2.

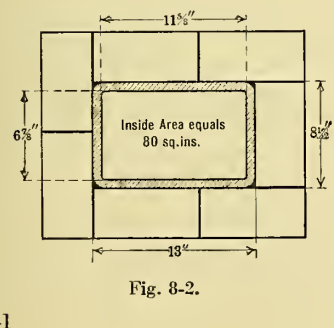


Fig. 8-2.

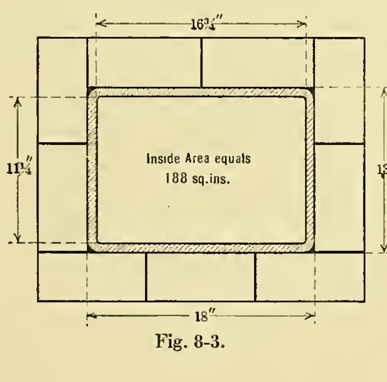


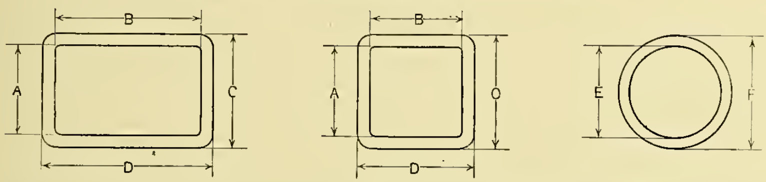
Fig. 8-3.

To get the greatest effectiveness, a definite amount of draft must be available. The actual amount required varies widely for different types of commercial cast-iron boilers, and, unfortunately, it is not always possible to know in advance which make of these boilers will be selected or may later be installed. It is, therefore, preferable to provide for excessive draft which may be controlled by damper, rather than to risk insufficient draft, the remedying of which is almost hopeless.

The curves in Figure 8-3 on page 8–1 indicate the probable capacity of chimneys of differing sizes and forms and various heights necessary to produce the proper draft for the average cast-iron heating boiler capable of generating steam at the given hourly rate.

It must, however, be borne in mind that the location of the building in relation to topography and surrounding structures may render a chimney absolutely inefficient, while another similar in every respect of height and cross-section, used for similar boiler and fuel, but favorably located, will be able to produce a superabundance of draft; also that the resistance due to thickness of coal bed, character and quality of fuel as well as resistance between the combustion chamber and chimney vary in different makes of boilers having similar ratings, and that these resistances form a large part of the total head for which chimneys are required.

Table 8-1. Dimensions of Flue Linings



AS MANUFACTURED BY

The Delaware Clay Products Co., Pittsburgh, Penna.

Rectangular and Square					Circular		
Sq. In. Free Area	A	B	C	D	Sq. In. Free Area	E	F
23	3¼	7¼	4½	8½	28	6	7¼
36	3⅛	11⅝	4½	13	38	7	8⅜
47	2⅞	16⅜	4½	18	50	8	9½
39	6¼	6¼	7½	7½	64	9	10 5/16
52	7 3/16	7 3/16	8½	8½	78	10	11¾
80	6 15/16	11 7/16	8½	13	113	12	14
110	6¾	16¼	8½	18	176	15	17¼
129	11⅜	11⅜	13	13	254	18	20½
188	11¼	16¾	13	18	314	20	22¾
256	16	16	18	18	380	22	25¼
					452	24	27¼

W. S. Dickey Clay Mfg. Co., Kansas City, Mo.

Rectangular and Square					Circular		
Sq. In. Free Area	A	B	C	D	Sq. In. Free Area	E	F
29	3¾	7¾	4½	8½			
61	7 13/16	7 13/16	8½	8½			
46	3¾	12¼	4½	13			
92	7⅝	12⅛	8½	13			
145	12 1/16	12 1/16	13	13			
127	7⅝	16⅝	8½	17½	125	12⅝	14¼
202	12⅛	16⅝	13	17½			
270	16 7/16	16 7/16	17½	17½			
					291	19¼	21½
					499	25 3/16	27½

Robinson Clay Products Co., Akron, Ohio

Rectangular and Square					Circular		
Sq. In. Free Area	A	B	C	D	Sq. In. Free Area	E	F
23	3¼	7	4¾	8⅝	28	6	7⅛
36	3 1/16	11¾	4¾	13¼	38	7	8½
60	3⅞	15½	4½	17	50	8	9
47	4½	10½	6	12	64	9	10½
33	5¾	5¾	7¼	7¼	78	10	12
52.5	7¼	7¼	8½	8½	113	12	14
80	6⅞	11⅝	8½	13	176	15	17⅛
104	6½	16	8½	18	254	18	20⅞
127	11¼	11¼	13	13	314	20	23
169	10¾	15¾	13	18	346	21	
240	15½	15½	18	18	380	22	
					452	24	27
					572	27	
					707	30	35
					855	33	
					1018	36	

Note. All dimensions are in inches and subject to slight variations.

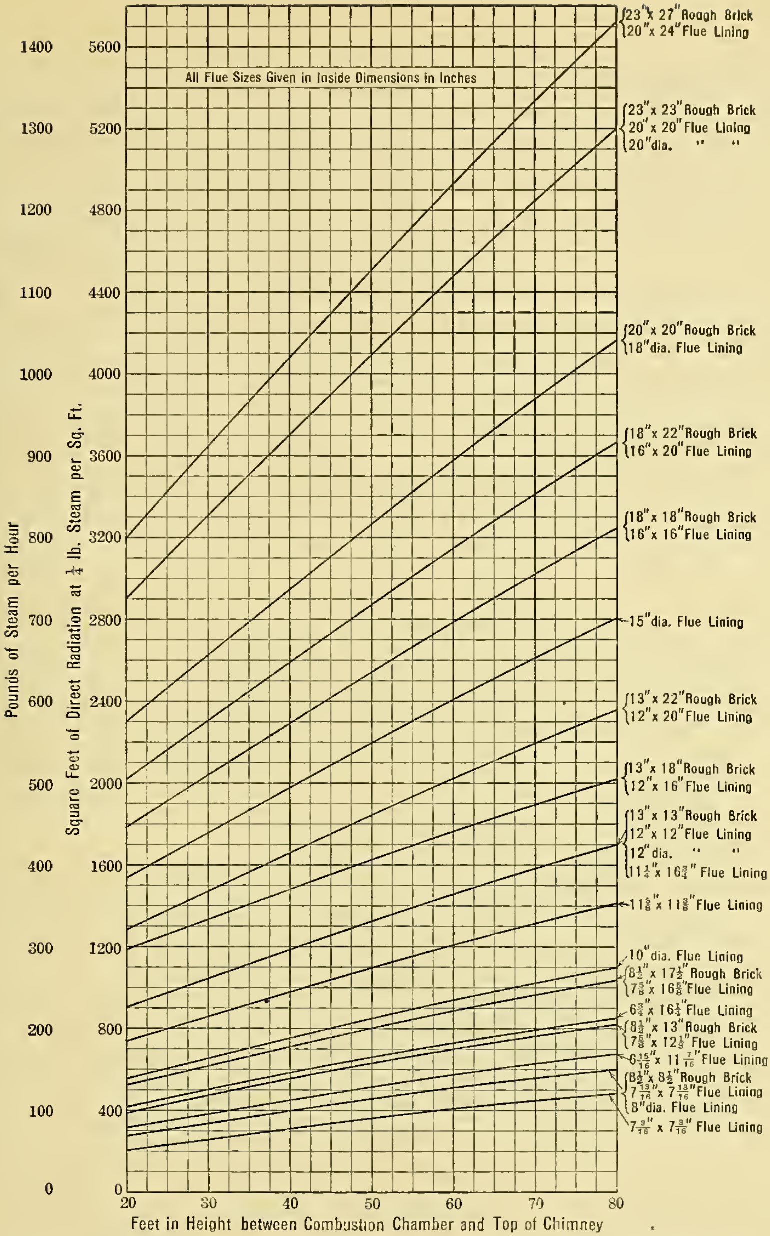


Fig. 8-3—Probable capacities of chimneys of different forms, sizes and heights to produce proper draft for average cast-iron boiler using anthracite coal.

B. Chimneys and Draft for Power Boilers*

The height and diameter of a properly designed chimney depend upon the amount of fuel to be burned, the design of the flue, with its arrangement relative to the boiler or boilers, and the altitude of the plant above sea level. There are so many factors involved that as yet there has been produced no formula which is satisfactory in taking them all into consideration and the methods used for determining stack sizes are largely empirical. In this chapter a method sufficiently comprehensive and accurate to cover all practical cases will be developed and illustrated.

Draft is the difference in pressure available for producing a flow of the gases. If the gases within a stack be heated, each cubic foot will expand, and the weight of the expanded gas per cubic foot will be less than that of a cubic foot of the cold air outside the chimney. Therefore, the unit pressure at the stack base due to the weight of the column of heated gas will be less than that due to a column of cold air. This difference in pressure, like the difference in head of water, will cause a flow of the gases into the base of the stack. In its passage to the stack the cold air must pass through the furnace or furnaces of the boilers connected to it, and it in turn becomes heated. This newly heated gas will also rise in the stack and the action will be continuous.

The intensity of the draft, or difference in pressure, is usually measured in inches of water. Assuming an atmospheric temperature of 62 deg. fahr. and the temperature of the gases in the chimney as 500 deg. fahr., and, neglecting for the moment the difference in density between the chimney gases and the air, the difference between the weights of the external air and the internal flue gases per cubic foot is .0347 pounds, obtained as follows:

Weight of a cubic foot of air at 62 deg. fahr. = .0761 pound
Weight of a cubic foot of air at 500 deg. fahr. = .0414 pound

Difference = .0347 pound

Therefore, a chimney 100 feet high, assumed for the purpose of illustration to be suspended in the air, would have a pressure exerted on each square foot of its cross-sectional area at its base of $.0347 \times 100 = 3.47$ pounds. As a cubic foot of water at 62 deg. fahr. weighs 62.32 pounds, an inch of water would exert a pressure of $62.32 \div 12 = 5.193$ pounds per square foot. The 100-foot stack would, therefore, under the above temperature conditions, show a draft of $3.47 \div 5.193$ or approximately 0.67 inches of water.

The method best suited for determining the proper proportion of stacks and flues is dependent upon the principle that if the cross-sectional area of the stack is sufficiently large for the volume of gases to be handled, the intensity of the draft will depend directly upon the height; therefore, the method of procedure is as follows:

(1) Select a stack of such height as will produce the draft required by the particular character of the fuel and the amount to be burned per square foot of grate surface.

(2) Determine the cross-sectional area necessary to handle the gases without undue frictional losses.

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The application of these rules follows:

DRAFT FORMULA—The force or intensity of the draft, not allowing for the difference in the density of the air and of the flue gases, is given by the formula:

$$D = 0.52\ H \times P \left(\frac{1}{T} - \frac{1}{T_1} \right) \text{ (Formula 8-1)}$$

in which

D = draft produced, measured in inches of water,
H = height of top of stack above grate bars in feet,
P = atmospheric pressure in pounds per square inch,
T = absolute atmospheric temperature,
T_1 = absolute temperature of stack gases.

In this formula no account is taken of the density of the flue gases, it being assumed that it is the same as that of air. Any error arising from this assumption is negligible in practice, as a factor of correction is applied in using the formula to cover the difference between the theoretical figures and those corresponding to actual operating conditions.

The force of draft at sea level (which corresponds to an atmospheric pressure of 14.7 pounds per square inch) produced by a chimney 100 feet high with the temperature of the air at 60 degrees fahr. and that of the flue gases at 500 degrees fahr. is,

$$D = 0.52 \times 100 \times 14.7 \left(\frac{1}{521} - \frac{1}{961} \right) = 0.67$$

Under the same temperature conditions this chimney at an atmospheric pressure of 10 pounds per square inch (which corresponds to an altitude of about 10,000 feet above sea level) would produce a draft of,

$$D = 0.52 \times 100 \times 10. \left(\frac{1}{521} - \frac{1}{961} \right) = 0.45$$

For use in applying this formula it is convenient to tabulate values of the product

$$0.52 \times 14.7 \left(\frac{1}{T} - \frac{1}{T_1} \right)$$

which we will call K, for various values of T_1. With these values calculated for assumed atmospheric temperature and pressure, Formula 8-1 becomes

$$D = K\ H. \quad \text{(Formula 8-2.)}$$

For average conditions the atmospheric pressure may be considered 14.7 pounds per square inch, and the temperature 60 deg. fahr. For these values and various stack temperatures K becomes:

Temperature Stack Gases	*Constant K*
750	.0084
700	.0081
650	.0078
600	.0075
550	.0071
500	.0067
450	.0063
400	.0058
350	.0053

Draft Losses—The intensity of the draft as determined by the above formula is theoretical and can never be observed with a draft gauge or any recording device. However, if the ashpit doors of the boiler are closed and there is no perceptible leakage of air through the boiler setting or flue, the draft measured at the stack base will be approximately the same as the theoretical draft. The difference existing at other times represents the pressure necessary to force the gases through the stack against their own inertia and the friction against the sides. This difference will increase with the velocity of the gases. With the ashpit doors closed the volume of gases passing to the stack are a minimum and the maximum force of draft will be shown by a gauge.

As draft measurements are taken along the path of the gases, the readings grow less as the points at which they are taken are farther from the stack, until in the boiler ashpit, with the ashpit doors open for freely admitting the air, there is little or no perceptible rise in the water of the gauge. The breeching, the boiler damper, the baffles and the tubes, and the coal on the grates all retard the passage of the gases, and the draft from the chimney is required to overcome the resistance offered by the various factors. The draft at the rear of the boiler setting where connection is made to the stack or flue may be 0.5 inch, while in the furnace directly over the fire it may not be over, say, 0.15 inch, the difference being the draft required to overcome the resistance offered in forcing the gases through the tubes and around the baffling.

One of the most important factors to be considered in designing a stack is the pressure required to force the air for combustion through the bed of fuel on the grates. This pressure will vary with the nature of the fuel used, and in many instances will be a large percentage of the tota draft. In the case of natural draft, its measure is found directly by noting the draft in the furnace, for with properly designed ashpit doors it is evident that the pressure under the grates will not differ sensibly from atmosphere pressure.

Loss in Stack—The difference between the theoretical draft as determined by Formula 8-1 and the amount lost by friction in the stack proper is the available draft, or that which the draft gauge indicates when connected to the base of the stack. The sum of the losses of draft in the flue, boiler and furnace must be equivalent to the available draft, and as these quantities can be determined from record of experiments, the problem of designing a stack becomes one of proportioning it to produce a certain available draft.

The loss in the stack due to friction of the gases can be calculated from the following formula:

$$\triangle D = \frac{fW^2CH}{A^3} \qquad \text{(Formula 8-3)}$$

in which

$\triangle D$ = draft loss in inches of water,
W = weight of gas in pounds passing per second,
C = perimeter of stack in feet,
H = height of stack in feet,

f = a constant with the following values at sea level:

.0015 for steel stacks, temperature of gases 600 deg. fahr.

.0011 for steel stacks, temperature of gases 350 deg. fahr.

.0020 for brick or brick-lined stacks, temperature of gases 600 deg. fahr.

.0015 for brick or brick-lined stacks, temperature of gases 350 deg. fahr.

A = area of stack in square feet.

This formula can also be used for calculating the frictional losses for flues, in which case, C = the perimeter of the flue in feet, H = the length of the flue in feet, the other values being the same as for stacks.

The available draft is equal to the difference between the theoretical draft from Formula 8-2 and the loss from Formula 8-3, hence:

$$d^1 = \text{available draft} = KH - \frac{fW^2CH}{A^3} \quad \text{(Formula 8-4)}$$

Table 8-0. Available Draft

Calculated for 100 ft. stack of different diameters assuming stack temperature of 500° F. and 100 lbs. of gas per H. P. For other heights of stack multiply draft by height ÷ 100

Horse Power	DIAMETER OF STACK IN INCHES 36	42	48	54	60	66	72	78	84	90	96	102	108	114	120
100	.64														
200	.55	.62													
300	.41	.55	.61												
400	.21	.46	.56	.61											
500		.34	.50	.51	.61										
600		.19	.42	.53	.59										
700			.34	.48	.56	.60	.63								
800			.23	.43	.52	.58	.61	.63							
900				.36	.49	.56	.60	.62	.64						
1000				.29	.45	.53	.58	.61	.63	.64					
1100					.40	.50	.56	.60	.62	.63	.64				
1200					.35	.47	.54	.58	.61	.63	.64	.65			
1300					.29	.44	.52	.57	.60	.62	.63	.64	.65		
1400						.40	.49	.55	.59	.61	.63	.64	.65	.65	
1500						.36	.47	.53	.58	.60	.62	.63	.64	.65	.65
1600						.31	.43	.52	.56	.59	.62	.63	.64	.65	.65
1700							.41	.50	.55	.58	.61	.62	.64	.64	.65
1800							.37	.47	.54	.57	.60	.62	.63	.64	.65
1900							.34	.45	.52	.56	.59	.61	.63	.64	.64
2000								.43	.50	.55	.59	.61	.62	.63	.64
2100								.40	.49	.54	.58	.60	.62	.63	.64
2200								.38	.47	.53	.57	.59	.61	.62	.64
2300								.35	.45	.52	.56	.59	.61	.62	.63
2400								.32	.43	.50	.55	.58	.60	.62	.63
2500									.41	.49	.54	.57	.60	.61	.63

Horse Power	DIA. STACK IN INCHES 90	96	102	108	114	120	132	144
2600	.47	.53	.56	.59	.61	.62	.64	.65
2700	.45	.52	.55	.58	.60	.62	.64	.65
2800	.44	.50	.55	.58	.60	.61	.61	.65
2900	.42	.49	.54	.57	.59	.61	.63	.65
3000	.40	.48	.53	.56	.59	.61	.63	.64
3100	.38	.47	.52	.56	.58	.60	.63	.64
3200		.45	.51	.55	.58	.60	.63	.64
3300		.44	.50	.54	.57	.59	.62	.64
3400		.42	.49	.53	.56	.59	.62	.64
3500		.40	.48	.52	.56	.58	.62	.64
3600			.47	.52	.55	.58	.61	.63
3700			.45	.51	.55	.57	.61	.63
3800			.44	.50	.54	.57	.61	.63
3900			.43	.49	.53	.56	.60	.63
4000			.42	.48	.52	.56	.60	.62
4100			.40	.47	.52	.55	.60	.62
4200			.39	.46	.51	.55	.59	.62
4300				.45	.50	.54	.59	.62
4400				.44	.49	.53	.59	.62
4500				.43	.49	.53	.58	.61
4600				.42	.48	.52	.58	.61
4700				.41	.47	.51	.57	.61
4800				.40	.46	.51	.57	.60
4900					.45	.50	.57	.60
5000					.44	.49	.56	.60

For Other Stack Temp. Add or Deduct Before Multiplying by $\frac{\text{Height}}{100}$

For 750° Fahr. Add .17 In.	For 650° Fahr. Add .11 In.	For 550° Fahr. Add .04 In.	For 400° Fahr. Deduct .09 In.
For 700° Fahr. Add .14 In.	For 600° Fahr. Add .08 In.	For 450° Fahr. Deduct .04 In.	For 350° Fahr. Deduct .14 In.

Table 8-1 gives the available draft in inches that a stack 100 feet high will produce when serving different horse powers of boilers with the methods of calculation for other heights.

Height and Diameter of Stacks—From Formula 8-4, it becomes evident that a stack of certain diameter, if it be increased in height, will produce the same available draft as one of larger diameter, the additional height being required to overcome the added frictional loss. It follows that among the various stacks that would meet the requirements of a particular case there must be one which can be constructed more cheaply than the others. It has been determined from the relation of the cost of stacks to their diameters and heights, in connection with the formula for available draft, that the minimum cost stack has a diameter dependent solely upon the horse power of the boilers it serves, and a height proportional to the available draft required.

Assuming 120 pounds of flue gas per hour for each boiler horse power, which provides for ordinary overloads and the use of poor coal, the method above stated gives:

For an unlined steel stack—diameter in inches = $4.68 \text{ h.p.}^{\frac{2}{5}}$. (Formula 8-5.)

For a stack lined with masonry—diameter in inches = $4.92 \text{ h.p.}^{\frac{2}{5}}$. (Formula 8-6.)

In both of these formulae h.p. = the rated horse power of the boiler.

From this formula the curve, Figure 8-4, has been calculated and from it the stack diameter for any boiler horse power can be selected.

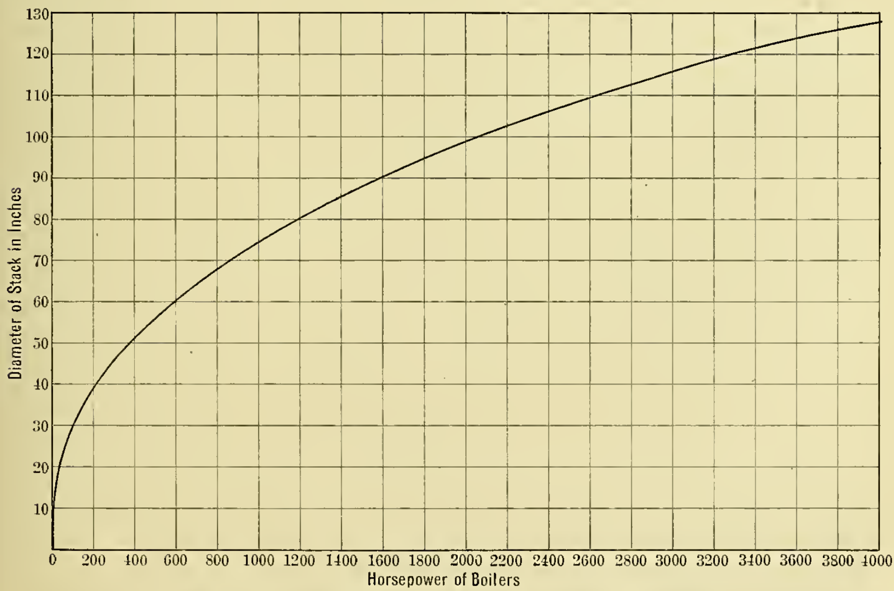


Fig. 8-4 Diameter of Stacks and Horse Power they will Serve

Computed from Formula (8-5). For brick or brick-lined stacks increase the diameter 6 per cent.

For stoker practice where a large stack serves a number of boilers, the area is usually made about one-third more than the above rules call for, which allows for leakage of air through the setting of any idle boilers, irregularities in operating conditions, etc.

Stacks with diameters determined as above will give an available draft which bears a constant ratio of the theoretical draft, and allowing for the cooling of the gases in their passage upward through the stack, this ratio is 8. Using this factor in Formula 8-2, and transposing, the height of the chimney becomes,

$$H = \frac{d^1}{.8K} \qquad \text{(Formula 8-7)}$$

Where H = height of stack in feet above the level of the grates,
d^1 = available draft required,
K = constant as in formula.

Losses in Flues—The loss of draft in straight flues due to friction and inertia can be calculated approximately from Formula 8-3, which was given for loss in stacks. It is to be borne in mind that C in this formula is the actual perimeter of the flue and is least, relative to the cross-sectional area, when the section is a circle, is greater for a square section, and greatest for a rectangular section. The retarding effect of a square flue is 12 per cent greater than that of a circular flue of the same area and that of a rectangular with sides as 1 and 1½, 15 per cent greater. The greater resistance of the more or less uneven brick or concrete flue is provided for in the value of the constants given for Formula 8-3. Both steel and brick flues should be short and should have as near a circular or square cross-section as possible. Abrupt turns are to be avoided, but as long easy sweeps require valuable space, it is often desirable to increase the height of the stack rather than to take up added space in the boiler room. Short right-angle turns reduce the draft by an amount which can be roughly approximated as equal to 0.05 inch for each turn. The turns which the gases make in leaving the damper box of a boiler, in entering a horizontal flue and in turning up into a stack should always be considered. The cross-sectional areas of the passages leading from the boilers to the stack should be of ample size to provide against undue frictional loss. It is poor economy to restrict the size of the flue and thus make additional stack height necessary to overcome the added friction. The general practice is to make flue areas the same or slightly larger than that of the stack; these should be, preferably, at least 20 per cent greater, and a safe rule to follow in figuring flue areas is to allow 35 square feet per 1000 horse power. It is unnecessary to maintain the same size of flue the entire distance behind a row of boilers, and the areas at any point may be made proportional to the volume of gases that will pass that point. That is, the areas may be reduced as connections to various boilers are passed.

With circular steel flues of approximately the same size as the stacks, or reduced proportionally to the volume of gases they will handle, a convenient rule is to allow 0.1-inch draft loss per 100 feet of flue length and 0.05 inch for each right-angle turn. These figures are also good for square or rectangular steel flues with areas sufficient to provide against excessive frictional loss. For losses in brick or concrete flues, these figures should be doubled.

Underground flues are less desirable than overhead or rear flues for the reason that in most instances the gases will have to make more turns where underground flues are used and because the cross-sectional area of such flues will oftentimes be decreased on account of an accumulation of dirt or water which it may be impossible to remove.

In tall buildings, such as office buildings, it is frequently necessary in order to carry spent gases above the roofs to install a stack the height of which is out of all proportion to the requirements of the boilers. In such cases it is permissible to decrease the diameter of a stack, but care must be taken that this decrease is not sufficient to cause a frictional loss in the stack as great as the added draft intensity due to the increase in height, which local conditions make necessary.

In such cases also the fact that the stack diameter is permissibly decreased is no reason why flue sizes connecting to the stack should be decreased. These should still be figured in proportion to the area of the stack that would be furnished under ordinary conditions or with an allowance of 35 square feet per 1000 horse power, even though the cross-sectional area appears out of proportion to the stack area.

Loss in Boilers—In calculating the available draft of a chimney 120 pounds per hour has been used as the weight of the gases per boiler horse power. This covers an overload of the boiler to an extent of 50 per cent and provides for the use of poor coal. The loss in draft through a boiler proper will depend upon its type and baffling and will increase with the per cent of rating at which it is run. No figures can be given which will cover all conditions, but for approximate use in figuring the available draft necessary it may be assumed that the loss through a boiler will be 0.25 inch where the boiler is run at rating, 0.40 inch where it is run at 150 per cent of its rated capacity, and 0.70 inch where it is run at 200 per cent of its rated capacity.

Loss in Furnace—The draft loss in the furnace or through the fuel bed varies between wide limits. The air necessary for combustion must pass through the interstices of the coal on the grate. Where these are large, as in the case with broken coal, but little pressure is required to force the air through the bed; but if they are small, as with bituminous slack or small sizes of anthracite, a much greater pressure is needed. If the draft is insufficient the coal will accumulate on the grates and a dead, smoky fire will result with the accompanying poor combustion; if the draft is too great, the coal may be rapidly consumed on certain portions of the grate, leaving the fire thin in spots and a portion of the grates uncovered with the resulting losses due to an excessive amount of air.

Draft Required for Different Fuels—For every kind of fuel and rate of combustion there is a certain draft with which the best general results are obtained. A comparatively light draft is best with the free-burning bituminous coals and the amount to use increases as the percentage of volatile matter diminishes and the fixed carbon increases, being highest for the small sizes of anthracites. Numerous other factors, such as the thickness of fires, the percentage of ash and the air spaces in the grates bear directly on this question of the draft best suited to a given combustion rate. The effect of these factors can only be found by experiment. It is almost im-

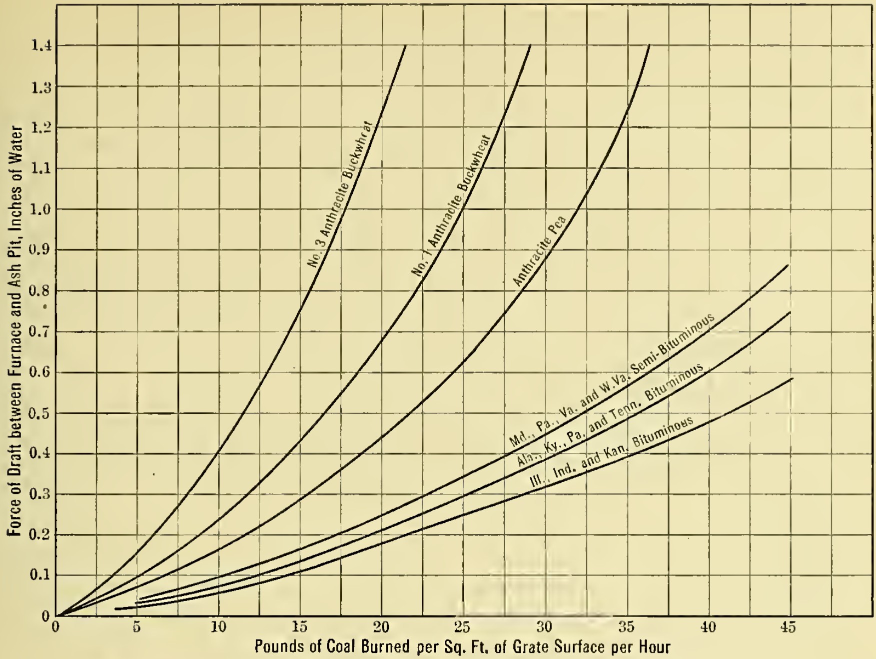


Fig. 8-5. Draft Required at Different Combustion Rates for Various Kinds of Coal

possible to show by one set of curves the furnace draft required at various rates of combustion for all of the different conditions of fuel, etc., that amy be met. The curves in Figure 8-5, however, give the furnace draft necessary to burn various kinds of coal at the combustion rates indicated by the abcissae, for a general set of conditions. These curves have been plotted from the records of numerous tests and allow a safe margin for economically burning coals of the kinds noted.

Rate of Combustion—The amount of coal which can be burned per hour per square foot of grate surface is governed by the character of the coal and the draft available. When the boiler and grate are properly proportioned, the efficiency will be practically the same, within reasonable limits, for different rates of combustion. The area of the grate, and the ratio of this area to the boiler heating surface will depend upon the nature of the fuel to be burned, and the stack should be so designed as to give a draft sufficient to burn the maximum amount of fuel per square foot of grate surface corresponding to the maximum evaporative requirements of the boiler.

Solution of a Problem—The stack diameter can be determined from the curve, Figure 8-4. The height can be determined by adding the draft losses in the furnace, through the boiler and flues, and computing from Formula 8-7 the height necessary to give this draft.

Example: Proportion a stack for boilers rated at 2000 horse power, equipped with stokers, and burning bituminous coal that will evaporate

8 pounds of water from and at 212 deg. fahr. per pound of fuel; the ratio of boiler heating surface to grate surface being 50: 1; the flues being 100 feet long and containing two right-angle turns; the stack to be able to handle overloads of 50 per cent; and the rated horse power of the boilers based on 10 square feet of heating surface per horse power.

The atmospheric temperature may be assumed as 60 deg. fahr. and the flue temperatures at the maximum overload as 550 deg. fahr. The grate surface equals 400 square feet. The total coal burned at rating $= \frac{2000 \times 34\frac{1}{2}}{8}$ = 8624 pounds. The coal per square foot of grate surface per hour at rating $= \frac{8624}{400} = 22$ pounds.

For 50 per cent overload the combustion rate will be approximately 60 per cent greater than this, or $1.60 \times 22 = 35$ pounds per square foot of grate surface per hour. The furnace draft required for the combustion rate, from the curve, Figure 8-5, is 0.6 inch. The loss in the boiler will be 0.4 inch, in the flue 0.1 inch, and in the turns $2 \times 0.05 = 0.1$ inch. The available draft required at the base of the stack is, therefore,

	Inches
Boiler	0.4
Furnace	0.6
Flues	0.1
Turns	0.1
Total	1.2

Since the available draft is 80 per cent of the theoretical draft, this draft due to the height required is $1.2 \div .8 = 1.5$ inches.

The chimney constant for temperatures of 60 deg. fahr. and 550 deg. fahr. is .0071 and from Formula 8-7,

$$H = \frac{1.5}{.0071} = 211 \text{ feet.}$$

Its diameter from curve in Figure 8-4 is 96 inches if unlined, and 102 inches inside if lined with masonry. The cross-sectional area of the flue should be approximately 70 square feet at the point where the total amount of gas is to be handled, tapering to the boiler farthest from the stack to a size which will depend upon the size of the boiler units used.

Correction in Stack Sizes for Altitudes—It has ordinarily been assumed that a stack height for altitude will be increased inversely as the ratio of the barometric pressure at the altitude to that at sea level, and that the stack diameter will increase inversely as the two-fifths power of this ratio. Such a relation has been based on the assumption of constant draft measured in inches of water at the base of the stack for a given rate of operation of the boilers, regardless of altitude.

If the assumption be made that boilers, flues and furnaces remain the same, and further that the increased velocity of a given weight of air passing through the furnace at a higher altitude would have no effect on the combustion, the theory has been advanced* that a different law applies.

* See "Chimneys for Crude Oil," C. R. Weymouth, Trans. A. S. M. E., Dec., 1912.

Under the above assumptions, whenever a stack is working at its maximum capacity at any altitude, the entire draft is utilized in overcoming the various resistances, each of which is proportional to the square of the velocity of the gases. Since boiler areas are fixed, all velocities may be related to a common velocity, say that within the stack, and all resistances may, therefore, be expressed as proportional to the square of the chimney velocity. The total resistance to flow, in terms of velocity head, may be expressed in terms of weight of a column of external air, the numerical value of such head being independent of the barometric pressure. Likewise the draft of a stack, expressed in height of column of external air, will be numerically independent of the barometric pressure. It is evident, therefore, that if a given boiler plant, with its stack operated with a fixed fuel, be transplanted from sea level to an altitude, assuming the temperatures remain constant, the total draft head measured in height of column of external air will be numerically constant. The velocity of chimney gases will, therefore, remain the same at altitude as at sea level and the weight of gases flowing per second with a fixed velocity will be proportional to the atmospheric density or inversely proportional to the normal barometric pressure.

To develop a given horse power requires a constant weight of chimney gas and air for combustion. Hence, as the altitude is increased, the density is decreased and, for the assumptions given above, the velocity through the furnace, the boiler passes, breeching and flues must be correspondingly greater at altitude than at sea level. The mean velocity, therefore, for a given boiler horse power and constant weight of gases will be inversely proportional to the barometric pressure and the velocity head measured in column of external air will be inversely proportional to the square of the barometric pressure.

For stacks operating at altitude it is necessary not only to increase the height but also the diameter, as there is an added resistance within the stack due to the added friction from the additional height. This frictional loss can be compensated by a suitable increase in the diameter and when so compensated, it is evident that, on the assumptions as given, the chimney height would have to be increased at a ratio inversely proportional to the square of the normal barometric pressure.

In designing a boiler for high altitudes, as already stated, the assumption is usually made that a given grade of fuel will require the same draft measured in inches of water at the boiler damper as at sea level, and this leads to making the stack height inversely as the barometric pressures, instead of inversely as the square of the barometric pressures. The correct height, no doubt, falls somewhere between the two values as larger flues are usually used at the higher altitudes, whereas to obtain the ratio of the squares, the flues must be the same size in each case, and again the effect of an increased velocity of a given weight of air through the fire at a high altitude, on the combustion, must be neglected. In making capacity tests with coal fuel, no difference has been noted in the rates of combustion for a given draft suction measured by a water column at high and low altitudes, and this would make it appear that the correct height to use is more nearly that obtained by the inverse ratio of the barometric readings than by the inverse ratio

8—12

of the squares of the barometric readings. If the assumption is made that the value falls midway between the two formulæ the error in using a stack figured in the ordinary way by making the height inversely proportional to the barometric readings would differ about 10 per cent in capacity at an altitude of 10,000 feet, which difference is well within the probable variation of the size determined by different methods. It would, therefore, appear that ample accuracy is obtained in all cases by simply making the height inversely proportional to the barometric readings and increasing the diameter so that the stacks used at high altitudes have the same frictional resistance as those used at low altitudes, although, if desired, the stack may be made somewhat higher at high altitudes than this rule calls for in order to be on the safe side.

The increase of stack diameter necessary to maintain the same friction loss is inversely as the two-fifths power of the barometric pressure.

Table 8-2. Stack Capacities, Correction Factors for Altitudes

Altitude Height in Feet Above Sea Level	Normal Barometer	R Ratio Barometer Reading Sea Level to Altitude	R^2	$R^{\frac{2}{5}}$ Ratio Increase in Stack Diameter
0	30.00	1.000	1.000	1.000
1000	28.88	1.039	1.079	1.015
2000	27.80	1.079	1.064	1.030
3000	26.76	1.121	1.257	1.047
4000	25.76	1.165	1.356	1.063
5000	24.79	1.210	1.464	1.079
6000	23.87	1.257	1.580	1.096
7000	22.97	1.306	1.706	1.113
8000	22.11	1.357	1.841	1.130
9000	21.28	1.410	1.988	1.147
10000	20.49	1.464	2.144	1.165

Table 8-2 gives the ratio of barometric readings of various altitudes to sea level, values for the square of this ratio and values of the two-fifths power of this ratio.

These figures show that the altitude affects the height to a much greater extent than the diameter, and that practically no increase in diameter is necessary for altitudes up to 3000 feet.

For high altitudes the increase in stack height necessary is, in some cases, such as to make the proportion of height to diameter impracticable. The method to be recommended in overcoming, at least partially, the great increase in height necessary at high altitudes is an increase in the grate surface of the boilers which the stack serves, in this way reducing the combustion rate necessary to develop a given power and hence the draft required for such combustion rate.

CHAPTER IX

Boilers

The boiler equipment is the production center of the heating system and the point where the bulk of the operating expense is centered. For this reason, a heating plant can be successful and economical only if the boiler equipment is of correct type, good material and workmanship, well proportioned from the standpoint of its work and ample in capacity.

Service from a heating system cannot properly be termed satisfactory unless the desired heating effect is secured without waste of fuel and without excess labor at the boilers, so it is the endeavor of this chapter to promote a better understanding of the boiler parts and what they should do.

Due consideration should be given to the proper selection of a boiler, not only as to size and capacity, but also as to its adaptability to the existing local conditions which, if not properly considered, may affect the success of the entire plant.

It is not intended in this discussion to cover any details of boiler construction, which properly come under the province of, and can best be solved by, the boiler makers themselves.

Steam boilers have been built in one form or another for nearly two hundred years, yet today they are the least understood of all the important elements which make up a power or heating plant.

Were no consideration to be given to the efficiency of the performance of a steam boiler, such as the number of pounds of water evaporated by a pound of fuel, or the relation of grate surface to heating surface, etc., the problem would be simple.

All the years of experience and the thousands of evaporating tests made have not produced any definite and reliable rule or formula for calculating either the amount of steam that will be generated per hour with a given fuel or the quantity of steam in pounds produced per pound of fuel burned in the furnace.

Lucke* says: "There is no absolute measure of boiler performance as to capacity or efficiency as a basis of comparison to measure the goodness of a boiler as a boiler; comparison must, therefore, be between one and another boiler, or one and another service condition; one boiler may be said to be better than another, or one condition more favorable and another worse, for the result desired, but hardly more than this is possible."

For commercial purposes, boiler capacities seem to be quite well standardized, boilers used for heating work being rated in capacity of square feet of steam radiation, and boilers for power work in boiler horse-power.

The boiler capacity rating in square feet is based on equivalent cast-iron direct radiation with a condensation rate of ¼ lb. steam per sq. ft. per hr.

The American Society of Mechanical Engineers in 1885 adopted a double definition of the "Boiler Horse-power" as follows:

(a) The evaporation of 34.5 lb. of water per hour from and at 212 deg. fahr.

* Engineering Thermodynamics.

(b) The absorption by water, between fuel conditions and that of the steam leaving the boiler, of 33,305 B.t.u. per hr.

A steam boiler consists of the following essential parts: A furnace in which the combustion of the fuel takes place; a vessel to contain water to be evaporated; a steam space where the steam is liberated and where the generated steam is contained; a heating surface to transmit the heat of the furnace to the water; a smoke pipe to carry away the products of combustion, and various attachments or fittings, such as gauges, damper regulators, safety valves, etc.

A proper relation of the first four parts to each other constitutes a successful heating boiler.

It is of prime importance that the furnace is of proper design as regards grate area, size of combustion chamber, ash pit, etc., to give most efficient operation, permitting the consumption of the maximum effective quantity of fuel per square foot of grate area. Further references will be made to importance of selecting the proper kind of grates for the various grades of fuel available in various localities.

The water space or the water-holding capacity of a boiler does not always receive enough attention. It should be remembered that the boiler which holds the greatest quantity of water at or near the normal water line for given size or capacity is the safest one to use, because in such a boiler the water line is not so readily brought down to and below the danger point, as would be the case where there is only about half the water-holding capacity.

An investigation of the various cast-iron boilers to which our remarks regarding the water-holding capacity particularly refer, will show that there is an astonishing difference in this particular feature. Selecting two boilers of the same capacity but of different makes, it will be found that the water-holding capacity at or near normal water line varies as much as 1 to 4. It stands to reason that the boiler from which 4 gallons of water can be withdrawn by lowering the water line ½ inch will be safer than the boiler which shows a lowering of the water line by ½ inch with loss of only one gallon.

Boiler manufacturers recognize more and more that if a boiler is to be successful the steam space should be liberal. The velocity with which the steam bubbles are separated from the water in the liberating space is extremely high. A boiler with limited steam-liberating surface will very likely lose its water under heavy load conditions because under the influence of this velocity particles are carried over with the steam into the piping system.

The heating surface of a boiler includes all parts of the boiler shell, flues, tubes, etc., which are covered by water and exposed to the hot gases. Any surface having steam on one side and hot gases on the other is superheating surface.

The American Society of Mechanical Engineers recommends that in measuring heating surface, the side next to the gases be used. Thus when estimating the heating surface of water-tube boilers, the *outside* areas of the tubes are measured, and for return-tubular or fire-box boilers the *inside* areas are measured.

The heat generated by the combustion of fuel permeates from the furnace through the heating surface to the water in the boiler. As the process of combustion proceeds, the heat liberated is immediately absorbed, partly by heat from the freshly added fuel, but mainly from the gaseous products of combustion. The absorption of heat by these substances causes a rise in their temperature and from these gases the heat is transmitted through the heating surfaces into the boiler water. This transmission of heat takes place in three distinct ways, each of which is governed by a definite law not applicable to the others.

Before the heat reaches the body of the boiler water, it changes its mode of travel at least twice. It is first imparted to the heating surface: (a) by *radiation* from the hot fuel bed, the furnace walls and the luminous flames, and (b) by convection from the hot moving gaseous products of combustion. Upon reaching the heating surfaces the heat changes its mode of transmission and passes through the soot, metal and scale to the inner surface, which is in contact with the water, purely by conduction. From the wet side of the heating surface the heat is carried into the boiler water mainly by convection.*

The water in the boiler can absorb only that heat called the "heat available for the boiler," which is above its own temperature. Heat below this temperature will not flow into the boiler and is, therefore, not available for use.

A commercial boiler absorbs only part of the available heat, which expressed as a percentage, is the true boiler effciency. This efficiency depends chiefly on the arrangement of the heating surfaces. Therefore, from point of economy in operation, the heating surface available and its arrangement should be carefully considered by the designer when selecting boiler equipment for a heating plant.

The true boiler efficiency, which is the only true measure of the boiler's ability to absorb heat, is expressed by the following equation:

$$\text{True boiler efficiency} = \frac{\text{Heat absorbed by boiler}}{\text{Heat available for boiler}}$$

The efficiencies ordinarily used in commercial boiler tests do not represent the true performance of the boiler under actual working conditions.

Boiler capacities as given in catalogues of manufacturers of heating boilers are based on the efficiencies obtained in the testing laboratories, and as stated are not representative of true conditions. In selecting a boiler for a heating plant, due allowance should be made to take care of this discrepancy by adding a factor of safety to compensate for the difference in laboratory and actual working conditions. This allowance, which may be called the safety factor to be added to the theoretical capacity, varies widely for the various types of boilers. Before determining the safety factor to be added to the commerical rating, the designer should carefully consider the type of boiler, the kind of fuel to be used, and the kind of attention the plant will receive, as all these bear on the performance and efficiency.

The necessity of providing an extra safety factor is recognized also by

* Bulletin 18, U. S. Bureau of Mines.

the heating trade and various trade associations that have established rules and regulations for the guidance of their members in determining boiler capacities.

The difficulty in obtaining the more desirable grades of coal has resulted in an increasing tendency to use coals which are more readily obtainable and lower in cost. The grates of the boilers should therefore be properly designed for the fuel which will most likely be used. Different authorities have a wide range of opinion as to the width of the air space that should be used between grate bar openings for a given grade of fuel.

Professor Gebhardt recommends an air space of ⅝ inch between the grate bars and bars ¾ inch wide for power boilers and for average bituminous coal.

For No. 3 buckwheat coal an air space of 3/16 inch and for No. 1 buckwheat 5/16 inch is recommended.

Grate areas are usually determined in proportion to the heating surface of the boiler, that is, for a given fuel, the grate surface and heating surface have a fixed ratio. For normal operation, a ratio of grate surface to heating surface of 1 to 35 to 45 develops the rated capacity of the boiler, while for fine coal or overload conditions, a ratio of 1 to 25 is desirable.

For return-tubular boilers and water-tube boilers, the following table shows the usual ratios of grate surface to heating surface and also the grate bar openings applying with these ratios when using soft coal fuel.

Table 9-1. Grate Surfaces for Soft Coals.

Coal	Grate Bar Openings		Ratio grate surface to heating surface.	
	Mine Run	Slack	Mine Run	Slack
Va., W. Va., Md., Pa.	½-in.	⅜-in.	1:55	1:50
Ohio, Ky., Tenn., Ala.	⅜ - ½	¼	1:50	1:45
Ill., Ind., Kan., Okla.	⅜ - ½	¼	1:45	1:40
Col. and Wyo.	⅜	¼	1:45	1:40

Determination of the amount of grate surface to be used under given conditions involves the available draft as well as the fuel to be used. The curves given in Figure 8-0, page 00, show how much draft is necessary for burning different coals at various rates of combustion.

The draft required to overcome resistances in the boiler is also given in Chapter 8, pages 00 and 00. These losses in the boiler and furnace must be deducted from the total available draft to determine the draft available for the fuel bed.

The capacity of the boiler and the B.t.u. to be developed being known, the number of pounds of coal to be burned can be readily computed. The total grate area required is found by dividing the total number of pounds of coal to be burned by the rate of combustion taken from Figure 8-0, Page 00.

Hand-fired, return tubular and water-tube boilers are readily operated at the rates of combustion in lb. of coal per sq. ft. of grate area given in Table 9-2.

Small boilers of the residence-heating type usually burn coal at rates ranging from 1 to 5 lb. per sq. ft. of grate surface per hr. and in larger heating

Table 9-2. Ratios of Combustion for Various Coals.

Anthracite	15 lb. per sq. ft. per hr.
Semi-Anthracite	16 " " " " " "
Semi-Bituminous	18 " " " " " "
Eastern Bituminous	20 " " " " " "
Western Bituminous	30 " " " " " "

boilers the ratio ranges from 4 to 12 lb. per sq. ft. of grate surface per hr.

These low rates of combustion are the result of demands for less frequent attention, in order that the man who fires the boiler may devote time to other work. In consequence, heating boilers are expected to do their work when fired once every hour or two or in residence heating, once in 6 to 8 hours, whereas power boilers are fired at regular intervals of 5 to 10 minutes.*

Another reason why heating boilers require different firing methods to burn bituminous coals successfully is that the space in the fire-box above the fuel bed is usually very much smaller than is the corresponding space in power boilers.* This space, known as the combustion chamber, is where the smoky gases driven off from the coal must become mixed with air and burn. The more rapidly the combustible gases are driven off from the coal, the larger must be the space necessary for burning them completely. The relatively small combustion space in heating boilers makes it important that the firing be done in a way to prevent the gases from being driven off too rapidly.†

The type of boiler to fit the given conditions most satisfactorily depends upon the physical conditions of the plant, as well as the type of heating system selected. The success of one depends upon the other. For this reason boiler selection is discussed also in Chapter 10, Selection of the Proper Type of Webster Heating System.

On account of the great variation of governing conditions, no attempt will be made here to discuss in detail the method of installation of the boilers or their connections.

Precautions should be taken in the design of the boiler plant to minimize bad effects from priming.

Liberal bleeder or drip connections from the boiler header, connecting directly to the return header, eliminate a great percentage of this trouble.

Priming in most cases is due to the presence of grease or oil in the boiler or to the presence in the water of certain alkalies which cause the water to foam or bubble, and be carried into the piping system by the steam. Before it can be expected to perform its functions uniformly, effectively and economically, a boiler must be thoroughly cleansed of oil, scale, dirt and other impurities. The priming of boilers is not confined to any particular type or make. The plant designer will safeguard the interest of the owner and himself as well, if he makes sure that bleeder connections are made to protect the boiler in case of priming and that his instructions about proper cleaning

* Technical Paper 180, U. S. Bureau of Mines.

† For further reference to the importance and effect of combustion space see Technical Papers 63, 80, and 137 of the U. S. Bureau of Mines.

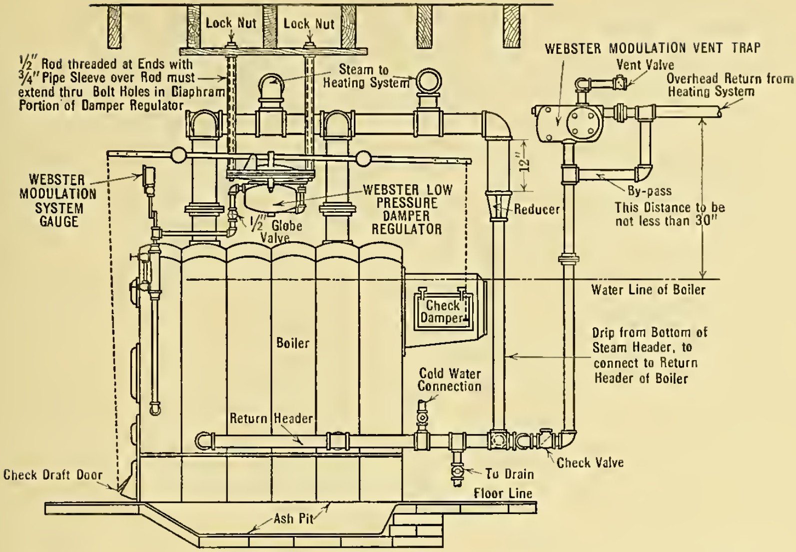


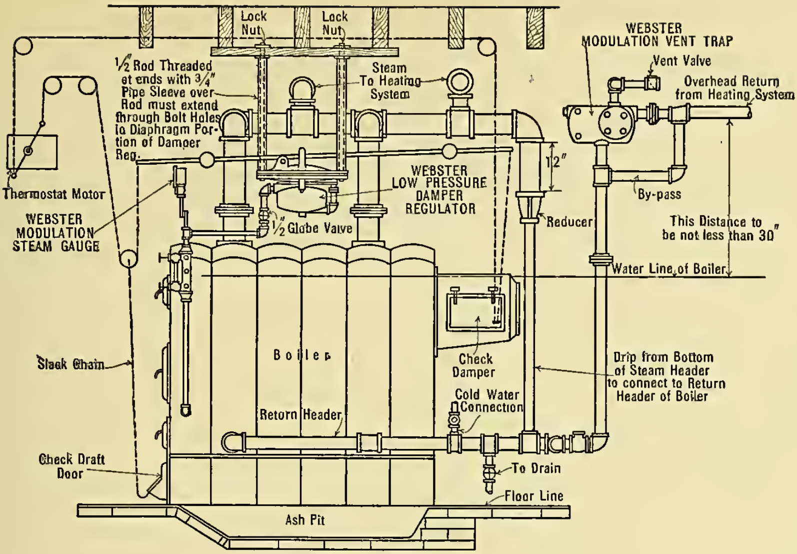
Fig. 9-1. Method of installing Webster Damper Regulator to a cast-iron sectional boiler.

out the boiler and the entire heating system are carried out in full by the heating contractor.

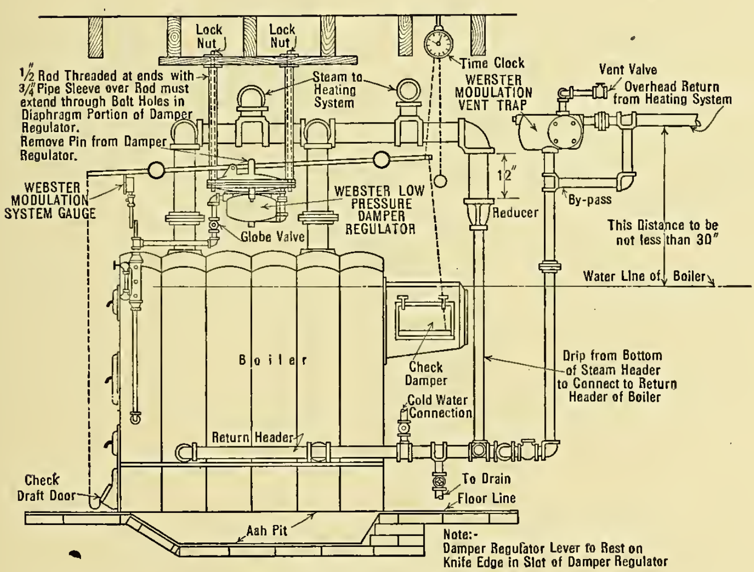
Damper control is an important feature of boiler operation. There are two classes of damper regulators, (1) those that move the damper for slight changes in the steam pressure, with a proportional movement due to the change in pressure and (2) those that operate the dampers between extreme positions when the steam pressure changes. The first is preferable from the standpoint of economical combustion.

As mentioned in Chapter 8, the fuel in a steam-boiler furnace is made to burn by passing through it a current of air, which supplies the necessary oxygen and carries away the products of combustion. A liberal supply of available air is therefore very important. Yet in many cases the space allotted to the boiler room is inside, small and without adequate air supply for combustion. Boiler rooms should be of ample size and depth to accommodate the boilers without crowding, and should have an abundant supply of air for both combustion and ventilation. The space in front of the boilers should be ample for convenience and comfort. A cramped boiler room is not only unsightly, but it also adds to the difficulty of taking care of the plant efficiently. The attendant, when firing, has to stand about 4½ or 5 feet from the front of the furnace and usually about 12 to 18 inches to the left of a straight line running through the centre of the furnace door. He should have ample room to swing his scoop from the coal pile into any part of the furnace.

Many a fireman is blamed for the poor economy shown by the plant he operates where the dissatisfaction should be at least partially charged to



With Thermostatic Control.



With Time Clock Control.

Fig. 9-2. Typical Applications of the Webster Damper Regulator to a Cast-iron Sectional Boiler in a Webster Modulation System.

the plant designer. It is difficult to keep skillful firemen in a small, poorly-kept boiler room.

The size and type of boiler to be specified and the evaporation the boiler will give are problems in which the advice of the boiler maker may well be considered. The boiler maker is usually quite willing to co-operate if provided with such data as the total radiation in square feet and pounds of condensation, total condensation of the steam and return lines in equivalent square feet of radiation and pounds, the quality and size of fuel available, the size and height of chimney and the firing period to be allowed.

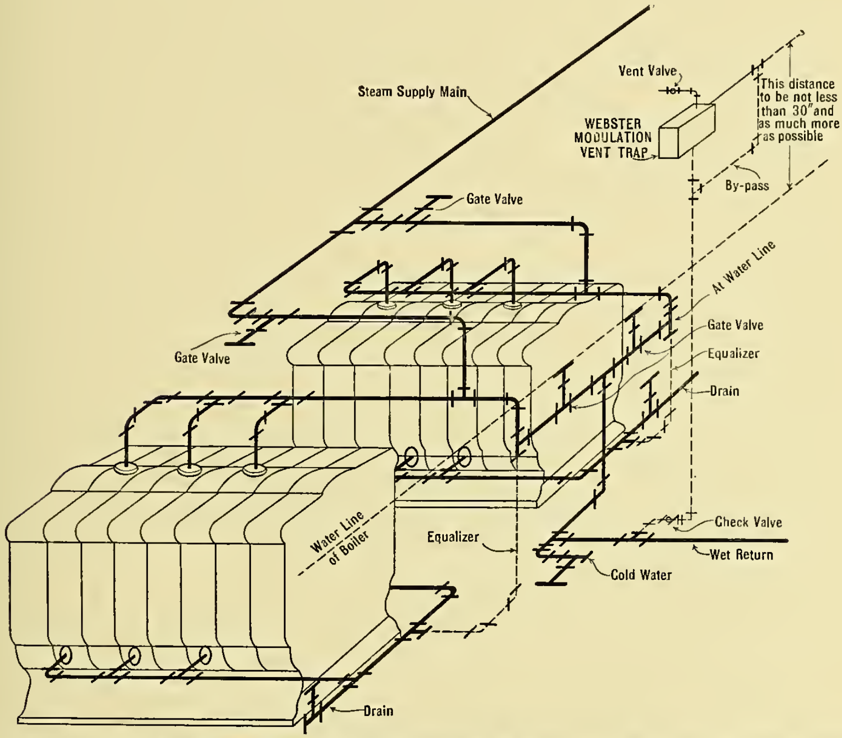


Fig. 9-3. Method of making connections to boilers operating in parallel. Check valve on vent discharge trap only.

CHAPTER X

Selection of the Proper Type of Webster Heating System

HAVING determined the heat requirements for a building or group of buildings, the decision as to the proper type of steam heating system involves selection from one of two main types: the Vacuum System and the Open-return or Modulation System.

A Vacuum System, broadly speaking, is one in which the steam enters the piping at or slightly above atmospheric pressure, and in which the air and water are mechanically removed at the other end of the system, at a pressure less than that of the atmosphere.

The Modulation System is one where the steam enters at a pressure slightly above atmosphere and the water and air of condensation flow by gravity to a point of disposal at which atmospheric or slightly lower pressure exists.

Very often the decision rests upon the source from which steam may be obtained with greatest economy. If power is to be developed in the building or nearby, usually the exhaust steam from the engines or turbines can be utilized for heating, and a Vacuum System is selected because it provides the more economical means of steam circulation and of returning the water of condensation to the boilers. Where power is never available or desirable as a means of circulation and where steam is generated at or obtained from a point of low pressure, a Modulation System is preferable. These two sets of conditions clearly indicate the proper selection of the type of heating system, but between them are many others where the desirability of either one or the other system is not so evident. Some conditions may even suggest a heating system which may be operated either on the vacuum or the modulation principle at will.

The flexibility of Webster Systems of Steam Heating for adaptation to the widely different operating conditions makes possible a correct Webster System for every type of building.

For instance, the Webster Vacuum System for a group of buildings spreading over considerable territory is quite different from that for a single compact structure, and a Webster Modulation System for a residence is often quite different from that for a hotel.

The character of the building or the purpose for which it is to be used, therefore, often determines the particular type of Webster System for a maximum of comfort and operating economy.

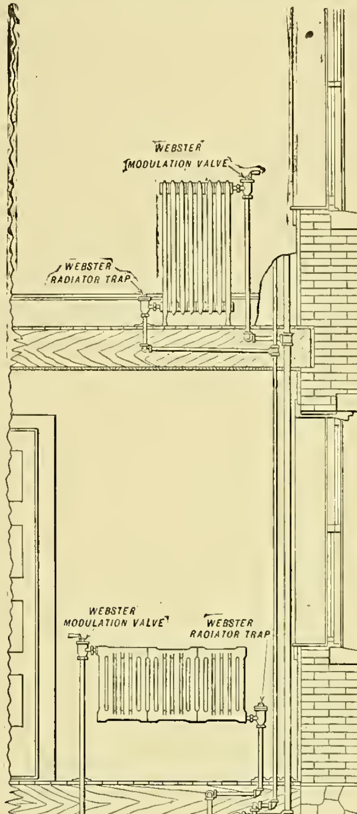
Grade may affect choice from among the various modifications of Webster Systems as the topography of the site may make the return of condensation too difficult except by the use of a vacuum or even by direct pumping.

The following pages describe the various types of Webster Heating Systems and the features which make each specially desirable for various classes of structures.

This is a highly efficient modern low-pressure system (see Figure 10-1) suitable for residences, store buildings, hotels and apartment houses, where live steam only is used for heating, either direct from heating boilers or from outside source.

The initial steam pressure is closely controlled by means of an extremely sensitive Webster Damper Regulator. Condensation is discharged automatically by gravity to the boilers or elsewhere through a Webster Modulation Vent Trap which operates without adjustment or attention even under fluctuating boiler pressures. The steam is admitted to each radiator through a Webster Modulation Valve which permits modulation of room temperature by simple hand manipulation. Condensation is discharged and air vented from each radiator through a Webster Return Trap, which maintains full heating efficiency of the radiation and eliminates the annoyances, difficulties and noises common to ordinary gravity steam heating systems.

Condensation and air from each radiator flow by gravity through a system of return risers and mains into the Webster Modulation Vent Trap, where the air is automatically vented, permitting the system under favorable boiler conditions to operate for long periods under partial vacuum, or "vapor," but also due to flexibility of



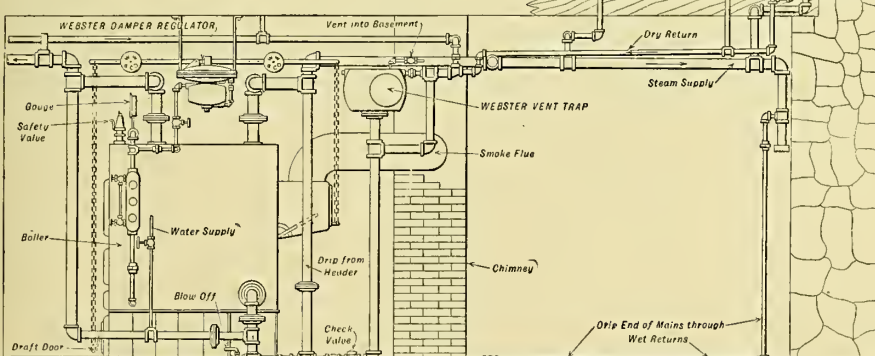


Fig. 10-1. Typical Arrangement of the Webster Modulation System of Steam Heating

the system permitting higher pressures to be carried in severe weather when a maximum amount of heat is required.

In addition to its saving of fuel and the feature of convenient independent temperature control for each room at the will of the occupant, the Webster Modulation System has a special advantage in its simplicity of operation. No expert attendance is required at any time and in most cases the attention which must be given to the operation of the system is merely the firing of the boiler at long and infrequent intervals.

The response of the Webster Modulation System to demands for changes in the rate of heating is almost immediate. A cool room or building can be heated quickly upon demand or if too hot, the temperature can be as easily and as quickly reduced.

In its simplest form, the Webster Modulation System consists of a cast-iron boiler, with its appurtenances, a system of supply piping, radiating surfaces with a Webster Modulation Valve at the inlet and a Webster Return Valve at the outlet of each, a system of return piping and a Modulation Vent Trap for removing air and for returning water to the boiler, all as previously described.

This is the Modulation System which would ordinarily be selected for a residence and usually for the smaller apartment, store, office and public building and for hotels and churches of moderate size.

Residences: The Modulation System with cast-iron boilers is most suitable from every standpoint for a residence, whether a thirty-room house or a five-room bungalow.

The designer of heating plants for residences, is in most cases confronted by two conditions which decide the type of boiler he shall use, *first*, smallness of the boiler room and, *second*, the low head-room in the basement. Both suggest the use of the cast-iron type boiler because of its compactness and low water line.

The prospective owner is particularly interested in the attention necessary for operation. Whether he attends to the heating system himself or employs someone else, he desires a plant requiring minimum attendance.

Except for the periodical feeding of coal and removal of ashes, the attention required by a Modulation System is negligible.

Apartment Buildings: The Modulation System is particularly adaptable here unless the building spreads over too much ground or the open-return-line system could not be properly graded without too much complication.

Apartment buildings are erected for the revenue which they will bring to their owners, and a heating plant which can be operated with greatest fuel saving and least janitor service is the best-paying proposition.

Control of the amount of steam admitted into each radiator independently gives the occupant of each room or apartment a convenient means of temperature regulation.

The small amount of attention required by a Modulation System gives the janitor of the building more time for his other duties.

Store Buildings: Store buildings of the smaller size, where no mechanical system of heating and ventilation need be provided, and where an open-

return-line system can be applied, take the same type of heating system as described for residences and apartments.

Office Buildings: Smaller and medium-sized buildings for office and other commercial purposes, telephone exchange buildings, etc., where the steam requirement is for heating purposes only and where the architectural features permit an open-return-line gravity heating system, can also utilize the Webster Modulation System to advantage.

The basement rooms as a rule are heated by radiating surfaces placed overhead, so that the return lines can be arranged for a gravity return to the Modulation Vent Trap and then to the boiler, eliminating all mechanical return-handling devices and their complications.

Inasmuch as any one who is able to handle a shovel can fire and operate the Webster Modulation System, after being given one lesson in its care and maintenance, this system will materially aid the owner in keeping his labor costs low.

Hotels: Big city hotels, being usually equipped with modern and complete mechanical equipment from which exhaust steam is available, can more profitably use the Vacuum System. However, great strides have been made in recent years by the country towns to provide more convenient hotel accommodations for the traveling public, and a great number of high-class hotel buildings which have proved excellent advertisements for their towns have come into existence.

The owner of such a hotel, while not in a position to equip with the many refinements of big metropolitan hotels, is anxious to have his guests provided with clean, comfortable and properly heated rooms, and is willing to pay the price for an efficient and economical heating plant.

The Webster Modulation System of Heating is particularly advantageous in this type of building, giving all that can be asked in heating effect, and enabling the janitor or so-called engineer, who is in many cases also the porter, bell boy and general utility man, to take care of his many other duties.

Churches: Churches where no mechanical ventilation is to be provided, and where all radiation can be placed high enough above the water line for gravity return of the water of condensation to the boiler, are properly equipped with the Modulation System.

Heating plants in churches as a rule do not receive the best of attention, and therefore the simpler the installation, the more satisfactory the service. The operation of the Modulation System in draining the condensation back to the boiler entirely by gravity also avoids the noise that usually accompanies the action of mechanical devices where the latter are employed for handling condensation.

Heating systems in churches are usually operated intermittently, and for this reason must be laid out with due precaution to avoid freezing—a condition which is met in the Modulation System by eliminating the use of wet returns.

Public Buildings: In this classification may be included schools, court houses, post-offices, libraries, etc., which are not to be provided with mechanical systems of ventilation, but where the ventilating systems are to be of

Fig. 10-2. The entry of a modern apartment building showing heat outlets in the side walls.

the indirect or direct-indirect gravity ventilation type. Such buildings have, as a rule, no other mechanical equipment besides the heating and ventilating plant. The Webster Modulation System with an open return line is recommended for these structures.

This class of buildings usually has intelligent men in charge of the operation and maintenance of the heating plant, so that the application of mechanical devices for handling the water of condensation would not be as objectionable as would be the case with the other types of buildings.

There is no advantage, however, in providing these mechanical devices. Their use should therefore be limited to cases where an open-line gravity return to the boiler does not work out satisfactorily.

Where public buildings, such as schools, are heated intermittently, proper provisions should be made to guard against freezing. Bottom supply connections should be used for all direct-indirect radiation.

Wet returns should also be eliminated as much as possible in such buildings, and where their elimination is not possible they should be properly protected against freezing.

With Fire Box Boilers: The foregoing discussion of the Webster Modulation System has favored cast-iron type boilers for buildings of moderate size.

Fire-box boilers can be used to advantage in buildings of somewhat larger proportions.

Rules and regulations in force in some communities specify maximum permissible size limits for cast-iron boilers and require that steel boilers of the fire-box type shall be used where the load requirements exceed the specified limit.

From a strictly engineering standpoint, good practice bases the limit of size of the cast-iron boiler upon the grate surface. The installation of cast-iron boilers which have a grate over 72 inches in length should not be permitted, as a grate over 6 feet in length makes difficulties in firing.

Street Steam: In localities where street steam is available with uninterrupted service guaranteed for the entire heating system, and where the rate does not exceed that at which steam can be generated in an individual plant, the installation of the Modulation System with street steam supply provides ideal heating for almost any type of building.

The reduced first cost of the heating plant and its installation and the fact that such a plant requires almost no operating attention, make this arrangement very attractive from the owner's standpoint.

The Service Company supplying the building provides the service steam line to which the heating contractor can make his connection. The water of condensation is discharged to the sewer through a meter in the return line, except where a flat rate per square foot of radiation is charged, in which case no meters are used.

This type of heating system can be installed in almost any type or size of building, except where too extensive area prevents satisfactory arrangement of the return line for gravity open-return circulation.

The buildings for which this type of heating plant is recommended require high-pressure steam for various purposes. Among these buildings are hospitals, Y. M. C. A. buildings, restuarants, etc. The steam supply comes from one or more high-pressure boilers. That for heating is reduced to suitable pressure by means of a pressure-reducing valve and is circulated through the heating system by gravity—that is, the water of condensation is returned to the vented receiver of an automatic steam-operated pump and receiver. The receiver has a float at its water level, the rise and fall of which controls the steam through an automatic valve to the pump.

The pump is operated by high-pressure steam, and its exhaust, after extraction of the oil by an oil separator, is utilized in the heating system.

Hospitals: The modern hospital has considerable steam-using equipment, such as sterilizers, blanket warmers, steam cookers, dish-washing machines, laundry machinery, etc., which requires steam at pressures ranging from 30 to 90 lb.

Reduction to the intermediate pressures and to the low pressure for heating purposes is effected by means of pressure-reducing valves.

The operating man in a hospital plant is in most cases a capable licensed engineer, but his duties are many and this fact should not be overlooked in the design of the plant. The heating system should combine simplicity in design, fuel efficiency, noiseless operation and flexibility of heating—all of which requirements are fully met by the Webster Modulation System.

Y. M. C. A. Buildings: These buildings resemble hotels in many respects, as, in addition to the recreational features, hotel accommodations are provided for members. Restaurant and cafeteria service are maintained, as well as swimming pools, Turkish baths, etc. in connection with which there is a demand for high-pressure steam in addition to the low-pressure steam needed for heating. For this reason all of the condensation cannot be returned directly to the boilers.

The heating system should be of a type which permits regulation of the supply of steam to the bedrooms, according to whether they are occupied or empty. The graduated control system of steam supply to the radiators by means of Webster Modulation Valve is therefore the most logical system to adopt. A steam-operated pump and receiver takes care of the returns from all the steam-using equipment and also from the Webster Modulation System.

Hotels and Restaurants: Assuming that the kitchen equipment is to be supplied with steam and that the mechanical equipment of the heating plant must be simple, a consideration of the various types of heating systems will readily suggest the Webster Modulation System with an open-return line to a pump and receiver, using reduced-pressure steam for heating and taking steam directly from the boiler at the pressure required for the other equipment.

This type of heating system in buildings of this class is limited only by the area covered by the structure. Buildings extending over considerable ground may require special provision for handling the return.

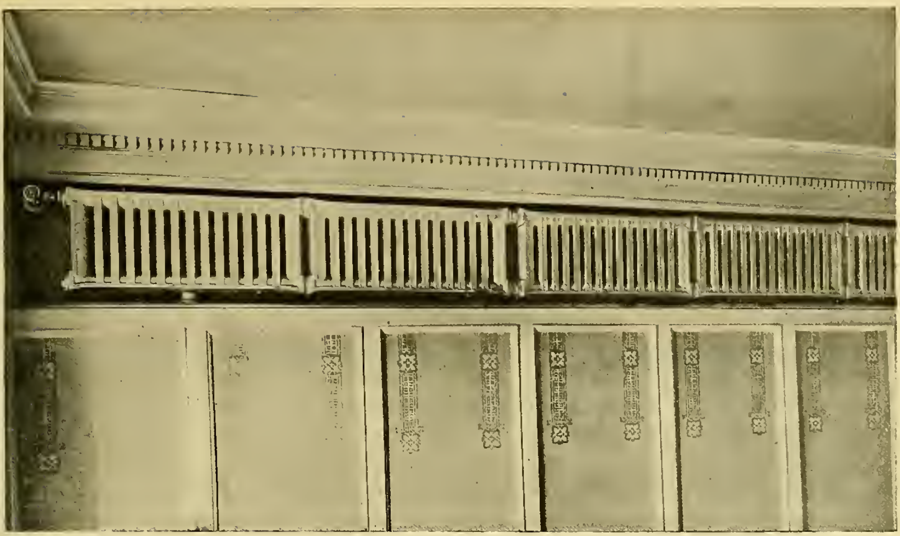


Fig. 10-3. Arrangement of cast-iron wall radiation in cove of ceiling in a grill room.

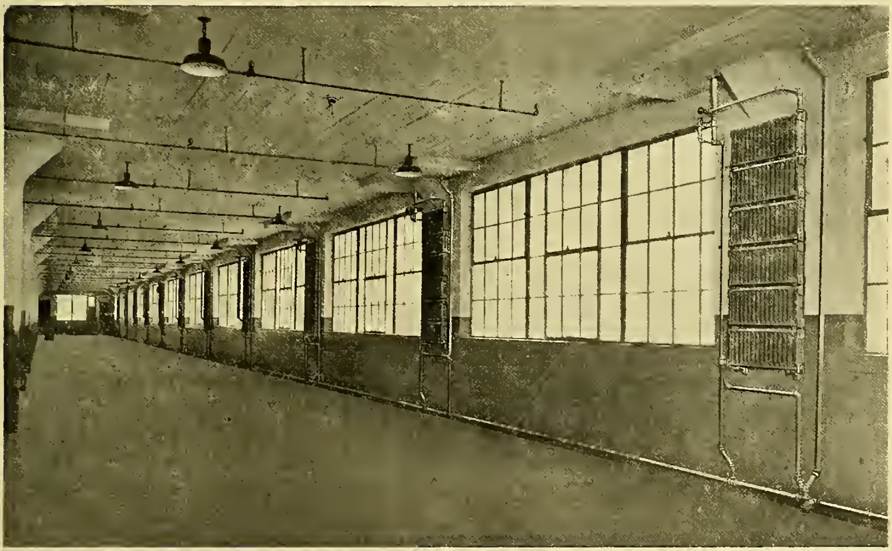


Fig. 10-4. Cast-iron wall radiation in garage. The radiation is placed to avoid being damaged by cars and to prevent injury to tires from beat.

Schools in Rural Districts: Rural school buildings seldom have electricity available for operating the fans of mechanical systems of ventilation, or for operating pumps supplying water from artesian wells.

A steam engine is necessary to operate the ventilating fan and a steam-operated pump is required for the water supply system.

The boilers in such cases are usually operated at about 30 lb. and the steam for heating purposes is reduced to about 1 lb. pressure. The exhaust steam from the engine and pump is utilized in the heating system, after extraction of the oil by passing the steam through an oil separator. A by-pass connection is provided in the exhaust pipe, which extends into the atmosphere, so that at such times as the heating system is not in operation the exhaust steam may be discharged into the air.

The Webster Modulation System for use under these conditions is designed with an open-return line and with a steam-operated pump and receiver which discharges the water of condensation back into the boiler.

The Webster Open Return Line System (without Modulation Valves): This type of heating system is identical in principle and method of operation with the Webster Modulation System previously described, except that standard type radiator valves are used in the supply connections to the radiators in place of Webster Modulation Valves. This system is recommended where an open-return-line system, either returning directly to the boilers or to a pump and receiver, can be used and where only large, open rooms are to be heated, as in department stores, loft buildings, warehouses, etc. Buildings of this type, with heated spaces usually containing a number of radiators, do not require Webster Modulation Valves in the supply connections to the radiators, inasmuch as a fair degree of temperature regulation may be obtained by shutting off one or more radiator units. A heating plant of this type costs less than a complete Webster Modulation System, and may be selected where a reduced first cost of installation is necessary.

If the building conditions permit the water of condensation to be returned to the boiler by gravity, the heating plant is very simple and requires little attention outside of boiler firing.

Conditions are sometimes met, especially in department stores, where the condensation cannot be returned to the boiler by gravity, in which case the installation of an automatic condensation pump becomes necessary.

Loft buildings used for light manufacturing purposes, requiring live steam for various uses, such, for instance, as in tailoring establishments and plating works, demand either a separate high-pressure boiler to supply steam for the process work, or an entire boiler plant operated at high pressure, with reduced pressure for the heating system supply.

The Webster Modulation System with Vacuum Pump Relay: This is a combination of the Webster Modulation and Vacuum Systems which may be operated as an open-return-line system, returning the water of condensation to the boiler by gravity, or as a Vacuum System, with electrically operated vacuum pump to remove the water of condensation and air from the heating system and to discharge the water into the

boiler. It is particularly advantageous in schools and theatres having mechanical systems of ventilation which are in operation only part of the time.

In a school building, for instance, the ventilating system is usually put into operation at about 8 o'clock in the morning and is shut down at 4 o'clock in the afternoon. While the ventilating system is in use, the plant operates as a Vacuum System. As soon as the ventilating system is shut down the vacuum pump may be stopped and the direct heating system is then operated as an open-return-line system discharging the returns through a modulation vent trap to the boiler by gravity. During the night the heating plant requires almost no attention.

As the vacuum pump is not operated 24 hours per day, the heating of a theatre may be accomplished in very much the same manner. In this instance, however, the ventilating system is in use in the afternoons and evenings, during which the plant is operated as a Vacuum System. After the close of the night performance, the change is made to operation as a Modulation System.

This type of system must always be so designed that the pressure drop through the entire system will not be less than the net static head between the modulation vent trap and the water line of the boilers.

The Webster Vacuum System of Steam Heating

Briefly, the Webster Vacuum System is a method of circulating exhaust or low-pressure live steam, or a combination of the two, with minimum initial or back pressure, and with entire freedom from water-hammer, air pockets, leaky air valves, and all of the other annoyances which are common with ordinary steam heating systems.

A partial vacuum is mechanically created and maintained by means of exhausting apparatus, which consists of steam-driven or power-driven vacuum pumps of suitable design and capacity.

A typical arrangement of the Webster Vacuum System is illustrated in Figure 10-3.

The exhaust steam from the engine passes through a Webster Oil Separator, dripped through Webster Grease Trap, thence to the heating system. A pressure-reducing valve with by-pass is provided to make up any deficiency in the volume of exhaust steam or to provide live steam for heating when the main engine is shut down.

The supply main is dripped as it enters the building, through a Webster Heavy-duty Trap, protected by a Webster Dirt Strainer. The steam supply risers in larger buildings may have to be dripped through additional Webster Traps of the proper size and type.

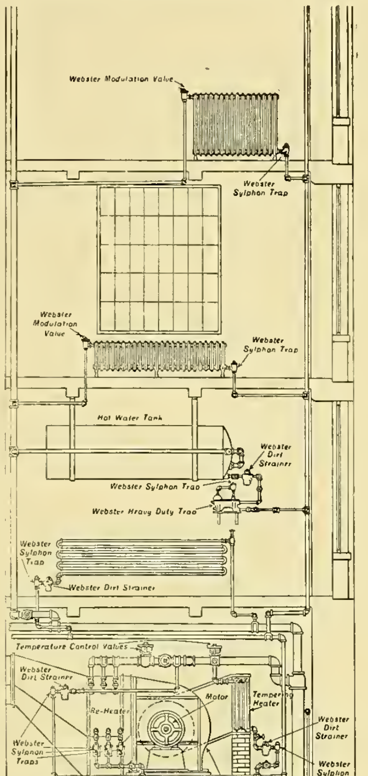
Steam is supplied to the various types of heating units through Webster Modulation Valves, although the system will work smoothly with automatic temperature control. Some of the radiator units are shown with ordinary supply valves. Each heating unit is drained through a Webster Return Trap into the return risers; and the larger heating coils are protected by Webster Dirt Strainers.

Steam is also supplied to tempering and reheating coils which are also drained at the return ends of each group through Webster Traps protected by Webster Dirt Strainers.

The returns all join and lead to a vacuum pump, protected by a Webster Suction Strainer. The steam supply to the pump is automatically controlled by a Webster Vacuum-pump Governor. Compound gauges mounted on a slate board and having connections to the heating main and the vacuum return line give an indication as to the internal steam conditions.

The vacuum pump discharges through a Webster Air-separating Tank to a Webster Feed-water Heater, usually of the Webster Preference Cut-out Type, with oil separator constructed to direct a sufficient quantity of exhaust steam toward the heater. The balance of the exhaust steam is available for the heating system. Any excess of exhaust steam over feed-water and heating needs escapes through a back-pressure valve. The heater may thus be cut out of service while the oil separator remains in use.

The ventilation scheme provides a supply of purified, humidified and heated fresh air for those rooms which it serves. The air is partially heated in passing over the tempering heater, and is drawn by the fan through the reheater into the main air supply duct. The supply of steam to the



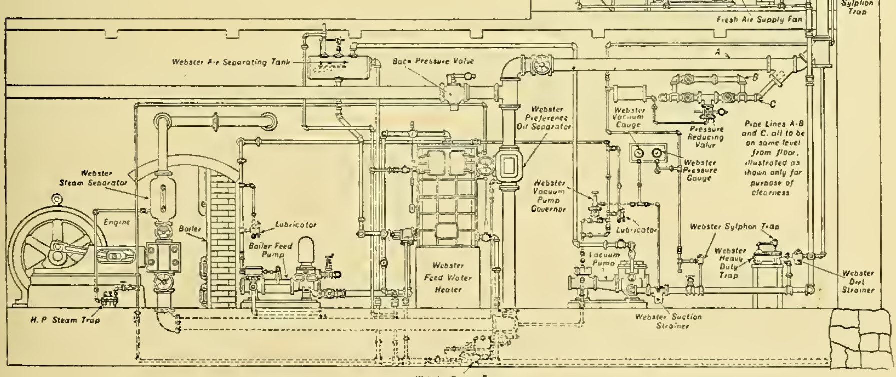


Fig. 10-2. Typical Webster Vacuum System of Steam Heating, showing arrangement of the power plant apparatus, mains, radiation surface and various accessories.

tempering heater and reheater coils is automatically governed by the temperature control valves.

The typical illustration, Figure 10-2, represents a Webster Vacuum System in a plant where steam is generated in high-pressure boilers for power purposes and where the exhaust steam is used for heating. It is not intended to portray a standard arrangement for all conditions.

A Vacuum System with low-pressure boilers requires an electrically operated vacuum pump for returning the water of condensation from the heating system to the boilers. This arrangement should be selected where a gravity return to the boiler cannot be arranged, and where the plant is of such a nature that an open-return-line system with an electrically operated condensation pump is not practical. It is also assumed that mechanical equipment, aside from that of the heating system, does not require high-pressure steam, so that the installation of low-pressure boilers is logical.

Electrically operated vacuum pumps, especially of the rotary type, have reached a high state of perfection, so that the attention required by the pump is reduced to a minimum.

The principal advantages from a properly installed and operated Webster Vacuum System are as follows:

1. The circulation of steam is quick, positive and uniform. All surfaces are heated in a relatively short space of time after steam is turned into the system.

2. Water-hammer in the piping is unheard of, due to the continuous relief of air and the positive removal of the products of condensation.

3. The radiators are maintained at 100 per cent heating efficiency due to the complete removal of air and water. The absence of air-valves on the radiator eliminates one of the most annoying features of many steam-heating systems.

4. Saving in operating cost is accomplished practically by eliminating back pressure upon steam engines. This either saves directly in fuel cost or permits the engine to do more work at the same expenditure of fuel.

5. Saving is effected through the ability, during mild weather, when the demands for heating are slight, to distribute a relatively small volume of steam throughout the system as needed, with a pressure at, or even slightly below, that of the atmosphere. This small volume of steam can be thoroughly distributed only where the Vacuum System principle is employed. In this country mild weather constitutes about 75 per cent of each heating season, "moderately" cold weather about 20 per cent and only 5 per cent can be classified as "severely" cold.

6. Saving of fuel results from utilizing the condensation and its contained heat as part of the boiler feed. The returns, being practically distilled water, are excellent for the boilers, as Webster Oil Separators remove the cylinder lubricant before the exhaust steam goes into the heating system. In some cases little new or make-up boiler water is needed.

To these advantages should be added comfort and convenience. More and better work is obtained from the occupants of properly heated build-

ings, and this too adds to the general feeling of satisfaction from the Webster System. Some of the savings are difficult to measure in actual dollars and cents, but they are nevertheless substantial.

Buildings for which the Webster Vacuum System is selected are usually of large proportions. Consequently the first and installation costs are less than for an open-return-line system on account of using smaller pipe sizes for both the supply and return lines.

Where lifts are necessary in the returns the Vacuum System is the only solution.

From the operating standpoint, the Vacuum System with an electrically operated vacuum pump of the rotary type is as simple as the open-return-line system with a condensation pump.

For a large building or a group of buildings the Vacuum System with low-pressure boilers is the logical choice, and it is particularly recommended for high buildings of any description, buildings occupying considerable area, buildings of any description in which lifts in the returns are necessary, and groups of buildings to be heated from a common boiler plant.

High Office Buildings: The pumps required for elevators and for water supply are usually steam-driven, even if the building does not have its own power plant. The exhaust steam from these pumps can and should be utilized in the Webster Vacuum System.

The Modern First Class Hotel: Most buildings in this class are provided with high-pressure steam boiler plants, either for generating their own electric power, or in case the electric current is purchased, for operating the pumping equipment, and furnishing the steam for kitchen and laundry purposes. The Vacuum System with steam-operated vacuum pumps is the proper type for heating such buildings.

Hospitals: As already mentioned in connection with the Modulation System (See page), these institutions require high-pressure steam for kitchen, laundry and sterilizing purposes, making it necessary to have a high-pressure boiler plant. Many of the larger hospitals have their own electric power plants, and use steam for operating the refrigerating machinery and pumping equipment. The heating system in such institutions should utilize the available exhaust steam, as is best accomplished in a Webster Vacuum System.

Manufacturing Plants: The selection of a vacuum system for a manufacturing plant is usually justified where high-pressure steam is needed for process work, and the conditions are such that cheap electric power is not available. In such cases high-pressure boilers, and the necessary electrical machinery for generating current are installed and exhaust steam is utilized for heating.

The Webster Conserving System: The Webster Conserving System (Figure 10-3) is specially designed for heating where the boilers are to be operated by unlicensed engineers, and where there are steam requirements for other than warming purposes.

Laws of various states prohibit steam pressures greater than 10 lb. per sq. in. in boiler plants which are in charge of unlicensed engineers. These laws have led to attempts, in plants where steam at from 5 to 10 lb. was

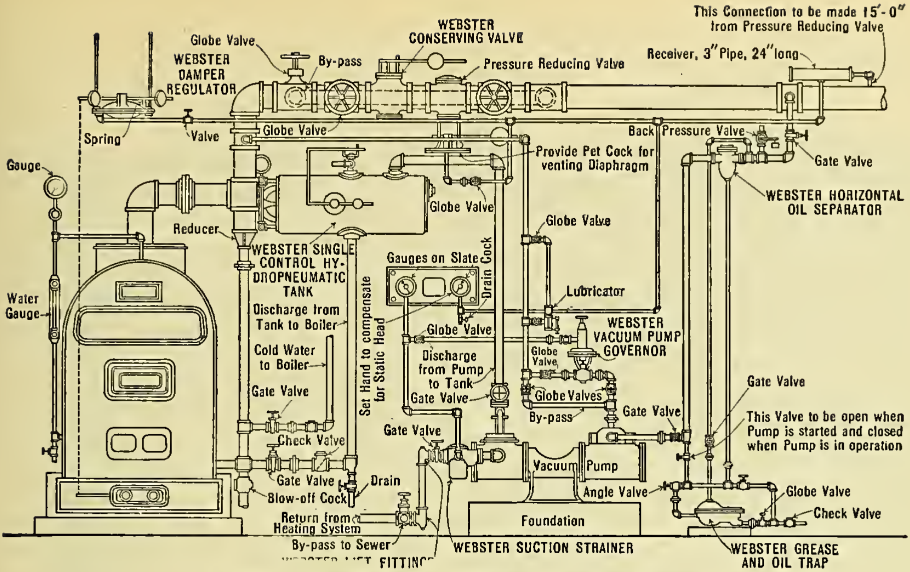


Fig. 10-3. Typical layout of a Webster Conserving System.

necessary for process work, to operate vacuum pumps by low-pressure steam, the pumps discharging the condensation through a receiver into the boilers.

Unfortunately it has proven almost impossible to control these low boiler pressures and the pump operation has been erratic. When the boiler pressure dropped and the pump stopped or failed to work properly, the condensation was not returned to the boilers. The water level was often lowered to a dangerous point and in many instances serious damage resulted to the boilers.

These difficulties are entirely overcome in the Webster Conserving System, the main feature of which is the Webster Conserving Valve, designed to prevent the admission of steam into the heating system until a certain predetermined steam pressure is reached. The pressure at which the valve will open is slightly above that required to operate the vacuum pump, so that the pump starts to operate before any steam is admitted into the heating system. If the boiler pressure drops, due to irregular firing or to other causes, so that the heating system receives no steam, or if the steam is purposely cut off from the heating system, the pump, taking its steam from the boiler header directly, continues to operate and thus insures the return of all condensation and the avoidance of damage to the boiler which might otherwise occur.

A pressure-reducing valve is installed on the low-pressure side of the conserving valve to prevent pressure from building up on the heating main beyond any desired point.

The exhaust steam from the vacuum pump is utilized in the heating system after the oil is extracted by means of a Webster Oil Separator. There is, therefore, practically no cost for power for steam circulation.

An additional feature of economy lies in the fact that the use of only one pump, acting as both vacuum pump and boiler-feed pump, minimizes the attention required for operating the system. The use of a single pump,

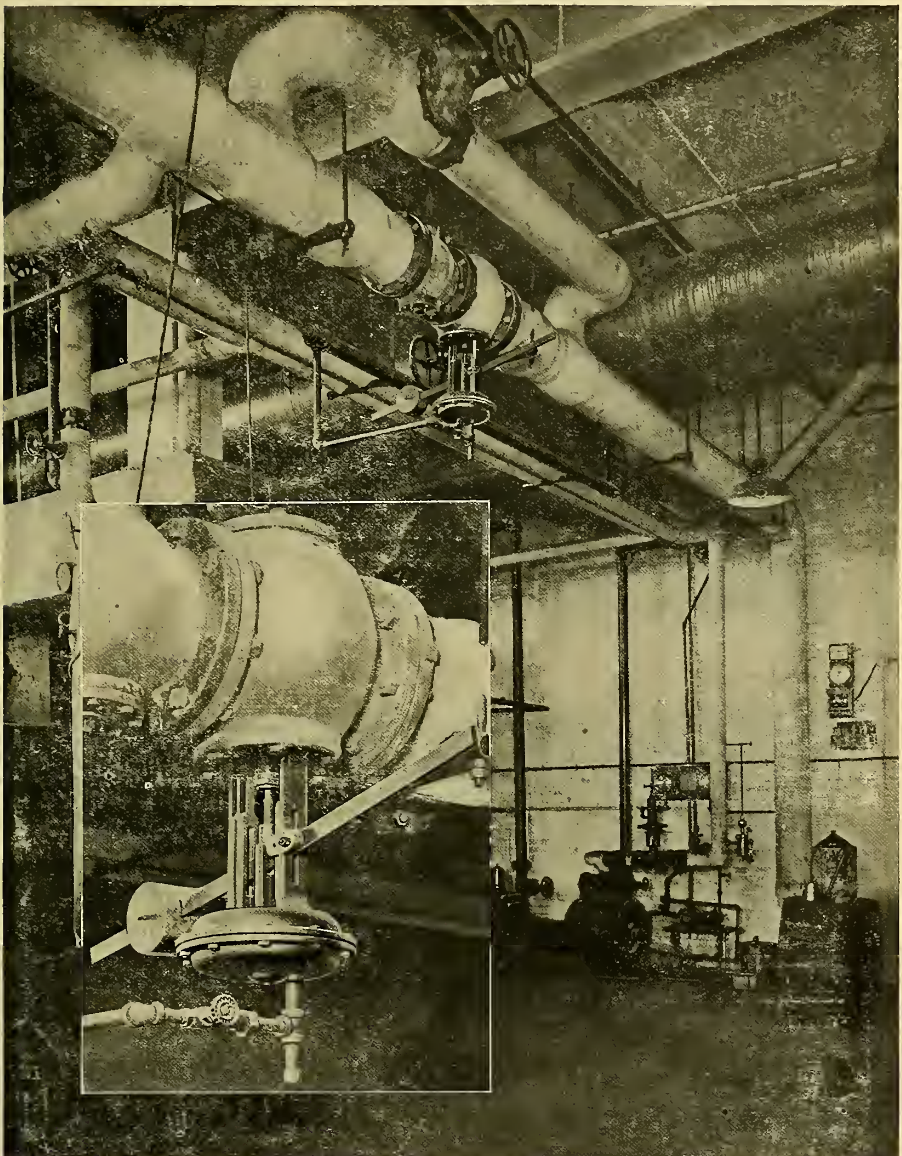


Fig. 10-4. Typical installation and close-up of the Webster Conserving Valve

Fig. 10-6. Cast-iron wall radiation arranged under the saw tooth of a factory.

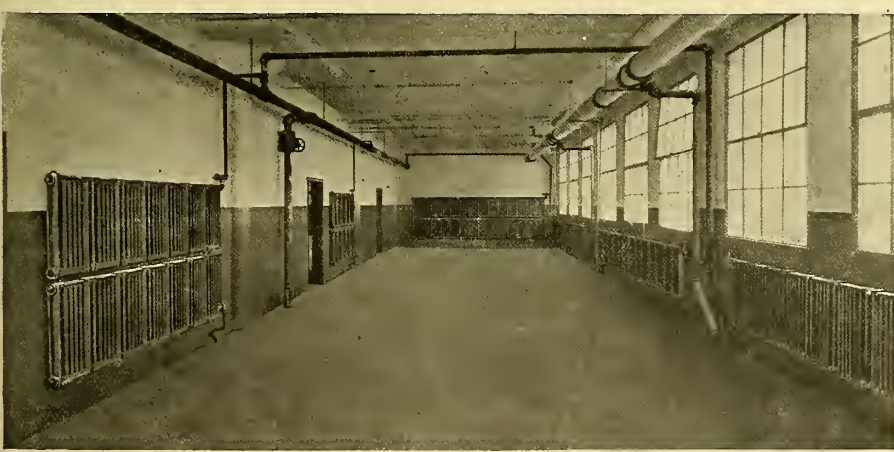


Fig. 10-7. Arrangement of cast-iron wall radiation on side walls of a factory building.

however, is recommended only where steam pressure maintained on the boilers is within 15 lb. per sq. in. For higher boiler pressures, the excessive pump capacity necessary to compensate for the high back pressure on the pump discharge makes a separate boiler-feed pump desirable.

The application of the Conserving System should be limited to plants where the steam pressure on the boiler ranges from 5 to 15 lb.

Application of the Conserving System to Large Industrial Plants: A study of steam engine performance, where the engine exhausts into the atmosphere or into the heating system against a back pressure slightly above that of the atmosphere, shows that engines working under such conditions actually convert only 5 to 10 per cent of the heat supplied to them into mechanical energy. The remaining 90 per cent of the heat originally supplied to the steam entering the engine is retained in the exhaust steam.

In some plants, power and heating loads are nicely balanced so that all the exhaust steam available from power units can be utilized for process work or heating purposes, in which event the 90 per cent of heat energy remaining in the exhaust steam is put to useful work. In such cases the engine may be considered as a pressure-reducing valve which reduces the pressure from that carried on the boilers to that required for heating and process purposes.

There are numerous industrial plants where the power load is greatly in excess of the heating load, so that the quantity of exhaust steam available is greatly in excess of that actually required. The surplus exhaust steam with its heat units must then be wasted.

Where these conditions exist the engines are often operated condensing instead of non-condensing, so that exhaust steam from the auxiliary machinery only is available. In most instances the quantity is not sufficient to supply the heating load, and the deficiency is made up by live steam supplied from the boiler through a pressure-reducing valve.

The work done by the pressure-reducing valve in reducing the steam from boiler pressure to that required in the heating system is converted into superheat on the low-pressure side of the valve. This work represents about 10 per cent of the total heat energy supplied to the steam. If this 10 per cent of heat energy can be utilized by conversion into mechanical energy, nearly ideal conditions will be approached.

Various attempts have been made in the past to improve the economy of power and heating plants by endeavoring to utilize the exhaust steam from the receivers of compound engines. This exhaust is bled into the heating system and the deficiency made up by admitting live steam into the receiver through a pressure-reducing valve. In determining the advisability of this form of application, the effect of the relations between heating and power load and the relative proportion of the cylinders so vitally affects the economy that in each instance special consideration has to be given to all elements entering.

The Webster Conserving System can be applied to this problem. In the same manner that the conserving valve is applied to conserve the pressure on the boiler by preventing the escape of its steam until a certain pre-

determined pressure is obtained, it can be applied to the receiver of a compound engine, opening and admitting steam at receiver pressure into the heating system, when the pressure on the receiver exceeds that which is necessary for the proper operation of the low-pressure cylinder, and closing when the receiver pressure drops below the point for which the conserving valve is set.

The quantity of steam taken from the receiver is made up by changing the cut-off on the high-pressure cylinder so that the high-pressure side acts as a pressure-reducing valve for the steam required for heating purposes. In expanding from boiler pressure to the receiver pressure, the heat energy given up in the expansion is converted into useful mechanical energy.

By means of the Webster Conserving System many existing power and heating plants may be brought to efficiency where they are otherwise wasteful of steam.

The Webster Hylo System: Where a number of buildings must be heated from a detached central plant, or where a building covers considerable ground, the source of steam supply and of vacuum cannot always be located to make a well-balanced system.

The largest building in the group may, for various reasons, be farthest from the source of supply, and may also be the lowest point in the system of return piping, thus making it doubly difficult to secure perfect heating and easy return of condensation. Nearby points may be favored with unnecessary "pressure difference."

Attempts have been made to solve this problem by running the supply and return mains in reverse direction, so that the point of highest pressure is the point of lowest vacuum and inversely, thus maintaining, in some degree, the same differential between supply and return pressures.

Where the largest building is at a low point away from the source of supply, it is obviously impracticable to solve the problem in this way. Furthermore, such a plan does not allow for extensions to or expansion of the plant, unless the new buildings can be located to suit the piping scheme irrespective of the manufacturing need.

This problem has been solved with unqualified success by Webster Hylo Vacuum Controlling Sets, which are installed at certain points in the return line to restrict the vacuum to just the amount necessary for proper circulation and drainage at nearby points where high vacuum is not needed. The high vacuum is carried to extreme or low points where high vacuum is required. The result is a well-balanced system with perfect circulation in all parts.

The operation of the vacuum pump is also improved to a marked degree as the degree of initial vacuum is reduced, making it unnecessary to use or waste cold water to condense the vapors arising from the hot water returned under high vacuum. Sometimes smaller pumps may be used, or the pumps may be operated at slower speed, with less wear and tear.

The Webster Hylo Sets consist of a Webster Hylo Trap, a Webster Hylo Vacuum Controller, Webster Hylo Vacuum Gauges, and when needed, a Webster Lift Fitting.

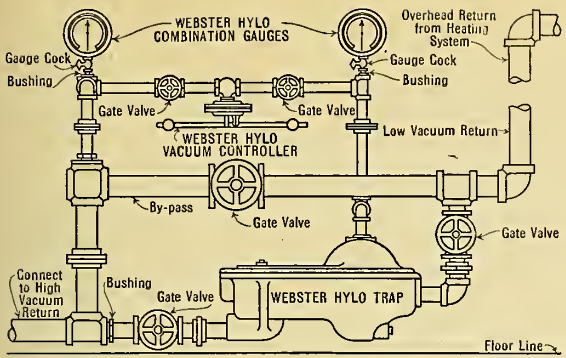


Fig. 10-5. Connections around Webster Hylo System equipment where the low-vacuum return main drops from overhead and discharges through a Webster Hylo Trap to the high vacuum return main.

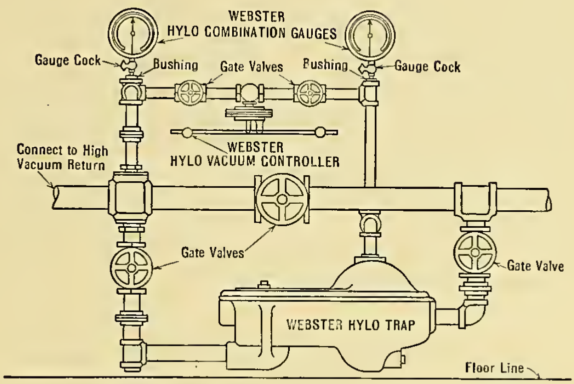


Fig. 10-6. Typical installation of Webster Hylo Trap, Controller and Gauges where high and low-vacuum returns are on the same level.

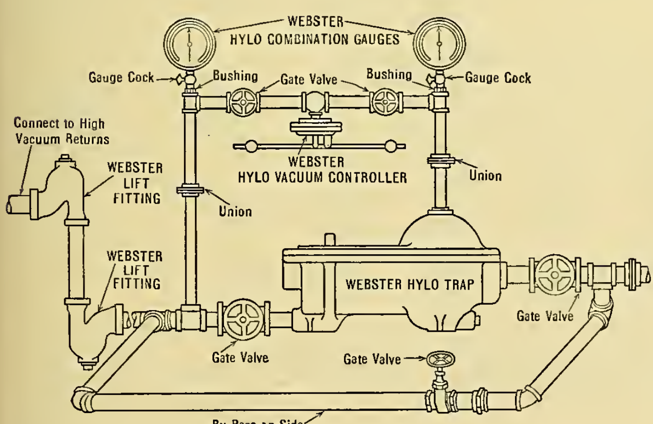


Fig. 10-7. Arrangement of the Webster Hylo Controller, Trap and Gauges where the low-vacuum return is lifted to the high-vacuum return.

CHAPTER XI

Pipe Sizes for Webster Systems

A. Flow of Steam Through Pipes: Flow of steam through piping is caused by difference in pressure, which diminishes continually from the source to the outlet, due to frictional resistance, deflection, contraction and expansion. Likewise there is a continual drop in temperature due to the transmission of heat through the walls of the piping.

Steam at initial pressure and density, but without material velocity, as in a boiler, requires a certain pressure drop, to impart initial velocity in the main. This drop varies with the velocity required, density of steam and shape of the orifice at entrance of the main. The pressure drop or head required for a given velocity, as of initial density at a point about three diameters beyond the entrance of a steam main, with sharp entrance edge, has been found from tests of the weight of low-pressure steam passing through a cylindrical sharp-edged orifice of length equal to three diameters. The pressure difference or head (h_1) necessary to produce such velocity (v_1) is fully 1½ times that found by the well-known velocity formula, $V = 2\ gh$.

It seems reasonable to assume that a like pressure drop is necessary to impart initial velocity within the heating main from a boiler or steam drum, as contrasted with the exhaust of an engine, reducing valve, etc.

Table 11-1 gives one and one-half times the pressure drop or head (h_1) in decimals of a pound and fractional ounces per square inch, based on the above assumption.

Table 11-1. Velocity Head at Entrance of Mains

In decimals of one pound and at fractional ounces per square inch. For various velocities as of initial density due to absolute initial steam pressures from 15 to 20 pounds per square inch.

Table 11-1. Absolute Pressure

Pressure Drop in Ounces per Sq. In.	Pressure Drop in Parts of Pound per Sq. In.		15	16	17	18	19	20
			VELOCITY IN FEET PER MINUTE					
3/16	3/256	.0134	2578	2507	2437	2373	2314	2238
3/8	3/128	.0234	3684	3582	3480	3390	3304	3131
3/4	3/64	.046875	5157	5014	4874	4746	4628	4476
1½	3/32	.093750	7368	7164	6960	6780	6611	6395
3	3/16	.187500	10315	10029	9744	9492	9256	8953
6	3/8	.3750	14736	14328	13920	13560	13224	12791
12	3/4	.750	20630	20059	19488	18984	18513	17907

Friction in Run: Steam having attained initial velocity at the entrance of the main by a pressure drop ($p_1 - p_2$), will require a further drop ($p_2 - p_3$) to overcome friction.

Various formulæ which agree quite closely as to larger sizes of pipe are in use; some of these do not appear to take in consideration the increased proportion of frictional surface to volume in the smaller sizes.

The formula here advanced has been found in practice to give satisfactory results.

$$W = 60\ C \sqrt{\frac{\frac{1}{V}(p_2 - p_3)\ d^5}{L \text{ plus } 50}} \quad \text{Formula 11-1}$$

in which

W = pounds per hour. $\frac{1}{V}$ = density in pounds per cubic foot.

d = inches internal diameter. L = length in feet. C = a variable dependent on diameter, and is as follows for various sizes of pipe.

Size of pipe	1	1¼	1½	2	2½	3	3½	4	5	6	7
Value of C	20	30	36	45	52	55	57	58	59	60	60.5

Size of pipe	8	9	10	12	14	16	18	20
Value of C	61.0	61.5	62.0	62.2	62.4	62.6	62.8	63

The following table has been computed from the formula

Table 11-2. Weight of steam flowing uniformly in one hour through straight level pipes 1000 feet long, with a loss of 1 lb. per sq. in. from given initial velocity within inlet end

No allowance is made for drop due to initial velocity or condensation losses in run.

1 = nominal size of pipe in inches.
2 = actual diameter of pipe in inches.
3 = linear feet per cubic foot internal volume.
4 = actual outside diameter of pipe in inches.
5 = actual inside area of pipe in square inches.
6 = length of straight pipe per square foot of external surface.
7 = square feet of external surface per linear foot of pipe.
8 = value of (Actual Diameter) $\frac{5}{2}$ expressing the square root of the fifth power of actual diameter.
9 = value of C in the equation.

P = absolute pressure.
S = cubic feet per pound of steam as of initial density.
$\frac{1}{V}$ = pounds per cubic foot of steam as of initial density.
L = latent heat of the steam.
Lbs. = pounds of steam flowing through pipe per hour.
$\frac{1000}{\text{B.t.u.}}$ = thousands of B.t.u. contained in entering steam.
V = velocity in feet per minute, as of initial density.

From Table 11-2, pressure drop for other lengths of run, other weights of steam or both may be estimated.

Table 11-2.

1. Nominal Size	2. Actual Dia. Inches	3. Linear Ft. per Cu. Ft. Internal Volume	4. Actual Outside Dia. Inches	5. Actual Inside Area Sq. In.	6. Length Straight Pipe per Sq. Ft. External Surface	7. Sq. Ft. per Linear Ft.	8. $(\text{Actual Dia.})^{5/2}$	9. "C" In Formula	P	15	16	17	18	19	20
									S	26.27	24.79	23.38	22.16	21.07	20.08
									$\frac{1}{V}$	.03806	.04042	.04277	.04512	.04746	.04980
									L	969.7	967.6	965.6	963.7	961.8	960.
1"	1.049	167.5	1.315	.86	2.9	.345	1.13	20.	Lb. 1000 B.t.u. V	8.16 7.91 600.	8.38 8.09 580.	8.64 8.32 564.	8.88 8.55 549.	9.12 8.76 537.	9.34 8.86 524.
1¼"	1.38	96.1	1.66	1.5	2.3	.434	2.235	30.	Lb. 1000 B.t.u. V	24.2 23.53 1015.	25.1 25.01 987.	25.7 25.66 958.	26.4 26.41 936.	27. 27.07 922.	27.7 27.73 891.
1½"	1.61	70.6	1.9	2.04	2.01	.497	3.28	36.	Lb. 1000 B.t.u. V	42.7 41.4 1320.	43.9 42.4 1280.	45.2 43.6 1240.	46.4 44.71 1210.	47.7 45.87 1180.	48.8 46.84 1150.
2"	2.067	42.9	2.375	3.36	1.61	.621	6.13	45.	Lb. 1000 B.t.u. V	99.8 96.71 1870.	102.4 99.3 1815.	105.5 101.9 1765.	108.3 104.4 1715.	111.2 106.9 1680.	114.0 109.5 1640.
2½"	2.469	30.15	2.875	4.78	1.33	.751	9.58	52.	Lb. 1000 B.t.u. V	180.4 174.5 2380.	185. 178.8 2300.	191. 184.2 2240.	196. 188.8 2180.	201. 193.3 2125.	206. 197.7 2080.
3"	3.068	19.5	3.5	7.39	1.09	.991	16.47	55.	Lb. 1000 B.t.u. V	327.6 317. 2790.	336.5 325. 2710.	347. 335. 2640.	356. 344. 2550.	365. 351. 2490.	374. 359.5 2440.
3½"	3.548	14.58	4.	9.89	.955	1.046	23.7	57.	Lb. 1000 B.t.u. V	488.4 473. 3120.	502. 485. 3030.	516. 498. 2930.	531. 511.7 2860.	545. 524.1 2790.	55.9 536.6 2730.
4"	4.026	11.3	4.5	12.73	.849	1.177	32.53	58.	Lb. 1000 B.t.u. V	681.6 660.1 3380.	702. 678. 3280.	722. 697. 3180.	742. 715. 3090.	761. 731.9 3020.	781. 749.7 2950.
5"	5.047	7.22	5.563	19.99	.686	1.457	57.17	59.	Lb. 1000 B.t.u. V	1218. 1179. 3840.	1250. 1207. 3730.	1288. 1242. 3620.	1324. 1275. 3530.	1358. 1304. 3450.	1393. 1337. 3330.
6"	6.065	4.99	6.625	28.89	.577	1.733	90.6	60.	Lb. 1000 B.t.u. V	1968. 1906. 4300.	2020. 1954. 4160.	2080. 2006. 4040.	2136. 2055. 3940.	2190. 2105. 3850.	2245. 2155. 3750.
7"	7.023	3.72	7.625	38.74	.501	2.	130.7	60.5	Lb. 1000 B.t.u. V	2864. 2770. 4660.	2940. 2841. 4520.	3026. 2920. 4380.	3110. 2992. 4270.	3190. 3067. 4170.	3270. 3139. 4070.
8"	7.981	2.88	8.625	50.02	.443	2.257	180.	61.	Lb. 1000 B.t.u. V	3978. 3855. 5010.	4080. 3948. 4850.	4200. 4050. 4710.	4320. 4160. 4590.	4430. 4260. 4490.	4550. 4368. 4340.
9"	8.941	2.29	9.625	62.72	.397	2.58	239.	61.5	Lb. 1000 B.t.u. V	5320. 5157. 5240.	5470. 5293. 5175.	5630. 5430. 5020.	5780. 5570. 4890.	5930. 5700. 4770.	6080. 5837. 4660.
10"	10.02	1.83	10.75	78.82	.355	2.82	317.7	62.	Lb. 1000 B.t.u. V	7116. 6900. 5703.	7320. 7083. 5530.	7540. 7270. 5370.	7750. 7460. 5240.	7930. 7625. 5110.	8150. 7824. 5000.
12"	12.	1.27	12.75	113.1	.299	3.3	498.8	62.2	Lb. 1000 B.t.u. V	11220. 10870. 6240.	11500. 11100. 6040.	11840. 11410. 5860.	12180. 11710. 5710.	12500. 12010. 5580.	12810. 12290. 5460.
14"	14.25	.904	15.	159.5	.255	3.90	766.5	62.4	Lb. 1000 B.t.u. V	17250. 16720. 6847.	17800. 17220. 6640.	18300. 17650. 6450.	18780. 18070. 6270.	19270. 18520. 6120.	19700. 18910. 5960.
16"	15.5	.765	16.	188.3	.239	4.16	945.9	62.6	Lb. 1000 B.t.u. V	21480. 20830. 7200.	22080. 21320. 6970.	22700. 21900. 6760.	23350. 22470. 6590.	23900. 23010. 6440.	24550. 23570. 6280.
18"	17.5	.601	18.	240.	.212	4.71	1281.	62.8	Lb. 1000 B.t.u. V	29100. 28220. 7660.	29860. 28860. 7410.	30750. 29670. 7200.	31600. 30400. 7020.	32400. 31100. 6830.	33300. 31970. 6700.
20"	19.5	.483	20.	298.	.191	5.23	1679.	63.	Lb. 1000 B.t.u. V	38280. 37100. 8100.	39350. 38080. 7850.	40500. 39100. 7630.	41600. 40100. 7420.	42700. 41000. 7250.	43800. 42050. 7080.

For a given weight of steam other than tabular for any given size of pipe, the pressure drop per 1000 feet corresponding to the given weight is the square root of the quotient obtained by dividing the given weight by the tabular weight.

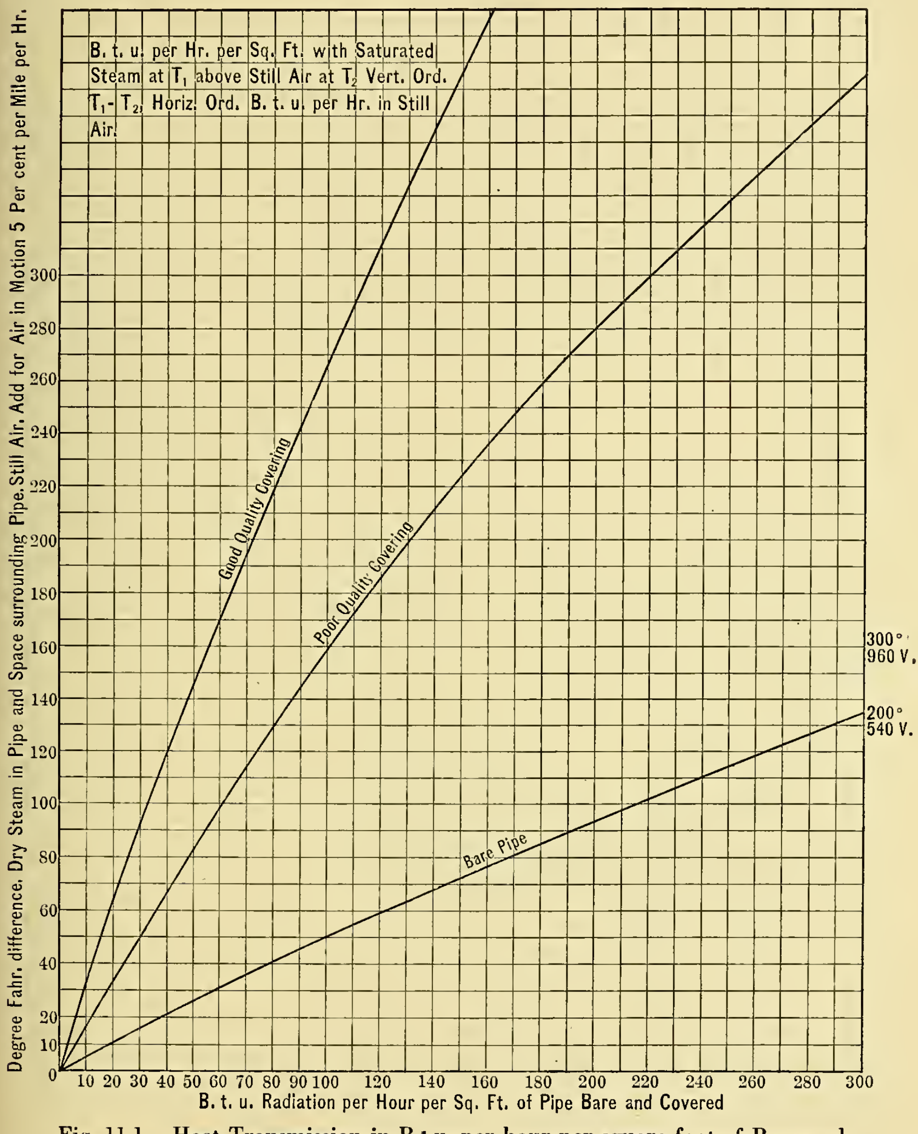


Fig. 11-1. Heat Transmission in B.t.u. per hour per square foot of Bare and Covered Pipe

This new pressure drop for 1000 feet increases or decreases in percentage as new run is more or less than 1000.

Condensation Loss: Through the entire length of run, there is a further loss of pressure, due to radiation and condensation. This loss is least in well covered mains with still air, at high temperature. Condensation in long runs of small pipe frequently causes the greatest loss of weight and occasions large pressure drop.

The following example is given to call attention to what is likely to happen if tabular steam values, in straight runs, be used to size mains supplying radiation through long runs of small pipe, even if the mains are well insulated. From Table 11-2 it will be seen that a 1½ in. pipe with a friction loss of 1/10 pound per 100 feet and an initial pressure of 16 lb. absolute will convey steam at an hourly rate of 43.9 lb. or 42,400 B.t.u. per hour.

By inspection of Figure 11-1, we find that if the difference in temperature between steam in the pipe and air surrounding it is 150 deg. fahr. and the pipe has a high-grade insulation, there is transmitted through that covering about 25 B.t.u. per lineal foot (½ sq. ft.), or 25,000 B.t.u. per hour for 1000 feet run. Therefore, about 60 per cent of the entering steam is condensed, and the balance flows at so low a pressure as to be of little value except under high vacuum.

For various differences between temperature of steam in the pipe and temperature of the surrounding space.

Effect of Deflection, Contraction and Expansion: Mains are seldom straight cylindrical pipe from end to end. Normally there are elbows, valves, branch outlets, reductions in size, separators, expansion joints, etc., each adding to frictional resistance and causing pressure drop.

Although not technically accurate, it has been found convenient in estimating, to express these resistances in units of the additional length of run of straight pipe that would produce an equal effect. Table 11-3, which

Table 11-3. Equivalent Resistance of Straight Pipe to Be Added to Run for Fittings

Size of Pipe in Inches	Gate Valve	Long Sweep Ell Run of Standard Tee	Medium Sweep Ell Reduced Run of Tee	Standard Ell Much Reduced Tee	Angle Valve	Short Bend	Side Outlet Tee	Globe Valve
	LENGTH IN FEET TO BE ADDED TO RUN							
2	2	3	4	5	9	11	17	19
2½	3	4	5	7	12	15	21	26
3	3	5	6	10	16	19	27	33
3½	4	6	8	12	19	22	32	39
4	5	7	9	14	22	24	36	45
5	7	9	11	18	27	30	44	58
6	9	11	14	22	32	36	51	70
7	10	13	17	26	37	41	56	82
8	12	15	20	31	42	47	63	94
9	13	17	22	35	47	52	68	104
10	15	20	24	39	52	57	76	117
12	18	24	30	47	62	68	91	140
14	20	26	33	53	71	79	105	160

is believed to be conservative and likely to produce results well within the tolerance necessary in so complicated a subject, is figured upon this basis.

This table is not claimed to be absolutely accurate, as fittings of different manufacturers vary in resistance in similar sizes and similar fittings vary in percentage of resistance. No very careful tests covering the entire range of flow of water, air and steam are available for data, but those that do exist have been studied and referred to in making up this table.

B. Pressure Drop: The necessity for pressure drop to create flow in heating systems has been explained in preceding pages. Modulation and vacuum heating systems differ in degree of this pressure drop rather than in principle.

*Pressure Drop in Vacuum Systems:** The reason for employing a vacuum system rather than a modulation system lies in the greater total drop than is obtainable from a given initial pressure P above atmospheric to a terminal pressure p which is less than atmospheric, thereby obtaining circulation through greater resistance due to long pipe runs and lack of grade for gravity flow of condensation.

Lowering the terminal pressure p by mechanical exhaustion in return mains (the vacuum system) provides for greater pressure drops through each of the series of resistance.

In good vacuum system practice, the total drop between source of supply through the inlet valve to the farthest radiator on the system should be that between available initial and atmospheric pressure, so that normally the pressure in the radiator will be at or very slightly below that of the atmosphere. The pressure drop p_4 of the return trap may usually be twice to three times that permissible in a well designed modulation system. The drop p_3 in the vacuum return lines, if graded in direction of flow, may equal that in the supply mains of the system under consideration, and if it be necessary to elevate the condensation at one or more vertical lifts in order to obtain horizontal grade toward the vacuum pump, this (within limits of temperature of condensation) may be obtained by increasing the displacement of air and vapor by the pump. In systems where the high vacuum necessary to lift the condensation at one or more points, would occasion a needlessly high vacuum in that portion of return system which has a gravity flow, the degree of vacuum may be reduced by means of special apparatus incorporated in the Webster Hylo Vacuum System which provides for continuous discharge of condensation and also for a reduction of degree of vacuum between the inlet and outlet of the apparatus. (See Chapter 23, page 00, for description.)

In general, owing to greater pressure drop, a vacuum system will not require as large mains, branches to, and inlet valves of radiation as needed for a modulation system. Likewise, the radiator traps and return mains may be smaller for similar sized units of radiation provided radiator traps of high efficiency are properly installed to prevent leakage of steam to return lines.

It is, as previously stated, good vacuum system practice to proportion mains between source and radiation for the pressure difference between initial and atmospheric as further described in this chapter.

* For illustration of symbols see Figure 11-2.

Return traps should be proportioned for between ½ and 1 pound pressure difference.

Return mains should be proportioned relatively to the steam mains selected for equal duty by the table of comparative sizes (Table 11-4), allowing additional areas, however, when there is probability of high temperature in the outlet end of returns, due to steam leakage of return traps or lack of vapor condensation occasioned by thoroughly insulated mains retaining the heat in the water passing through the radiator traps.

At least one size larger return main should be used where high vacuum for lifts increases volume of vapors and gases to be removed.

Such degree of partial vacuum should be carried by properly proportioned pump displacement as to cause a partial vacuum equal to the selected pressure difference (p_4) through the most remote return traps on the system. In proportioning pump displacement for vacuum systems, the most complex problem is that of proper allowance for the amount of vapor and air. Pressure below atmosphere in any part of the system is liable to induce invisible air leaks. For full efficiency of radiation the temperature of condensation passing through return traps must be close to that due to the steam pressure in the radiator.

This hot water flowing into the lower pressure in return lines partially flashes into vapor of high specific volume, as may be determined by inspection of the re-evaporation chart, Figure 21-00, Chapter 21, and referring the percentage there found to the volume occupied by the ascertained weight when at the pressure in the return pipe. Some of this vapor will be condensed on the way to the pump, the amount depending on whether or not returns are insulated and on the efficiency of that insulation. It must also be borne in mind that when the temperature of water of condensation exceeds that of vapor at the vacuum pressure, a portion of this water will flash into vapor occupying many hundreds of times its volume as water and practically impossible of mechanical displacement by a pump.

Inleakage of air through even minute imperfections in piping causes an increase of volume to be handled proportionately as the absolute temperature of the air at inleak is to the absolute temperature in the return system,

Table 11-4. Normal Relation of Return Mains and Risers to Supply Mains Caring for Equal Amounts of Steam in Vacuum Systems

Horizontal Supply Main	Horizontal Return	Vertical Return
1¼ and less	¾	¾
1½ and 2	1	¾
2½	1¼	1
3 and 3½	1½	1¼
4, 4½ and 5	2	1½
6 and 7	2½	2
8 and 9	3	2½
10	3½	3
12	4	3½
14 and 15	4½	4
16 and 17	5	4½
18 and 20	6	5

Gravity drip vertical outlet at heel of risers 2½ and under less than 12 stories high ¾ in. Over 12 stories or over 2½ riser 1 in. Vertical outlet increasing in horizontal run to 1¼.

Horizontal gravity drips graded at least ¼ in. in 10 feet are usually capable of caring for the number of ¾ or 1 in. outlets as follows:

Number of ¾ or 1 in. outlets which may be carried on one horizontal run when graded ¼ in. in 10 feet, provided radiation on steam riser does not drain as in one pipe system.

Size Horizontal	No. of ¾″ Outlets	No. of 1″ Outlets
1¼	12	6
1½	18	12
2	30	18
2½	60	36
3	100	50

plus expansion from that volume at atmospheric pressure to that of vacuum pressure.

As explained in Chapter 14 on Vacuum Pumps, it is frequently possible to take advantage of some condensing medium such as cool air for ventilation, or water, for cooking and washing, boiler feed, etc., which must be warmed, and use this medium for cooling and condensing the air and vapor to decrease its volume on the way to the pump.

The basic proportional sizes of returns to mains recommended in the above discussion are given in Table 11-4.

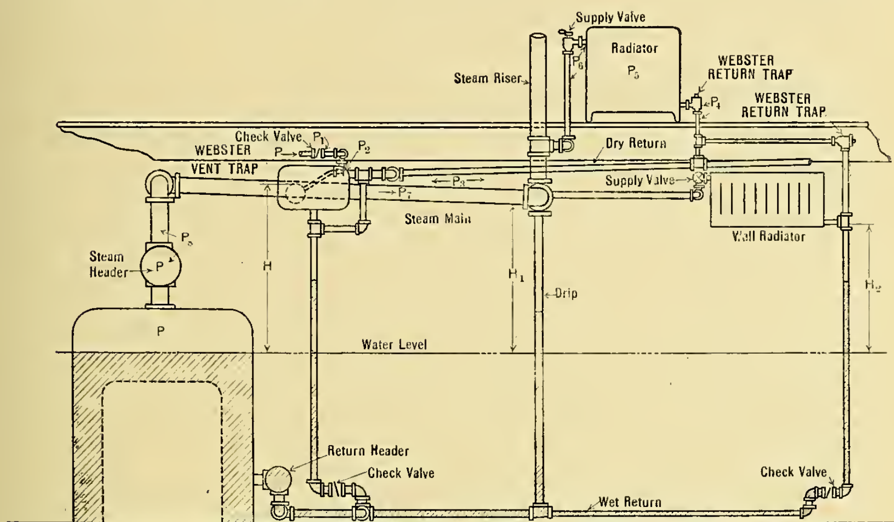


Fig. 11-2. Diagram of Modulation System layout to Illustrate Pressure Drop

Pressure Drop in Modulation Systems: The typical modulation system, as illustrated in Figure 11-2, when operating at normal rate, requires sufficient pressure against the valve piece of the check valve p_1 to cause it to open against the atmospheric pressure. Representing atmospheric pressure as p and this excess pressure as p_1 the expressions $p + p_1 =$ pressure at entrance of check valve.

To cause the air to flow from the vent trap through the vent valve orifice requires a pressure difference, which may be represented by p_2, varying with velocity of flow. Therefore, pressure in the vent trap becomes $= p + p_1 + p_2$. To cause the air to flow from outlet of the radiator trap through return main to the vent trap, there must be another pressure difference, represented by p_3, dependent on velocity of flow; also another pressure difference through orifice of radiator trap p_4. Therefore, pressure p_5 in the radiator at the time of air displacement by steam from the boiler must equal the sum of $p_4 + p_3 + p_2 + p_1 + p$. Of these last expressions p is relatively constant with gauge at 0 lb. The flow through the check valve p_1 is nearly constant, being mainly that pressure difference necessary to

overcome the gravity of the clapper and adhesion of wet surfaces of seat. The variable due to the volume of air passing is so slight, owing to low velocity, that it may usually be neglected.

The range of pressure p_1 of a check valve suitable for a modulation system is from 1/20 to 1/12 lb. per sq. in.

The pressure drop through vent valve orifice p_2 is a variable—greatest during initial heating-up period when a large volume of cool air is expelled from the heating system, and least during normal heating when velocity is that slight amount due to entrained air in condensation passing from the radiation. Air-vent traps are rated on basis of flow of initial air in 40 minutes in a system with 1/16 lb. per sq. in. differential pressure through the vent-trap valve.

For less than rated capacity, either the time or pressure factor or both may be less; for instance, with p_2 constant, one-half the amount of radiation would require one-half the time period.

The pressure drop in the return main p_3 is also a variable: greatest during initial heating, and dependent on length of run and maximum velocity. In a well-proportioned system, p_3 should never exceed 1/20 lb. per sq. in. differential between the farthest radiator trap and the vent trap, and during normal heating it is so slight as to be almost negligible.

The pressure drop through a radiator trap p_4 is also a variable, least and almost negligible during initial expulsion of air from radiation, at which time the trap-valve orifice is wide open. As the radiator warms up and condensation flows through trap orifice with the last of the contained air, p_4 gradually becomes greater. It becomes maximum when condensation, at or near steam temperature is flowing at the full rating of the return trap for a given p_4 of 1/16 lb. per sq. in., which pressure has been selected from tabular ratings of return traps. It is good practice not to have p_4 exceed 1/16 lb. per sq. in. where it is advisable to carry less than ½ lb. pressure on the boiler and ⅛ lb. where a pressure of 1 or 2 lb. can be carried.

Representing the pressure difference necessary for flow, initially of air and subsequently of steam, from the radiator branch through the inlet or modulation valve to the radiator, requires another variable p_6, 1/32 lb. per sq. in. at full rating, least (in a properly designed Modulation Valve full open), during initial expulsion of air, and greatest when the valve is partly closed for modulation effect, at the selected rating of this valve, for a given p_6.

p_7 is usually assumed for a system of mains, risers, branches, and runouts, designed from data on flow of steam in Main Table 11-0, to carry the maximum normal quantity of steam in a given time from the main heat pipe near the boiler to the inlet valve of farthest radiator, with this pressure drop p_7.

The quantity of steam assumed in the preceding paragraph, carried through the selected main heat pipe close to boiler, involves a velocity consequent on the area of pipe and volume flowing in unit time. To impart this velocity to the steam from condition of practically quiet in the steam space of the boiler, and to offset the resistance of the orifice, or the reduction in effective area, another pressure difference p_8 is required. This pressure

difference or velocity head may also be ascertained from inspection of Table 11-1. "Velocity Head" for the given main heat pipe (selected under p_7) delivering steam at maximum normal quantity, for the heating system under consideration.

It follows from consideration of the above that the pressure in the boiler P at time of maximum normal heating effect must be the sum of $p + p_1 + p_2$ etc., including p_8 as follows:

p constant at atmospheric pressure.
p_1 at least intermittent at that time.
p_2 negligible at that time.
p_3 negligible at that time if return has proper grade.
p_4 tabular if full rated value in radiation is on furthest unit.
p_5 pressure drop in radiator, negligible at that time.
p_6 tabular if full rated value in radiation is on farthest unit.
p_7 from assumption in design from flow of steam in main. Table 11-2
p_8 that required for velocity head under above assumption. Table 11-1

The heating-up period will vary in accordance with initial pressure in the source of steam supply. Usually some time is required to raise steam to the normal pressure P, and during that time air will be expelled and steam flow into the radiation at varying rates due to the increasing pressure through the increasing resistance of $p_1 + p_8$. If steam is constantly supplied during the heating-up period at pressure P (as might be the case when a central plant is the steam source), the condensation rate in the radiation due to absorption of heat by the metal will be as far in excess of normal as the sum of maximum $p_1 + p_2 + p_3 +$ an intermediate p_4., deducted from $P - p$, will produce a pressure difference (p_d) to cause initial velocity and flow through mains at a rate substantially proportional as p_d is to p_7, provided initial velocity equal to p_8 has been previously imparted to the steam within the entrance of the main.

The intermediate p_4 referred to in the above paragraph is caused by the partial extension of the thermostatically moved valve piece in the return trap, from full open and minimum resistance when chilled condensation commences to pass, to nearly closed and full resistance, when the radiator is completely filled to the return trap with steam at a temperature corresponding with its pressure.

Modulation systems when operated at less than normal condensation may continuously circulate at pressure materially lower than the normal P, or may be intermittently operated at a pressure less than p, provided the air has first been expelled by a higher operating pressure. Under such conditions, however, the system will gradually become air-bound and cease to circulate.

In designing modulation systems, all gravity drip points should be provided with a hydraulic head (H_1) of at least 2½ feet for each pound per square inch of $p_7 + p_8 +$ frictional resistance in run of gravity drip and resistance of check valve between gravity drip and boiler when the boiler is generating steam at its full capacity to supply cold radiation.

If the gravity drip be taken from radiation located below the dry return, with thermostatic air vent up to the dry return, then the resistance of

any additional branch main, radiator, valve and check valve on gravity drip, must be added to $p_7 + p_6$, etc., given above, to determine whether H_2 is sufficient.

The hydraulic head in inches of water on the check valves will vary with the make, weight and angle of the clapper and the size of pipe tapping. This head is seldom less than 3 inches with the clapper at an angle of 10 degrees from vertical and may run up to 18 inches and higher with vertical-lift valve pieces.

In installing radiation with gravity drip for condensation as above, it is important that the branch connections and valve to such radiation have sufficient free area when in use, to cause little or no reduction in pressure in the radiator, from that in the main. A partially closed inlet valve might cause such a reduction in pressure, when added to the other resistances, that there would not remain sufficient total pressure in the radiator, when added to the available H_2, to overcome the pressure P plus the check valve resistance in gravity drip; in consequence of this, condensation would build up in the column H_2, seal the radiator outlet and finally cause the radiator to become water-logged, possibly draining at a partial condensation rate, through the air vent into the dry return line.

The closing level of the air-vent trap, should be located at such a height above the water line of the boiler that a hydraulic column fully equal to the resistance of its check valve and drain pipe plus normal P is produced.

This, however, is not as important as to have H_1 and H_2 ample. An air pressure will accumulate in this vent trap due to closing of the vent outlet, when Column H is not sufficient to overcome resistance of drip line and excess P plus pressure in boiler. This air pressure will continue to build up with vent closed, until the built-up pressure with the assistance of Column H overcomes the resistance of the boiler pressure. Then Column H will fall, the air vent will open and allow escape of some air, thereby relieving part of pressure in the vent trap. Column H will again rise, closing the air vent, and this cycle will be repeated. When intermittent venting is repeated for a sufficient length of time under excess pressure P without admitting raw feed water containing gases, all the air will be expelled from the radiation.

Such a system will continue indefinitely to circulate, due to a pressure difference which will be fully equal to that of its normal H; that is, the pressure in the vent trap will be less than the pressure in the boiler, by an amount equal to an hydraulic column of height H less the resistance of the check valve on the drip of this column.

The only difficulty with excess pressures P in modulation systems, designed for pressure P and open vent at head H, occurs from rapid raising of steam and generation of excess P before the initial air has been expelled. Under such conditions complete circulation will not be obtained as rapidly as if steam had been raised more slowly.

As stated in the discussion of "Pressure Drop in Vacuum Systems," the return mains should be proportioned relatively to the steam mains selected for equal duty. This principle applies also to modulation systems.

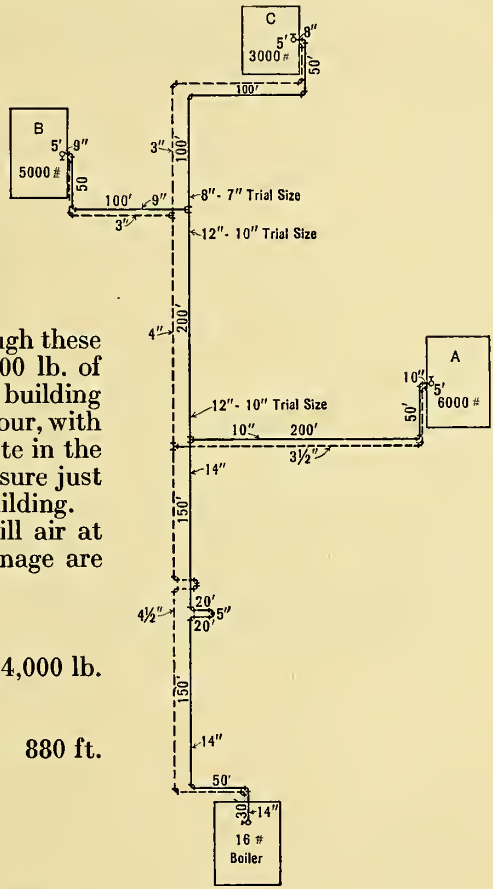
The basic proportional sizes of returns to supply mains recommended are given in Table 11-5:

Table 11-5. Relative Proportions of Steam Supply and Return Mains in Modulation Systems

Supply Main	Dry Return Main	Return Riser	Wet Return
1	¾	¾	¾
1¼	1	¾	1
1½ and 2	1¼	1	1¼
2½	1½	1¼	1½
3 and 3½	1½	1½	1½
4	2	1½	1½
4½ and 5	2½	2	1½
6	3	2½	2
7 and 8	3 and 3½	3	2
9 and 10	4 and 4½	3½	2
12	5	4	2½

C. Sizing of Piping: The use of the foregoing tables in sizing piping may best be explained by following examples.

Vacuum System: Assuming a central steam generating plant for a group of buildings as shown in Figure 11-3.



In the problem here presented are a boiler house and three detached buildings A, B and C, connected by a system of well-covered mains in a tunnel. Through these mains it is desired to convey 6000 lb. of steam to building A, 5000 lb. to building B, and 3000 lb. to building C, per hour, with a pressure drop from 16 lb. absolute in the boilers to or near atmospheric pressure just beyond the main valve in each building.

A good grade of covering, still air at about 70 degrees and proper drainage are assumed.

1. We find the total steam requirement per hour of buildings A, B and C to be . . . . 14,000 lb.
2. The longest run of the main is to building C, which without allowances is. . . . . 880 ft.
3. By referring to Table 11-2, we find under the Column of 16 lb. absolute that for a 1000 ft. run and 1 lb. drop in pressure a 14-in. main will convey . . . . 17,800 lb.

4. Length of trunk main from boiler house to first branch main	425 ft.
5. Referring to Table 11-3, we find the allowance for a 14-in. angle valve to be 71 and for a 14-in. flanged ell 53 —Total for 1 angle valve and 6 ells = 71 plus (6 x 53) =	389 ft.
6. Trunk run with allowances, 425 plus 389 =	814 ft.
7. Branch main to building A must carry	6000 lb.
8. From Table 11-2 we find under column of 16 lb. absolute pressure that for a 1000 ft. run and 1 lb. drop in pressure a 10-in. main will convey	7320 lb.
9. Length of branch run to A	255 ft.
10. Branch A allowances, 1 outlet, 2 ells and 1 globe valve, which according to Table 11-3 = 53 plus (2 × 39)plus 117	248 ft.
11. Total branch run A 255 plus 248	503 ft.
12. Beyond the first branch B and C require 5000 plus 3000 lb.	8000 lb.
13. From Table 11-2 we find that a 12-in. main will carry 11,500 lb. and a 10-in. will carry 7320 lb. For a trial we take the latter	7320 lb.
14. Length of 10-in. run net	200 ft.
15. Allowance for 1 14 × 10 run-reducing tee from Table 11-3	24 ft.
16. Total run 200 plus 24	224 ft.
17. From 10-in. tee inlet, building B — requires net	5000 lb.
18. From Table 11-2, we find that under the same condition as above a 9-in. main will carry	5470 lb.
19. Net run of branch main to B	155 ft.
20. Allowance for 2 – 9-in. ells and 1 globe valve from Table 11-3 = (2×35) plus 104	174 ft.
21. Total 9-in. run of branch to B = 155 plus 174	329 ft.
22. Steam requirement of building B	3000 lb.
23. From Table 11-2, we find that under conditions as above a 7-in. main will carry	2940 lb.
24. Net run of 7-in. main to building C	255 ft.
25. Allowance for 1 reduction, 3 ells and 1 globe valve from Table 11-3, 17 plus (3×26) plus 80	177 ft.
26. Total run of 7-in. main, 255 plus 177	432 ft.

To determine the condensation loss in the mains we convert the length of the runs to square feet of external surface from Column 6, Table 11-2, as follows:

27. Line 14-in. pipe $\frac{425}{.255}$ = 1660 sq. ft.

28. Line 10-in. pipe $\frac{255}{.355}$ = 718 sq. ft.

29. Line 10-in. pipe $\frac{200}{.355}$ = 563 sq. ft.

30. Line 9-in. pipe $\frac{155}{.397}$ = 387 sq. ft.

31. Line 7-in. pipe $\frac{255}{.5}$ = 510 sq. ft.

32. Total external surface 3838 sq. ft.

33. Add 5% for fittings to the above, which will make the total external surface 4030 sq. ft. Referring to Figure 11-1, we find from the given condition of 140 degrees temperature difference (212° — 72°) that with the assumed good quality covering we will have a heat emission of 50 B.t.u. per sq. ft. of external surface, or a total heat emission of 4030×50 . . . . 201,500 B.t.u.

34. Each pound of steam containing approximately 970 B.t.u. in latent heat, the 190,450 B.t.u. represents $\frac{201,500}{970}$ 208 lb.

Allotting this total condensation loss in pounds to each of the separate runs we have

for the 14-in. main line.	90 lb.
for the 10-in. main line.	40 lb.
for the 10-in. main line.	30 lb.
for the 9-in. main line.	20 lb.
for the 7-in. main line.	28 lb.

35. To the line loss add the requirements of buildings A, B and C. . . . . 14,000 lb.

36. Total steam requirements. . . . . 14,208 lb.

37. We find in Table 11-2 that the steam-carrying capacity of a 14-in. main 1000 feet long, with steam at 16 lb. absolute and a drop in pressure of 1 lb., is 17,800 lb. per hr. The length of run in our problem and also the quantity of steam to be supplied are smaller and the pressure drop is also consequently reduced.

The steam required, 14,205 lb. per hr., is 80% of the capacity, 17,800 lb. per hr. $\left(\frac{14208}{17800}\right) = .80$

Trunk run with allowances 814 ft. divided 1000 (the basic run) = .814 of the length for 1 lb. drop

Drop in trunk main = $\sqrt{.80} \times .814 = .89 \times .814 =$ . . . . 0.724 lb.

38. Pressure drop to A in the 10-in. pipe

Following the same line of reasoning as to the relation of steam requirement in percentage of capacity and length of run in the percentage of basic run, we obtain a pressure to A (required 6000 lb. plus 40 lb. condensation loss)

$$\frac{6000 + 40}{7320} = .824$$

Length of run, 503 divided by 1000 = .503 of basic length.

Drop of pressure to A = $\sqrt{.824} \times .503 = .905 \times .503$ . . .455 lb.

39. Total drop to building A, including drop in trunk main = .455 plus .72. . . . . 1.175 lb.

40. We arrive at the actual pressure drop to B and C in the same manner:

Capacity of 10-in. main, 7320 lb.

Steam requirement in B and C, including condensation losses $= 8000 + 30 + 20 + 28 =$ .... 8078 lb.

$\frac{8078}{7320} = 1.10$

Length of run, 225 ft. = .225 of basic run

Pressure drop in 10-in. line $\sqrt{1.1} \times .224 = 1.05 \times .224 =$ .235 lb.

41. Pressure drop to B in the 9-in. line becomes

$\frac{5000 + 20}{5470} \times \frac{329}{1000}$ or $\sqrt{.917} \times .329 = .96 \times .329 =$ ..... .316 lb.

42. Pressure drop to C is obtained by the same procedure as explained above and becomes

$\frac{3000 + 28}{2940} \times \frac{432}{1000}$ or $\sqrt{1.03} \times .432 = 1.015 \times .432 =$ ... .438 lb.

in which 30 lb. condensation loss is added to the steam requirement in the building in the same manner as the condensation losses which are added in the other calculations, and

43. By referring to Table 11-2, we find S the cubic feet per pound of steam, which for 16 lbs. absolute is very nearly 24.8.

Converting the total steam required in pounds per hour to cubic feet per minute, we have

$$\frac{14207 \times 24.8}{60} = \frac{352.333}{60} = 5872 \text{ cu. ft. per min.}$$

By referring to Table 11-2, Column 3, we find the linear feet per cubic foot of volume, which for a 14-in. pipe is .904. Multiplying 5872 by .904 we obtain the velocity in ft. per min. of the steam in the 14-in. main, V = ... 5300 ft.

44. We must now determine the pressure drop to impart initial velocity, and by referring to Table 11-1, we find for a 5300-ft.-per-minute velocity and 16-lb. absolute pressure a velocity head of very nearly......................... 0.047 lb.

45. The series drop to building B, therefore, is the sum of $.72 + .047 = .767 + .235 + .316 =$ .... 1.318 lb.

46. The series drop to building C becomes $.767 + .235 + .438 =$ .... 1.440 lb.

47. The series drop to building A is $1.175 + .047$........... 1.222 lb.

As stated at the beginning of our problem, it was desired to have a pressure drop from 16 lb. absolute to or near atmospheric pressure in the buildings, and we find that the pressure drop to building C is slightly above the 1.3 lb. drop desired, which shows that the 10-in. main and the 7-in. main (which were assumed for trial) should be increased to 12 in. and 8 in. respectively.

Based on the increased sizes, the pressure drops will be re-calculated.

48. The pressure drops in trunk main and the series drop in the main to building A will be assumed to remain as given, as we will neglect the small increase in the pressure drop in the trunk main due to the slight increase in condensation losses in the 12-in. and 8-in. mains as compared with the losses in the 10-in. and 7-in. mains before considered.

49. Length of 12-in. run to B and C.	200 ft.
Length of 8-in. run to C.	255 ft.
Exposed surface of 12-in. main $= \frac{200}{.299} =$	670 sq. ft.
Exposed surface of 8-in. main $= \frac{255}{.443} =$	570 sq. ft.
The condensation loss in the 12-in. main will be.	35 lb.
and in the 8-in. main.	32 lb.
50. Net run of 12-in. main.	200 ft.
51. Allowance for 1 reducing tee.	30 ft.
52. Total run of 12-in. main 200 plus 30.	230 ft.
53. Steam requirement to B and C, 8000 lb., to which we add the condensation losses in the 12-in., the 9-in. and the 8-in. lines, or 35 + 20 + 32 + 8000.	8087 lb.
54. From Table 11-2 we find the capacity of a 12-in. main at 16 lb. absolute through 1000 ft. of pipe, and with 1 pound drop in pressure, to be 11,500 lb., and on the basis of a total run of 12-in. main of 230 ft. we have a pressure drop in this line $= \sqrt{\frac{8087}{11,500}} \times .230 = \sqrt{.703} \times .230$.	.193 lb.
55. Net run of 8-in. main to building C.	255 ft.
56. Allowance for 1 reduction, 3 ells and 1 globe valve, from Table 11-3, $20 + (3 \times 31) + 94 =$	207 ft.
57. Total run of 8-in. main, 255 + 207.	462 ft.
58. Steam requirement to building C, 3000 lb., to which we add the condensation loss of 32 lb. = ...	3032 lb.
59. From Table 11-2, we find that the capacity of an 8-in. pipe with a pressure drop of 1 lb. in 1000 ft. of pipe is 4080 lb., and consequently the pressure drop in 462 ft. of 8-in. pipe supplying 3032 lb. of steam is $\frac{3032}{4080} \times \frac{462}{1000} =$ $\sqrt{.74} \times .429 = .85 \times .462 =$	.393 lb.
60. The series drop to building B now becomes the sum of $.724 + .047 + .193 + .316 =$.	1.280 lb.

as compared with the previous pressure drop of 1.318.

61. Considering the changed series drop to building C, we find same to be

$.724 + .047 + .193 + .393 =$.	1.357 lb.

As all of the pressure drops come so close to the maximum assumed drop of 1.3 lb., 14-in. trunk main, a 10-in.

branch to building A; 12-in. to B and C with a 9-in. line leading to B and 8-in. to building C, are the closest possible in commercial pipe sizes.

Long computations such as above are required only in connection with extensive distributing systems where the cost of one size larger pipe becomes important.

For general use in sizing mains and branches of mains in buildings for radiation, separate tables based on 70 per cent of the values from Table 11-2 will cover an ordinary amount of valves and fittings (if globe valves are excluded), and if used with discretion will prove sufficiently accurate.

For convenience in laying out general work, Tables 11-6 to 11-9, based on these values, are computed and appended at the end of this chapter.

The sizing of run-outs requires special consideration, however, and will be discussed in Chapter 12, on "Critical Flow of Steam."

For sizing the return mains we refer to Table 11-4. Based on the sizes of steam supply mains as obtained in the foregoing example, we arrive at the following conclusion:

Return from Building C.........3 in.
Return from Building B.........3 in.

Connect the two 3-in. returns from buildings B and C into one 4-in., to which the return from building A, which is 3½ in., connects, and increase the return at this point to 4½ in.

The 4½-in. return is continued full size to the boiler house.

For sizing branch return mains and run-outs the same procedure is followed, that is, the size of the return is based on the size of supply selected for an equal duty.

Modulation System: In sizing piping for modulation systems, long computations such as described under vacuum systems are not necessary. The Tables 11-6 to 11-9 appended at the end of this chapter are sufficiently accurate for ordinary conditions.

The total quantity of steam to be supplied per hour at the time of maximum normal heating effect being a known factor and the total maximum pressure drop in the heating system being determined for this period, the pressure drop in the supply main must be so chosen that the pressure to be carried on the boiler will exceed by a safe margin the sum total of resistances between the boiler and the outlet of the vent valve.

For an illustration, assume a typical modulation system which requires 500 lb. of steam per hour for maximum normal heating effect. The length of run is assumed to be 300 ft. and the boiler pressure is not to exceed ½ lb. gauge.

To find the proper size of supply main to meet these conditions, the pressure drops from p to p_6 as described in the discussion of "Pressure Drop in Modulation Systems," must be determined, before the permissible pressure drop p_7 in the supply main can be ascertained.

During maximum normal heating effect we find the pressure drop from p to p_6 to be as follows:

p = constant at atmospheric pressure =	0.000 lb. gauge	
p_1 = pressure drop through vent check valve (intermittent at that period) = 1/20 lb. =	0.050 "	"
p_2 = pressure drop through vent valve orifice (negligible at that time) =	0.000 "	"
p_3 = pressure drop in return main is negligible if return has proper grade =	0.000 "	"
p_4 = pressure drop through orifice of radiator trap, which for the given condition will be the maximum tabular value of 1/16 lb =	0.060 "	"
p_5 = pressure drop through radiator is negligible at that time =	0.000 "	"
p_6 = pressure drop through radiator valve will be the maximum tabular value for the given period—1/32 lb. =	0.031 "	"
Total drop p to p_6 =	0.141 "	"
The pressure to be carried on the boiler = ½ lb.	0.500 "	"
Pressure drop p to p_6 =	0.141 "	"
Difference of pressure available	0.359 "	"

Bearing in mind that in addition to the pressure drop p_7 in the supply main, we must consider also the pressure drop p_8 to impart initial velocity, we readily see that a pressure drop of ¼ lb. in the supply main would be unsafe and we, therefore, select the ⅛ lb. drop in the supply main p_7 as the basis for determining the size of pipe required.

We find by referring to Table 11-6 that a 5 in. main is necessary to supply 500 lb. of steam with ⅛ lb. drop in pressure in a run of 300 ft.

We now have to determine the head or pressure drop p_8 necessary to impart initial velocity to the steam.

From Table 11-2, we find S, the cubic ft. per pound of steam at 15.3 lb. absolute (assumed boiler pressure) is very nearly 26.27.

Converting the total steam required in pounds per hour into cubic feet per minute

$$\frac{500 \times 26.27}{60} = \frac{13135}{60} = 218.9, \text{ or, say, } 219 \text{ cubic feet.}$$

By referring to Table 11-2, column 3, we find the linear feet per cubic foot volume, which for a 5 in. pipe is 7.22.

Multiplying 219 by 7.22 we obtain the velocity in feet per minute of the steam to be.. 1582 ft.

We now determine the pressure drop p_8 necessary to impart initial velocity and by referring to Table 11-1 we find for a 2500-ft velocity, a pressure drop of 0.0134 lb., which for a 1582-ft. velocity would be approximately 0.008 lb. per sq. inch.

The total pressure drop between the boiler and the outlet of the vent valve then becomes:

Pressure drop $p - P_6$ as stated before =	0.141 lb. gauge	
Pressure drop p_7 in main ⅛ lb. =	0.125 "	"
Pressure drop p_8 to impart initial velocity =	0.008 "	"
Total pressure drop $p - P$ =	0.274 lb. gauge	

We find an effective differential in pressure between the boiler pressure and the pressure losses in the sytem of .500 – .274 = .226 lb. gauge, for maintaining circulation in the system during the period of maximum normal heating effect.

This proves that for the above condition, the ⅛ lb. drop in pressure in p_7 is the proper basis for selecting the table to be used, and this being determined, the intermediate sizes of the main and branches are taken from same.

The sizing of run-outs requires special consideration as described in detail in Chapter 12, "Critical Flow of Steam".

The sizing of returns involves the same procedure with modulation systems as outlined before in the discussion of sizing of piping for vacuum systems. The size of the return depends on the size of supply for an equal duty. By referring to Table 11-5, we find that the size of return corresponding to a 5 in. supply main is 2½ in., which is the size we select.

Taking care of the condensation in the steam main at the far point is often found necessary in modulation systems in which case the pipe sizes must be increased toward the end of the run, beyond the tabular values, to take care of the reduction in effective area of the pipe due to the condensation being carried along with the steam.

A further reason for increasing the sizes of the pipes toward the end of the run is to compensate for the air carried along with the steam in the pipes, which, if not properly relieved, will retard the circulation of steam to a great extent.

Air relief connections must be provided at the ends of the runs, through thermostatically actuated return traps into the nearest dry return, in all cases where gravity drips are made into a wet drip line.

Table 11-6. Flow of Steam at 16 lb. per sq. in. Absolute Initial Pressure through Mains of 200 to 1000 feet of Run

⅛-LB. DROP IN PRESSURE

LENGTH OF RUN IN FEET

Pipe Size	200	300	400	500	750	1000
1	4.7	3.89	3.36	3.0	2.45	2.13
1¼	14.38	11.73	10.13	9.06	7.4	6.43
1½	24.85	20.3	17.57	15.75	12.81	11.13
2	58.1	47.6	41.16	36.82	29.96	26.04
2½	104.3	85.7	74.2	66.5	54.11	46.97
3	190.4	156.1	135.1	120.4	98.35	85.4
3½	284.2	233.1	211.6	179.9	14.7	127.4
4	397.6	325.5	281.4	252.	205.1	178.5
5	707.	581.	501.2	448.7	365.4	317.8
6	1141.	938.	805.	724.5	590.8	513.1
7	1666.	1365.	1176.	1057.	861.	745.5
8	2317.	1890.	1638.	1463.	1190.	1036.
9	3101.	2534.	2198.	1960.	1596.	1386.
10	4151.	3395.	2926.	2632.	2142.	1862.
12	6524.	5341.	4620.	4130.	3360.	2919.
14	10080.	8260.	7140.	6398.	5208.	4522.
16	12495.	10220.	8820.	7910.	6454.	5635.
18	16940.	13860.	11970.	10710.	8715.	7500.
20	22260.	18270.	15750.	14140.	11480.	10010.

Table 11-7. Flow of Steam at 16 lb. per sq. in. Absolute Initial Pressure through Mains of 200 to 1000 feet of Run

¼-LB. DROP IN PRESSURE

LENGTH OF RUN IN FEET

Pipe Size	200	300	400	500	750	1000
1	7.	5.48	4.74	4.24	3.47	3.
1¼	20.39	16.53	14.33	12.79	10.46	9.06
1½	35.42	28.7	24.92	22.26	18.20	15.75
2	82.6	66.99	57.75	51.8	42.56	36.68
2½	149.1	121.1	104.3	93.45	76.3	66.36
3	271.6	219.8	190.4	170.1	139.3	120.4
3½	404.6	328.3	284.2	254.1	207.9	179.9
4	565.6	459.2	397.6	354.9	290.5	252.
5	1008.	819.	707.	633.5	518.	448.
6	1624.7	1316.	1141.	1022.	840.	721.
7	2373.	1918.	1666.	1484.	1218.	1050.
8	3290.	2660.	2310.	2065.	1687.	1463.
9	4410.	3570.	3099.	2772.	2268.	1960.
10	5887.	4788.	4151.	3710.	3038.	2625.
12	9254.	7490.	6510.	6174.	4760.	4130.
14	14350.	11620.	10080.	8995.	7360.	6447.
16	17780.	14420.	12460.	11130.	9100.	7910.
18	24080.	19460.	16940.	15120.	12390.	10710.
20	31710.	25760.	22260.	18880.	16310.	14070.

Table 11-8. Flow of Steam at 16 lb. per. sq. in. Absolute Initial Pressure through Mains of 200 to 1000 feet of Run

½-LB. DROP IN PRESSURE

LENGTH OF RUN IN FEET

Pipe Size	200	300	400	500	750	1000
1	9.38	7.72	6.66	6.04	4.91	4.24
1¼	28.39	23.46	19.73	18.26	14.79	12.79
1½	49.28	40.6	35.07	31.78	25.76	22.26
2	114.1	94.5	81.69	105.	60.06	51.8
2½	207.2	170.8	147.7	133.1	108.5	93.45
3	378.	311.5	268.8	243.6	197.4	170.1
3½	562.1	464.1	401.1	362.6	294.	254.1
4	784.	649.6	560.7	507.5	410.9	354.9
5	1400.	1155.	994.	903.	730.8	633.5
6	2261.	1860.	1610.	1456.	1183.	1022.
7	3297.	2723.	2352	2380.	1722.	1484.
8	4571.	3780.	3262.	2947.	2387.	2065.
9	5425.	5054.	4368.	3955.	3206.	2772.
10	8190.	6762.	5852.	5985.	4291.	3710.
12	12880.	10640.	9170.	8309.	6748.	6174.
14	19950.	16450.	14210.	12845.	10430.	8995.
16	24710.	20440.	17640.	15946.	12950.	11130.
18	33460.	27580.	23870.	21560.	17360.	15120.
20	44100.	36400.	31430.	28420.	23030.	19880.

Table 11-9. Flow of Steam at 16 lb. per sq. in. Absolute Initial Pressure through Mains of 200 to 1000 feet of Run

1-LB. DROP IN PRESSURE

LENGTH OF RUN IN FEET

Pipe Size	200	300	400	500	750	1000
1	13.3	10.97	9.52	8.51	7.	6.04
1¼	40.32	33.19	28.73	25.7	21.13	18.26
1½	70.	57.4	49.91	44.59	36.68	31.78
2	162.4	134.4	115.5	103.6	85.4	108.5
2½	294.	242.2	210.	187.6	154.	133.7
3	535.5	441.	382.2	341.6	282.1	243.6
3½	798.	658.	568.4	509.6	420.	362.6
4	1113.	917.	795.2	714.	586.6	507.5
5	1988.	1638.	1442.	1281.	1043.	903.
6	3220.	2646.	2296.	2051.	1686.	1456.
7	4672.5	3850.	3339.	2989.	2457.	2380.
8	6489.	5355.	4634.	4151.	3402.	2947.
9	8680.	7140.	6202.	5558.	4564.	3955.
10	11620.	9590.	8295.	7420.	6125.	5285.
12	18270.	15050.	13090.	11690.	9590.	8309.
14	28350.	23310.	20160.	18060.	14770.	12845.
16	35140.	28910.	25060.	22400.	18410.	15946.
18	47600.	39200.	33950.	30380.	25025.	21560.
20	62650.	51520.	44100.	33900.	32760.	28420.

CHAPTER XII

Critical Velocities in Radiator Run-outs

MUCH speculation and uncertainty exists as to the possibility of supplying steam to radiation through run-outs in which the steam condensed in the run-out must drain back against the flow of steam to the radiator.

If the flow velocity of steam is higher than the critical point, the condensation will be swept up the grade to the elbow where the vertical rise to the radiator valve occurs and there cause a churning noise and obstruction to circulation.

At first thought it would seem a simple matter to arrange a series of tests from which the critical velocity at grade could be accurately determined. In fact, however, so many variables enter the problem that a series of careful tests extending over a long period of time will be necessary to obtain results within a satisfactory degree of tolerance.

The critical velocity evidently varies not only with grade and steam density, but also with the amount of condensation. It follows (1) that the length of run-out and character of covering are material factors, because the longer the pipe the greater the volume of condensation with the same difference in temperature between interior and exterior of pipe, and (2) that a well-covered pipe will permit much less condensation per lineal foot of run than an uncovered one when in the same surrounding space temperature and air-flow conditions.

In practice, consideration must also be given to the liability of back flow of condensation from the radiator. This would occur in a one-pipe system of radiators if the bottom of the inlet pipe were at lower level than that of the outlet, as in cases of concentric tapping with inlet larger than outlet. It follows that where eccentric bushing of inlet and outlet is insisted upon, and also where run-outs are short or have few turns and are well insulated throughout, a higher critical flow may be obtained.

A series of carefully conducted tests of various sized bare pipes, 18 feet long, reasonably straight, and set at accurate grades, in a room where temperature averaged 70 deg. indicated critical velocities of steam at initial density as shown in Table 12-1.

Table 12-1. Critical Velocities of Steam in Pipes Surrounded by 70 deg. fahr. Room Temperature and Well Insulated

	Velocities in feet per minute		
Grades in 10 feet	¾″-1″-1¼″ Pipe	1½″ Pipe	2″ Pipe
⅛ in.	360	470	840
¼ in.	460	640	1070
½ in.	640	910	1330
¾ in.	830	1090	1470
1 in.	1020	1210	1570
1½ in.	1380	1320	1730

The highest velocity through run-outs occurs when steam is first turned on to cold radiators, at which time the flow is probably fully $^1/_5$ cu. ft. per min. per sq. ft. of average cast-iron radiation. It would, therefore, seem advisable, when permissible grade is limited, to limit the amount of such radiation to that shown in Table 12-2.

Table 12-2. Advisable Limits of Radiation for Various Sizes of Run-outs Where Only Small Grade is Permissible

From tests made on run-outs composed of 18 feet of uncovered pipe in a surrounding air temperature of 70 deg. fahr.

Pipe Diam.	¾″	1″	1¼″	1½″	2″
Grade in 10 feet			Radiation in square feet		
⅛ in.	7.1	11.0	19.2	33.3	98.
¼ in.	8.8	13.7	23.9	45.3	125.
½ in.	12.2	19.1	32.3	64.5	155.
¾ in.	15.8	24.7	43.1	77.3	171.
1 in.	19.5	31.5	53.0	86.0	183.
1½ in.	25.2	41.0	71.0	94.0	201.

Where the run-out has less frictional length, or where pipe is well insulated, greater ratings may be permissible. Where slow initial circulation and noise are not objectionable, these ratings may possibly be doubled. It is hoped through further tests and a digest of the results to be able to place before the engineering profession formulæ from which more accurate results may be estimated, although it cannot be hoped to eliminate the uncertainties due to inaccurate grading, rough pipe ends and other structural defects. Accurate results in any case are entirely dependent upon conditions being fully equal to those predicated.

CHAPTER XIII

Capacities and Ratings of Webster Apparatus

CAPACITY is a basis obtained from tests under one set of conditions from which ratings are deduced for other operating conditions.

The term *CAPACITY* is used in "Steam Heating" to denote the number of pounds of condensation per hour (W_1) which at uniform flow will pass through the specified apparatus when the pressure is maintained at 1 lb. per sq. in. (P_1) above that of the atmosphere and the pressure at the outlet is that of the atmosphere (P_2).

Having obtained the capacity of any unit of steam-heating apparatus under these standard conditions, *RATINGS* may be estimated within a very small error, for other stated conditions of pressure difference, time, or the amount of heat content in the steam at given initial pressure.

For any other pressure difference ($P_3 — P_4$) not differing greatly in amount from the standard pressure difference ($P_1 — P_2$), the quantity of discharge (W_2) varies from the quantity (W_1) discharged under standard conditions in proportion to the square roots of the pressure differences; that is

$$W_2 = W_1\sqrt{\frac{P_3 - P_4}{P_1 - P_2}}$$

or so nearly as to be within the normal errors of test.

The distinction which should be made between *capacity* and *rating*, especially where *rating* is expressed in some indeterminate value like "square feet of radiation," can best be emphasized by examples.

Assume a radiator trap, the capacity of which, with a drop from 1 lb. pressure above atmospheric in the radiator and trap, to atmospheric pressure in the trap outlet, has been found by tests to be 60 lb. of condensation per hr.

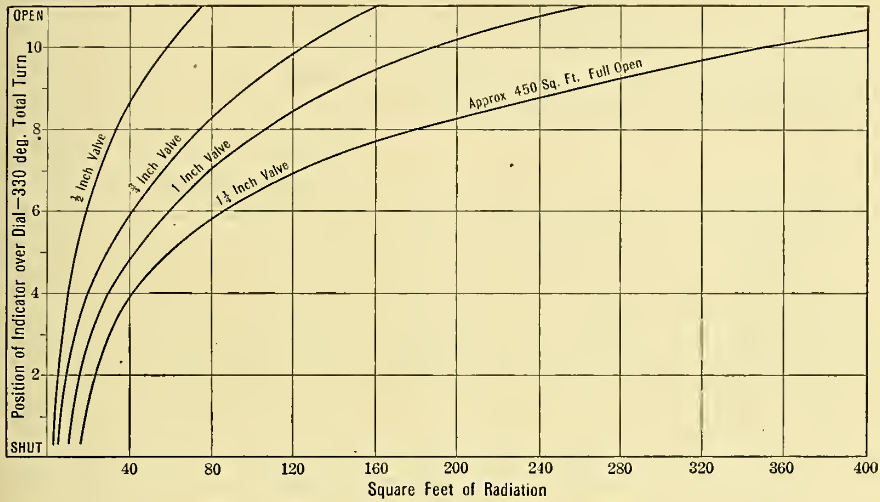
Example 1. At what should this trap be *rated* in square feet of radiation on a coil in a room of 60 deg. average temperature, when the steam pressure in the coil is 4-lb. gauge and the vacuum, at the trap outlet, is 10-in. or 5-lb. gauge?

Answer: The pressure difference through the trap would then be 4 + 5, or 9 lb. The flow through the trap would be as the square root of 1 is to the square root of 9, or three times the capacity of the trap at standard 1-lb. pressure difference. This figures out 180 lb. per hr.

Each pound of steam at 4-lb. gauge pressure gives off in condensing in a coil about 963 B.t.u. of latent heat, a total of 963 × 180 or 173,340 B.t.u. per hr. Under the temperature due to 4-lb. gauge pressure the coil would probably give off 324 heat units per sq. ft. of surface. Therefore, the *rating* of this trap under the above conditions would be 324 divided into 173,340, or 535 sq. ft. of direct radiation.

Example 2. At what would this same trap be *rated* in *square feet of radiation* on the same kind of a coil similarly placed when supplied with steam at ¼-lb. gauge, and exhausting to atmospheric pressure at the outlet?

Answer: The pressure difference through trap being as stated, ¼ lb. per sq. in., the flow through trap will be as the square root of 1 is to the square


OPEN
SHUT
10
8
6
4
2
Position of Indicator over Dial—330 deg. Total Turn
40
80
120
160
200
240
280
320
360
400
Square Feet of Radiation
½ Inch Valve
¾ Inch Valve
1 Inch Valve
1¼ Inch Valve
Approx 450 Sq. Ft. Full Open

root of $\frac{1}{4}$, or $\frac{1}{2}$ the rate at 1 lb. difference in pressure, or 30 lb. of steam per hr. Each pound of this steam will give off in condensing about 969 B.t.u. of latent heat or $969 \times 30 = 29{,}070$ B.t.u. per hour.

Under the temperature due to $\frac{1}{4}$-lb. gauge pressure, the coil would probably give off 300 B.t.u. per sq. ft of surface. Therefore the *rating* of the trap under the conditions of this example would be 29,070 divided by 300 = 96.9 sq. ft. of direct radiation.

In Example 1, the *rating* in sq. ft. of radiation is more than five times that in Example 2, the difference being due to the effect of differences in pressure on the same trap, which in both cases had the same *capacity*.

WEBSTER MODULATION SUPPLY VALVES. Careful consideration should be given to the following facts concerning ratings of this type of apparatus:

The capacity of a modulation valve should be based on the quantity of steam expressed in pounds per hour, or the equivalent B.t.u. of latent heat therein at 1-lb. pressure above atmospheric pressure which will flow through the valve when the outlet is at atmospheric pressure.

This capacity may be referred to as the number of square feet of radiating surface which would absorb the total latent heat of the steam flowing into the surface in a given time, at the commencement of which the temperature of the metal of the radiation and the room were at a stated degree below the normal room temperature.

The condensation in radiation is greatest during the warming-up period, and a large part of the latent heat of the entering steam will be absorbed in supplying the heat requirement of the metal (specific heat). Possibly one-half of the normal B.t.u. of heat flow due to radiation and convection will also occur during this period. It follows that the longer the heating-up period, other things being equal, the greater the portion of the capacity which may be expressed in the rating. The consensus of opinion is that for a radiator, the basis should be a 20-minute heating-up period from 40-degree room temperature.

As the weight per square foot of the usual types of radiating surface varies more than 2 to 1, the weights cannot consistently be averaged until at least divided into classes, as in Table 13-1.

Table 13-1. Classes of Heating Surface and Their Radiation Values

Radiation Average Class	B.t.u. Per Sq. Ft. Specific	B.t.u. Per Sq. Ft. Radiation	Total B.t.u. in $\frac{1}{3}$ Hour	Hourly B.t.u. Rate	Unit Cap. in Sq. Ft.
Cast-iron direct	148	42	190	570	1.7
$1\frac{1}{4}$-inch Steel Pipe	102	50	152	456	2.13
Sheet Steel	45	50	95	275	3.16

The basis of rating should therefore be the average class (unit capacity in square feet) divided into the capacity of the wide-open modulation valve, thus arriving at the maximum rating of the particular valve for the specified class at 1-lb. differential. The normal average flow to a heated cast-iron radiator is about 250 B.t.u. A properly designed modulation valve, when .6 open should supply the radiator with 5/12 of the full open flow, which is the approximate need for full modulation effect. The balance, or 7/12 of the opening, is thus available for a quick warming-up period (20 minutes) when the valve is full open.

Due to the wide difference in area between standard pipe sizes, a valve of say 1 in. size must be used on all different sizes of radiators between its own maximum rating and that of the next smaller, or ¾-inch valve. The 1-in. valve wide-open will therefore produce a much more rapid heating-up effect when connected to a radiator but little too large for a ¾-in. valve, and the full modulation effect will be reached much before the valve is 0.6 open, at which the maximum radiator would be under full modulation effect. This problem is best solved by putting a restrictive valve piece in those valve bodies which are used on the lower half of the range. This limits the flow at 0.6 open to about half way between the maximum for that particular valve and the maximum of the next smaller size. In this way, a valve having a total range of 45 to 78 sq. ft. of radiation at 0.6 open can be limited to 45 to 60 sq. ft. of radiation, thus gaining the whole 0.6 range for controlling the degree of modulating effect, instead of commencing to modulate only after about 2/3 closed and having but the remaining 1/3 of the total movement for graduating the modulating effect.

Fig. 13-1

Ratings of Webster Type W Modulation Valve. Based upon a differential of one pound at the valve

The capacities of each Webster Type W Modulation Valve at various positions of the pointer, both with and without the restricted valve piece, are indicated in Figure 13-1, which will assist in selection of a valve of the proper size for any set of conditions.

Initial steam pressure alone is not a correct basis for valve rating or sizing. It is safer by far to allow for the maximum possible drop in line pressure when figuring the inlet pressure at the valve. Similarly, allowances must be made for variation in return line pressure, especially with vacuum systems.

"Pounds above atmosphere" or "gauge pressures" are apt to be misleading. Inlet and outlet pressures are best figured in "pounds absolute."

The condensation rate of radiation varies with the type of radiation or

coil, its location, and the difference between outside and room temperatures, and allowance must be made accordingly.

The above facts, which hardly admit of argument, are the basis of the design and application of the Webster Modulation Valves. These valves, selected and applied according to the ratings given below, will be more efficient and will give better economy than any other manually controlled radiator inlet or supply valves.

Table 13-2. Ratings of Webster Type-W Modulation Supply Valves

In pounds of condensation and B.t.u. per hour

Size of Valves	MODULATION SYSTEM				VACUUM SYSTEM			
	Low (1 oz. dif.)		High (2 oz. dif.)		Low (¼ lb. dif.)		High (1 lb. dif.)	
	Lb.	B.t.u.	Lb.	B.t.u.	Lb.	B.t.u.	Lb.	B.t.u.
½″	11	10363	15	14380	21	20725	43	41450
¾″	23	22160	32	30760	46	44320	91	88640
1 ″	38	36700	53	50940	76	73400	151	146800
1¼″	64	62325	89	86500	129	124650	257	249300

Webster Return Traps: Both the Webster Sylphon Return Trap and the Webster No. 7 Return Trap are rated on the basis of the quantity of condensation which they will pass under stated conditions. For convenience, these ratings are given in B.t.u. and pounds per hour.

Due to the fact that these traps when cold are fully open, the warming-up period of a radiator is dependent entirely on the rate of flow of steam units to the radiator through the supply valve and its connecting piping.

The thermostatically actuated members of Webster Sylphon and No. 7 Return Traps are sensitive to very slight changes of the temperature of the surrounding medium. The motion of the members is due to difference in pressure and temperature on a hermetically sealed charge, partially liquid, partially gas and vapor, which responds to changes in temperature with material changes in volume and pressure, which provides a powerful force to actuate the valve piece.

Table 13-3. Ratings of Webster Return Traps in Pounds of Condensation and B.t.u. per Hour at Various Pressure Differences

Size of Trap	Type of Trap	MODULATION SYSTEM				VACUUM SYSTEM			
		Low (1 oz. dif.)		High (2 oz. dif.)		Low (¼ lb. dif.)		High (1 lb. dif.)	
		Lb.	B.t.u.	Lb.	B.t.u.	Lb.	B.t.u.	Lb.	B.t.u.
½″	512 and 712	10	9700	14	13580	19	18430	38	36860
	522 and 722	16	15035	22	21049	31	30070	62	60140
	533 and 733	47	45348	65	63487	94	90695	187	181390
	544 and 744	94	90696	131	126974	188	181390	375	362780
	545 and 745	188	181392	262	253948	375	362780	750	725560

Note: Webster Water-seal Traps in the few cases where they are used are rated same as the Sylphons and No. 7 Traps.

The low modulation rating in this table is based upon a differential of 1 ounce through the trap, and covers a modulation system where the boiler is to be operated on vapor pressure, as in a residence. The high modulation rating is based upon a 2-ounce differential through the trap and represents a modulation system where higher pressure may be carried, as where

the difference between water line in the boiler and the water line in the vent trap is 30 inches or more.

The low vacuum rating represents ¼-lb. differential through the trap and would approximate a vacuum system with atmospheric pressure at the radiator and a vacuum of 2 in. in the return line. The high vacuum rating is based upon 1-lb. differential through the trap and would cover a vacuum system with atmospheric pressure at the radiator and about 5-in. vacuum in the return line.

Webster Heavy-duty Return Traps: This trap is for use where large quantities of condensation are to be handled at any temperature. It has a cone-shaped float-operated valve piece seating on a sharp-edged orifice, the seat being below the low-water line of the trap. The air entering the trap is allowed to pass to the return line, either through a hand-adjusted orifice or through a connection controlled by a thermostatically operated trap discharging through a cored passage to the return line.

Table 13-4. Ratings of Webster Heavy-duty Traps in Pounds per Hour at Various Pressure Differences Through the Valve

No allowance made for pressure drop in the connecting piping between radiation and trap or from trap through run-out to return.

Size of Trap	PRESSURE DIFFERENCE							
	½ Lb.	1 Lb.	2 Lb.	3 Lb.	4 Lb.	5 Lb.	10 Lb.	15 Lb.
0016	700	1000	1400	1700	2000	2200	3150	3900
016	1250	1800	2500	3050	3600	4000	5700	7000
116	2100	3000	4200	5100	6000	6700	9500	11700
216	5600	8000	11200	13600	16000	17900	25300	31100
316	10500	15000	21000	25500	30000	33500	47400	58400

Webster Series 20 Modulation Vent Traps: Ratings of Series 20 Modulation Vent Traps are based on 6000 cu. ft. per hr. velocity of air flow through a vent orifice of 1 sq. in. from 1 lb. above to atmospheric pressure.

It is assumed that 50 sq. ft. of cast-iron radiation with connecting supply lines contain 1 cu. ft. of space. The air which must be discharged from this space before steam may enter contains about 13.2 cu. ft. per lb.

The velocity, $(V) = C\sqrt{2\,gh}$, in which $C = 0.7$

Table 13-5. Ratings of Series 20 Modulation Vent Traps

Size	Cubic Feet of Air per Hr. at 1-Lb. Dif.	Square Feet of Dir. Rad. per Hr. at ¼-Oz. Pressure	Square Feet of Dir. Rad. in 40 Min. at 1-Oz. Pressure	Square Feet of Dir. Rad. per Hr. at 1-Oz. Pressure
120	1176.	7350	9800	14700
220	2652.	16575	22100	33150
320	4710.	29437	39250	58875

The vent outlets from all three sizes of vent traps are of the same size, *viz.:* 1¼ in.

The vent opening of No. 120 Trap should be bushed to ¾ in. and fitted with a ¾-in. special vent valve.

The vent opening of No. 220 Trap should be bushed to 1 in. and fitted with a 1-in. special vent valve.

The vent opening of No. 320 Trap should remain full size and be fitted with 1¼-in. vent valve.

CHAPTER XIV

Vacuum Pumps and Auxiliary Equipment for Webster Systems

VACUUM PUMPS are used in Webster Heating Systems:

1. To remove air and other products of condensation from the return main where these products cannot be expelled to atmosphere by gravity or internal steam pressure alone.

2. To induce circulation by reducing the pressure in the return main, thereby increasing the pressure differential.

3. To assist in the complete disposal of the products of condensation.

Experience indicates two successful types of pump for this service, namely, reciprocating steam-driven and rotating electric-driven. The steam-driven pump has efficiency and economy in its favor where steam at 30-lb. or greater absolute pressure is continuously available and the pump exhaust and its contained heat may be fully utilized in the system. The electric-driven pump is generally most efficient where exhaust steam from the engines and other sources is continuously available in greater quantity than is necessary to supply the heating system—in other words, where the exhaust from the vacuum pump to waste would be a loss. The electric-driven pump is also preferable when the available live steam supply is at too low a pressure to operate a steam-driven pump.

Many rotating pumps in which both air and water were handled in one chamber have deteriorated very rapidly in service largely because of the grit always present in the condensation. Rotating pumps with one pump chamber handling air and vapor and another containing a centrifugal impeller for handling the water have proved practical.

Many variables enter the problem of ascertaining the proper size of pump for a given heating system. In the final analysis, good judgment based on wide experience in applying a table of probable pump displacement is of far greater value than any theoretical formula.

Even for a close approximation it is necessary to know enough about the heating plan in addition to "the square feet of equivalent radiation" to be able to estimate the probable maximum volumes in unit of time of both water and elastic fluids of condensation; also the necessary degree of vacuum at the pump and the discharge head against which condensation must be delivered.

The volume of water condensation varies in different installations fully 40 per cent per square foot of equivalent direct radiation. The volume of elastic fluids—air, water, vapor, steam and gases from impurities—also varies with the initial and terminal pressures, with the efficiency of the radiator traps, with the degree of prevention of inward leakage of air, with the probable cooling effect in the return, and with the character of the impurities in the boiler-feed.

Lifts (see Figure 14-1) in the return call for lower terminal pressure with consequent greater expansion in volume of the elastic fluids, thus calling for greater pump displacement, and, therefore, should be avoided wherever possible.

Discharge head on reciprocating pumps handling water and air has the

effect of increasing clearance and slip and thereby decreasing the effective displacement.

A discharge head of more than one added atmosphere on reciprocating pumps is best handled by separating the water and gases and removing them independently through two separate pumps.

For slip in reciprocating wet-vacuum pumps it is seldom safe to allow less than ⅙ of the displacement, although a newly packed pump may show much less.

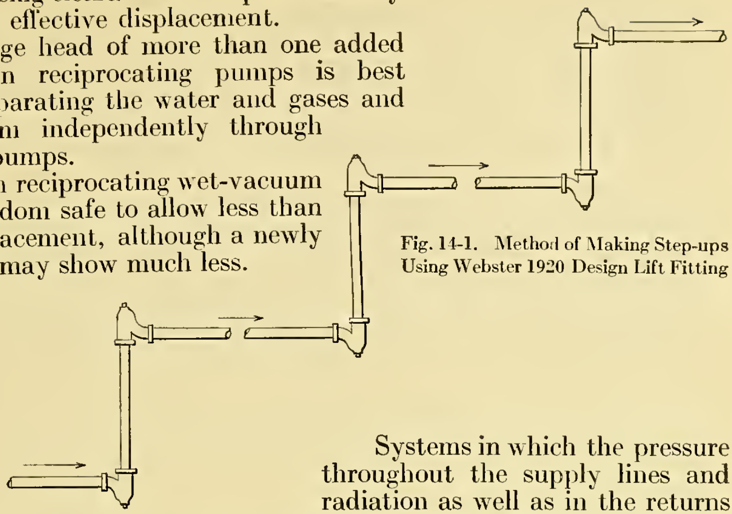


Fig. 14-1. Method of Making Step-ups Using Webster 1920 Design Lift Fitting

Systems in which the pressure throughout the supply lines and radiation as well as in the returns is normally less than that of the atmosphere are subject to invisible inleakage of air around the valves and fittings. Such systems require increased displacement also, because of the greater volume of elastic fluids due to low terminal pressure necessary for circulation.

Cooling and consequent reduction in volume of the elastic fluids in return presents an element of considerable magnitude and uncertainty.

Well-insulated return pipes, also large volumes of condensation entering the main return close to the vacuum pump, require greater displacement than would the same radiation with returns in which a considerable portion of the vapors had the opportunity to condense between the radiation and the pump.

Clearance reduces effective displacement in all pumps. The clearance for a given cylinder diameter in reciprocating pumps of some makes is approximately the same in short-stroke as in long-stroke pumps. Commercial sizes of reciprocating vacuum pumps vary in ratio of bore to stroke between 1 to ¾ and 1 to 2; it follows that a pump of the latter proportion has greater efficiency per displacement than the short-stroke pump because of smaller percentage of clearance.

Experience with reciprocating steam-driven vacuum pumps indicates that for most favorable conditions the use of water cylinders of less displacement than eight times the normal volume of water of condensation is seldom safe. With radiation divided into small units, a ratio of at least 10 to 1 will be required.

Ratings for the rotating combination units should be based substantially on a 10 to 1 ratio of the combined displacement of water and air cylinders, the ratio of these cylinders to each other being about 2 of water displacement to 8 of air. In these pumps the displacement of water must be

high on account of the constant speed, while a low proportion of air displacement may be taken because of the high efficiency of the air chamber as compared with reciprocating pump cylinders which have greater clearance.

The speed and displacement in rotating pumps are normally constant, unless expensive variable speed motors are used, whereas in reciprocating steam-driven pumps the piston speed may be varied through a wide range. The temptation with the latter is to gain displacement by excessive piston speed.

The time element of stopping and starting the piston and valves twice to a cycle should, therefore, be considered, and the piston speed should bear some relation to length of stroke. Commercial sizes of reciprocating vacuum pumps usually have stroke equal to bore, and have, therefore, been selected in Table 14-1 for presenting displacement as a logical basis for estimating, within a reasonable tolerance, the proper size of pump to handle the condensation from the wide range of conditions found in vacuum heating systems.

Separate columns of this table give minimum connections of the pump and accessories and factors for conditions other than those assumed as normal.

Column 2 gives the basic pounds of condensation and B.t.u. per hour for pumps having stroke equal to bore in inches shown on same line in Column 1.

The basis ratings for each diameter of pump are calculated as shown in following example.

Column 1. Diameter of pump (d) 4″
Square root of diameter d 2
Piston travel per hr.
$1200\ \sqrt{d}$ $= 2400$
Area in sq. ft. of d 0.0873
Gross displacement $= 1200 \times \sqrt{d} \times 0.0873 =$ 209.5 cu. ft. per hr.
Pounds of condensation per hour =
$\frac{1}{10}$ gross displacement in cu. ft. = 20.95
60 pounds = 1 cu. ft. less $\frac{1}{6}$ for slip = 50 pounds

Column 2. Basic pounds of condensation = 20.95 x 50 = 1047.5
average B.t.u. per pound of condensation 970

Column 2. Basic B.t.u. = 1047.5 x 970 = 1,016,075

Column 4. In this column are factors for converting basic ratings of Column 2 into ratings for pumps with stroke greater or less than bore, the proper multiplier for Column 2 in Column 4 having been found in Column 3 under the quotient of stroke divided by bore.

Example: Assume 4 in. diameter and 6 in. stroke. Find the basic rating.
Divide 4 into 6, quotient = 1.5.
Inspection of Column 3 shows 1.5 on a line with 1.19 in Column 4.
Multiplying basic rating, 1047.5, by Column 4 = 1.19.
Basic rating of 4 in. x 6 in. pump = 1246 lb.

Column 5. Approximate size of return graded 1 ft. in 300 to pump, which when half filled would deliver net pounds of condensation

Table 14-1. Data on Vacuum Pumps and Their Auxiliary Equipment

Diameter in Inches Water Cylinder	Condensation Pounds per Hour	Condensation B.t.u. per Hour	Per Cent. of Normal Efficiency	Quotient of Stroke÷Bore	Minimum Size Return and Strainer. Basis—Pipe ½ Full, Slope 1" in 300'	Minimum Size Discharge of Pump. Basis—Pipe ½ Full, Slope 1" in 20'	Sq. Ft. Cross Section Separating Tank. Basis—2100 Lbs. Condensation = 1 Sq. Ft.	Hydro-Pneumatic or Plain Air Separating Tank: Diameter	Hydro-Pneumatic or Plain Air Separating Tank: Length	Tanks having Automatic Control by Steam or Water, and Capacity for 5 Minutes Storage Space: Diameter	Tanks having Automatic Control by Steam or Water, and Capacity for 5 Minutes Storage Space: Length	Size of Steam Supply and Vac. Governor
No. 1	No. 2		No. 3	No. 4	No. 5	No. 6	No. 7	No. 8	No. 9	No. 10	No. 11	No. 12
3"	509	493730	2.0	138	1¼"	¾"	⅓	4"	12"	18"	24"	½"
4"	1047	1015590	1.9	134	1½"	1"	½	6"	12"	18"	36"	½"
5"	1825	1770250	1.8	131	2"	1¼"	6/7	8"	18"	18"	36"	½"
6"	2890	2803300	1.7	127	2½"	1½"	1.37	8"	24"	18"	48"	½"
7"	4230	4103100	1.6	123	{2½", 3½"}	{1½", 2"}	2.0	12"	24"	{24", 24"}	{36", 48"}	¾"
8"	5930	5752100	1.5	119	{3", 3½"}	2"	3.0	12"	36"	24"	48"	¾"
9"	7950	7711500	1.4	115	{3½", 4"}	2"	4.0	18"	30"	{24", 24"}	{60", 72"}	¾"
10"	10340	10029800	1.3	112	{4", 4½"}	2½"	5.0	18"	48"	30"	48"	1"
12"	16300	15811000	1.25	110	{4½", 5"}	2½"	8.0	24"	48"	30"	60"	1"
14"	23900	23183000	1.20	108	{5", 6"}	3"	11.4	{30", 24"}	{48", 72"}	{36", 36"}	{60", 72"}	1¼"
16"	33500	32495000	1.1	104	{6", 7"}	3½"	16.0	{36", 36"}	{60", 72"}	{36", 42"}	{96", 72"}	1¼"
18"	45000	43650000	1.0	100	7"	4"	21.4	{42", 36"}	{60", 96"}	42"	96"	1½"
20"	58500	56745000	0.9	96	{7", 8"}	4½"	28.0	{42", 42"}	{72", 96"}	48", 48"	{72", 96"}	1½"
22"	74300	72071000	0.8	91	8"	4½"	35.0	48"	96"	48"	96"	2"
24"	92200	89434000	0.75	89	{8", 10"}	5"	44.0	48"	120"			2"
26"	112500	109125000	0.70	87	10"	6"	56.0	60"	120"			2½"
28"	136500	132405000	0.60	82	12"	6"	65.0	72"	120"			2½"
30"	161400	156558000	0.50	78	12"	6"	77.0	96"	120"			2½"
32"	189300	183621000	0.4	72	14"	7"	90.0	96"	120"			3"
34"	220000	213400000	0.3	67	14"	7"	105.0	96"	144"			3"
36"	254000	246380000	0.25	65	14"	8"	121.0	96"	144"			3"

NORMAL SYSTEM

(A) Steam above atmosphere at farthest radiator
(B) Units of 20 to 25 square feet
(C) Standard screw-down radiator valves
(D) No large volume of condensation at steam temperature
(E) No lifting of returns
(F) Returns insulated
(G) Run of mains less than 500 feet
(H) All drip points have Webster Sylphon or No. 7 Traps

FACTORS FOR CONDITIONS OTHER THAN ABOVE BUT WEBSTER TRAPS USED	Per Cent. Normal	Equiv't to Normal
	No. 13	No. 14
Ample condensation on returns	50	2.0
Blower stacks and water heaters in large units	66	1.5
120 Sq. Ft. Units,—Returns covered; 90 " " " " exposed	75	1.33
90 " " " " covered; 70 " " " " exposed	80	1.25
70 " " " " covered; 55 " " " " exposed	84	1.20
55 " " " " covered; 40 " " " " exposed	89	1.125
40 " " " " covered; 20 or 25 Ft. " " exposed	100	1.0
20 or 25 Ft. " " covered; 10 Sq. Ft. " " exposed	111	0.90
Water Seal Traps—Large units	125	0.80
" " " —80 sq. ft. units	143	0.70
" " " —40 " " "	166	0.60
" " " —15 " " "	200	0.50

Factors for approximate reduction of square feet of radiation into pounds of condensation per hour:

Factory pipe coils	0.33
Factory wall radiation	0.30
Low cast-iron radiators—1 and 2 column	0.27
Medium height cast-iron radiators—1 to 3 column	0.25
High cast-iron radiators—2 to 4 column	0.23

(Column 2) by gravity ($Q = ac\sqrt{rs}$, in which q = quantity discharged per second, a = cross-sectional area of pipe, c is a constant, r = hydraulic radius and s = slope of pipe).

Column 6. Approximate size of pump delivery pipe from pump to tank, based on pipe half filled and slope of 1 ft. in 20.

Column 7. Air separating surface required in cross-section of tank based on 1 sq. ft. to 2100 lb. of condensation.

Column 8 and 9. Diameter and length of tank for air separation only, as in Hydro-pneumatic or plain tanks.

Column 10 and 11. Tanks in which condensation automatically operates valves, either for admission of cold make-up water or the speed of boiler-feed pump connected to tank. These tanks are to be capable of storing the normal water of condensation pumped to them in five minutes.

Column 12. Normal size of steam pipe and vacuum governor to pump used with boilers of 75 to 125 lb. steam pressure.

Column 13 and 14. Factors for conditions of heating system other than those assumed as normal.

The normal basis being:

(a) That such system be continuously supplied with steam at pressure above atmosphere and 2 to 3 in. vacuum in return at farthest radiator.

(b) That average cast-iron radiation units of 20 to 25 square feet direct are used. If units average larger than 25 square feet direct and other conditions are normal, there will be less air to be handled and the displacement may be less.

(c) That all units of radiation have standard radiator valves of screw-down type. If there is no inleakage around inlet valves, less displacement is required.

(d) That draining of horizontal mains is by gravity drip to hot-well or receiver. If no large volumes of condensation at or near steam temperature enter return near the pump, a smaller volume of vapor is to be handled.

(e) That there are no lift points in return. Lifts require an increased degree of vacuum and a greater volume of displacement.

(f) That returns as well as supplies are insulated. Returns not insulated or having some form of cooling coil decrease the normal volume of elastic fluids and require less displacement.

(g) That total run is less than 500 feet from the source of steam supply to the farthest radiator. The initial degree of vacuum to obtain a terminal of 2 or 3 inches at farthest radiator normally increases with length of run and calls for greater displacement.

(h) That all units of radiation have on their drip connections Webster Thermostatic Traps of Sylphon or No. 7 Type. If the traps on the drip ends of radiation leak steam at any normal variations in differential pressure through the trap, the volume of elastic fluids to be displaced increases with the steam leak and the required degree of vacuum.

To use this table where the B.t.u. basis of each class of radiation is shown in the calculation for heat losses, and the condition of the system is

other than outlined under Normal Systems, multiply the B.t.u. in each class by the factor in Column 13 or divide by Column 14 for that class. Find sum of all classes when so factored and look in Column 2 for the nearest basis rating in B.t.u. to the above factored sum. The corresponding diameter in Column 1 will be that of the pump required.

The sum of factored quantities divided into the basic rating of given diameter will give index in Column 3 and factor in Column 4 times given diameter will give proper stroke.

If the result does not fit stock sizes obtainable, select a stock pump of diameter and stroke which when factored by Column 4 will give a basic rating at least equal to the factored sum of heat losses.

In using the table to ascertain a pump corresponding to the radiation in square feet, convert the square feet of each class of radiation into pounds per hour or B.t.u.; multiply the pounds or B.t.u. per hour for each class by the factor in Column 13 for that class. Find sum of all classes when so factored and look in Column 2 for the nearest basic rating in pounds to the above factored sum. The corresponding diameter in Column 1 will be that of the pump required.

The method of finding the required stroke of reconciling to stock sizes is to divide required pounds or B.t.u. into basis lb. or B.t.u. and apply quotient to Column 3 to find proportion of stroke to bore in Column 4.

Problem: Find the proper size of water end to handle the condensation from Buildings A and B and C wherein all returns are exposed.

Bldg. A.	Blast coils condensing	5000 lb. per hr.	
	Closed heater condensing	1000	
		6000 x 970	= 5,820,000
Bldg. B.	100 pipe coils 130 sq. ft. each approximately 375 B.t.u. per sq. ft.		= 4,850,000
Bldg. C.	200 direct-indirect radiation, 50 sq. ft., each approximately 290 B.t.u.		= 2,910,000

Factoring these B.t.u. results by Columns 13 or 14 we have:

A = 5,820,000 x .66 = Blower stack and water heater	3,880,000
B = 4,850,000 x .75 = Coils over 120 sq. ft.	3,637,500
C = 2,910,000 x .89 = Radiators, about 50 sq. ft.	2,589,900
	10,107,400
Column 2. Nearest diameter under 10 in.	10,029,800

We note that the basic rating of a 10-in. bore comes nearest to the sum of factored quantities, and we, therefore, select a pump with a 10-in. bore.

$$\frac{\text{Factored B.t.u.}}{\text{Basic B.t.u.}} = \frac{10107400}{10029800} = 1.07 \text{ say } 1.0 \text{ (Column 3)}$$

Factor in Column 4 corresponding = 100% = 1

Bore x Factor (Column 3) = Stroke = 10 x 1 = 10-in. stroke

We, therefore, use a 10 in. x 10 in. water end, which is a stock size.

Looking in Column 5 and Column 6	Minimum return 4 to 4½ in. for long run	= 4½ in.
	Minimum discharge 2½ in. to an open tank	= 2½ in.

Columns 8 and 9. Size of open tank min. 18 in. x 48 in., stock size.

Column 12. Vacuum governor if high pressure steam available, 1 in. size.

Note: See example of three buildings in Chapter 11. Main, 14-in; Return, 4½ in.

Proportioning of Steam Ends of Reciprocating Vacuum Pump: It is seldom safe to use a factor of less than 100 per cent in proportioning area of steam cylinder to area of water and air cylinder. Carelessness in setting up the packing in water and air pistons is prevalent and to be expected.

Necessity for pump to keep going when steam pressure is lower than that predicated is another reason for use of above factor.

For ascertaining the minimum area of the steam piston the following rule may be applied: Multiply the area of the water cylinder by the sum of the maximum vacuum and discharge head expressed in pounds per square inch, and divide by one-half the available gauge pressure of steam *at the pump*.

It must be remembered that the available pressure at the pump is always lower than that at the boiler. Therefore, for a safe approximation of steam piston area one-third of the boiler pressure times the area of steam cylinder in square inches should equal the pressure in water end (vacuum plus discharge pressure) times the water piston area in square inches, as expressed in following equation:

$$A^s = \frac{A^w \text{ x } (V \text{ plus } D_p) \text{ x } 3}{B_p}$$

in which

A^s = Area of steam piston in square inches.

A^w = Area of water piston in square inches.

V = Suction pressure expressed in pounds = Vacuum in inches divided by 2.

D_p = Discharge pressure in pounds per square inch.

B_p = Boiler pressure in pounds per square inch.

Note: All pressures by gauge.

Power-driven Reciprocating Vacuum Pumps: Lack of available steam pressure to operate the piston in reciprocating vacuum pumps requires that some other source of power must occasionally be utilized. Where this is the case, a reciprocating pump is in many cases unsuitable because of the difficulty in handling the varying load during each stroke and because no satisfactory means for controlling the displacement to maintain the desired degree of vacuum has yet been devised for this type of pump.

To move the reciprocating piston in the water cylinder by means of a connecting rod and crank, the latter necessarily rotating at low speed, entails gearing or an extremely large pulley and countershafting. Inasmuch as the torque varies from almost nothing at the ends of the stroke to a high maximum at about three-fourths stroke, back-lash, noise and wear of gears or slapping and slip of belts are to be expected unless a heavy fly-wheel is used, and in any instance the power consumption is excessive.

Variable-speed motors are sometimes utilized for driving, but are expensive, and at best give only two or three steps of displacement, which must be selected either manually or by complicated and delicate electrical controllers.

There is nothing to commend in intermittent control. Constant speed and displacement with a vacuum breaker to admit air when the load is below normal is probably nearest to a satisfactory arrangement where power-driven reciprocating vacuum pumps are used.

Disposal of Vacuum Pump Discharge: Conditions vary to such an extent that good judgment is the only safe guide in determining the best method for the disposal of the vacuum pump discharge. In no case should the head against the discharge of reciprocating pumps exceed 15 pounds unless the pump stroke materially exceeds the bore and thus reduces the bad effect of clearance. Usually one of the following seven methods will best apply.

1. *Discharge to Waste:* Disposal by discharge to waste involves loss of all the valuable heat and water, but in rare cases this is permissible.

2. *Discharge through Air-separating Tanks:* Where first thought seems to indicate disposal to waste, it will in many cases be found possible to deliver the water and air into a separating tank, or stand pipe sufficiently elevated for the water, after separation, to flow by gravity to some point of valuable use, such as boiler, feed-water heater, etc.

Where due to structural conditions, a suitable elevated location cannot be found, the effect of head may be obtained by use of a Hydro-pneumatic Tank as described under heading No. 4.

3. *Discharge to Open Vent Tanks:* Open vent tanks, otherwise called plain separating tanks, normally serve the purpose of releasing the entrained air from the discharge of the vacuum pump. (See Figure 14-2.)

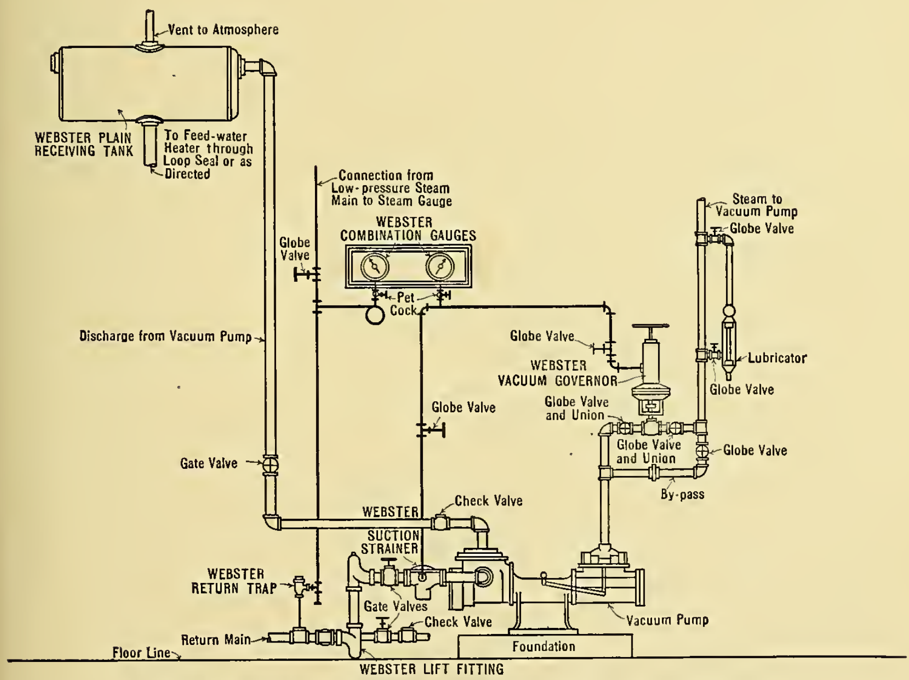


Fig. 14-2. Method of Connecting Vacuum Pump to a Plain Receiving Tank

This air removal requires the generous water surface area of either a tank of large horizontal cross-section, rather than one of large vertical sectional area, or a tank with a large vertical head and enough sectional area to permit of low-velocity downward water flow while entrained air is floating to the surface against the water current, as in a stand pipe. For removal of air, one square foot of horizontal cross-section has usually been found sufficient for each 2100 lb. of water per hour. A stand pipe, with diameter equal to that of the pump cylinder, is usually sufficient, although a more logical rule is to make the cross-sectional area of the stand pipe bear some direct relation to the amount of condensation from which the air is to be separated, and to the height of column of water through which the air bubbles must rise against the flow of liquid.

The fact that the discharge of reciprocating wet-vacuum pumps is a mixture of water and air favors the use of a freely vented separating tank wherever a suitable location may be obtained. This is at such height that the pressure produced by the water column will be sufficient to overcome that in the low-pressure boiler, feed-water heater (see Figure 14-3), or other point of disposition.

Fig. 14-3. Typical application of Webster Water-control Receiving Tank in connection with an open feed-water heater. The heater should be set on a foundation of sufficient height (a vertical rise of not less than three feet) between the pump outlet of the heater and the suction valves of the boiler-feed pump

The effective column or head between the pump-discharge valve and the inlet of the separating tank will be less than that of solid water by the volume of air contained in the mixture. The contents in separating tank and discharge pipe therefrom will be water only. It is, therefore, possible with pump discharge properly proportioned and provided with lift fittings, vertical rise pipe to tank, etc., to obtain a gravity head in the tank discharge above the level of the pump valve deck, considerably greater than the pressure in the pump cylinder necessary to lift the valves and discharge the condensation to the elevated return tank.

4. *Discharge to Hydro-pneumatic Tanks:* As the name indicates, hydro-pneumatic tanks bring the elastic pressure of the liberated air to act on and supplement the head, in the discharge of the water of condensation. A float-controlled valve is placed on the air outlet of the separating tank, and so arranged that when the water of condensation has not sufficient head to flow by gravity to the point of use, the air will be confined in upper part of tank. As the pump continues to deliver water and air to the tank

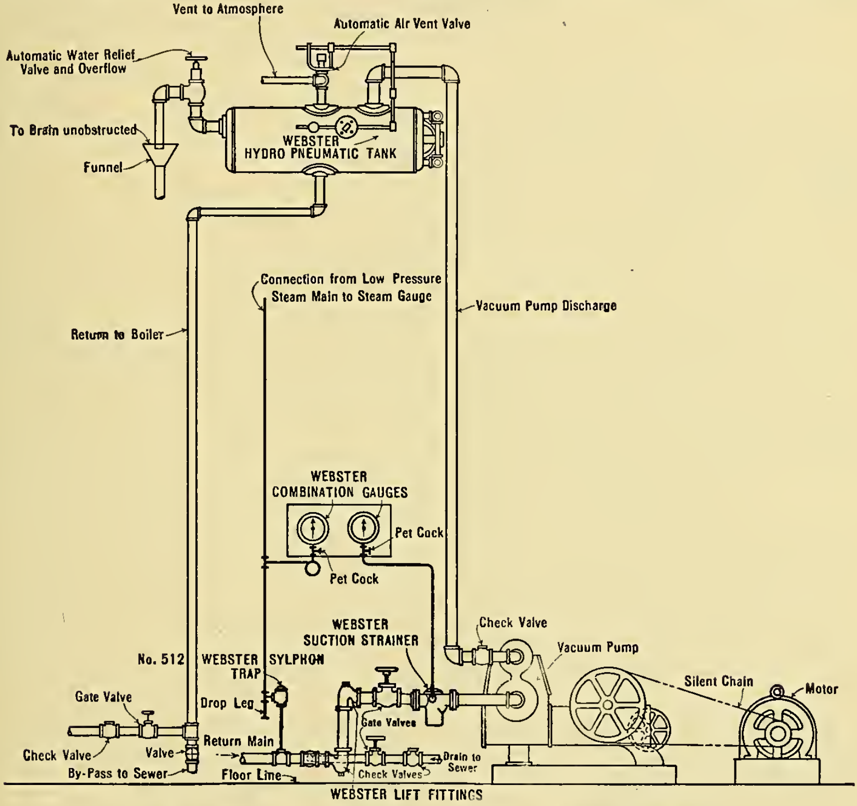


Fig. 14-4. Method of Connecting Geared Type Vacuum Pump and Webster Single-control Hydropneumatic Tank

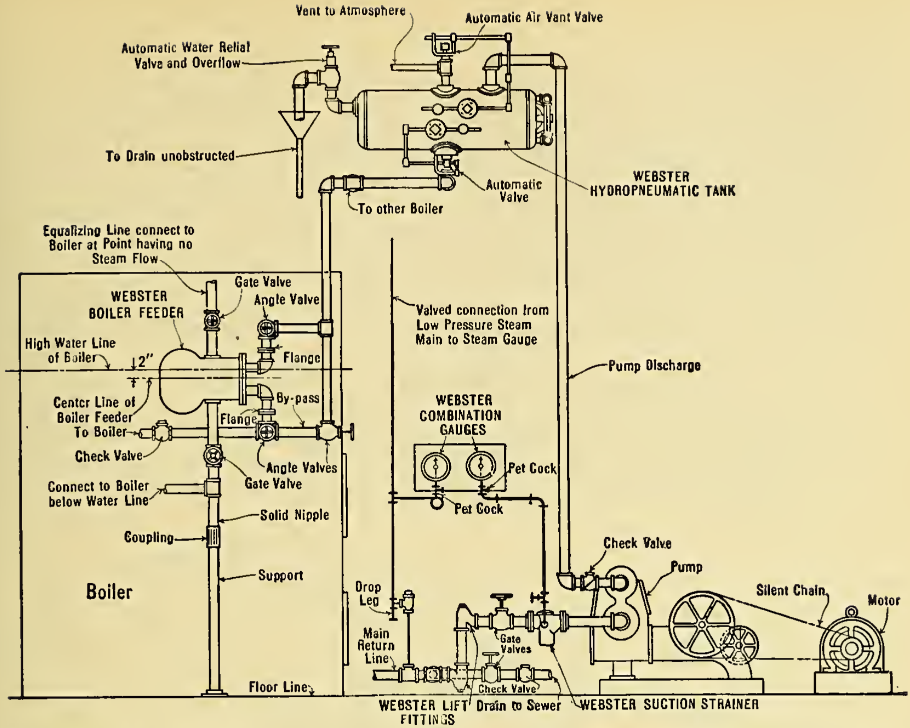


Fig. 14-5. Typical Connections to Vacuum Pump, Double-control Hydro-pneumatic Tank and Boiler Feeder

(see Figure 14-4) the pressure inside the tank increases until sufficient to discharge the water, thus lowering the water line and eventually permitting escape of the surplus air through the float-controlled air valve.

The discharge of condensation to low-pressure boilers, in which the pressure may at times be less than that of the atmosphere, requires another float in the hydro-pneumatic tank (see Figure 14-5) to control the valve on the tank water discharge and keep this pipe closed at such times as there might be danger of air flowing from the tank to the boiler.

The hydro-pneumatic type of tank is used only where an open tank cannot be located at a height sufficient to provide gravity head to discharge the tank contents against the maximum pressure in the heater or boiler, or where there are large variations between the maximum and minimum pressures to be overcome. Where the hydro-pneumatic tank is used merely as a substitute for an open separating tank, little advantage may be taken of the high density of the pump discharge.

The confined air pressure in the hydro-pneumatic tank plus the gravity head in the tank discharge pipe must be sufficient to cause flow to the place of disposition. This confined air pressure plus the column of mixed air and water in pump discharge to the tank is the total head against which the pump must act.

Where pressure on the heater, boiler, etc., varies materially from time to time, but in general is near the minimum, a substantial saving in energy may be obtained by using a hydro-pneumatic tank instead of a plain tank set at higher elevation to overcome the peak pressure in the boiler or heater. The use of a plain tank under these conditions keeps the pump operating constantly against the maximum head, where a hydro-pneumatic tank set lower operates as a plain tank whenever the gravity head in the tank is sufficient to cause flow at the low elevation, and employs the combination of air pressure and gravity head (with air vent closed) only at times of peak load. Only for this short time is the air pressure load added to the pump discharge.

5. *Discharge to Loop Seal on Tank Outlet to Heater or Boiler:* The disposal of water of condensation from a return tank to a feed heater (see Figure 14-6), boiler, or other receptacle, in which there may be greater pressure than that of the atmosphere, requires guarding against back flow of steam, air or whatever other elastic fluid may be present at the outlet.

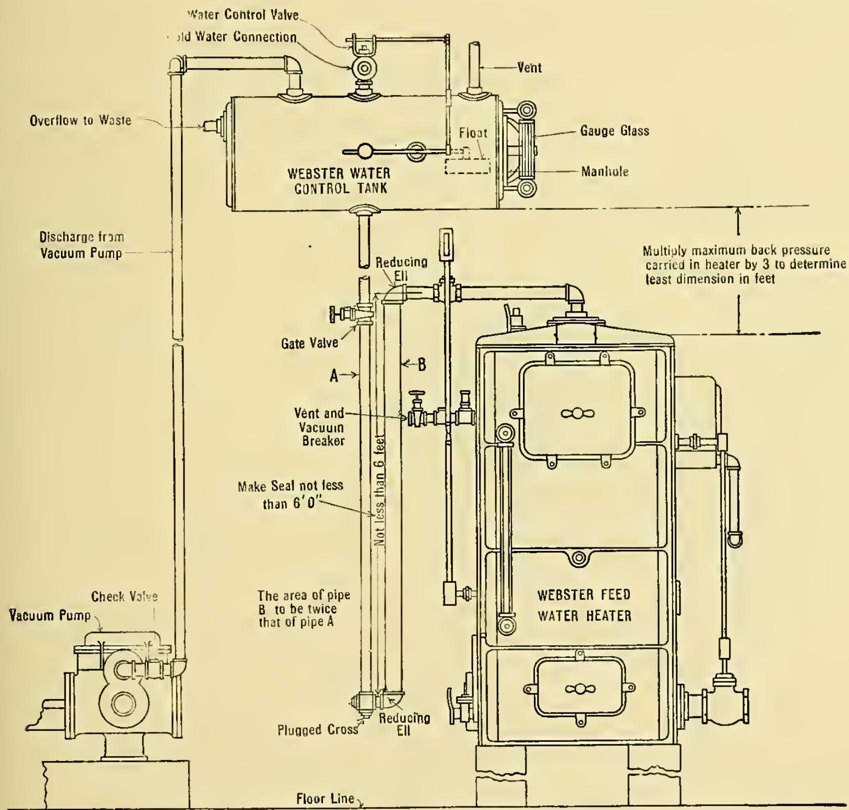


Fig. 14-6. Vacuum Pump Connections to Open Heater Using Single Control Hydro-pneumatic Tank

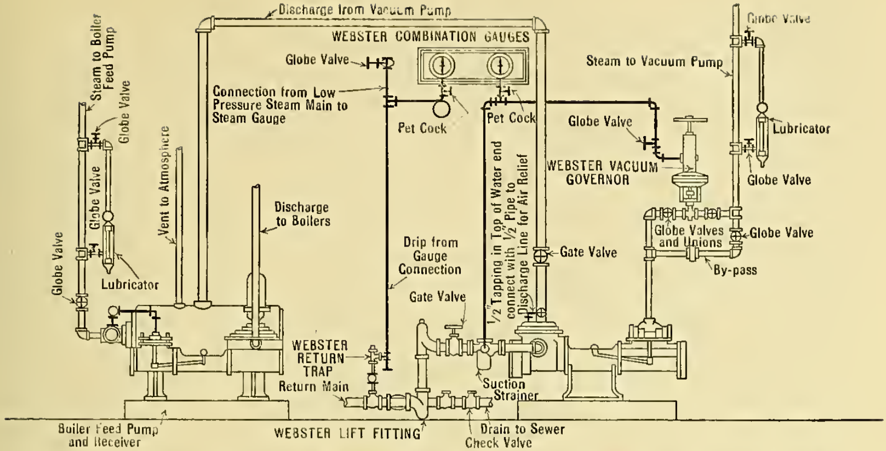


Fig. 14-7. Method of Connecting Vacuum Pump and Automatic Boiler-feed Pump and Receiver

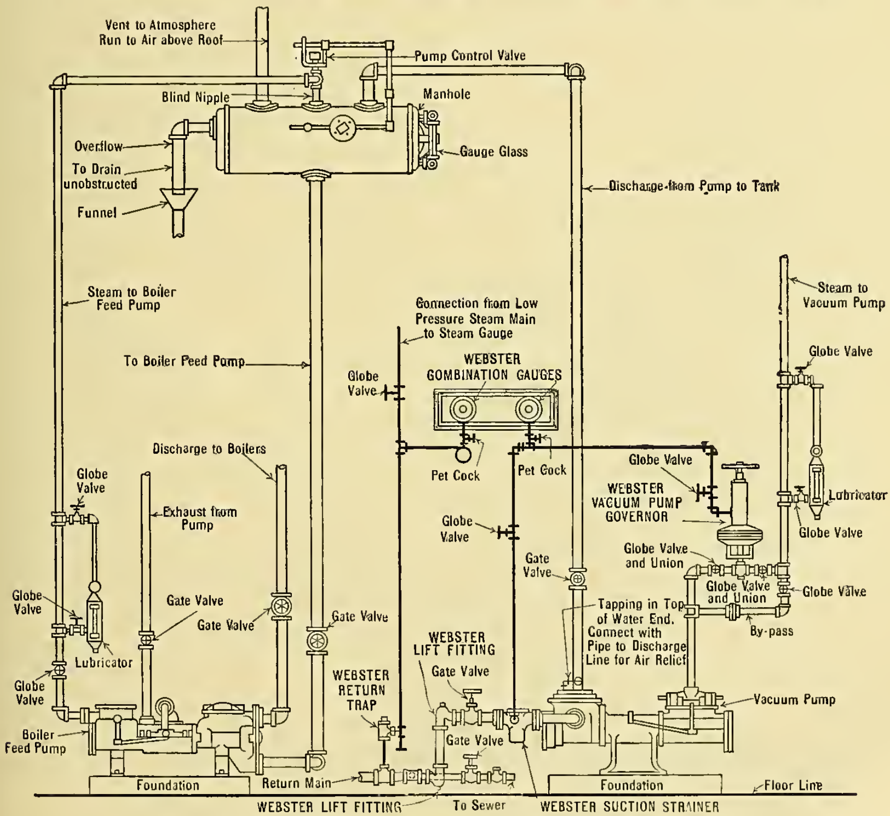


Fig. 14-8. Method of Connecting Vacuum Pump, Boiler-feed Pump and Webster Steam Control Receiving Tank

A loop seal has been found most suitable for this purpose, provided the seal is made long and contains ample volume in the vertical leg on the pressure side. A variable pressure when increasing tends to force the level of water down in the leg on the pressure side and up in the leg toward the tank. If there is not sufficient water in the loop, the water will become displaced, and the seal broken before enough of a water column has been built up in the leg from the tank. The column will then blow into the return tank and the steam or other elastic fluid will continue to blow while its pressure is above that at the tank outlet.

The fact that water in the tank is ready to seal the loop below will not avail as long as there is a difference in pressure between the tank and boiler sufficient to blow a comparatively short slug of water back into tank. The only way to restore the seal is first to equalize the pressure on both legs. A good practice is to proportion the leg on the pressure side to hold twice the contents of the pipe from the tank to the bottom of the seal.

6. *Discharge to Receiver and Boiler-feed or Tank Pump:* Where the head on the delivery side of steam-driven vacuum pumps exceeds 15 pounds, it is good practice to deliver the condensation to a vented receiver (see Figure

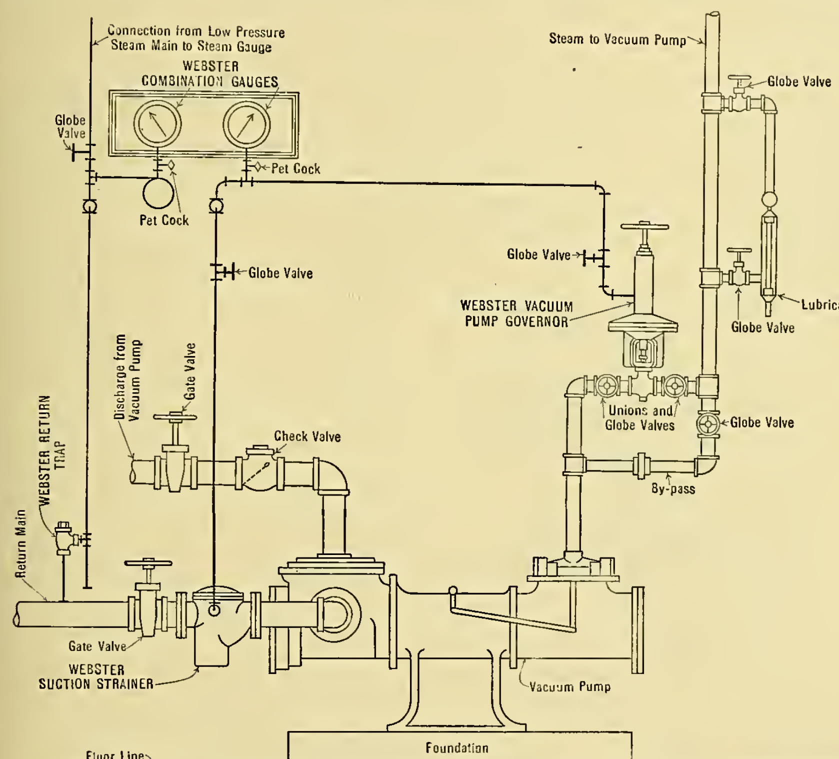


Fig. 14-9. Method of Making Connections to Steam-operated Vacuum Pump

14-7) located close to the level of the vacuum pump outlet. This receiver should be connected to a separate steam or power-driven water pump which is capable of delivering against the maximum head. (See Figure 14-8.) If this pump is steam-driven, its displacement should be controlled by a throttle valve, actuated by the water line in the receiving tank; if power-driven, the effective displacement may best be controlled by a by-pass valve between pump suction and delivery, and actuated by a float on the water line in the receiver.

7. *Discharge to Dry-vacuum Pump Receiver and Water Pump:* This combination proves very effective under conditions of high delivery head where the main return can be arranged to flow by gravity to a closed receiver, which in turn is sufficiently elevated above the location of water pump to provide a head of 2 to 3 pounds on the pump inlet valves:

The dry-vacuum pump being free from dirt and abrasive material, may have close clearance and fairly high efficiency. It may be located above and take its suction from the top of the receiver, and frequently some form of condenser may be arranged in the suction line to absorb and utilize otherwise wasted heat from the air and water vapor and at same time materially reduce the volume of vapor to be handled.

The receiver, if properly designed, forms a receptacle for the grit and impurities which would otherwise injure the water pump; and it also affords space for a float governor for controlling the water pump by the varying volume of return water.

Excessive vacuum in a receiver will cause trouble in the water pump. For this reason, a vacuum governor should always be used to control the dry-vacuum pump and to hold the vacuum well within the pre-determined limits.

Suction Strainers: The worst of the grit and dirt from condensation should be retarded and removed before entering the pump where it would score the water cylinder. Strainers (see Figure 14-9) with readily removed baskets for use on the main vacuum return line were first designed and recommended by Warren Webster & Company 23 years ago. The original Webster design with little modification has been almost universally adopted.

Vacuum Governors: In steam-driven pumps, control of displacement by the degree of vacuum maintained in the return line may be effectually accomplished by throttling the steam supply. (See Figure 14-10.) Simple forms of diaphragm-actuated throttle valves will control the degree of vacuum in the main return within sufficiently narrow limits for all practical purposes.

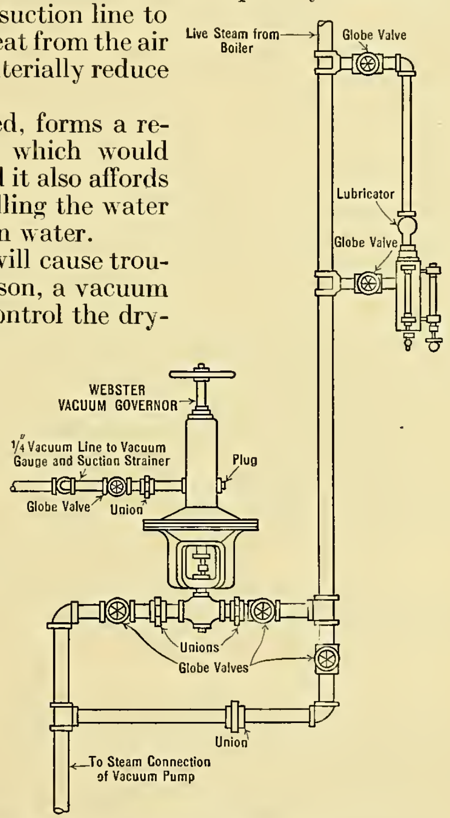


Fig. 14-10. Connections for a Webster Vacuum-pump Governor

CHAPTER XV

Applications of Webster Systems to Slashers and to Cloth and Paper Drying Apparatus

SLASHERS are used in the textile industry for sizing and drying warps or yarns before they are placed in looms to be woven into cloth. In these machines, steam is supplied usually to two cylinders, of five and seven feet diameter, over which the yarn passes to be dried after sizing.

Ordinarily the steam supply and the drainage connections are on opposite heads of the cylinders, the connections passing through the cored shafts upon which the cylinders revolve. Steam is carried through the mains to the slasher at high pressure and before it enters the cylinders is reduced to between 5 and 12 pounds per square inch by a pressure-reducing valve. The steam pressure in the cylinders must of course always be above that of the atmosphere as the rapid drying of the materials demands that the surface temperature of the cylinders shall be above the atmospheric boiling point.

Owing to the light weight of the metal used in construction of slashers, vacuum breakers, usually three in number, are provided in the head of the discharge side of each cylinder. These open when a partial vacuum occurs in the cylinder and prevent collapse of same.

The condensation is raised to its point of removal from the slasher by means of troughs or buckets, usually three in number, attached to the inside cylindrical surface. A pipe attached to each bucket carries the condensation to the hollow cylinder shaft and thence through the bearing to the outside. From there the condensation goes through the Webster Traps, etc., to the point of disposal.

The Webster System for draining slashers provides the most efficient drying effect with least attention to the drainage equipment. It has succeeded in overcoming entirely the frequent delays and slowing down of the manufacturing processes previously experienced with other devices.

As will be seen in Figure 15-1, each cylinder is equipped with a Webster Return Trap, a Webster Dirt Strainer and a bull's-eye sight glass.

The Webster Return Trap permits the free passage of air and water and closes against the discharge of steam. The Webster Dirt Strainer protects the trap from dirt and the sight glass enables the operator to see whether or not the drainage system is functioning.

A by-pass is provided around the drainage apparatus. When starting up, the by-pass may be opened for a few minutes to permit the quick discharge of air. After starting the slasher is drained automatically through the Webster equipment.

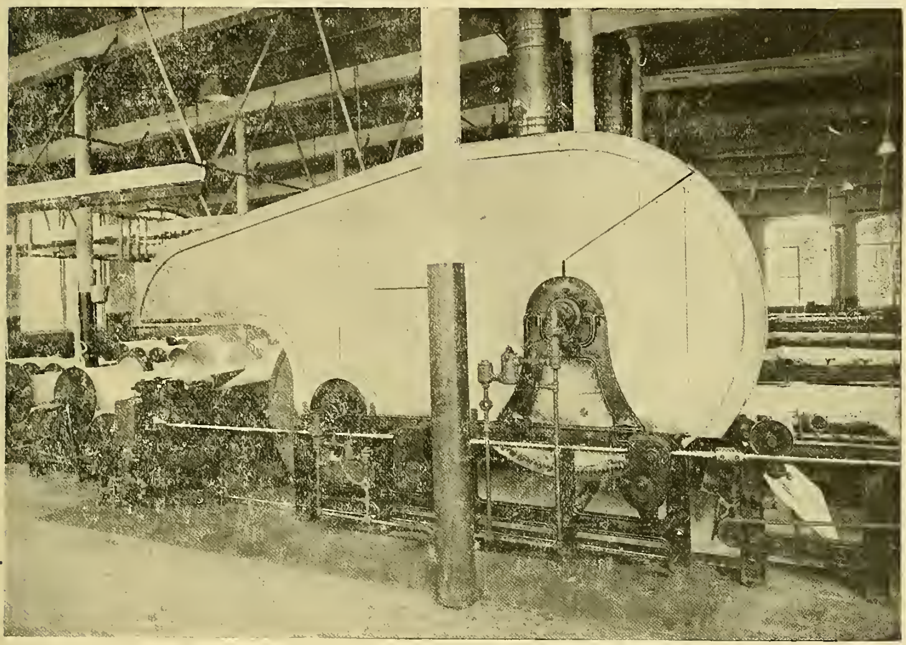
A pressure sufficiently above that of the atmosphere must be carried in the cylinder to dry the goods and this is sufficient to discharge the condensation and air through the Webster Trap, if free vent to atmosphere is maintained. There is no advantage in connecting the discharge of the traps to a vacuum pump if sufficient vertical distance is available to allow a proper fall for the condensate to flow by gravity to an open receptacle.

The condensation rate with this type of slasher will vary from 400 to 600 pounds per hour.

One of the best-known American manufacturers of slashers states in his catalog:

"We strongly recommend the use of Warren Webster & Co.'s apparatus for slasher drainage.

"Steam traps can be furnished if desired but we recommend the use of the Webster System in preference, as higher economy will certainly maintain a higher rate of production and its simplicity lessens the liability of stoppage to which a system of steam traps is apt to be subject after a few years of use.



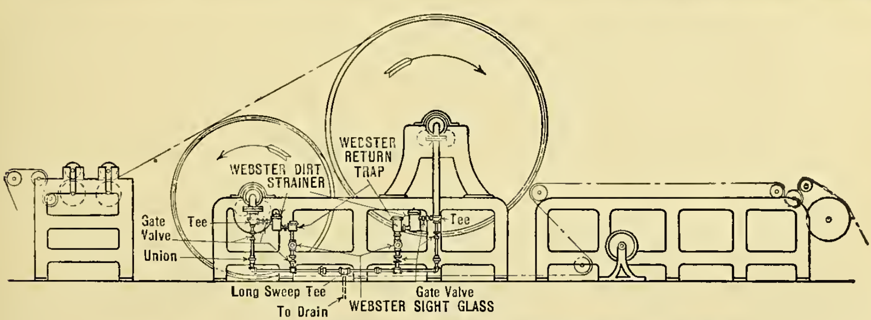


Fig. 15-1. Typical application of Webster Vacuum System to a slasher.

" The Webster System as compared with a steam-trap system insures steady, instead of intermittent, drainage and practically an entire absence of condensation in the cylinder with all consequent advantages."

Cloth and Warp Drying Machines—Except in details, the process of draining drying machines of both vertical and horizontal types (shown in Fig. 15-2) is the same as for slashers.

Each cylinder is provided with troughs or buckets which, as the cylinder revolves, empty through a pipe to a hollow shaft and through the journal to the return duct.

The housings of the machine and the brackets supporting the cylinders are cored to provide ducts for conveying steam to the cylinders and con-

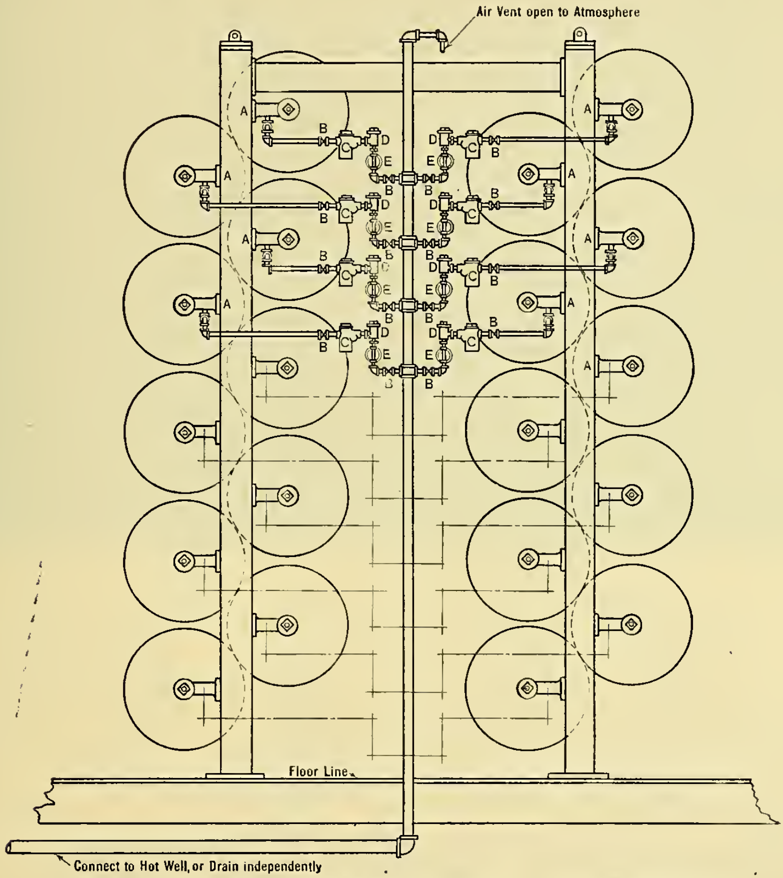


Fig. 15-2. Application of the Webster System to a vertical drying machine.

"A" Solid copper gasket inserted between bracket and housing. A copper gasket having hole equal in area to that in the bracket must also be placed between the bracket and housing on the inlet side to keep cylinder alignment true. "B" Gate Valve. "C" Webster Dirt Strainer. "D" Webster Return Trap. "E" Webster Bull's-eye Sight Glass.

densation away from them. The frame on one side acts as a supply pipe while that on the other side acts as a return. Steam at a pressure of 15 pounds per square inch or less is admitted to the housing and passes through the brackets and the journals to the cylinders. To prevent collapse, vacuum breakers are installed in the cylinder heads, usually on the discharge end.

Frequently it is advisable to make two or three separate steam supply connections to each housing, as the area of the cored opening in housing is

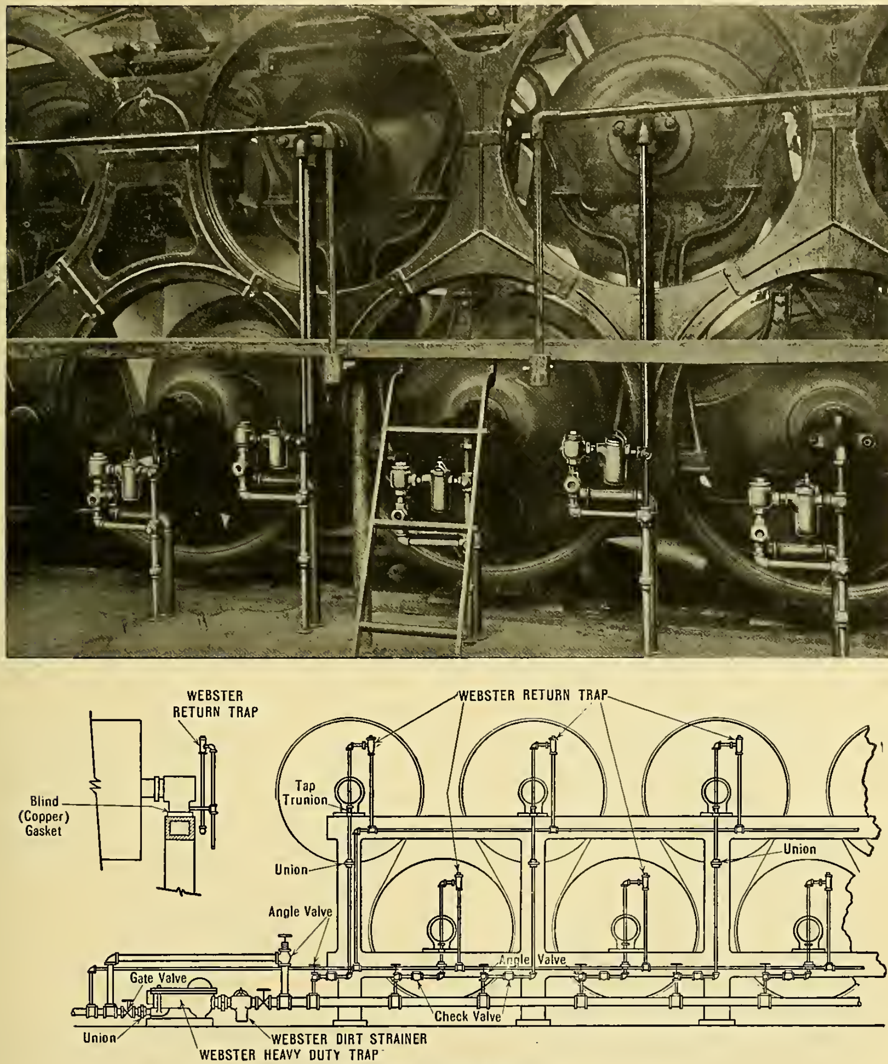


Fig. 15-3. Application of the Webster System to paper machine.

too small to convey the required amount of steam without too great a pressure drop.

The duct in the housing through which the products of condensation pass can best be drained by the use of one or more Webster Heavy-duty Traps provided with thermostatically controlled air by-pass.

Paper Machines—Two types of machines of particular interest are used in the manufacture of paper—Cylinder Machines and Fourdrinier Machines. Both require the evaporation of large quantities of water from the paper after the pulp has been pressed and the web has formed.

After passing through the presses the paper usually contains about 45 per cent of water. This moisture is reduced to about 5 per cent, depending upon the thickness of sheet and the finish desired, by passing the paper over a series of drying cylinders, the inside surfaces of which are heated by either exhaust or live steam at low pressure or a combination of the two.

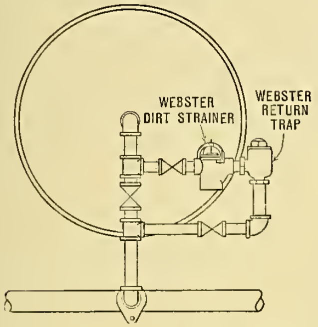


Fig. 15-4. Method of draining cylinder of a paper machine using Webster Return Trap and Webster Dirt Strainer. These connections are suitable for operation with either vacuum or gravity discharge.

Usually the steam-supply header runs parallel with the machine, close to the floor, a hole being bored in the header and connected by a pipe to the cored journal on the cylinder.

The return header runs either above or below the steam header and has the same kind of connections as the supply.

The drying cylinders vary in size and length. For the purpose of removing the water, one type of cylinder is equipped with buckets and another with what is termed a siphon pipe. Cylinders equipped with buckets discharge the condensation only when in motion, while those equipped with

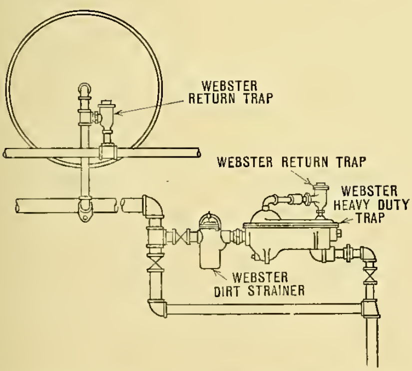


Fig 15-5. Method of draining cylinder of a paper machine using Webster Heavy-duty Trap and Webster Dirt Strainer and a Webster Return Trap for air vent discharging into dry returns.

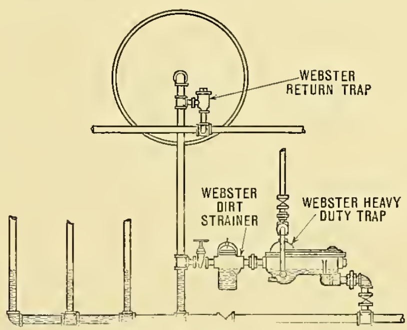


Fig. 15-6. Method of draining cylinder of paper machine for gravity discharge where a water line is to be maintained, using Webster Heavy-duty Trap with balanced steam connection, Webster Dirt Strainer and a Webster Return Trap for vent discharging into dry returns.

siphon pipes discharge whenever water accumulates, provided there is sufficient pressure in the cylinder or vacuum in the return line to give the necessary differential.

The condensation per square foot of exposed drying surface of the cylinders depends upon the speed of operation and the thickness and width of the paper on the cylinders. The stock from which the paper is made, together with the amount of water extracted by the press rolls, also has a direct bearing upon steam consumption. The condensation will average about 1½ pounds per square foot of total roll surface and naturally is greatest at the wet end of the machine.

The drainage from the cylinders may be removed either by gravity or by means of a vacuum pump, whichever is desirable.

Usually with the Webster System of drainage, a Webster Return Trap with its Webster Dirt Strainer and By-pass is provided for each cylinder as shown in Figures 15-3 and 15-4. All traps discharge into a main return which leads to the point of disposal—which is a feed water heater or hot well—open to the atmosphere for the removal of air.

Webster Heavy-duty Traps are sometimes used instead of Webster Return Traps (Figure 15-5) especially where the presence of a water line is desirable in the return (See Figure 15-6).

CHAPTER XVI

Applications of Webster Systems to Vacuum Pans and Similar Apparatus

IN processes of manufacture where boiling of the product at a low temperature is necessary or desirable, a special application of the Webster System has been devised for removing air and water of condensation.

One of the important uses for vacuum pans is in the milk-condensing industry and in the following pages this particular application of the Webster System is discussed. However, the principles and the Webster apparatus are equally applicable to other processes such, for instance, as the manufacture of sugar, salt, candy or tartaric acid.

The development and growth of the milk industry has reached a point in the last few years where it is now necessary, due to keen competition, to use not only the most modern and efficient machinery in the process of milk manufacture, but to install modern power equipment and a perfect system of steam circulation in order to insure the commercial efficiency of the plant.

It is essential that each pound of steam (live or exhaust) shall do the maximum of useful work and that all water of condensation shall be returned to the boiler.

There are numerous uses for exhaust steam in the modern condensory, such as heating of boiler feed water, heating of water for general use and in the heating system of the building, but as a rule these require only a small portion of the amount of steam available from the exhausts of the engine, compressors, pumps, etc.

In a condensory of say 100,000 lbs. capacity of milk daily, there will be available at least 200 h.p. of exhaust steam, not over 20% of which is required for any of the above uses. The remaining 160 h.p. of exhaust steam is available for use in the vacuum pans.

The usual practice in the past has been to use live steam in the heating coils of the vacuum pan at a pressure of about 15 to 20 lb. gauge, reducing to this pressure from the high-pressure mains. Very often excess exhaust steam from the engines has been wasted to the atmosphere, being considered a by-product of the engine room with little value excepting for its uses in the boiler room. Exhaust steam at 5-lb. gauge pressure contains about 88 per cent of the heat content of the live steam used to develop power and is just as effective in the coils of the vacuum pan as live steam reduced to the same pressure.

To make use of exhaust steam at 5-lb. gauge pressure where live steam was used in the vacuum-pan coils, only slight changes are necessary. Occasionally the sizes of coil connections must be increased to the size of the coils themselves and where the steam pressure is decreased, a slight additional amount of heating surface in the coils will be required on account of the lower temperature of the steam at this pressure. In some plants where exhaust steam has been substituted for live steam without changes in the heating surfaces, a slight additional time was required to condense the batch of milk. In most cases this increase was not more than ten minutes.

The usual control valve connections, that is, the double globe valve and a gauge attached to each coil connection, will be the same for use with the exhaust steam as with the live steam.

The return connections for use with the exhaust steam are very simple. A single Webster High Differential Heavy-duty Trap (see page [redacted] Chapter XVI) with a by-pass is connected to each coil outlet. These traps discharge to the return main leading to a vacuum pump in the boiler room. It is essential that each coil shall be drained separately into the vacuum return main in order that the pan operator may have absolute control of the steam pressure in each individual coil.

It is necessary when condensing milk to vary the pressure in these coils at will. In some instances the pressure in certain coils must be reduced to atmosphere, while the pressure in other coils is increased to as high as 5 lbs. per square inch in order to cause a positive circulation of milk within the pan. Without this positive control of circulation it is impossible for the pan operator to properly manipulate the process.

It is also imperative that the water and air of condensation shall be removed immediately from the coils of the vacuum pan and that this shall be accomplished independently of any conditions which may affect the operation of the general exhaust steam system in the plant.

It is advisable to use an independent pump and return line for the vacuum pans and not to depend upon other similar equipment which may be used for heating the building. The return line should have a gradual gravity pitch to the vacuum pump and should be so arranged with by-passes and valves that in case the vacuum pump should become inoperative for any reason the return condensation may be discharged by gravity. There must necessarily be no pockets of any nature in this return line.

A maintained vacuum of 6 to 8 inches at the outlet of the trap is usually sufficient to insure at all times a positive circulation of steam and the instant removal of all water and air of condensation.

Not only are much better results obtained by the certainty of this circulation, but in many cases where exhaust steam has been substituted for live steam a marked improvement in the flavor of the product has been noted.

The great saving in steam consumption in a condensory when equipped with the Webster System will usually pay for the entire installation within a few months. However, a careful analysis must be made of the existing conditions of an old plant or the requirements of a new condensory before any exact arrangement can be determined. There is no other single improvement to a condensory that will approach the saving obtainable through the economical use of exhaust steam.

Figure 16-1 shows an older type of connection for vacuum pans, in which high-pressure steam only is used. The pressure is reduced from 125 lbs. per sq. in. boiler pressure to 15 or 20 lbs. per sq. in. for use in the pan.

The outlet connections are pipes without valves or checks, leading to a header which is piped to a tank located beneath the pan. The tank is a receptacle for water and air by condensation. The air is vented through the small vent valve while the water is drained to a high-pressure positive

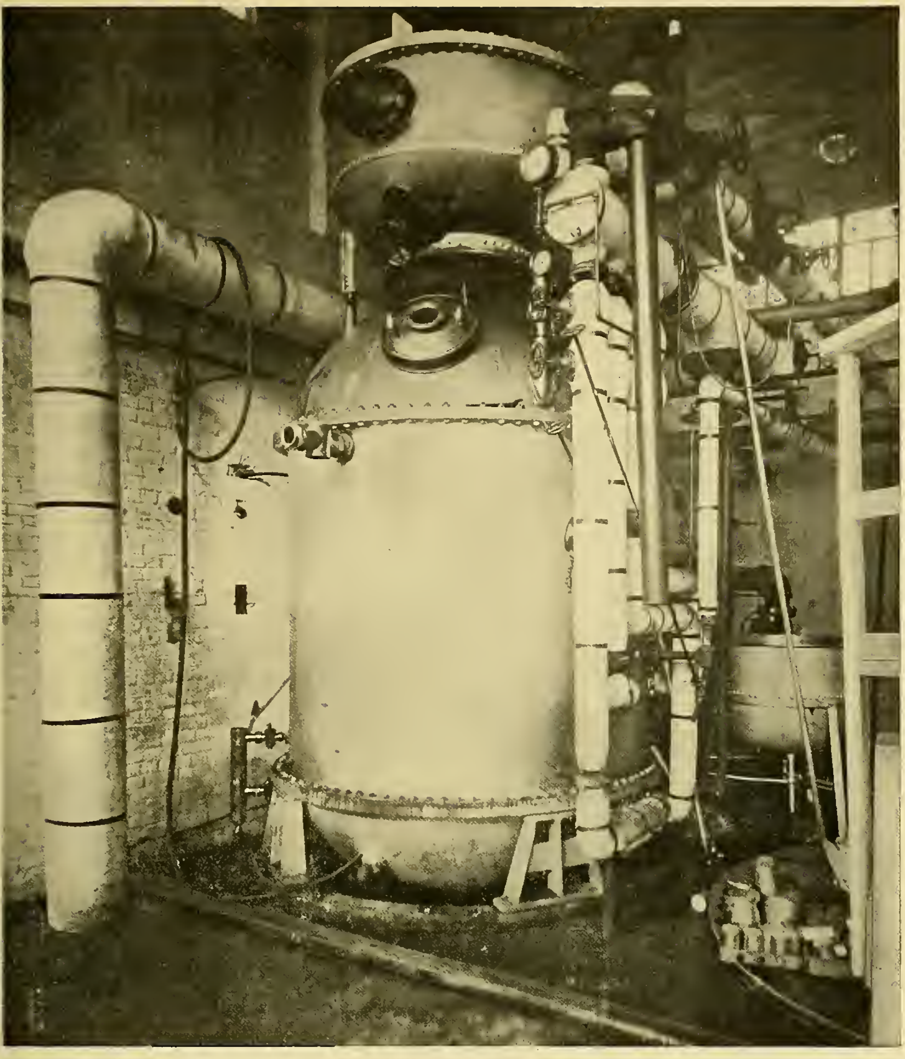


Fig. 14-1. Milk Condenser.

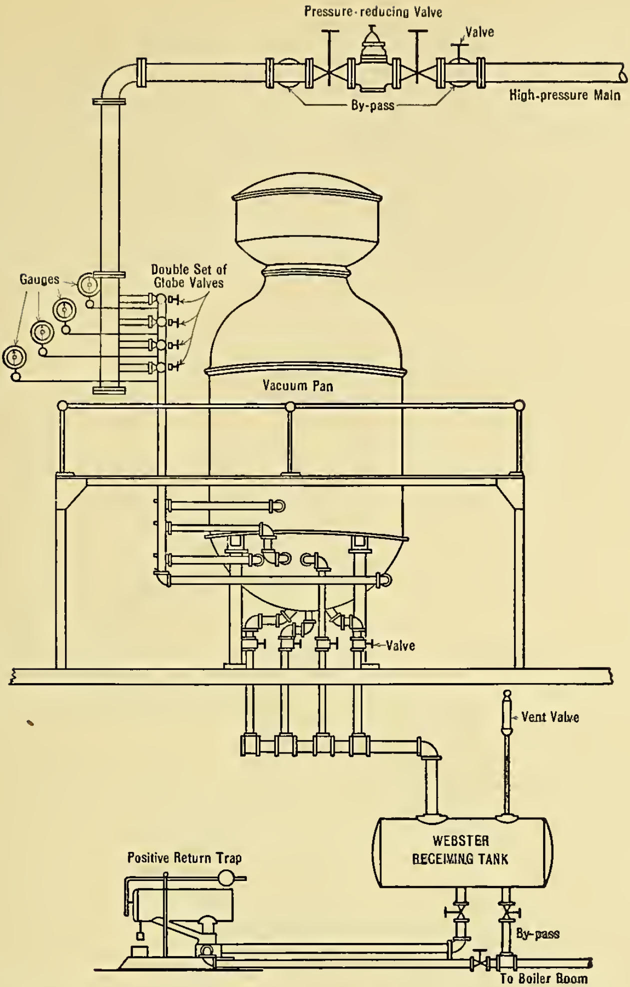


Fig. 16-1. **Drainage System for a vacuum pan using a positive return trap and receiving tank**

return trap which discharges the water to an open hot well or to a feed water heater.

The difficulties encountered in this construction will be short-circuiting of the steam from one return to the other and the impossibility of maintaining independent or separate pressure control on each coil in the pan.

The system of piping, however, is in common use in most of the smaller condensories at the present time.

Figure 16-2 shows another construction where the inlet connections are

similar to those in Figure 16-1, but where the outlet connections are controlled by means of gate valves and check valves which discharge into a common return line. This return line is run direct to a pump and receiver which discharges the water back to the boiler. A great many installations are somewhat similar to this and it is evident that there is a great deal of waste of steam due to the inability of the operator to properly throttle the controlling valves on the outlet connections.

Figure 16-3 shows the approved application of the Webster System.

The exhaust-steam piping includes a Webster Steam and Oil Separator and an auxiliary connection from the high-pressure main with pressure-reducing valve. It is essential that the pressure-reducing valve shall be of

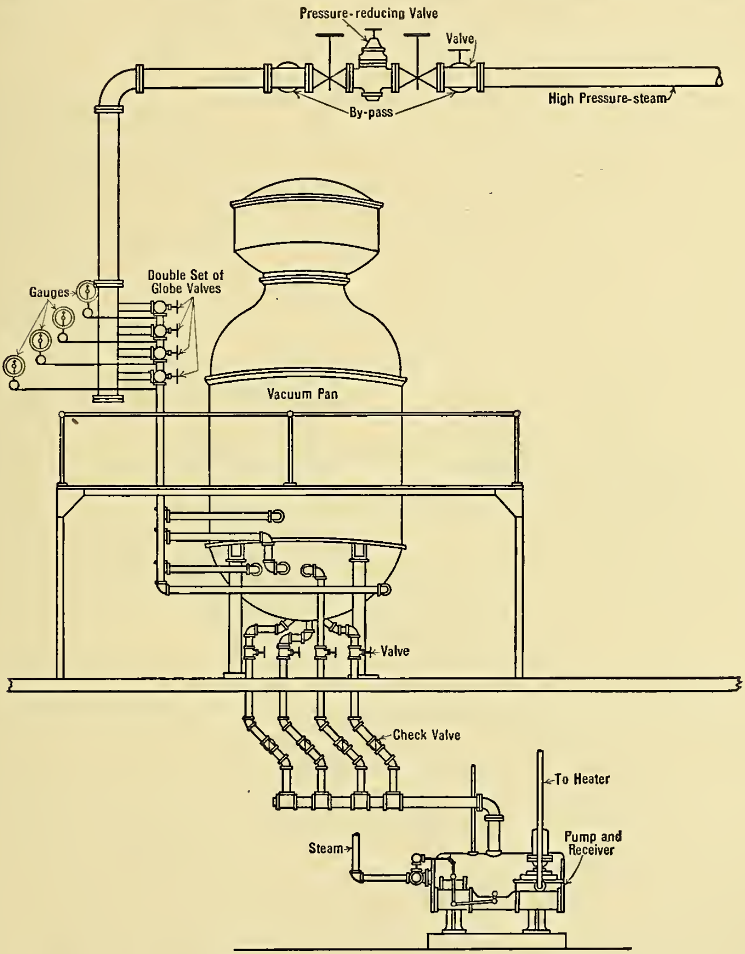


Fig. 16-2. Drainage System for a vacuum pan using a pump and receiver

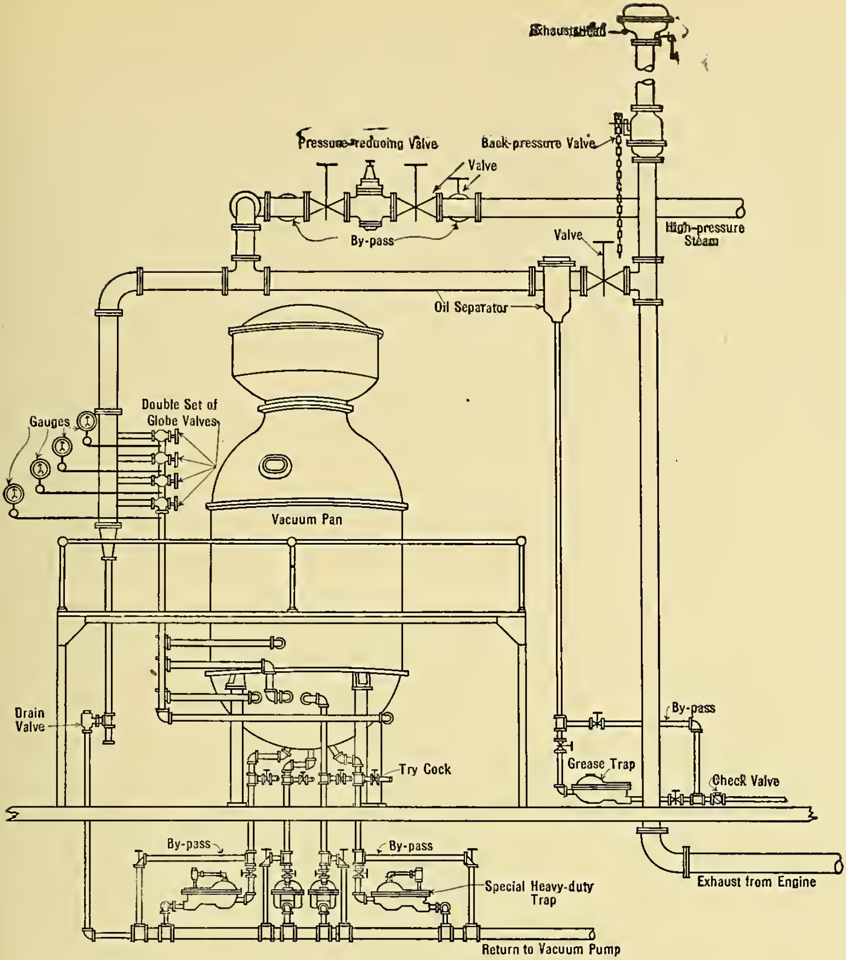


Fig. 16-3. Approved manner of applying the Webster System to a vacuum pan

such construction that it will maintain constantly the pressure which is desired when it is necessary to use live steam for condensing. The back-pressure valve must be of such construction that it is impossible at any time to exceed 10 lbs. per square inch pressure on the low-pressure mains.

The outlet connections from the vacuum pan are run direct to the Webster High Differential Heavy-duty Traps, which are provided with by-passes and thermostatically controlled air lines and are connected directly to the vacuum return line, which is run through a Webster Suction Strainer to the vacuum pump. These outlet connections also must be equipped with small try-cocks in order that the operator may test the working condition of any coil in the pan at any time.

CHAPTER XVII

Application of Webster System to Lumber and Other Kiln-Drying Problems

PROPER seasoning and drying of raw lumber is a first essential to well-finished products in any wood-working industry.

This basic condition makes the dry kiln or room a most important feature, for as proven by experience in many instances lumber that was found defective when worked would have been satisfactory if proper methods had been applied for drying. Very careful attention should therefore be given to the design of the drying room, the character of apparatus used and the heating medium employed.

The method to be employed in drying will depend entirely upon the condition of the product when put in the kiln. Green lumber, or lumber having a high percentage of moisture, will require a different method of procedure, and a longer time to dry than lumber which has been air dried. Hard woods such as oak or hard maple usually require a longer time than soft woods.

Saw mills should determine the percentage of free moisture by test and so mark each pile of lumber when first piled in the yard. Later, when it is sold, the lumber should be tested again and the two records given to the factory or other purchaser.

Factories should test and mark the lumber when first received, and if it is piled in the yard to be kiln dried later, it should be tested before going to the kiln and again before removal, these records being placed on file.

Oak or any other wood that shows 25 to 30 per cent of moisture when going into the kiln will take longer to dry than it would if it contained 15 to 20 per cent. This indicates the importance of testing the lumber before putting it into the kiln, as well as when taken out.

Kiln-dried lumber piled in storage rooms without any heat will absorb 7 to 9 per cent of moisture, and when so stored, should be tested for moisture before being manufactured. Dry storage rooms should be provided with heating coils and should be properly ventilated.

It is unusual to work through the factory lumber which has more than 5 or 6 per cent of moisture, or less than 3 per cent.

Green lumber contains a certain amount of free moisture in excess of the fibre saturation point. If the lumber is partially air dried, a considerable amount of the free moisture may have been removed. The fibre saturation point is generally about 25 to 30 per cent.

A primary test will reveal whether or not the lumber contains more than this amount, and subsequent tests at intervals of from two to three days will reveal the progress of the drying within the kiln and will inform the operator when to change the conditions within the kiln.

The process required for the drying of lumber in kilns is properly divided into four parts, as follows:

First—The primary treatment, during which all dampers are closed, 100 per cent humidity is maintained and the stock is warmed through without drying.

Second—The initial drying period, during which the conditions of temperature and humidity within the kiln are advanced sufficiently to reduce the moisture content to 25 per cent.

Third—The intermediate drying period, during which drying conditions are still more advanced to reduce the moisture content to 10 per cent.

Fourth—A final drying period, during which extreme conditions are used to further reduce the moisture content to the percentage desired.

Improper drying methods will usually result in one or more of the following conditions:

(1) Percentage of moisture not correct for working. (2) Case hardening. (3) Hollow-horning or honey-combing. (4) Molding.

The operator should make careful test readings to determine the moisture content both before and during the drying of the lumber.

Records from such tests will give data on which to base his treatment of the stock. Tests should be made at stated intervals of 48 to 72 hours during the drying period. For this purpose test boards from which samples may be taken should be inserted in the kiln. A good solid heavy piece as a sample, or better still, two or more sections out of as many different boards taken out of the pile one-third the distance from the bottom, will yield an average or representative test for moisture content. With two or more tests for moisture showing varying results, it is safer to use readings showing the highest moisture content rather than the average of the pieces.

At the same time, tests should be made for case hardening. If the lumber becomes case hardened, it practically stops the drying process, or at least slows it to a great extent. Frequently this results in hollow-horning cupping, internal strains and many other evils which affect the stock throughout the manufacturing process.

Almost all "working" which occurs in furniture, or other wood articles, is due to stresses which developed in the wood during the seasoning period. These stresses may be determined by two simple tests and eliminated before the stock leaves the kiln.

The manufacturers of the different makes of dry kilns furnish detailed instructions for the various tests on which the successful operation of their kilns depend.

The final condition of the lumber required in different factories varies with the purpose for which the lumber is used. For instance, in wagon work, many manufacturers do not use lumber containing less than 10 to 12 per cent of moisture; in auto body work, for open bodies, 6 to 8 per cent is considered proper; for closed bodies, 5 to 6 per cent. Furniture manufacturers generally dry down to 4 to 6 per cent., while wheel manufacturers dry the spokes as nearly bone dry as possible, but do not dry the felloes below 8 per cent, the theory being that when the wheel is made the spokes may absorb moisture and make a snug fit.

A modern kiln is usually constructed with brick side walls and a roof of tile or cement covered with roofing felt, tar and gravel. The doors are of special design to allow for easy loading and unloading, and to prevent, as much as possible, air leakage and loss of heat. Ventilating flues are provided in the side walls for supplying air and removing same as desired.

The heating medium usually employed is steam at varying pressures, depending upon the kiln temperature desired. The temperature within the kiln is controlled by means of a thermostat operating a valve in the pipe supplying steam to the coils.

A system of steam spray pipes is provided under the material to be dried for increasing the humidity as desired and to assist in warming the stock. The percentage of humidity in the kiln may be automatically controlled by means of a humidistat operating a valve controlling the supply of steam to the spray pipes.

Where steam, whether exhaust from engines and auxiliaries, or taken direct from the boilers, is used as a heating medium, the success of the drying equipment depends upon the manner of carrying this steam to the heating units, the proper drainage of the supply mains, the circulation of the steam through the heating units and the removal of air and water of condensation.

All manufactures of drying equipment utilizing steam as a heating medium recognize the importance of these features. One of the largest manufacturers of drying equipment in the United States says in its book of instructions:

"Where troubles have been experienced, investigations have shown that they are generally due to one or more of the following conditions:

"Poor steam service.

"Pressure not constant.

"Wet steam due to improper condensation drainage.

"Insufficient steam pressure.

"Poor drainage from traps.

"Improper design of supply and drainage piping.

"Traps allowing steam to blow through into the main drainage line, holding back kiln drainage.

"Traps on heating units not functioning properly.

"Traps stopped with scale or dirt.

"Trouble is often caused by faulty design in making steam connections to kilns.

"All steam lines must pitch in the direction of steam flow. Automatic drain traps must be provided at all low points on these lines in order that there may be absolutely no condensation lying in the lines at these places, and that steam may enter the kiln dry and at a high temperature. Failure to provide proper methods of drainage will result in reduced volume and temperature of steam and correspondingly low temperatures and poor service in dry kilns."

The important features in connection with the steam supply and drainage system can be enumerated as follows.

(1) Pressure of steam supply.
(2) Manner of conveying steam to coils.
(3) Method of draining main steam supply.
(4) Character of design of heating units.
(5) Method of air removal from heating units.
(6) Method of removal of condensation from heating units.

(7) System of drainage piping.

(8) Ultimate disposal of water of condensation.

Items one, seven and eight will be governed materially by the conditions existing at the plant where kilns are to be used, and as these conditions vary with the character of the plant, this discussion will be limited to the requirements of the kiln only.

The pressure of steam supply, so far as the operation of the kiln is concerned, will depend upon the temperature required within the kiln. If a maximum kiln temperature of not more than 160 deg. fahr. is required, satisfactory results can be obtained by the use of exhaust steam from engines and auxiliaries at a pressure not to exceed 1½ lb. gauge pressure. The same results will be obtained, of course, by using steam direct from the boiler reduced to a corresponding pressure by means of reducing valves.

Where temperatures greater than 160 deg. fahr. are required it will be necessary to increase the pressure of the steam accordingly. In good practice the temperature of the steam must not be less than 60 degrees higher than the temperature desired in the kiln.

The size of the steam supply mains will depend upon the volume of steam to be delivered, and the drop in pressure allowable. This may be determined with the help of the tables in Chapter XI in this book after a decision has been reached as to the total heat requirements of the kiln and the distance of the kiln from the source of steam supply. The same principles apply for the installation of steam mains to the kilns as would apply for the installation of steam mains for any other purpose.

Extreme care should be given to the drainage of the steam main at the point of entrance to the kiln. It is advisable that water of condensation from the main shall be relieved from the bottom into the return and that steam for kilns shall be taken from the top of the main rather than to allow the condensation to drain through the coils. The supply main may enter the kiln from a point above the coils used for heating, or from a point below them. The manner of providing for the drainage of these mains is shown by Figures 17-1 and 17-2.

Manufacturers of drying equipment have devised numerous types of heating units but practically all have standardized on those constructed of pipe. The coils are placed either vertically along the side walls of kiln, or horizontally in a space provided underneath the material to be dried. In the latter instance they are usually installed in a horizontal position, although some manufacturers prefer coils placed vertically.

Naturally, the problem of removal of air and condensation is not so great where small units are used. The advantage of more equal heat distribution is claimed for the large unit laid horizontally, but this is not fully realized unless the removal of air and condensation is complete. Practical experience has demonstrated that incomplete removal of air and condensation has not only caused unequal heat distribution throughout the kiln, but a drop in temperature of from 25 to 50 per cent.

The types of heating units which are universally used and the manner of applying the Webster specialties for proper air removal and drainage of condensation are shown in Figures 17-3 and 17-5. These sketches and de-

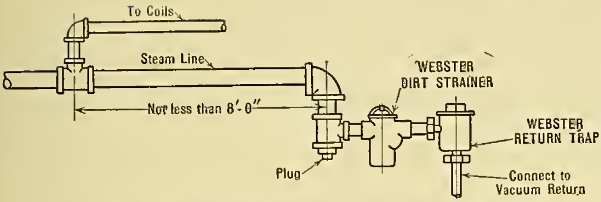


Fig. 17-2. Method of draining underground main

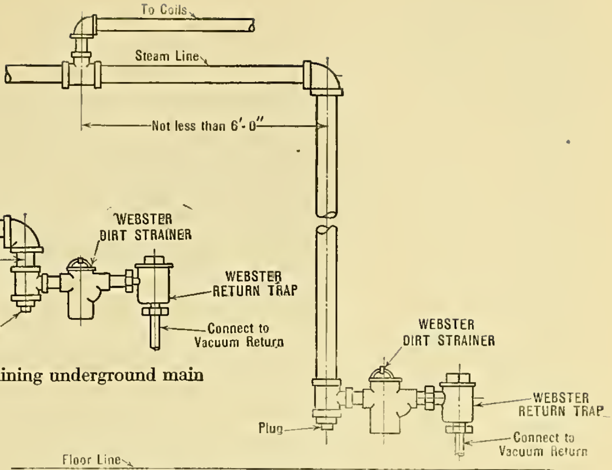


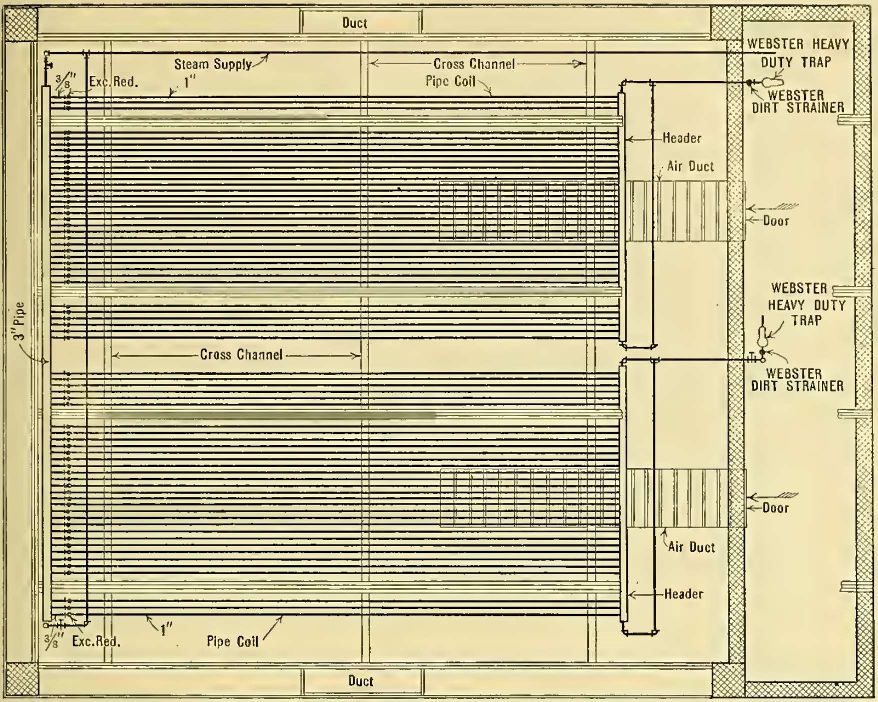
Fig. 17-1. Method of draining end of overhead main

tails are self-explanatory, but attention is called to the necessity of providing dirt strainers to each drain connection to coils. These strainers should be of easy access for cleaning, as the temperature in kiln is usually high and disagreeable to work in for any length of time.

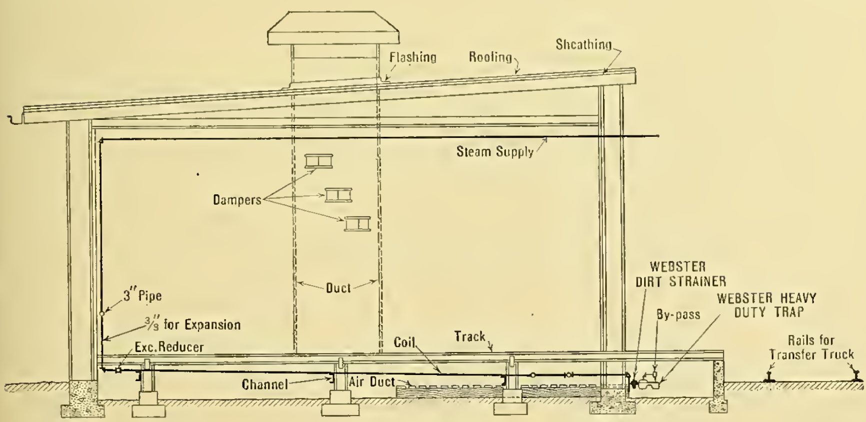
Attention is called to the location of the drain traps. These should be readily accessible also. Where thermostatic traps are used they should be located where they will not be subject to the high temperatures of the kiln. This is usually accomplished by extending drain connections to the extreme front or rear of the kiln and placing the traps near the floor.

On small units as shown in Figure 17-7, where thermostatic traps are used, additional provision for the removal of air is unnecessary, but where a large volume of condensation accumulates, additional provision for air removal is essential and heavy-duty traps should be used. Where the heating unit is of the continuous-header type, as shown in Figure 17-4, the air removal can be accomplishd by the use of a heavy duty trap, equipped with a thermostatically actuated air by-pass within the trap. Where it is desired to drain the condensation from two or more coils to one heavy-duty trap, or where the return header of the coils is of special construction divided into two or more sections, and the condensation from all sections is drained by one trap it is essential for the proper removal of air to equip each return header, or each section of the return header, with a thermostatically actuated return trap. The outlets of these traps are connected into the main vacuum return line beyond the discharge connection of the heavy-duty trap, as shown by Figure 17-10.

The discharge from all heavy-duty traps and thermostatically actuated return traps used in connection with kilns may be connected into a common return line, but it is preferable that this return line from kilns shall be extended independently from the kilns to the vacuum pump, rather than to connect it into returns from the heating system of the manufacturing plant



Plan



Elevation

Fig. 17-3. Sections thru a typical Dry Kiln with coils of the continuous-header type using Webster Heavy-duty Traps for drainage

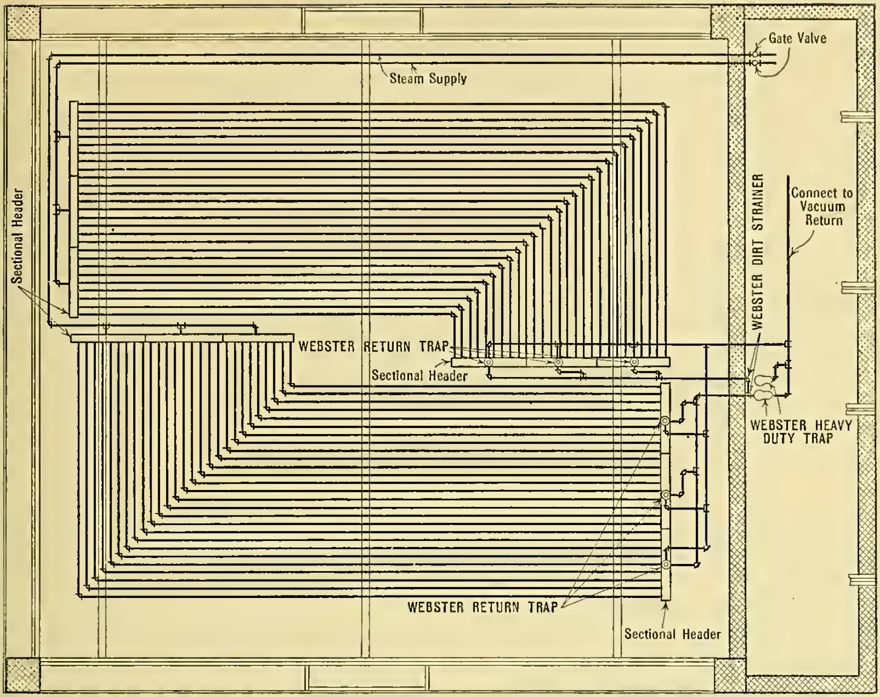


Fig. 17-5. Typical section through a Dry Kiln using coils of the sectional-header type

Fig. 17-4. Method of draining coil of continuous-header type with drain connections on each end of header. Detail of connection at discharge ends of header. (See Fig. 17-3)

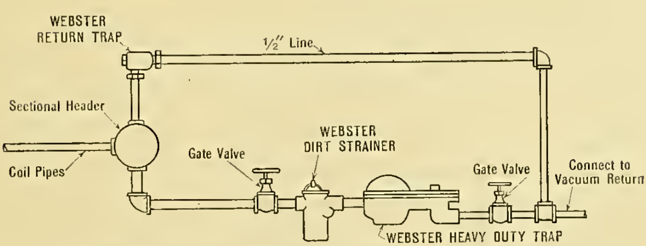
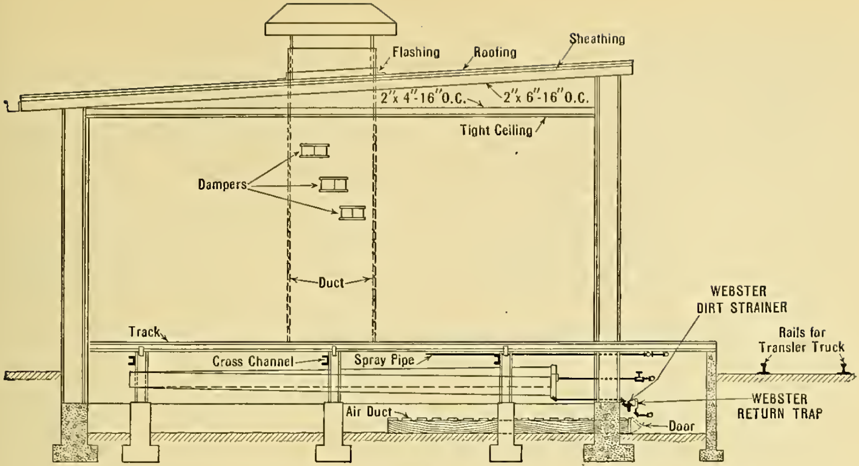
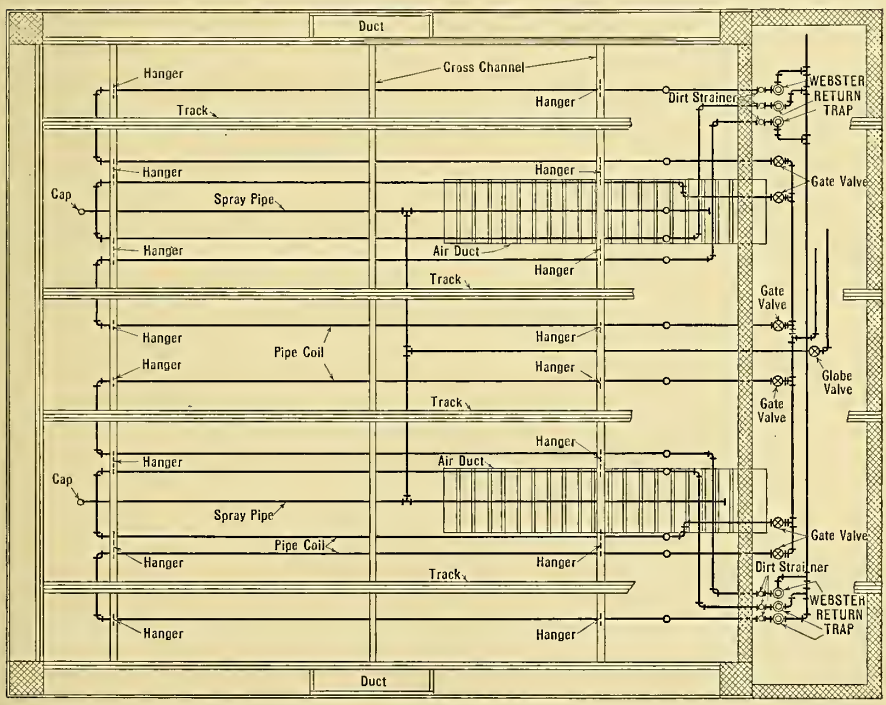


Fig. 17-6. Connections around Webster Heavy-duty Trap Draining Coils with sectional headers. (See Fig. 17-5)



Elevation



Plan

Fig. 17-7. Sectional drawings of a typical small Dry Kiln using individual traps for drainage of coils

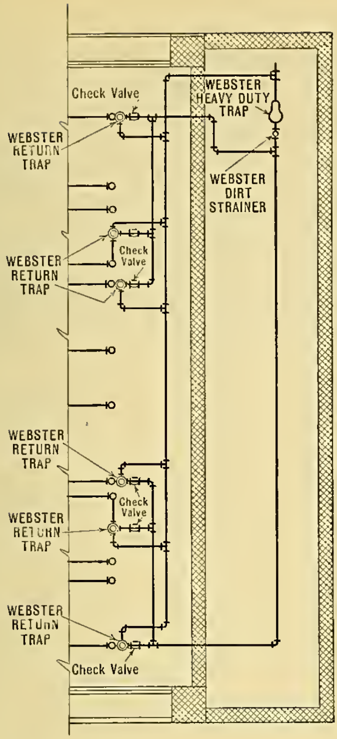


Fig. 17-8. Showing the connections where two or more coils are drained through one Webster Heavy-duty Trap

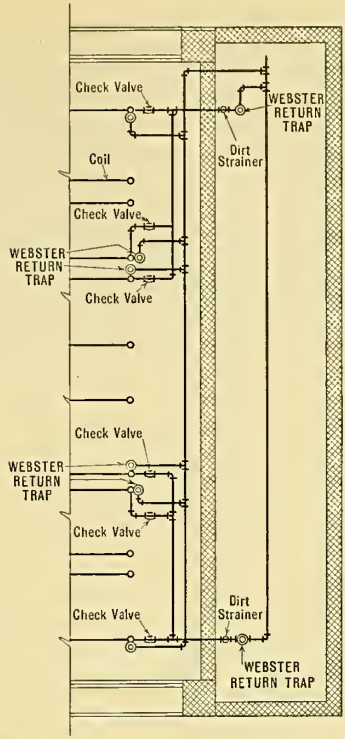


Fig. 17-9. Showing the connections where two or more coils are drained through Webster Return Traps

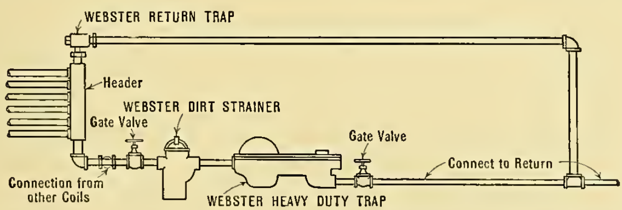


Fig. 17-10. Detail of Connections around Webster Heavy-duty Trap (See Fig. 17-8)

or other equipment. The condensation rate from the kilns will fluctuate, depending upon the temperature within the kiln, the nature and condition of the product being dried and the outside temperature. Consequently, at times when the air removal and condensation rate from the kilns is high, trouble may be experienced with the operation of other equipment if connected to the same return line. Also, if the same efficient equipment is not used in connection with the heating system or other equipment, as used in

connection with the kilns, the poor operation of the heating system or other equipment will naturally reflect in unsatisfactory operation of the kilns.

The amount and location of radiation installed within the kiln will depend upon the location of the kiln, the temperature desired within the kiln, the steam pressure, and nature of product to be dried. This constitutes a special branch of engineering and engineers thoroughly familiar with this class of work should be consulted.

The method for figuring the total radiation required by a given dry kiln will not vary from the descriptions given in detail in Chapter V, except that during the warming-up period an additional heat factor is required to care for the moisture content of the lumber or other material being dried.

CHAPTER XVIII

Application of the Webster System to Railroad Terminals and Steamship Piers

THERE are many uses for thermostatically actuated return traps where the pressures carried are greater than in heating-system work. Instances involving operation under steam gauge pressures of from 15 to 100 pounds are described in this and following chapters.

The requirement, in all cases, is that the return trap shall discharge the water and air of condensation without waste of steam and that the fixture being heated shall be maintained at maximum efficiency.

In these special installations, certain fundamentals must be observed to secure successful operation. The first requires that the thermostatically actuated traps must discharge directly to the atmosphere or to a return line in which atmospheric pressure is maintained.

This latter condition may be obtained by venting the return line free to the atmosphere. In some cases the same result is secured by discharging the returns into a flash tank, the vent of which is connected to the low-pressure heating main, while the condensation is cared for through the usual type of return traps to the vacuum return.

Railroad Terminals—One of the greatest causes of delay in the daily movement of hundreds of trains into and out of terminals where there is freezing weather is the difficulty in keeping the switches clear of snow and ice.

Many terminals have therefore adopted the method (Figure 18-1) of placing steam-heating coils between the ties, under the switches. Due to the unusual exposure, these coils and their supply lines are operated under 60 to 80 pounds gauge pressure in order to prevent freezing. The dripping of these lines and coils presents a double problem: First, the water and air

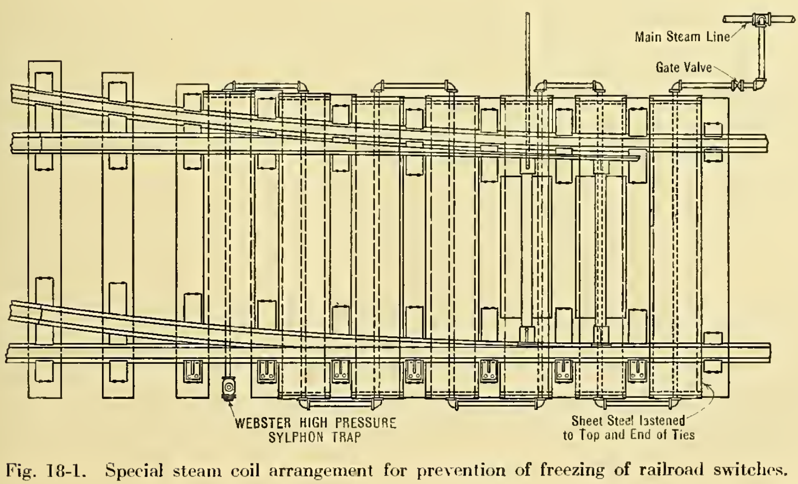


Fig. 18-1. Special steam coil arrangement for prevention of freezing of railroad switches.

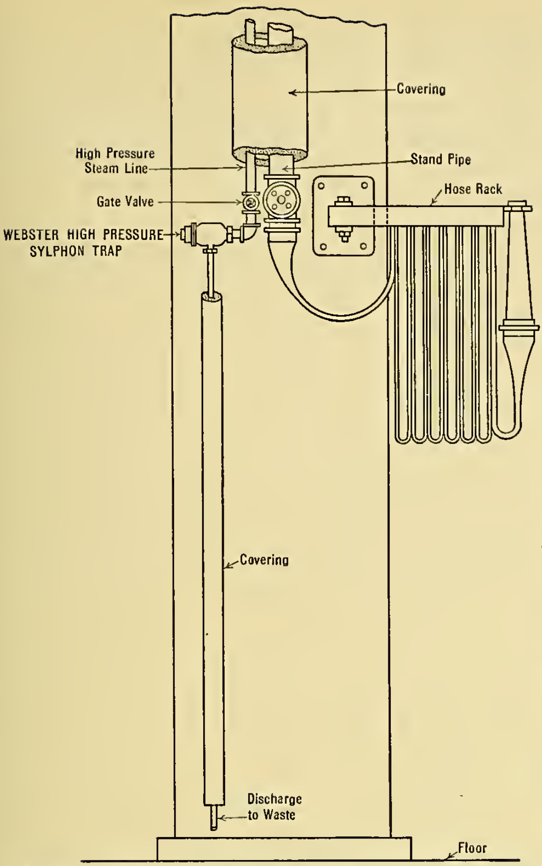


Fig. 18-2. Method of prevention of freezing for fire protection lines. The water and steam pipes are encased in the same insulation and the steam pipe is drained by a thermostatic return trap

of condensation must be freely discharged onto the roadbed, and Second, the condensation must not form steam clouds that might obscure nearby switch signals.

A type of thermostatically actuated return trap which answers these requirements has been developed by Warren Webster & Company after many tests and experiments. This return trap is fitted with Monel-metal seats and valve pieces to withstand the wire-drawing effects of steam at high pressure differential. The thermostatic member is placed on the outboard or atmospheric side of the trap, and as the trap is generally placed in the rock ballast of the road bed, its exterior is usually given a special finish to give it protection against the elements.

Railroad terminals are also equipped with extensive systems of water lines for fire protection purposes and these lines, too, must be kept from freezing. The method of prevention (Figure 18-2) found most satisfactory is to run a steam line, carrying from 60 to 80 pounds gauge pressure, parallel with and close enough to each water line that both steam and water lines can be encased in the same insulating covering. Where the water lines terminate, as at hydrant valve outlets, the same dripping of the steam lines and the same thorough removal of condensation with absence of steam cloud are required as with the yard switches.

The same type of return trap is used in both cases.

Steamship Piers—Steamship piers in cold climates are somewhat similar to railroad terminals in that the fire lines must be protected. In addition, heat is required for a large number of small enclosures scattered throughout for housing the pier clerks.

Piers are so constructed that the water of condensation from the coils heating the water lines and the clerk houses cannot be easily returned.

The practice is to discharge the condensation overboard through the deck of the pier. The return traps must, therefore, keep the lines clear of condensation to avoid the possibility of freezing and at the same time avoid the waste of uncondensed steam.

Webster Return Traps of similar construction to those previously described for railroad terminals are successfully used for this work.

CHAPTER XIX

Application of Webster System to Sterilizers, Cooking Kettles and Similar Apparatus

HOSPITAL EQUIPMENT—All hospital equipment, such as sterilizers for surgical instruments, bandages and dressings, blanket warmers, etc., requires steam at more than the usual heating pressures. As these fixtures are comparatively small consumers of steam, being operated at gauge pressures of 15 to 100 pounds, and as they are situated at different parts of the building, it is usual to run a special set of steam supply and return lines for them so that steam may be available at any time throughout the year.

For the purpose of insuring rapid removal of condensation and air from each fixture, a Webster Return Trap of similar construction to those described in the preceding chapter is placed on the return of each unit. The operating temperature of the thermostatic members of these traps is close to that of steam at atmospheric pressure; hence it is necessary to provide sufficient exposed piping between the fixture and the trap to allow the condensation to cool down to the operating temperature of the return trap. This exposed piping is termed cooling surface.

Each return from trap before connecting into the common discharge line of similar traps should have a check valve between the trap and the

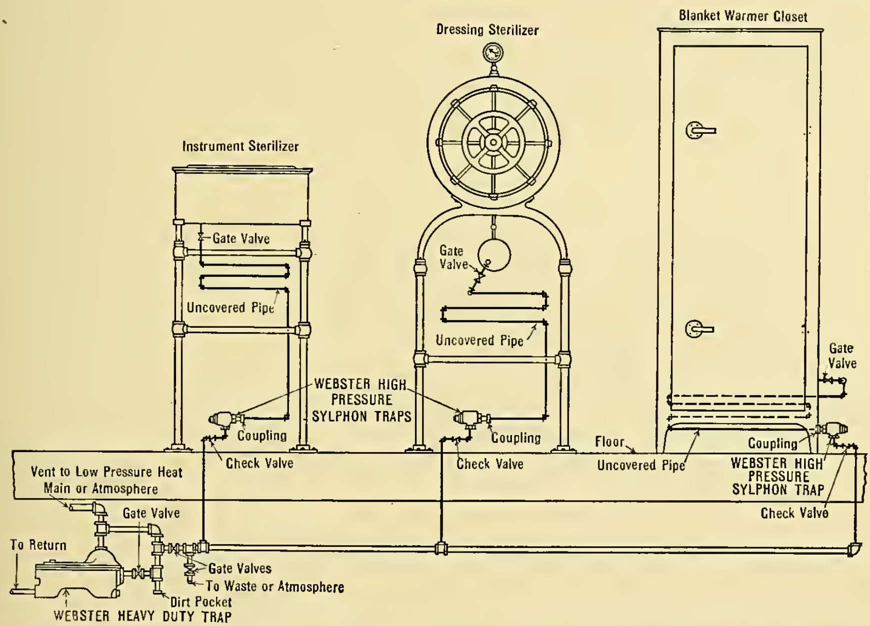


Fig. 19-1. Application of the Webster System to instrument sterilizer, dressing sterilizer and blanket warmer closet in a hospital

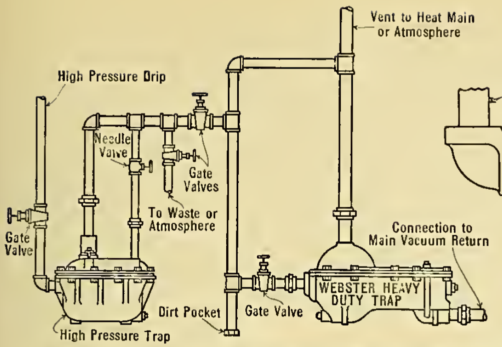


Fig. 19-3. Method of discharging high-pressure drips or returns from high-pressure apparatus into low-pressure heating mains and vacuum return mains through a Webster High-pressure Steam Trap

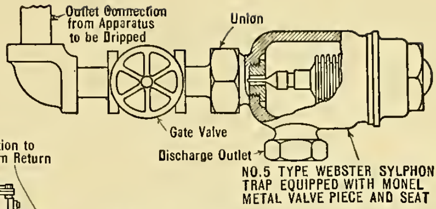


Fig. 19-2. Connections for return trap where the operating pressure exceeds 15 pounds per square inch

return, as well as a hand shut-off valve between fixture and trap as shown in Figure 19–2. As stated in Chapter 17, where a common return line is used on such service to carry condensation from several traps, it is necessary that this line shall be vented free to the atmosphere, or in cases where possible, to the low-pressure heat main (Figure 19–3). In no case should the discharge of these traps be connected directly to a vacuum return as the vacuum would unbalance the operating member of the trap and cause it to give unsatisfactory results.

Cooking Kettles, Plate Warmers, Bain-Maries, Coffee Urns and Other Kitchen Equipment—This equipment requires practically

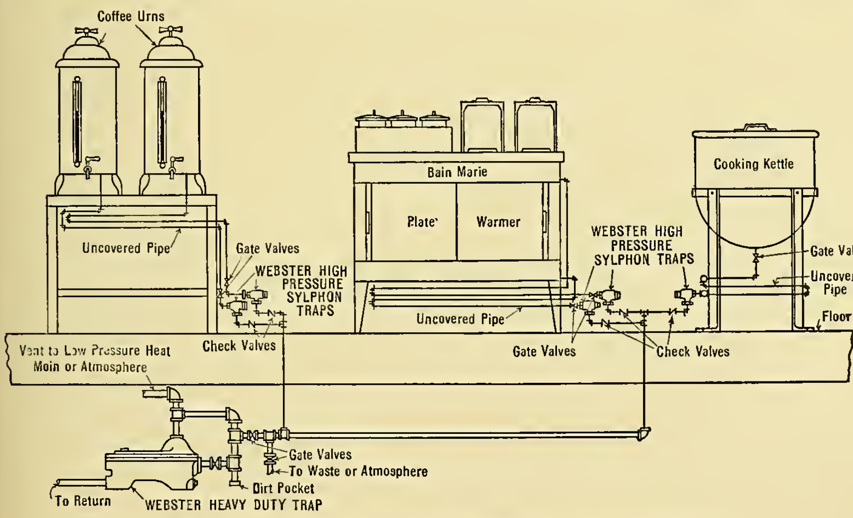


Fig. 19-4. Application of the Webster System to kitchen equipment

the same treatment as that of hospitals, and the same general statement about arrangement of return lines applies.

In food-product factories where the cooking equipment is much more extensive, a special form of Webster Float-controlled Return Trap with thermostatically actuated vent is used. This particular type is called the Webster High-differential Heavy-duty Trap. For details of these traps see Chapter 13, page 00. These traps are also used for removing the condensation and air from the steam coils of vacuum pans in evaporating processes for sugar, milk salt, tartaric acid, candy, and the like.

It is important in all applications to high-pressure duty that the maximum initial steam pressure to which the trap may be subjected does not exceed 50 lbs. gauge pressure, and that the maximum condensation rate shall be known.

CHAPTER XX

Applications of Webster Systems to Greenhouses

THE heating of greenhouses is a special field, owing to the peculiar characteristics of the buildings and the necessity for even interior temperatures.

Commercial greenhouses are more exacting in their heat requirements than are public or private conservatories. Constant maintenance of the most desirable temperatures is essential in commercial houses to bring the crop to salable maturity in the shortest possible time and to keep the quantity of first-class product at a maximum throughout the season. A single serious temperature drop for a comparatively short interval may stunt the crop beyond recovery to normal condition within a month's time, and even slight temperature variation renders some kinds of plants more susceptible to certain destructive fungi.

The heat regulation should to an extent be flexible, so that by applying more or less heat to compensate for loss of sunlight in cloudy weather the crop can be forced or retarded to come to maturity at the time when salable at the biggest profit. The blossoming of Easter lilies, for instance, requires absolute regulation to within a period of a very few days, and failure to meet the time limits results in an almost total loss. The same principle is utilized during the period of uncertain sunshine between November and February to keep the daily production of the majority of varieties of cut flowers more uniform.

Due to the high rate of heat transmission through the glass of which greenhouse enclosures are largely made, the heating system must be capable

Fig. 20-1. Conservatory of the Missouri Botanical Gardens

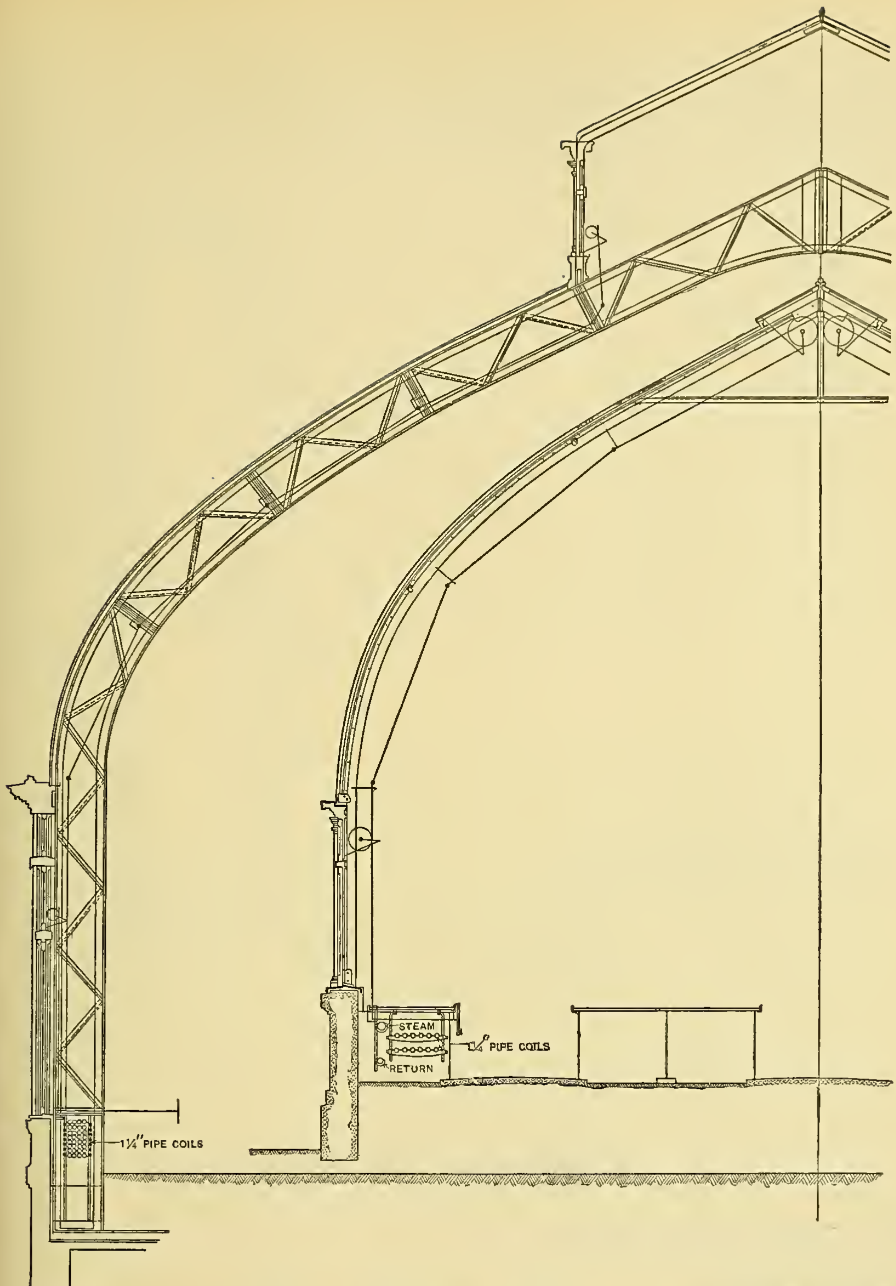


Fig. 20-2. Elevation of half of houses A and B (see Fig. 20-3), Conservatory, Missouri Botanical Gardens. Other halves of these houses are symmetrical with the parts shown

of quick response to the demands for extra heat during nights, cloudy and cold days, and particularly when a sudden cold wind springs up. The system must also be capable of assisting the ventilators by quickly reducing the heat given off by coils, etc., during the days or parts of days when the heat from the sun's rays tends to increase the interior temperature beyond the point desired.

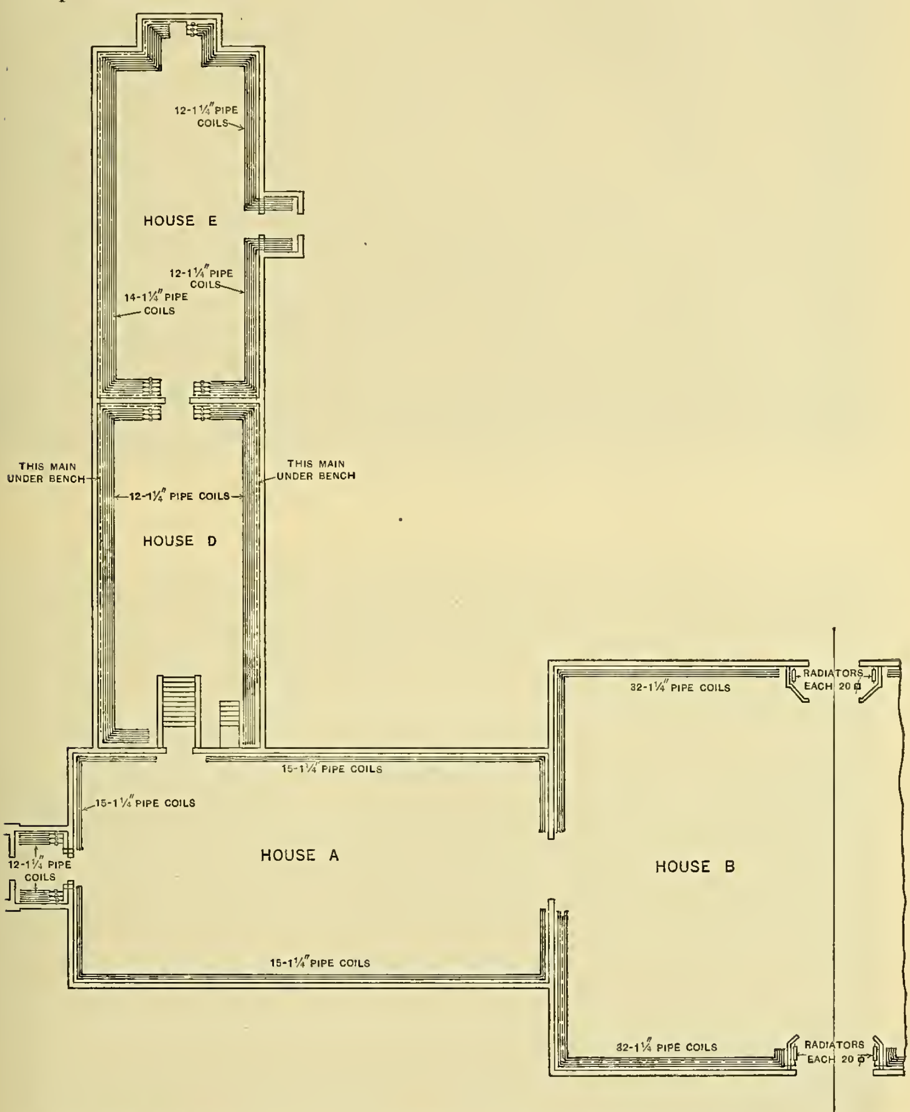


Fig. 20-3. Plan of half the Conservatory of the Missouri Botanical Gardens, showing layout of heating coils

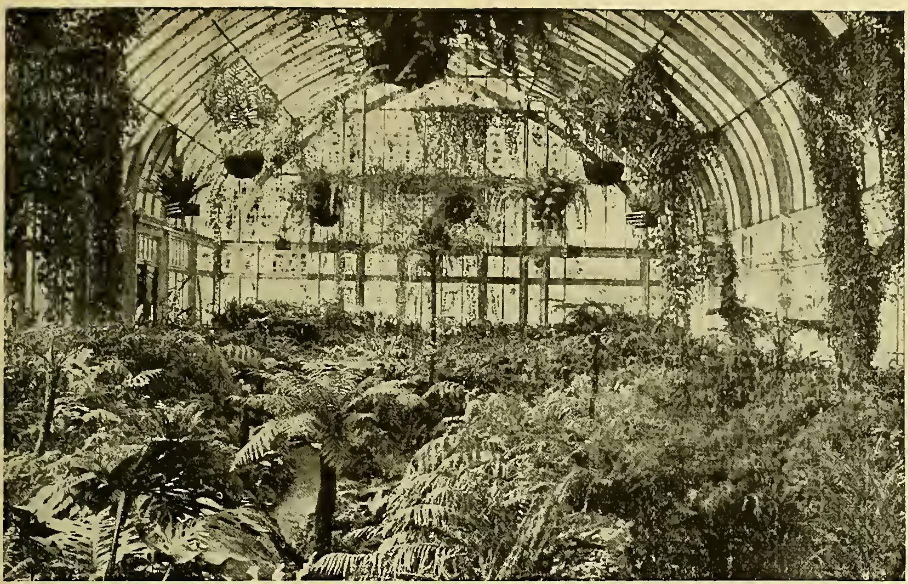


Fig. 20-4. Fern House of the Missouri Botanical Gardens

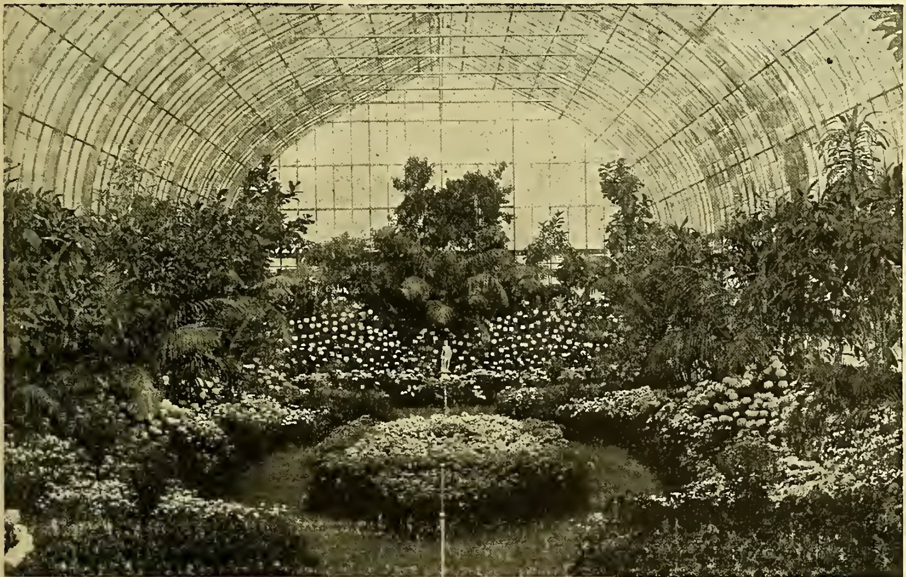


Fig. 20-5. Floral Display House of the Missouri Botannical Gardens during chrysanthemum show

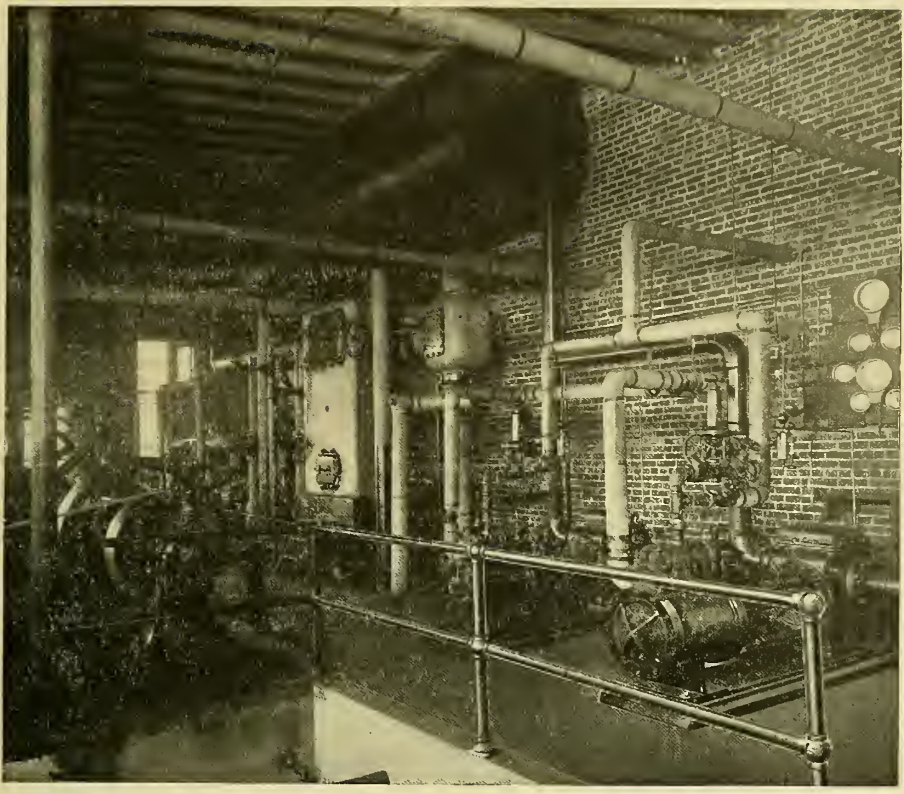


Fig. 20-8. Part of the power plant of the Davis Gardens, showing the feed-water heater and vacuum pumps of the Webster Heating System

Up to within a comparatively few years ago, hot water was considered the best medium for circulation in the heating coils of greenhouses. However, as the size and importance of greenhouses have increased, a medium with quicker response in heat flow to better meet the many changes in outside temperature and wind velocity and direction, has become necessary. Steam has proven ideal for this work when the conditions of the individual plant were understood.

The arrangement of the heating coils in different types of greenhouses varies to suit the particular plants or vegetables grown and to meet the needs of forcing, propagation, etc.

The Conservatory group of the Missouri Botanical Garden at St. Louis, Missouri, consisting of the Palm, Economic, Cycad, Succulent and Fern Houses, shown by Figures 20-1 to 20-5, are heated by the Webster Vacuum System of Steam Heating. These five greenhouses are part of the 125-acre Botanical Garden presented to the public by Mr. Henry Shaw at his death in 1889.

Fig. 20-6. One of the ten 600 by 80-ft. greenhouses of the Davis Gardens. Heated by the Webster Vacuum System

Fig. 20-7. Crosswise view at the center of one of the cucumber houses of the Davis Gardens, showing arrangement of heating coils around the beds

Eleven thousand species of plants grow in this Garden. The Palm House contains 150 kinds of palms, such as date, cocoanut, sugar, Panama and rattan. The Economic House has a variety of tropical and sub-tropical plants, such as rubber, spices, drugs, dyes and coffee. The Cycad House is arranged in Japanese style and contains representatives of all known genera of cycads, as well as a collection of tropical evergreens. The Succulent House contains species of all the plants found in the deserts of the world. The Fern House has a very complete collection of the numerous ferns and their allies.

Different atmospheric conditions are required in each of these Houses; ferns, for instance, would not live in the dry air needed by the cactus. The Webster System is maintaining the required temperatures throughout every

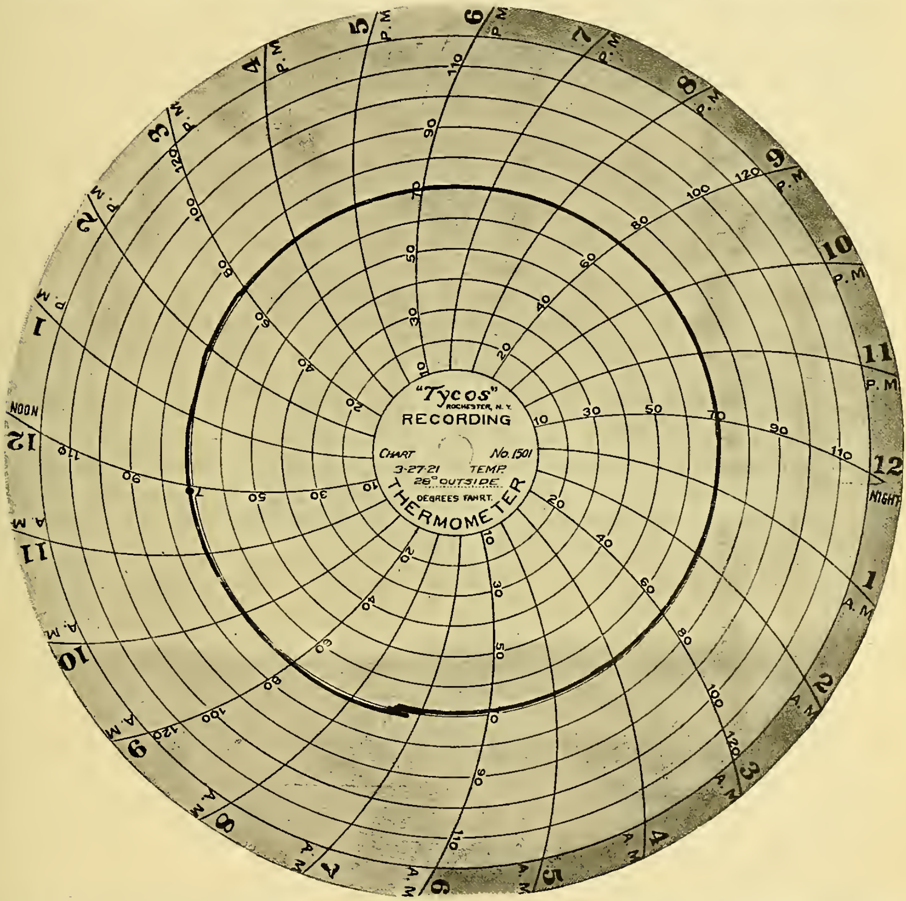


Fig. 20 9. Typical temperature chart from one of the greenhouses of the Davis Gardens, Terre Haute, Ind. The outside temperature on the day the chart was taken averaged 28 deg. fahr. The variation in inside temperature was less than 3 deg. in 24 hours

part of these conservatories, and in most locations the permissible variation in temperature is limited to five degrees. How quickly the heating system must respond to sudden outside temperature changes to keep the interior temperature within the 5-degree variation limit may be inferred from the fact that the palm house is 60 feet high from floor to peak.

The heating coils are banked on the side walls of the Houses as shown in Figure 20-2, and the arrangement of the coils is shown in plan, Figure 20-3. Steam is supplied from a central heating plant under pressure and is reduced at the Conservatory, the heating system operating at from 1 to 2 lb. gauge pressure. The returns flow to the power house, where the main vacuum pumps discharge the condensation to an open tank, from which it is pumped to the boilers.

The J. W. Davis Company, of Terre Haute, Indiana, operates the largest hothouse vegetable growing plant in the country, this plant consisting of ten greenhouses, each 600 feet long by 80 feet wide. Some idea of the magnitude of these houses may be conceived from the fact that for heating alone an 1800 h.p. steam generating plant and sixty miles of coils and piping are required.

The main product of the Davis Company is hothouse-grown cucumbers, of which 12,000 dozen are shipped each week, but the output includes also flowering plants grown for the sale of both cut flowers and the plants themselves. The stock of flowering plants, among other things, includes hundreds of thousands of cyclamen.

The temperature requirements of these greenhouses are even more exacting than those of the Missouri Botanical Garden, as the chart, Figure 20-9, taken from the recording thermometer, shows.

The steam for heating is taken from a 95-lb. steam line running through the connecting corridors, and the pressure is reduced in each greenhouse for the Webster Vacuum Heating System, which operates at 5-lb. pressure. The condensation is carried through a vacuum return back to the power plant, where it is delivered by the main vacuum pumps through a tank to a Webster Feed-water Heater and from there pumped to the boilers.

CHAPTER XXI

Methods of Testing Return Traps in the Laboratory

LABORATORY tests may be conducted for widely different purposes. Those discussed in this chapter are the usual ones for determining the commerical efficiency of different types and makes of traps.

All of the operating conditions possible or probable in an actual heating system cannot be artificially produced in the laboratory, nor is it practical to carry out tests long enough or upon sufficient numbers of samples to learn all facts which become evident in practice. Furthermore, as the whole heating system—its design and installation—has its effect upon the efficiency of the devices entering into it as parts, any laboratory tests for efficiency can indicate only the results which are probable when the devices are properly used in practice.

Too much stress should not be laid, therefore, upon the comparative performances of any two makes of traps during laboratory tests. Knowledge of performances in actual installations of many heating systems, maker's ability and care in manufacturing, shop tests, inspection and proper engineering application of the traps are of great importance to the investigator who wishes to make commercial use of his study of such devices.

However, as laboratory tests have their useful place in commercial investigation, this volume would be incomplete without descriptions of the following methods and apparatus which have been found practicable for reproducing as nearly as possible the conditions which exist in practice. Mention is also made of common but improper methods of testing which should be avoided because of the erroneous data which they produce.

Usually the object of a laboratory test of a Return Trap is to determine one or all of the following characteristics:

1. Effect of the trap upon radiator efficiency.

2. Efficiency of the trap for the removal of air and water of condensation and for conservation of the steam and vapor.

3. Behavior of the trap without special adjustment to meet the varying conditions of pressure and vacuum in normal practice.

4. Durability of the trap through a long period of use.

5. Construction features of the trap, particularly the amount of valve movement, which indicates the ability to get rid of dirt and pipe scale.

The results of tests by many investigators, of radiator and trap efficiency, have varied widely and often been misleading, largely because the methods of testing have been faulty and partly because the devices themselves have not always been manufactured to operate uniformly.

Most tests of which the results have been published have been faulty through failure to cover a wide enough variety of test conditions, through limitation of the time period for each test to a few minutes instead of hours, and through considering and testing only one or two samples of any one device, instead of six or more selected by the investigator from the manufacturer's stock bins.

Tests for Heating Efficiency: The heating efficiency of a radiator depends upon physical conditions within the radiator which are affected by the action of the return trap. The radiator, among a number of common size and type, which maintains the highest average temperature when tested under the same conditions is the most efficient.

The greatest possible steam economy is obtained where this efficiency is highest; that is, where steam is being condensed to the greatest extent possible within the radiator and the trap passes the least amount of steam or vapor into the return pipe.

The highest radiator efficiency can be obtained only where the discharge is sufficiently and properly restricted to prevent steam from blowing into the return. Also the air released from the steam in the radiator must be allowed to settle to the lower parts, from which it can enter the trap and be discharged.

A return trap, in addition to restricting the discharge, must effectively accomplish the following:

1. The discharge of all water of condensation as formed. Otherwise water accumulates in the radiator, prevents free discharge of air and also reduces the amount of surface effective for emitting heat from the steam.

2. The discharge of all air as well as water from the radiator immediately upon their reaching the discharge outlet.

3. Thorough prevention of the discharge of steam to the return.

To accomplish these requirements the valve of a return trap must open or close within a very narrow range of temperature, above or below that of steam at pressure—irrespective of variations in steam pressure—and must adapt itself to such changes of pressure and corresponding steam temperature as may be met in practice.

A brief review of the various types of return traps will facilitate a better understanding of tests and the results which are desired.

All return traps commonly used in low-pressure or vacuum steam heating practice may be classed as float traps, differential traps or thermostatic traps.

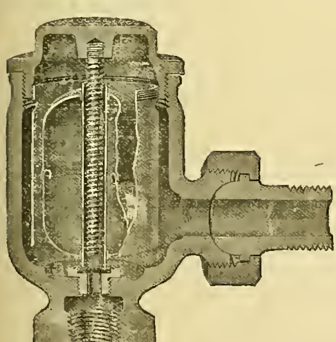


Fig. 21-1

Float Traps may have either sealed floats or inverted open buckets as the means of operation, and in either case, the float is raised by incoming condensation to uncover the valve seat through which water is discharged. Air escapes into the return pipe through an air port, which must be located above the highest water level in the trap. The air port is controlled in some makes by thermostatic devices to prevent leakage of steam to the return.

Tests upon a float trap may generally be expected to show considerable leakage of steam to the return unless the air port is thermostatically controlled, or if the air port is so

controlled the small port and its mechanism may be vulnerable to the effects of dirt and rust. Such traps, however, will be found to have large water discharge capacities and some of the various makes can be used to advantage where widely varying volumes of water must be discharged without respect to temperature.

A differential trap depends for operation upon the difference in pressure at the inlet and at the outlet. In its simplest form, it is a check valve which is closed when the difference in the pressures ahead and back of the clapper is insufficient to overcome the weight of the clapper, Inasmuch as no special means are provided for discharge of air, such a valve may be expected to leak steam to the return under any conditions of higher differential pressure and to stay closed with consequent air binding and water logging of the radiation where the pressure differential falls below that for which the valve is adjusted.

Another form of differential trap is shown in Figure 21-2. Water entering the valve body raises the float, thus closing the air port by means of the valve piece attached to it. A higher pressure in the lower part of the trap B than that existing in the chamber A results in the operation of the piston which raises the valve from its seat by means of the connecting valve stem. As the condensation is discharged, the water level lowers and causes the float to fall, thus uncovering the air port, and equalizing the pressures on opposite sides of the piston. The weight of the operating parts and the force of the spring then closes the valve. This trap may be expected to show fairly good results in laboratory tests but it is not satisfactory under the usual operating conditions in which dirt and scale are always present.

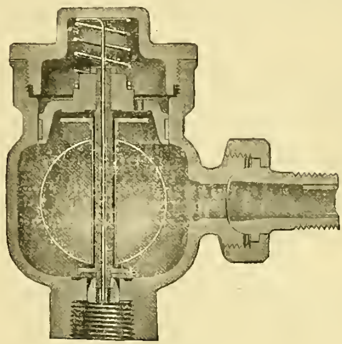


Fig. 21-2

A Thermostatic Trap depends for its operation upon the difference between steam temperature at the pressure in radiator and the temperature of the condensate to which it is exposed.

Many devices have been made which depend upon the expansion and contraction of metals or composition, or which make use of a Bourdon Tube. As a class these have failed because there is not enough difference in area between the inside and outside of the spring to produce the required force at normal difference in temperature between steam and air vapor at a given exterior pressure. This and other faults, such as the necessity for adjustment for varying pressure conditions and slowness in operation, have led to the abandonment of thermostatic types by most manufacturers.

Of all types of Return Traps, the only ones in general use today are those which depend for movement of the valve piece upon the change of vapor pressure of fluids confined within a flexible chamber when subjected to different exterior pressures and temperatures. The volatile fluids contained in the flexible chamber vaporize to a greater or less pressure depending upon the

temperature of the steam, vapor, water or air which surround the chamber. The expansion or contraction of the chamber moves the valve piece which is attached to the free end of the chamber.

These traps are, generally speaking, of either the "inboard" type where the thermostatic member is exposed to the temperature and pressure of the steam, water and air as it exists at the radiator outlet, or of the "outboard" type which depends for operation upon the conditions existing between the valve piece and the entrance to the return piping beyond the trap.

To be effective for the inboard type, the thermostatic member must expand and contract through a distance sufficient to open and close the valve under the influence of the extremely small differences of temperature which exist during normal operation. Most traps of the inboard type are inefficient because of the very short "stroke" which can be realized with the inelastic disc construction generally utilized for the flexible chamber, this defect showing in operation as inability of the trap to rid itself of dirt and scale.

Traps of the outboard type are affected by the pressure and temperature of the return. They are in proper adjustment only at one definite pressure and temperature and out of adjustment at all other normal combinations of pressure and temperature. They cannot be adjusted even for these normal variations in radiator pressures and vacuum in the return, and as a result usually water-log and air-bind the radiator by staying closed when high temperature and pressure differentials exist, or stay open and blow steam under conditions of low differential.

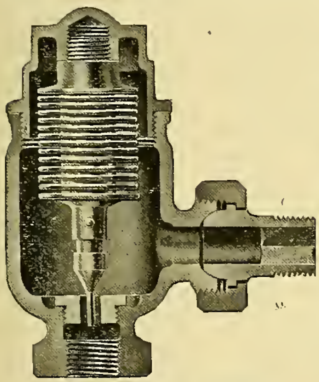


Fig. 21-3. The Webster Sylphon Trap

The Webster Sylphon Trap shown in Figure 21-3, has best met all of the requirements of theory, test and operation during the years since its first use in practice.

It is a thermostatic trap of the inboard type and as such is affected in operation only by the temperatures and pressures existing in the radiator.

The multifold design of the thermostatic member or Sylphon Bellows gives it great elasticity and consequent ample movement in response to changes of temperature and pressure in the medium surrounding it.

The Webster Sylphon Trap is filled with a liquid which makes the trap self-compensating for differences in operating pressures of steam within the radiator. It operates effectively and without adjustment to rid the radiator of water and air and to prevent discharge of steam, whether the pressure differential between radiator and return is ⅛ lb. or 15 lb. per in.

Its construction, using a conical valve piece closing upon a sharp-edged seat, assures positive self-cleaning. Dirt and scale cannot lodge between the valve and seat to cause the trap to leak steam to the return.

Because of careful workmanship and frequent factory tests and inspection, Webster Sylphon Traps in laboratory tests of a number of units may

be expected to show very uniform results—in fact, so uniform that successful operation of these traps in large numbers in any properly designed heating system can be foretold from the laboratory results.

The Webster No. 7 Trap is constructed with a different form of flexible member but its operating characteristics are essentially identical with and fully as satisfactory as those of the Webster Sylphon Trap.

It has been stated that a trap must not leak steam to the return, but in this connection there should be no confusion between steam discharged through a trap and vapor rising from hot condensate. Though their appearance during certain forms of visual tests are much alike, they are two entirely different things, and if confused with each other, as is sometimes done, wrong conclusions will result.

Visibility is deceptive. A great amount of moisture in the atmosphere and favorable light conditions both add to the visibility. The air discharged from an efficient trap is saturated with water at discharge temperature and this water mixing with air at room temperature looks like steam, while the discharge of a trap utterly deficient in air removal shows only the vapor of re-evaporation.

The water of condensation contains total heat in excess of that in water of condensation at lower pressure. This excess heat boils off some of the condensation into steam. The amount so boiled off is entirely dependent on excess of total heat in outflowing condensate above total heat of water at lower pressure.

If steam passes out with condensate, a steam of greater total heat is dissipated. A fully efficient trap releases the condensation at or near steam temperature and radiator pressure, into a return of lower pressure. All heat above that consistent with lower pressure then generates vapor. This vapor passes to the vapor receiver in a test. A certain amount of vapor per pound of condensation is normal and any excess of vapor above the normal is steam leakage.

The condensate from a higher pressure into a lower pressure will never be at a higher temperature than that due to steam at the lower pressure. The balance of the heat in the outflowing condensate will flash part of the water into steam.

These points are emphasized to show the fallibility of visibility test to show the efficiency of return traps.

Very rough tests, Figure 21-4, are often made by connecting a trap to a radiator, letting it discharge to the atmosphere, and noting its operation. With this test particularly, the erroneous distinction between the vapor from steam and from re-evaporation often leads to a wrong conclusion. Further, such a test can show only how the trap behaves for a condition far different from those of actual operation. The effects of the return piping connections and the pressure conditions therein have such a great effect upon operation that the results of rough tests of this nature should never be accepted as conclusive.

Tests of similar apparatus are often made to determine the comparative value of two traps, using the amount of water discharged from each trap

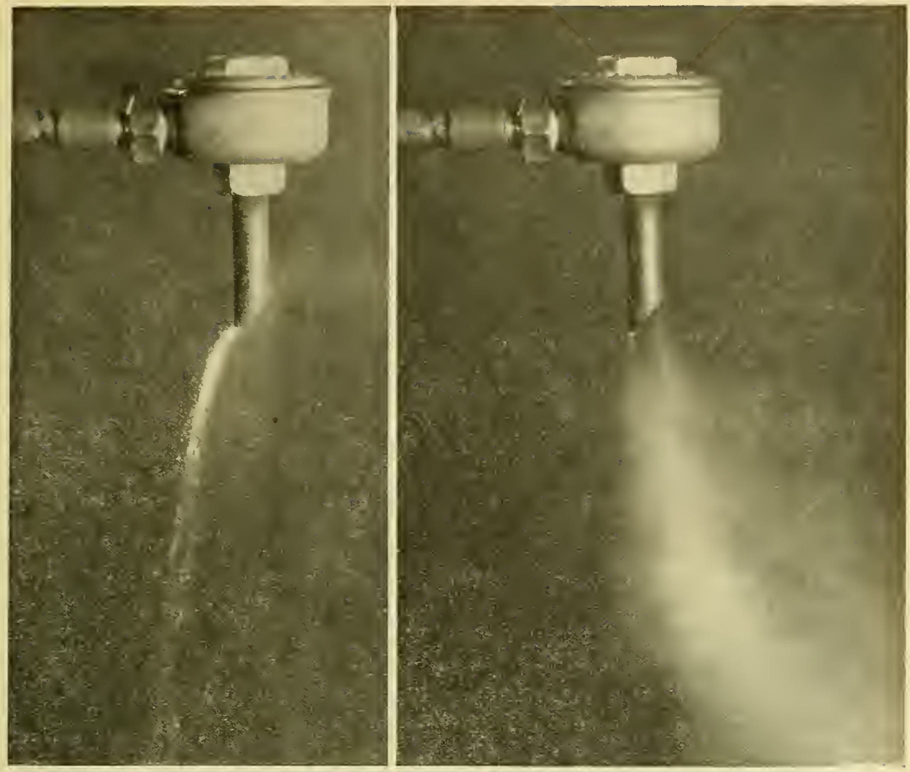


Fig. 21-4. Visibility test showing at the left a trap which is discharging condensation without steam leakage, and at the right a trap in which steam is leaking through the outlet valve

after runs of equal duration as a measure of their desirability. In such cases the condensate dripping from the trap is carefully weighed.

It is evident upon consideration that such tests demonstrate nothing regarding the performance of the traps while no water is held up in the radiator, and even then there is no assurance that the same comparative result will be obtained when the trap is under actual operating conditions.

Other tests have been made to determine the vacuum which could be maintained at the discharge end as a criterion of the comparative worth of traps. For such tests, the apparatus consists of a radiator, a return trap, a return connection to a vacuum pump, and devices for maintaining constant pressure of steam supply to the radiator and for operating the pump at a constant speed.

The trap with which a higher degree of vacuum is maintained by this test is considered to be the better. With little or no attempt to determine the extent to which the radiator is air- and water-bound in the comparative tests, the data obtained has frequently led to a wrong choice and unsatisfactory results when the trap was operated in a heating system.

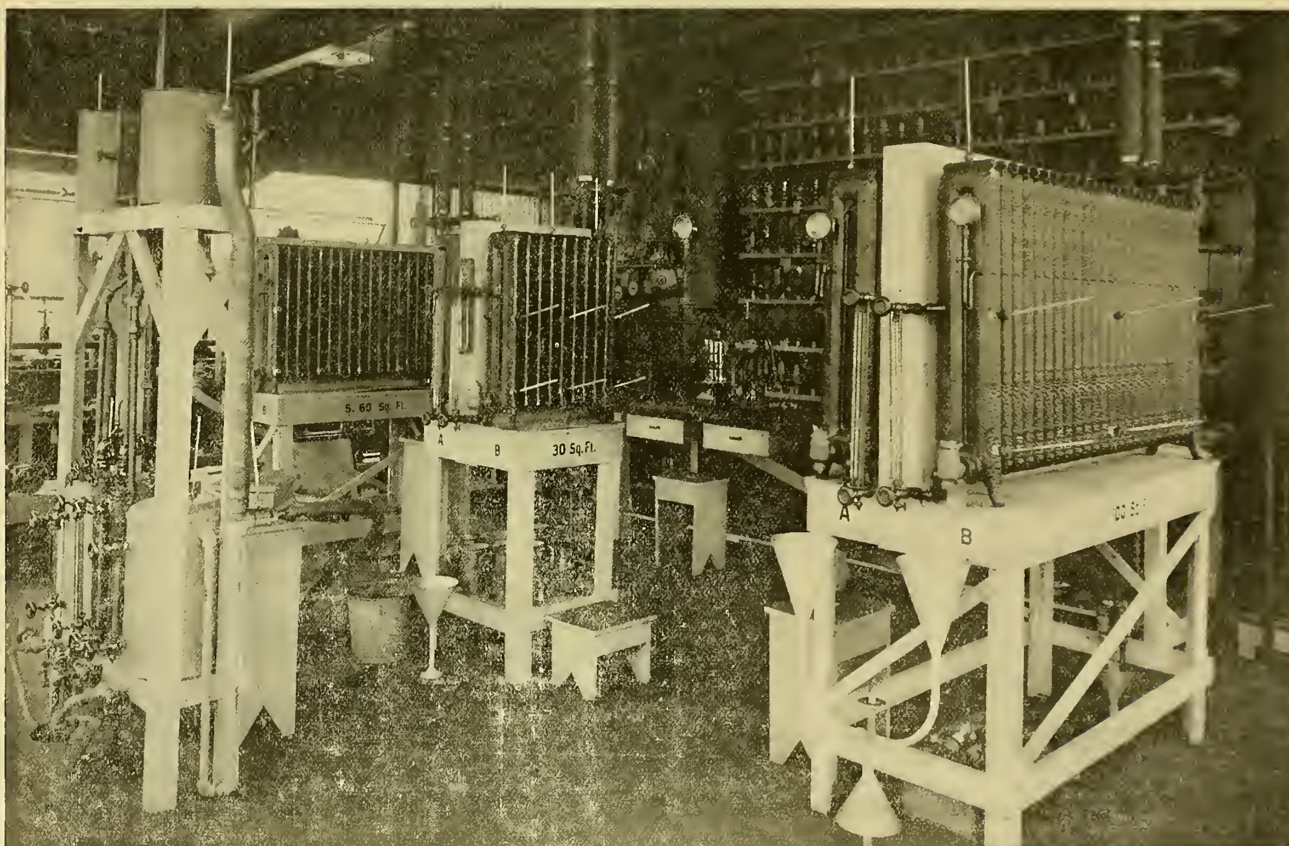


Fig. 21-5. Part of the testing laboratory of Warren Webster & Company

These few devices and methods are the ones commonly used for determining comparative worth of return traps where only the most easily procurable testing apparatus is available. Like other scientific investigations more careful methods will lead to more reliable results and with proper apparatus and thoughtful procedure it is entirely practicable to obtain test data which can be relied upon as accurately forecasting the success which may be expected from the use of any return trap in an actual heating system.

The first thought for any reliable test should be to create laboratory conditions as nearly as possible like those met in actual practice. Coincidently, the apparatus should be designed to provide exactly like and simultaneous test conditions where traps are tested for comparison, and of course, appliances for measuring the results must be carefully placed and adjusted. Then, by following a proper test, planned to exhaust the various possibilities of different operating conditions, results are secured which can be accepted as conclusive.

A complete testing outfit is illustrated in Figure 21-5. Sets of two radiators are installed exactly alike, so that the operating conditions for comparative tests of any two traps are as nearly the same as possible.

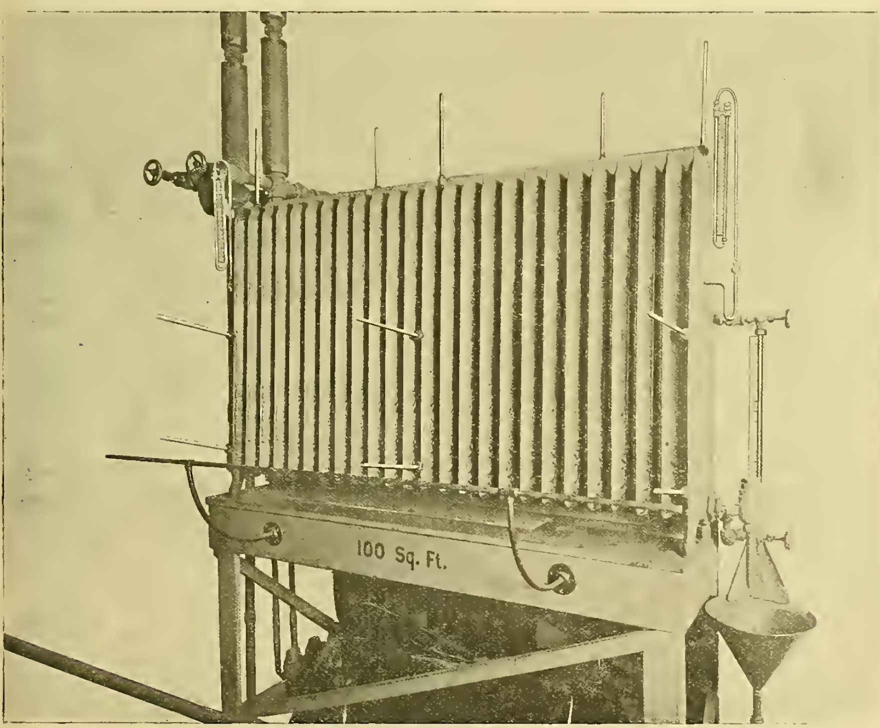


Fig. 21-6. Showing position of thermometers for determining internal conditions at various parts of the radiator

The test radiators should be: (1) of the steam type to provide most difficult conditions for trap operation, (2) of height normally met in practice, and, (3) of a size approaching the maximum capacity of the trap at the minimum pressure differential contemplated for the test.

Each radiator is provided with thermometers set in wells, as shown in Figure 21-6. Three thermometers in the loop at the supply end, three in the loop at the center, and three in the loop at the return end of each radiator give a fair enough indication whether or not air is present in any portion of the radiator.

The water gauge glass with connections tapped into the highest and lowest parts of the last loop of each radiator, serves to show whether water accumulates in the radiator, and if so, the amount, or whether the trap freely discharges all water of condensation.

Pressure gauges, preferably mercury columns, are connected to each end loop to show the pressures at both supply and return ends of the radiators. The average pressure and temperature must be known to determine the heating efficiency.

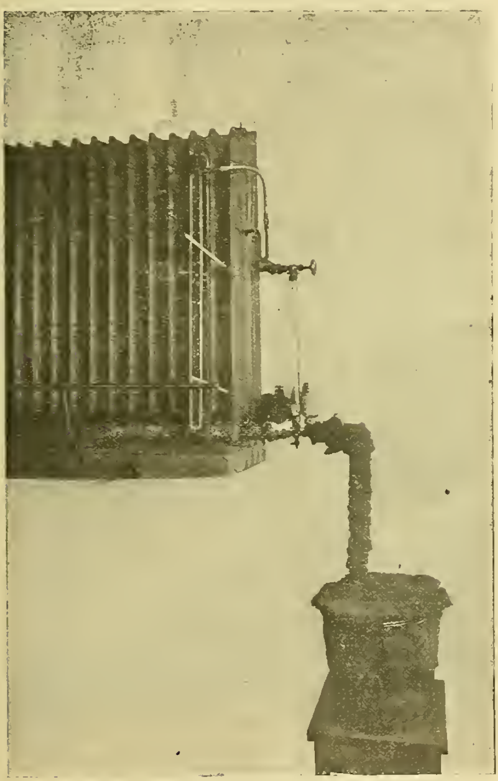


Fig. 21-7. Arrangement of bucket calorimeter for determining the quantity of trap discharge

The apparatus must be supplemented with other devices, as will be described later if a complete test is desired. However, where part knowledge quickly gained is acceptable, indication of the following characteristics of return traps can be obtained by using only the twin radiators:

A. Ability to discharge all water of condensation, as indicated by the water glass.

B. Effectiveness in discharging all air, as shown by the thermometers.

C. The time required after turning steam on the cold radiator before the radiator is completely heated throughout.

D. The rate of condensation for the radiator itself after it is completely heated, or the rate of discharge of the trap can be determined by weighing the condensate collected during a measured period.

All of the data refers of course to the action of the traps with steam at the average pressure in the radiator and discharging direct to atmosphere. The

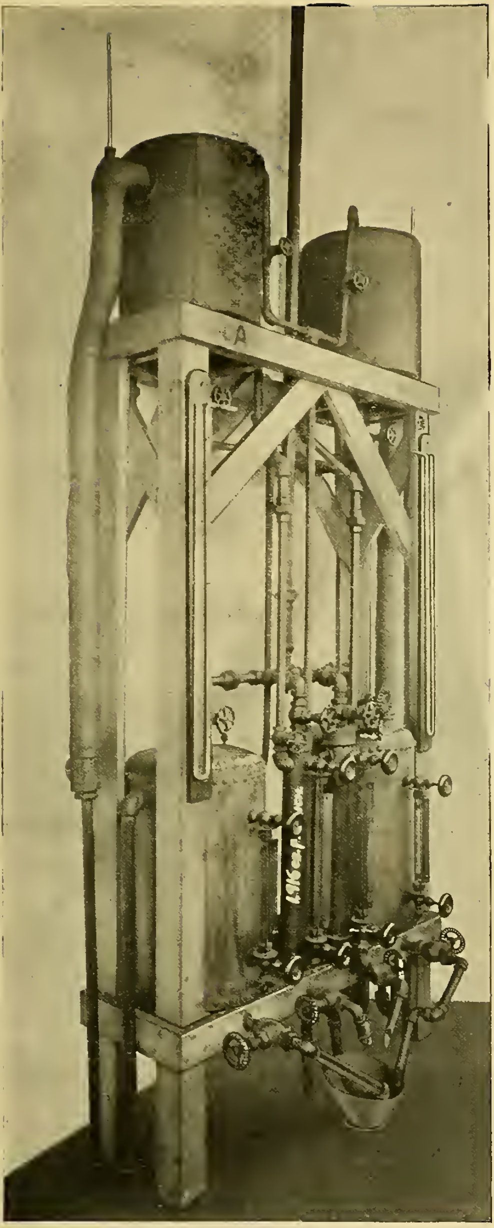


Fig. 21-8. Twin measuring apparatus for determining separately the amount of discharge from each of two radiators

results obtained will vary greatly from those of test conditions, more closely approaching actual operation.

A determination of the amount of steam or vapor as well as of water discharged through a trap can be made by attaching a comparatively simple bucket calorimeter, as shown in Figure 21-7. The discharge from the trap condenses in the coil and discharges to and is mixed with the water in the bucket.

The observations in such a test are:

Time of run.

Barometric pressure.

Steam pressure.

Temperatures of water at beginning and end of test.

Weight of water before and after test.

The "heat balance" calculated from this data will show whether steam has leaked through the trap; that is, the increase in B.t.u. in the cooling water should indicate only an amount accountable for by the absorption of latent heat from the steam condensed in the coil.

Figure 21-8 illustrates a form of apparatus for twin operation in combination with the two radiators for separately measuring the water and vapor discharged from the return traps.

The discharge from each trap is led to a separating tee, from which the water passes downward through a seal into the measuring vessel, the air and vapor rising to the condensing coil and thence to another tank in which the condensed vapor is measured.

The tanks which collect water are large enough for a run of an hour or more, are provided with gauge glasses and scales, have pipe connections at the top leading to a dry vacuum pump, and have the necessary drain and pet cocks for breaking vacuum when emptying the tanks. The smaller tanks are provided with similar equipment and connections.

Thermometers and mercury columns are installed, the former to give the necessary temperature readings from each tank of water, and the latter, readings of the vacuum at which the test is being run.

Connection is made from the outlet of each trap to the separating tees by means of steam hose. This connection and all pipe connections from the tanks to the pump must of course be absolutely tight, as air inleakage would impair the results by carrying water to the condensing coils. All metal piping at ends of steam hose must be covered with wool felt or similar non-conductive material.

The steam pressure at the inlets of the radiators is kept constant by means of pressure-reducing valves and the pump vacuum by means of vacuum-pump governors, as shown in Figure 21-9.

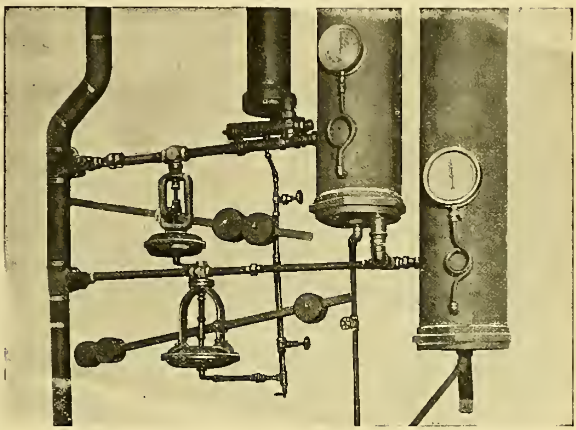


Fig. 21-0. Pressure-reducing valves for maintaining constant pressure at radiator inlets during test

Very complete tests of return traps can be made with this apparatus, and results will closely approximate the operating characteristics of the same traps in actual heating service.

Complete testing of traps involves the following considerations:

1. Selection of trap. The traps to be tested should be selected from the manufacturer's stock bins, or if this is impractical they should be purchased on the market.

2. Number of traps to be tested. A fair determination of the average performance of any make of trap requires that at least six units, or more if possible, be placed under the same test conditions. Only by this procedure can the standard of performance be determined within reasonable limits of uniformity.

3. Duration of each test. The first two traps of any given make under any given pressure and vacuum conditions should be tested during a run of at least five hours. Subsequent tests may be shortened to as little as one hour, depending upon the time taken in the earlier tests for the traps to show all of their operating characteristics. The longer time for first tests has been found by experience to be justified by the failure of certain types of traps to show their deficiencies within a shorter period.

4. Pressure and vacuum conditions. Each trap should be tested under each one of the following combinations.

1 lb.	pressure	in	radiator	and	5 in.	vacuum	in	return.
5 "	"	"	"	"	5 "	"	"	"
10 "	"	"	"	"	5 "	"	"	"
1 "	"	"	"	"	10 "	"	"	"
5 "	"	"	"	"	10 "	"	"	"
10 "	"	"	"	"	10 "	"	"	"
1 "	"	"	"	"	15 "	"	"	"
5 "	"	"	"	"	15 "	"	"	"
10 "	"	"	"	"	15 "	"	"	"

These combinations are frequently met in practice, the higher pressures and vacuums representing unfavorable conditions, such as small sizes of supply pipes, small lifts or small piping in the return.

5. Frequency of readings. Usually the thermometers, water gauges, barometer, pressure gauges, etc., need be read only at intervals of 5 to 10 minutes after the radiators have become thoroughly heated. Where two traps are tested on the twin apparatus, corresponding instruments should be read simultaneously.

At the conclusion of the test, the data is worked into terms of efficiency and capacity of the traps.

The *heat efficiency* is the relation expressed in per cent of the average temperature throughout the radiator to the temperature of the entering steam. To obtain average heat throughout the radiator, it is usual to add the temperatures shown by thermometers in the end loops only once, while the temperatures in the center sections are given added value by being considered twice. In other words, if there are nine thermometers as shown in Figure 21-6, the sum of the temperatures is divided by 12 to obtain the average. A low heat efficiency indicates either air- or water-binding or both. If no water shows in the radiator gauge glass at time of low efficiency, it may be safely assumed that the return trap is not successfully ridding the radiator of air.

The relation of the weight of condensed vapor collected in the smaller tank of Figure 21-8 to the total weight of condensate and condensed vapor collected in both the smaller and larger tanks is the "vapor efficiency" of the trap. Corrections for re-evaporation may be made if desired, but when tests are run with low pressure in the radiator and low vacuum in the return, the correction factor may be neglected without appreciable error.

Where the gauge glass shows that the trap is holding condensation in the radiator, the condensation should accumulate during entire test period. The condensation is then drawn into the larger condensate tank (Figure 21-8) by creating a vacuum there by shutting steam from the radiator and opening an air vent on the radiator. The weight of water actually passed by the trap during the test period compared as a percentage to the total of water passed during test, plus the water accumulated in the radiator is the "water-discharge efficiency" of the trap.

Tests along the lines indicated, and with apparatus described, provide practically all the data which is usually investigated, although other useful and important data may be obtained with additional equipment.

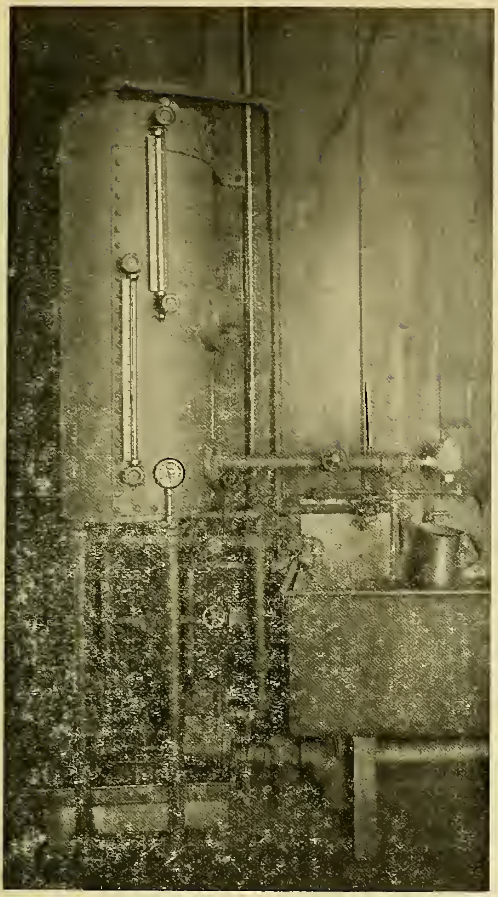


Fig. 21-10. Apparatus for determining flooded capacity of traps

For instance, it may be desirable to determine the maximum discharge capacity of a given trap when operating at stated pressure differential. This pressure may be the one at which the trap is to be operated in a proposed heating system, and it may or may not be the pressure differential which the manufacturer has selected as a basis for rating his trap.

Such tests must take into consideration the fact that with thermostatic traps, the temperature of the condensate has a major effect upon the rate of discharge. As the temperature of the condensate in an actual installation is comparatively high, it is specially desirable that apparatus used for determining flooded capacities of traps shall provide for suitable heating of the water to be discharged.

Such apparatus is illustrated in Figure 21-10. The tank is "closed" and is provided with steam injection devices for heating the water to any desired temperature. In addition to water head, discharge pressure can be placed upon the trap by compressed air. The water when discharged falls into a second tank which is open to the atmosphere, and from which it is returned to the supply tank by means of the pump.

This apparatus makes possible the determination of the proper rating for any trap in pounds of water per unit of time for any pressure differential or any condensate temperature which may be met in practice.

Another interesting test, designed to indicate the ability of a trap to rid itself of the dirt and scale met in operation in a heating system, can be made with the apparatus shown in Figure 21-11. In this test, a mixture of cylinder oil and core sand, sifted through a 1/10 in. mesh screen, is inserted at the tee in the outlet piping, in doses of usually about a quarter teaspoonful.

Upon opening the valve in the outlet connection, the mixture passes to the trap and its effect upon operation is carefully noted.

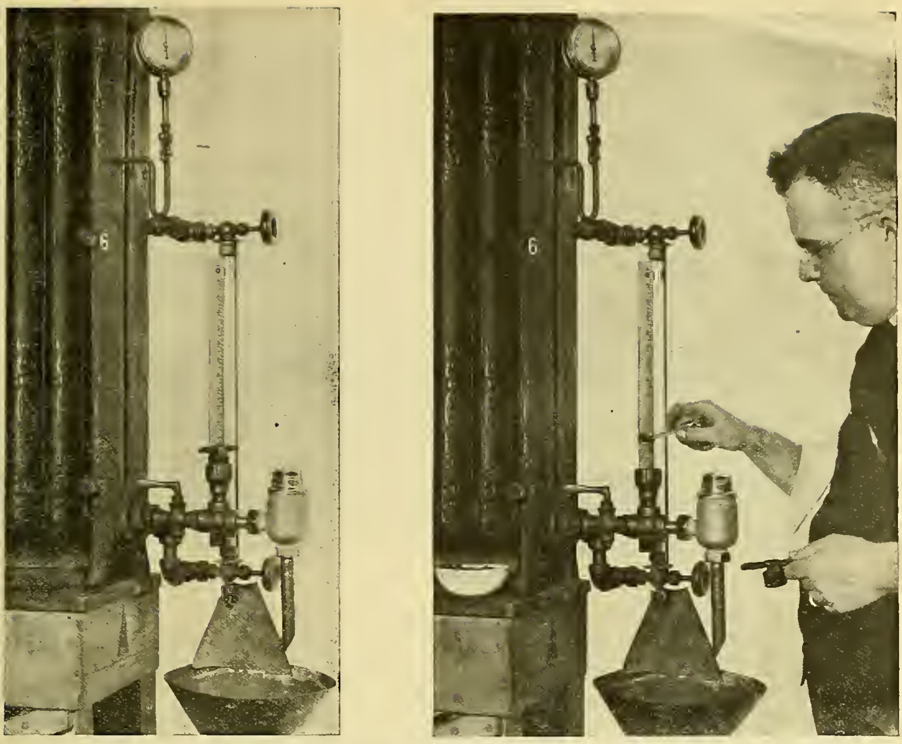


Fig. 21-11. Apparatus for determining the ability of a trap to rid itself of dirt and scale

A repetition of this dosing with dirt about ten times gives a fair test for the trap. The number of times out of the ten in which the trap closes and opens properly to hold steam and to discharge air and water indicates the ability of the trap to operate efficiently under the trying conditions of dirt and scale which are most serious during the initiation of a new radiator or newly erected heating system.

A vibrating machine driven by a motor for demonstrating the relative durability of the thermostatic members of return traps is illustrated in Figure 23-12. The stroke of the machine can be adjusted to equal the distance through which the thermostatic member will expand and contract during operation in a heating system. A counter for determining the number of strokes is provided.

The trap using the member which will withstand the greatest number of strokes through its individual distance of operation, before failure of the material occurs, is the one likely to have the longest life in actual service before repairs are required. This test gives merely indication of the durability of the trap. Other operating conditions besides the movement of the member affect the wearing qualities, but unfortunately, these conditions cannot readily be duplicated in short-time laboratory tests.

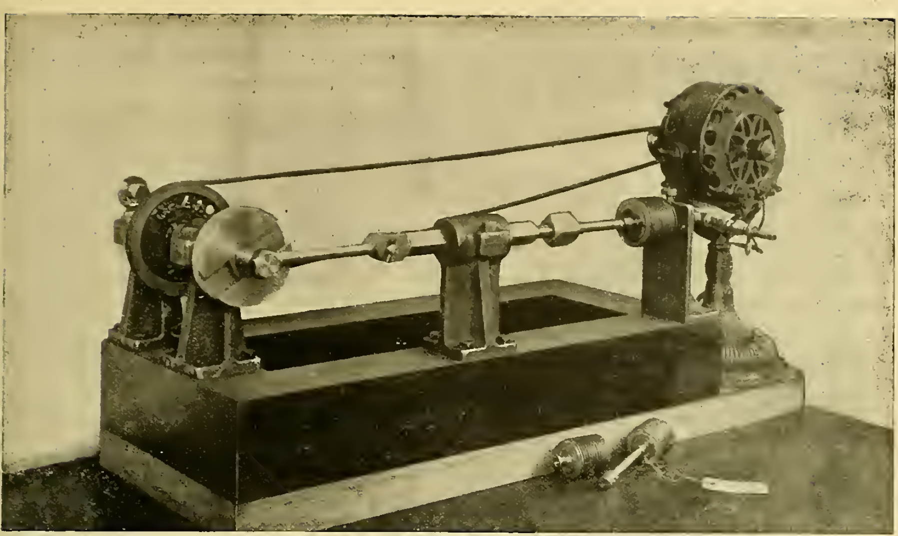


Fig. 23-12. Vibrating machine for demonstrating the relative durability of various types of thermostatic members of return traps

Enough has been said to show that valuable data regarding the probable performance of return traps can be obtained in the laboratory where suitable apparatus is available and where suitable test methods are carefully applied. However, the long-time test of devices in actual heating systems is the best guide for determining the relative value of return traps, and further, the efficiency of a good return trap can be fully realized only when the heating system itself is properly planned and operated.

RE-EVAPORATION: The idea which seems to prevail that water at a temperature higher than the boiling point of the space into which it is discharged should not make steam when so discharged is erroneous.

The amount of steam so generated is readily determined by the chart, Figure 21-13, provided the initial temperature of the water and the pressure of the space into which it is discharged are known.

Many times highly efficient radiator traps are condemned for leaking steam, due to the observed vapor of re-evaporation noted at their discharge outlet, and less efficient traps have been commended because of absence of such vapors.

The absence of vapor at the discharge is in reality an indication that the trap is holding back condensation and entrained air until the temperature of the discharge is materially less than that of steam at the pressure of the outlet. The consequence of such holding back is a partially air-bound and water-logged radiator, with less than full efficiency in the radiation.

Again, there are instances where in tests of radiator trap efficiency all the steam leakage recorded has been ascribed to re-evaporation when, due to high differential of initial and terminal conditions, all or a greater part of the observed steam should have been ascribed to re-evaporation.

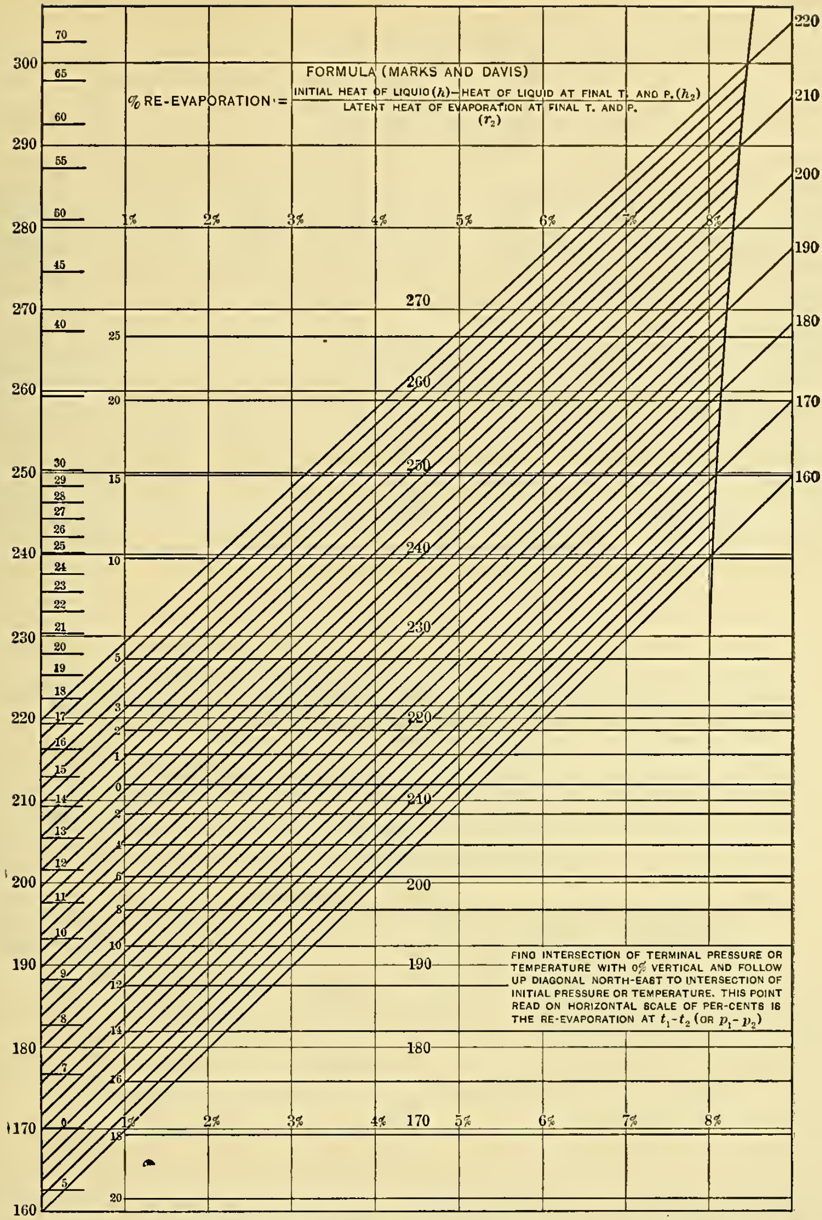


Fig. 21-13. Re-evaporation chart for determining the percentage of water re-evaporated from any temperature between 300 and 170 deg. fahr. into water vapor of a lower temperature and corresponding pressure

CHAPTER XXII

Installation Details

MANY of the methods of pipe connections which have been developed by Warren Webster & Company during the past thirty-three years, and which have become standard practice, are shown in this chapter and elsewhere in connection with descriptions of specific apparatus. Most of the ones previously illustrated as Webster Service Details are familiar to the profession and trade. These drawings, which indicate the general arrangement of the pipe, fittings and Webster apparatus, when used, have been revised from time to time and as shown here represent the latest and best thought. They are not to be used as exact layouts of piping, as each individual application presents its own special conditions. No effort has been made to indicate the necessary unions or right and left nipples required for the connections, as these requirements for any case would naturally be best determined by the detail of the layout or by the steamfitter at the job, based upon his skill and upon materials available.

Details Applicable to Both the Webster Vacuum System and the Webster Modulation System

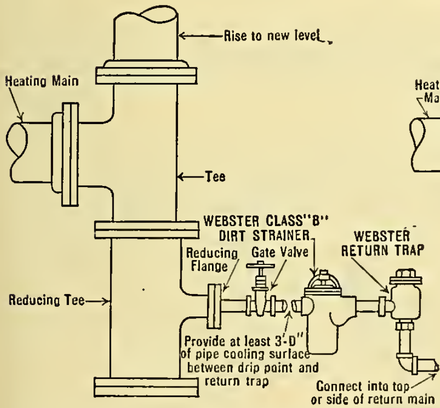


Fig. 22-1. Application of a Webster Return Trap on a low-pressure heat main. at a low point where the main rises. A sufficient length of uncovered pipe must be provided between the drip point and the return trap.

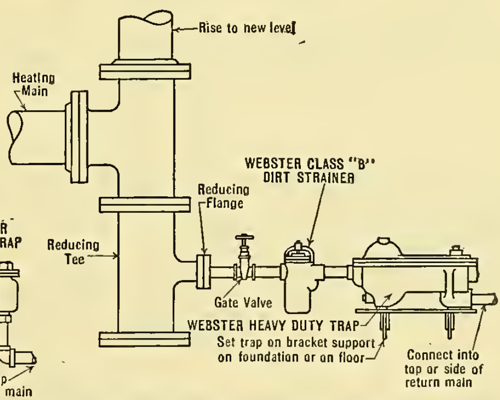


Fig. 22-2. The drainage of a low-pressure heat main at a low point, where the line rises, is of such importance that special attention is warranted. This diagram shows a large main with drip through gate valve, Webster Dirt Strainer and Webster Heavy-duty Trap.

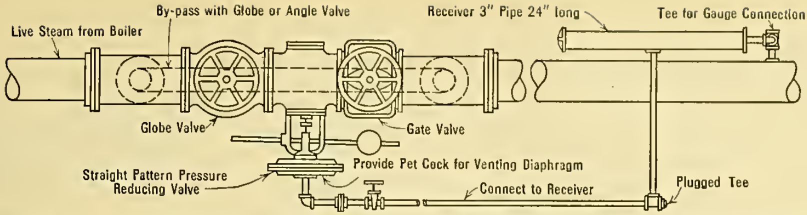


Fig. 22-3. Connections for a steam pressure-reducing valve. The control pipe from the low-pressure side of the line must be taken from a point far enough from the valve to insure that the pressure will have been fully expanded. The use of the receiver facilitates a constant static pressure on the diaphragm of the reducing valve.

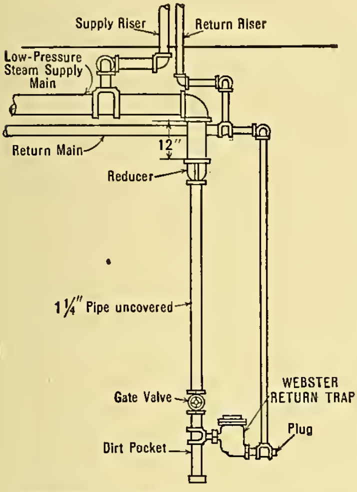


Fig. 22-4. Method of dripping supply risers through a Webster Return Trap into vacuum return line; the vertical leg acts both as cooling surface and dirt packet.

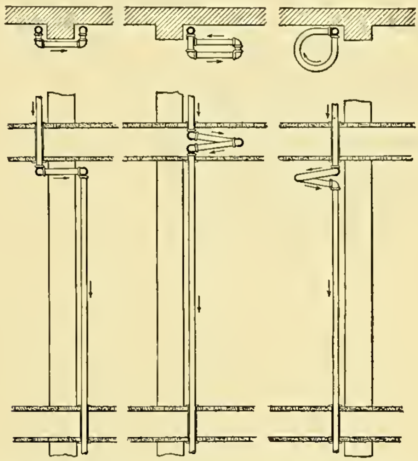


Fig. 22-5. Three methods of making loops to provide for expansion movement in risers. The expansion of supply and return risers should have careful study.

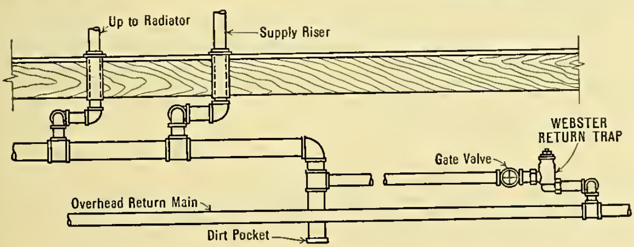


Fig. 22-6. Arrangement for dripping the end of a supply main which also carries the condensation from the upfeed risers. Overhead return main. The return trap is located at a point four feet or more from the point dripped.

Fig. 22-7. Arrangement for dripping a down-feed riser into an overhead return main, showing the uncovered horizontal cooling pipe.

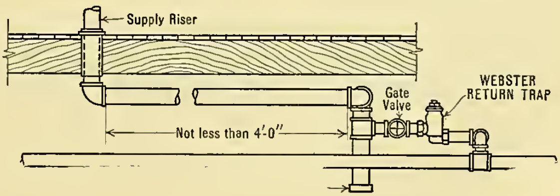


Fig. 22-8. Dripping the heel of a down-feed supply riser, where provision must also be made for down thrust or expansion.

Fig. 22-9. The end of an up-feed system supply main where provision must be made for the drip as well as the condensation from the risers. The return is located along the floor and the vertical line to return trap can be used as a cooling leg.

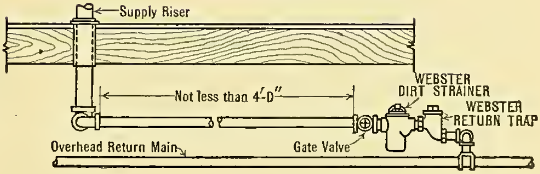


Fig. 22-10. Arrangement for dripping down-feed risers into an overhead return line. Cooling pipe used with a Webster Dirt Strainer located at the entrance to the return trap.

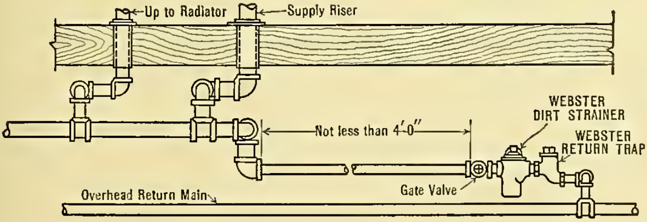


Fig. 22-11. Arrangement for dripping the end of an overhead supply main through Webster Dirt Strainer and Return Trap into an overhead return main.

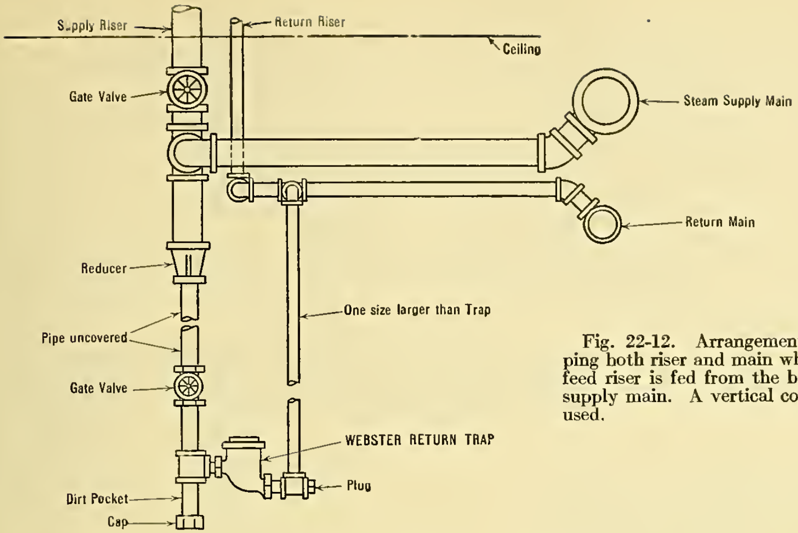


Fig. 22-12. Arrangement for dripping both riser and main where an up-feed riser is fed from the bottom of a supply main. A vertical cooling leg is used.

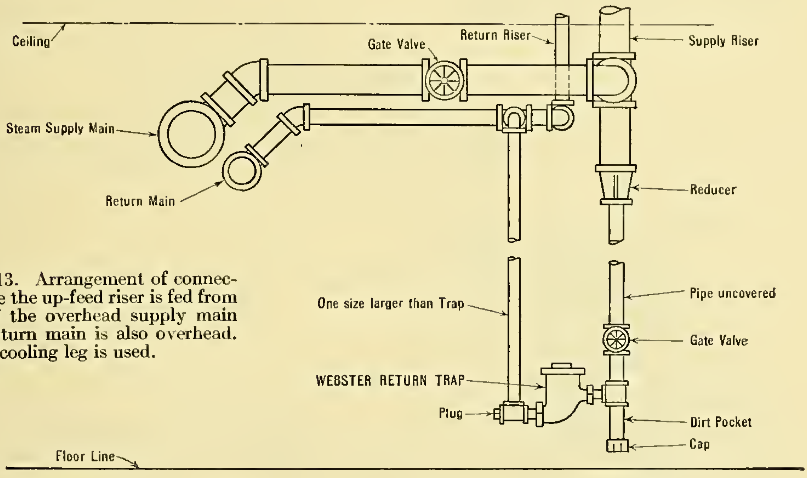


Fig. 22-13. Arrangement of connections where the up-feed riser is fed from the top of the overhead supply main and the return main is also overhead. A vertical cooling leg is used.

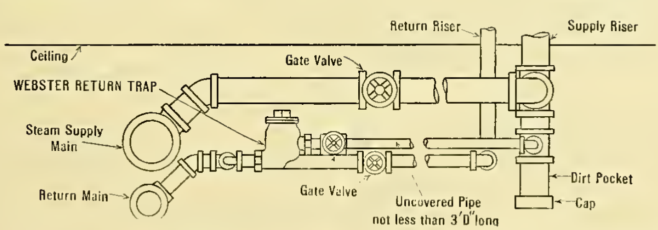


Fig. 22-14. Where it is not possible to run a vertical cooling leg on the drip of the riser, cooling surface in the form of a horizontal pipe may be employed as shown.

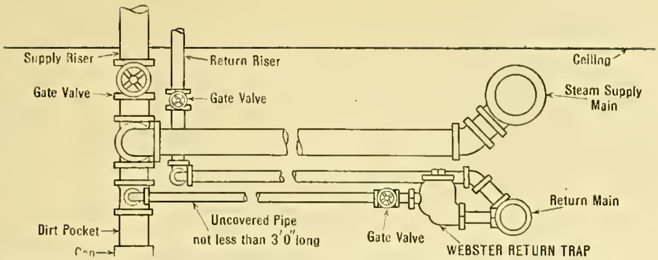


Fig. 22-15. Drip of main and up-feed riser using horizontal cooling surface.

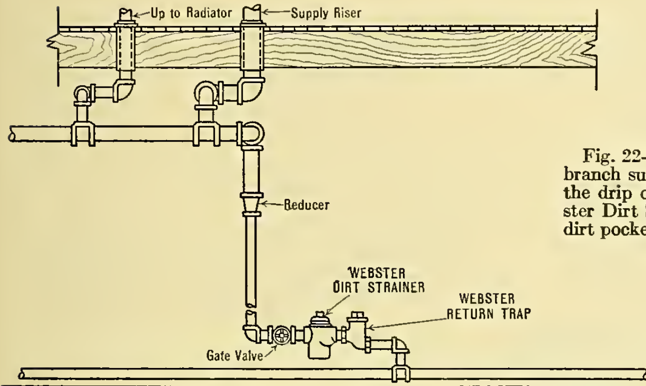


Fig. 22-16. The drip of the end of a branch supply main which also carries the drip of down-feed risers. A Webster Dirt Strainer is used in place of a dirt pocket.

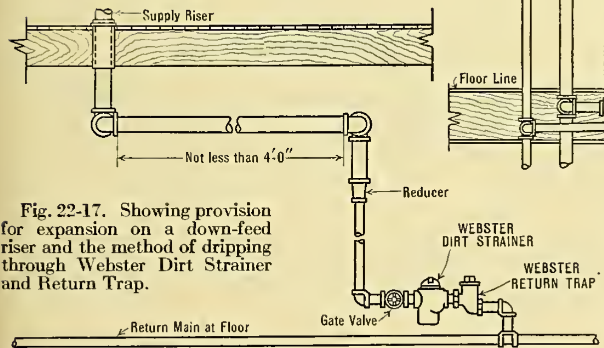


Fig. 22-17. Showing provision for expansion on a down-feed riser and the method of dripping through Webster Dirt Strainer and Return Trap.

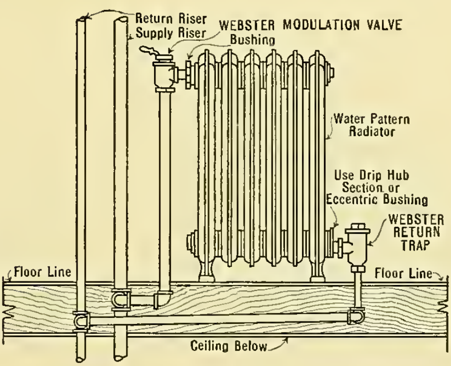


Fig. 22-18. Arrangement of connections to a radiator where the branch run-outs are in the floor construction.

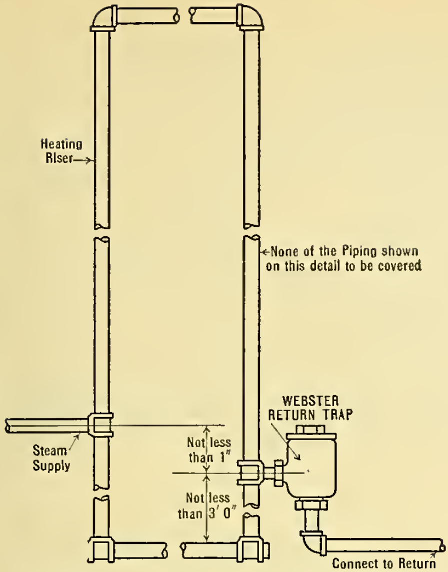


Fig. 22-19 A small amount of heating surface is often desired in certain classes of buildings in bath rooms, etc., without involving the expense of separate radiators. Where these rooms are one above the other a heating riser may be used with connections as shown in this diagram.

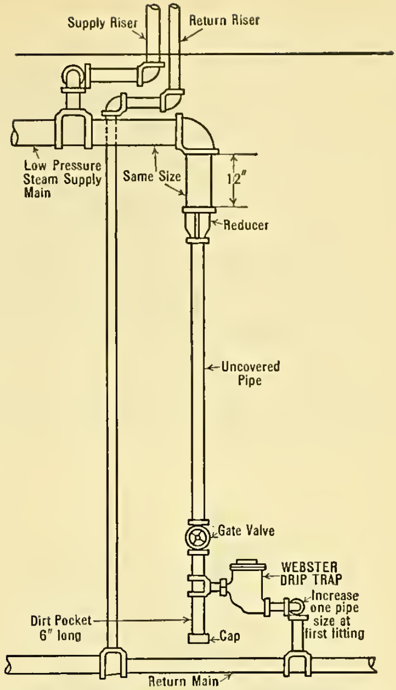


Fig. 22-20. The dripping of the end of an overhead steam supply main where the return line is carried along near the floor. The uncovered vertical line to the return trap acts as a cooling leg.

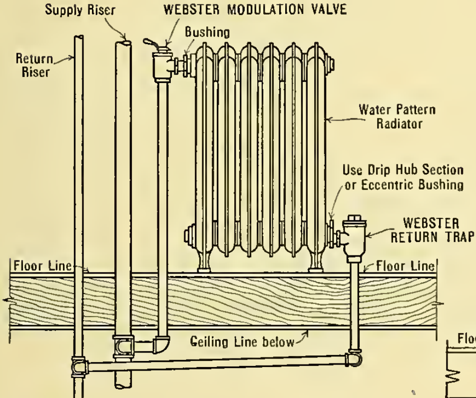


Fig. 22-21. Arrangement of connections to a radiator in a factory or loft building where there is no objection to branch run-outs on the ceiling of the floor below.

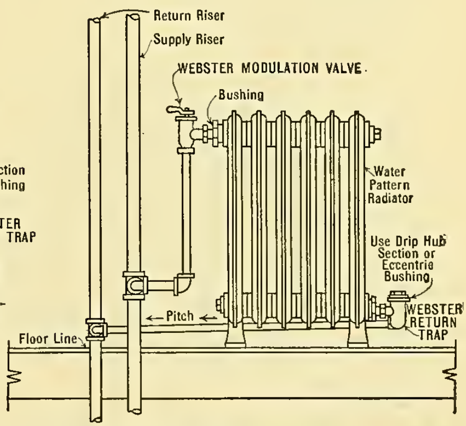


Fig. 22-22. Arrangement of connections to a radiator with all branch run-outs exposed in the room.

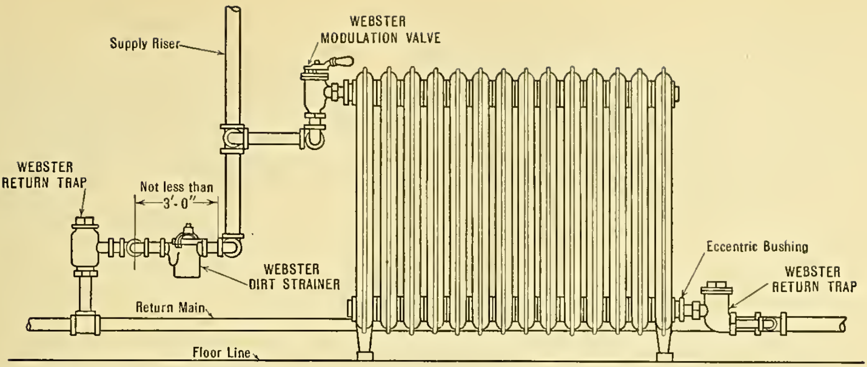


Fig. 22-23. Arrangement for removing a considerable amount of condensation from a down-feed riser. A drip goes through a Webster Dirt Strainer and Return Trap, the connection to lowest radiator being made above the drip point.

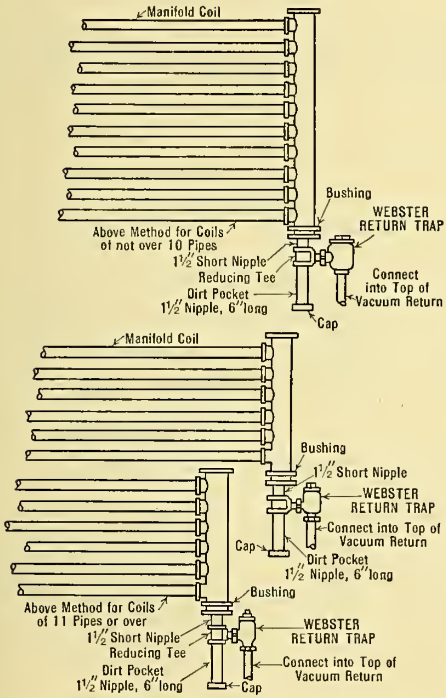


Fig. 22-24. Drip connections to the return headers of manifold coils. Coils of ten pipes or less have one return header and those of over ten pipes are usually split and provided with two headers.

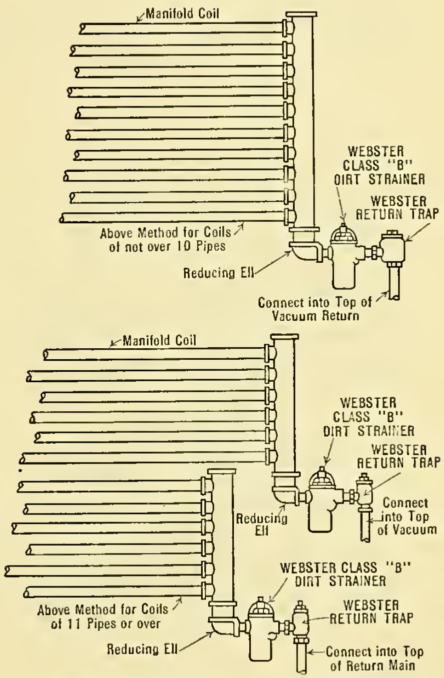


Fig. 22-25. Arrangement of headers similar to Fig. 22-24, but showing the use of the Webster Dirt Strainer at the entrance of the return traps.

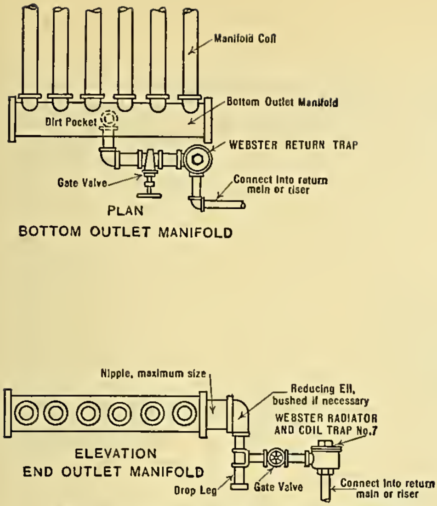


Fig. 22-26. With drop leg to catch dirt.

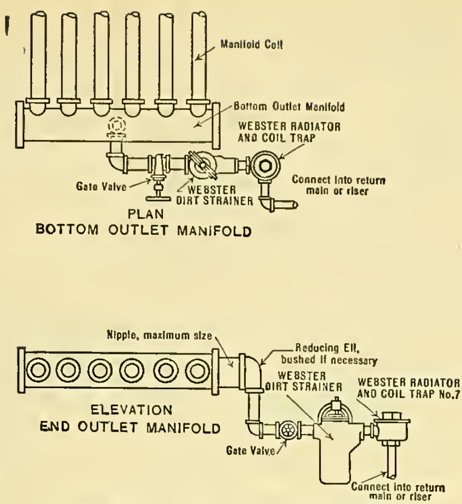


Fig. 22-27. With Webster Dirt Strainer.

Return connection to a flat overhead coil where (above) a bottom-outlet manifold and where (below) an end-outlet manifold is used. Dirt is collected by drop leg (Fig. 22-26) or by a Webster Dirt Strainer (Fig. 22-27).

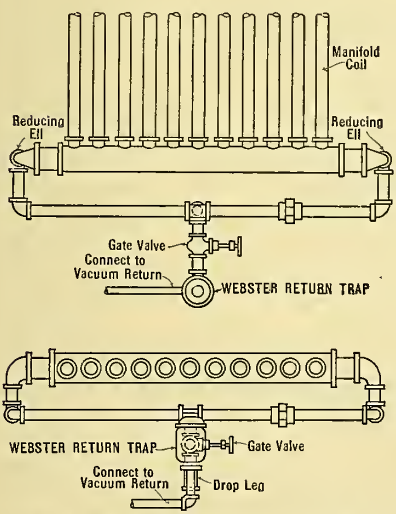


Fig. 22-28. With drop leg to catch dirt.

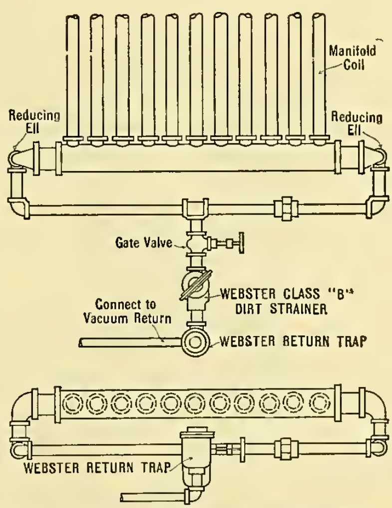
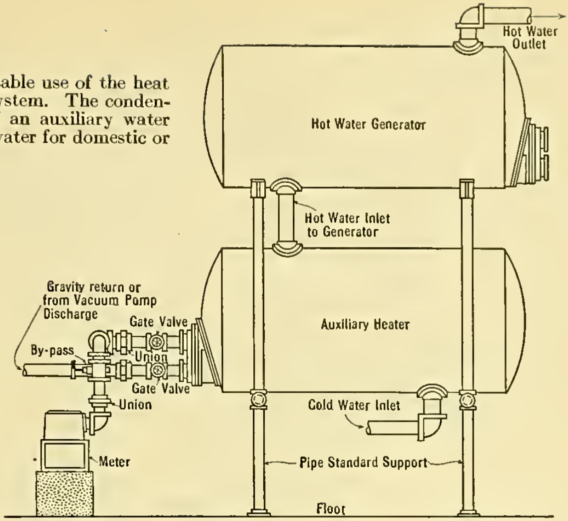


Fig. 22-29. With Webster Dirt Strainer.

Wide, flat, overhead coils should have return connections taken from both ends of the return manifold. Dirt is collected by a drop leg (Fig. 22-28) or by a Webster Dirt Strainer (Fig. 22-29).

Fig. 22-30. Arrangement for profitable use of the heat in the condensation from a heating system. The condensation is passed through the coils of an auxiliary water heater and its heat so transferred to water for domestic or manufacturing use.



NOTE—Additional details applicable to the Webster Modulation System will be found on page 00.

Details Applicable to the Webster Vacuum System Only

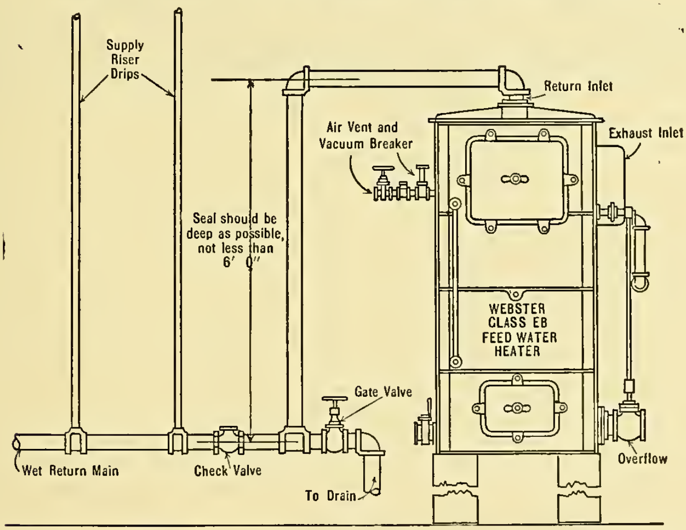


Fig. 22-31. Under certain conditions the condensation from the heels of downfeed risers can be removed by connecting the separate gravity drip or wet-return line to the return inlet of a Webster Feed-water Heater. In this instance, the static head between the top of the heater and the lowest radiator connection must exceed the pressure in the heater. Suitable connection of the return line to the heater is shown in the diagram.

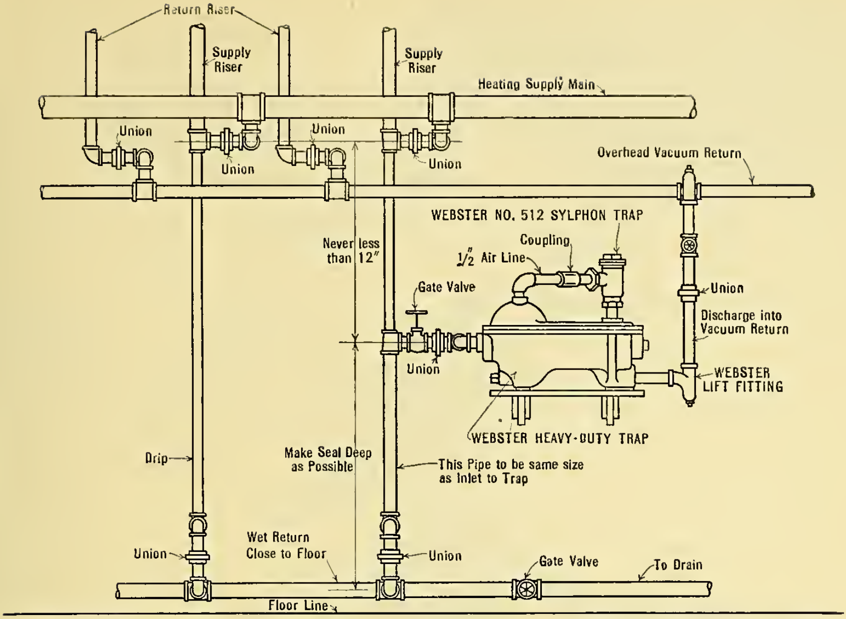


Fig. 22-32. Where the drips of risers and mains are carried through a separate gravity drip line near the floor and it is desired to deliver the condensation into the overhead vacuum return line through a Webster Heavy-duty Trap, the arrangement shown has proven most satisfactory.

Fig. 22-33. In the usual down-feed system where the drips of risers are cared for by a separate gravity drip line run near the floor and where the condensation is to be delivered to the overhead vacuum return line through a Webster Heavy-duty Trap, the method shown should be followed.

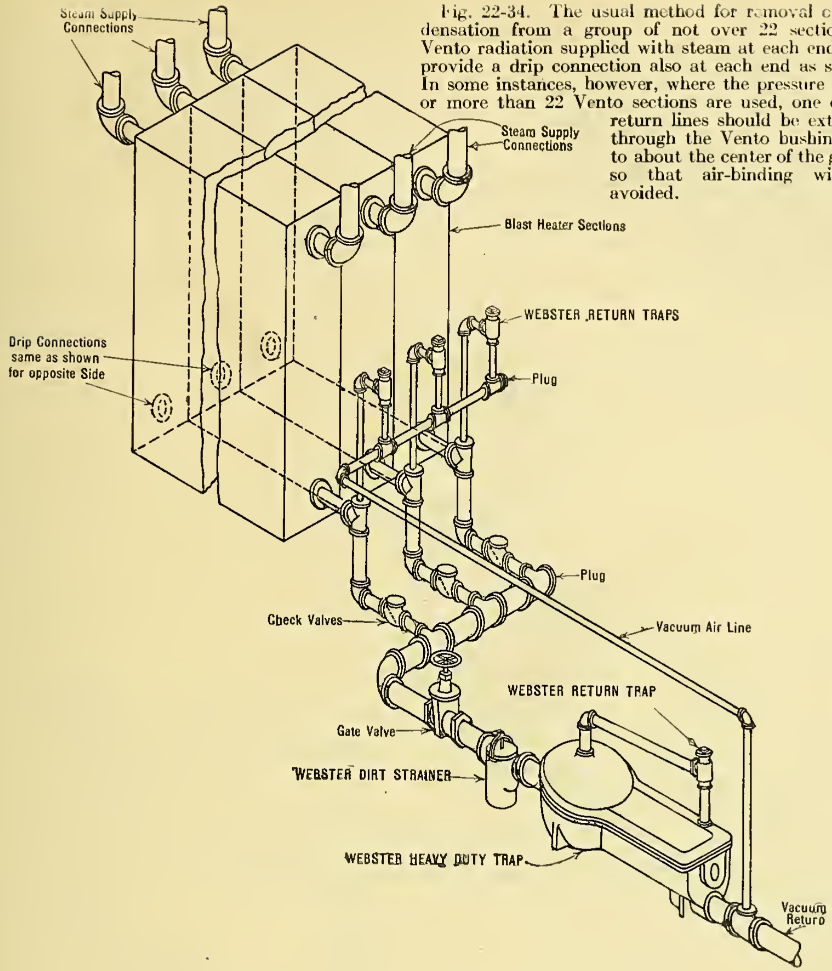


Fig. 22-34. The usual method for removal of condensation from a group of not over 22 sections of Vento radiation supplied with steam at each end is to provide a drip connection also at each end as shown. In some instances, however, where the pressure is low or more than 22 Vento sections are used, one of the return lines should be extended through the Vento bushing and to about the center of the group, so that air-binding will be avoided.

Fig. 22-35. The approved method of draining condensation from the coils of a hot-water service heater to the vacuum return line through gate valve, Webster Dirt Strainer and Webster Heavy - duty Trap.

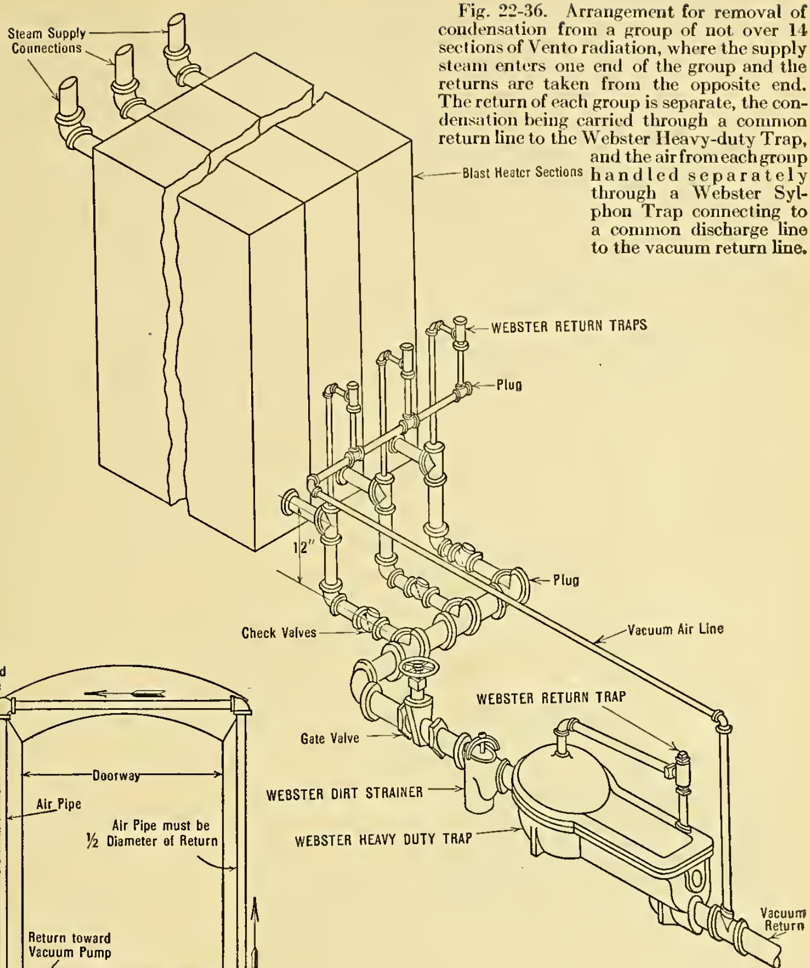


Fig. 22-36. Arrangement for removal of condensation from a group of not over 14 sections of Vento radiation, where the supply steam enters one end of the group and the returns are taken from the opposite end. The return of each group is separate, the condensation being carried through a common return line to the Webster Heavy-duty Trap, and the air from each group handled separately through a Webster Sylphon Trap connecting to a common discharge line to the vacuum return line.

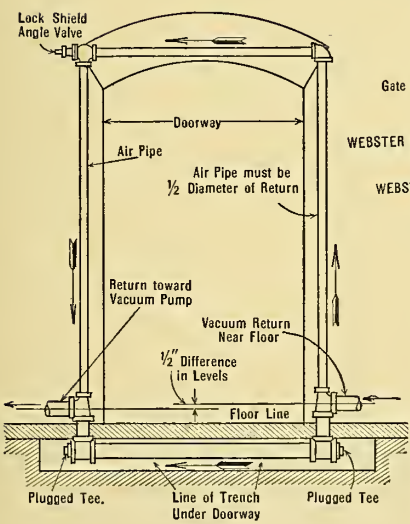


Fig. 22-37. Arrangement of piping where a vacuum return line is carried along the wall near the floor and passes doorways or other openings. The water is carried under the opening and the air is passed through the line over the opening.

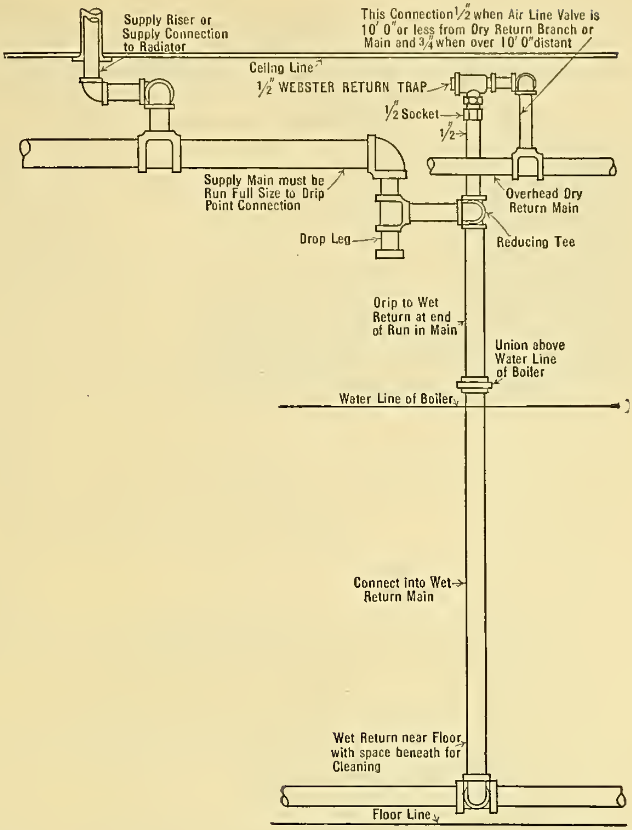


Fig. 22-38. Where a drip is required, at the end of a heating main, the air should usually be vented through a Webster Sylphon Trap into the dry return, as shown in this diagram.

Fig. 22-39. The dry return in a Webster Modulation System, due to its required grade, must sometimes get down into the head room, in which event it may be drained into the wet return and elevated to a higher level. Certain fundamentals must be observed in doing this. The most important is that at the point where the change in elevation occurs, the dry return must never be closer than 6 inches to the level of the inlet to the Webster Modulation Vent Trap.

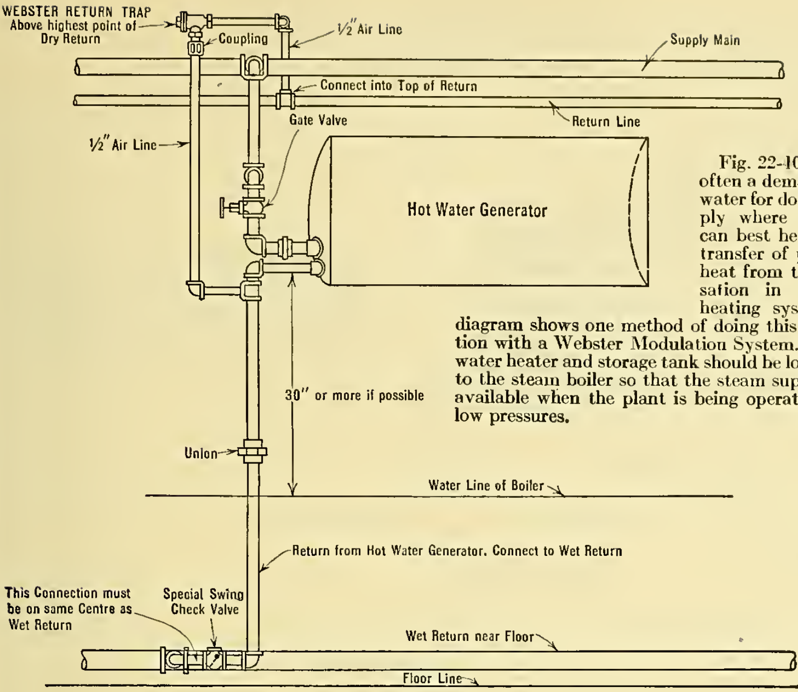


Fig. 22-40. There is often a demand for hot water for domestic supply where this water can best be heated by transfer of part of the heat from the condensation in the steam heating system. The diagram shows one method of doing this in connection with a Webster Modulation System. The hot-water heater and storage tank should be located close to the steam boiler so that the steam supply will be available when the plant is being operated at very low pressures.

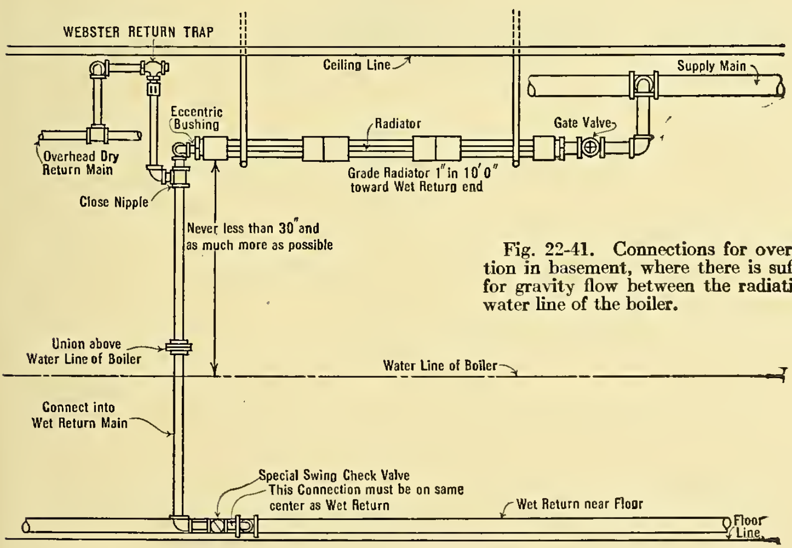


Fig. 22-41. Connections for overhead radiation in basement, where there is sufficient drop for gravity flow between the radiation and the water line of the boiler.

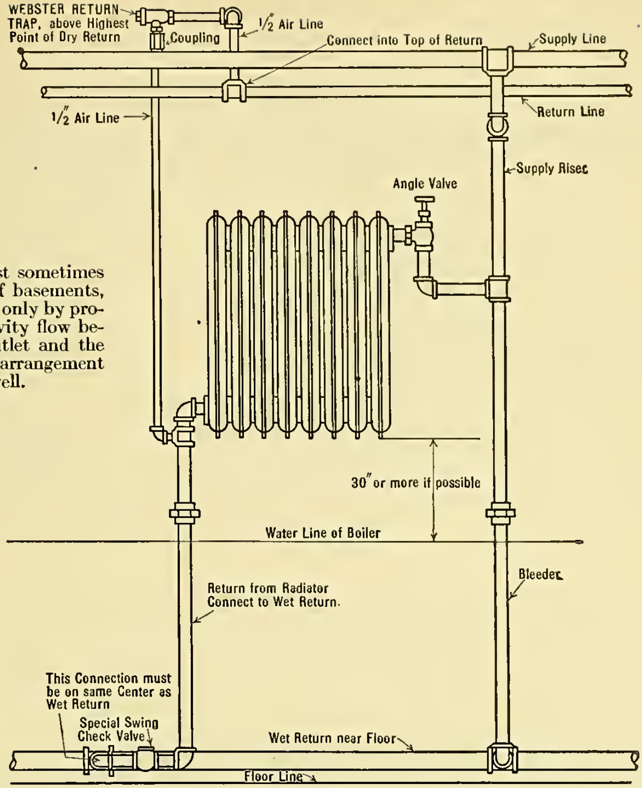


Fig. 22-43. Radiation must sometimes be placed on the side walls of basements, where steam can be circulated only by providing sufficient head for gravity flow between the radiator return outlet and the water line of the boiler. The arrangement shown handles this problem well.

CHAPTER XXIII

Appliances for the Webster Systems of Steam Heating

WEBSTER Appliances used as parts of Webster Heating Systems are illustrated and briefly described in the following pages.

These appliances include:

Webster Oil Separators
Webster Grease and Oil Traps
Webster Dirt Strainers
Webster Return Traps
Webster Heavy-duty Traps
Webster Double-service Valves
Webster Lift Fittings.
Webster Suction Strainers
Webster Vapor Economizers
Webster Vacuum-pump Governors
Webster Air-separating Tanks and Receivers
Webster Modulation Valves
Webster Modulation Vent Traps
Webster Damper Regulators
Webster Expansion Joints
Webster Pipe Anchors
Webster Hydro-pneumatic Tanks
Webster Low-pressure Boiler Feeders
Webster Conserving Valves
Webster Hylo Vacuum Controllers
Webster Hylo Vacuum Traps
Webster High-pressure Traps
Webster High-differential Heavy-duty Traps

Webster Return Traps for Automatically Removing Water of Condensation and Air from Heating Units: The return trap, to be perfect in operation, should—

(*a*) Allow the condensation to escape at a temperature slightly below that of the steam.

(*b*) Drain the radiator thoroughly by gravity, without the assistance of pressure or vacuum. A water-logged radiator loses efficiency because part of the heating is being done by the water condensed from steam, which is at lower temperature, and because a water-logged radiator is also an air-bound radiator.

(*c*) Permit continuous removal of air. An air-bound radiator loses efficiency because the steam cannot completely fill it.

(*d*) Automatically close to prevent loss or waste of steam.

(*e*) Work within the widest necessary range of pressure and vacuum variation.

(*f*) Require no adjustment under such variations.

(*g*) Be noiseless in operation, if used in rooms where noise is objectionable.

(*h*) Be so designed that the valve will close even when dirt may be present in normal quantities.

(*i*) Be durable and require little or no attention or repairs.

The efficiency of the radiator will depend upon how nearly the return trap meets these requirements.

A return trap working sluggishly will not only hold back the water, but will "bottle up" the air and air-bind the radiator, thus defeating the very purpose of a vacuum system.

As different methods must at times be employed in connection with

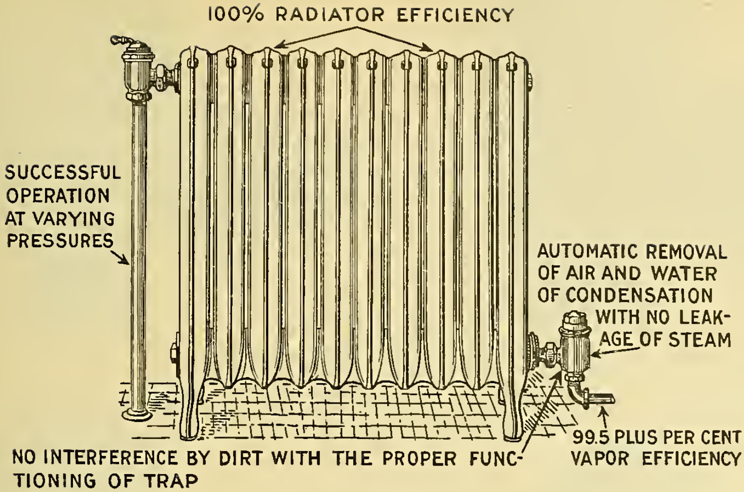


Fig. 23-1. The requirements of a perfect radiator trap.

direct radiators, blast sections, riser drips, main drips, dripping hot-water generators, factory coils, etc., Webster Return Traps are made in several forms, at least one of which will meet the requirements of any installation.

The type and capacity of the trap required depends upon the point of application, the amount of air and water to be removed, the character of the heating surface and the pressure and vacuum carried. Suggestions as to the proper sizes of traps for specific conditions will therefore be of value and freely given upon request.

The Webster Sylphon Trap

The Webster Sylphon Trap has been specially designed to meet the requirements for a perfect radiator trap. It maintains the highest possible efficiency within the heating surface by the removal of all of the products of condensation, and as this is effected without loss of steam, it is economical in the highest degree. The economy is especially apparent when reduced-pressure live steam is used in whole or in part, or where under present operating conditions it may be necessary to waste large quantities of cold water to cool the heating system returns before they enter the vacuum pump.

The operating member consists of a Sylphon bellows, which carries a conical-shaped valve piece, closing against a sharp-edged seat. The bellows member is supersensitive, operating to close or open the valve port by the slightest change in the temperature of the surrounding medium, and is the most durable form of thermostatic device so far known. The multiple construction of the seamless brass folds forming the bellows distributes the strain of movement and increases the life of the operating member. The increase in the pressure on the outside of the bellows is compensated by the increase in pressure on the inside of the bellows.

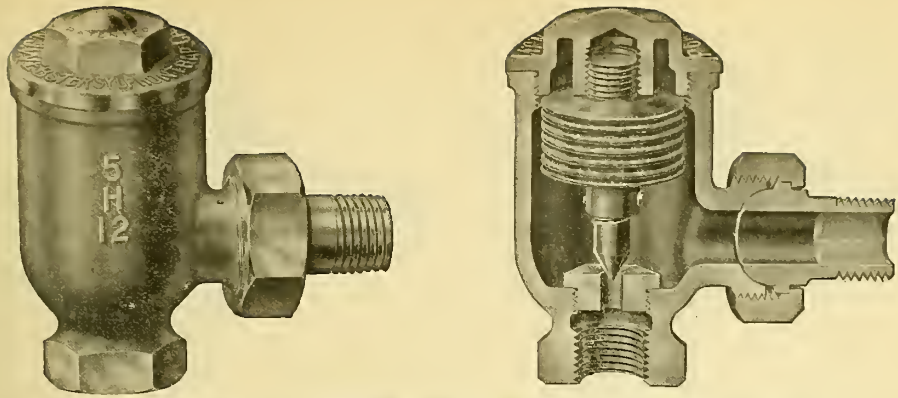


Fig. 23-2. No. 512 Model H Webster Sylphon Trap. Size of pipe connections, ½-inch.

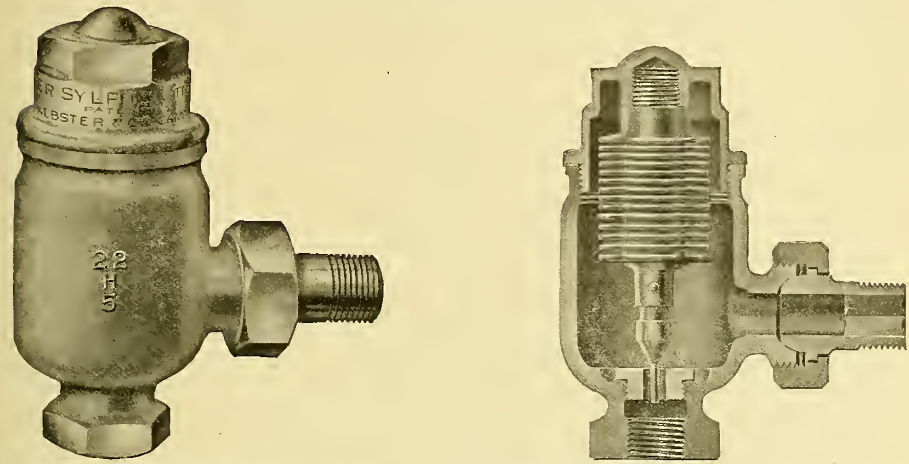


Fig. 23-3. No. 522 Model H Webster Sylphon Trap. Size of pipe connections, ½-inch. Nos. 512 and 522 differ in capacity rating and lift of valve, No. 522 being of greater capacity.
No. 523 has same size body mechanism and capacity as No. 522, but has ¾-inch pipe connections to meet unusual specifications in that respect.

The sensitiveness of this member is due to the flexibility of the walls of the bellows to movement in the desired direction and the small amount of movement of each fold when acted upon by the pressure surrounding and also that generated within the bellows. The sum of the small movement of each of the many folds gives a greater total lift of the valve than any other device for similar purpose.

The conical valve piece and sharp-edged seat give increased capacity for discharge of water, and the valve does not become inoperative due to presence of dirt and scale.

The Webster Sylphon Trap will close quickly and positively when steam reaches the bellows, while the water and air will be freely withdrawn or dis-

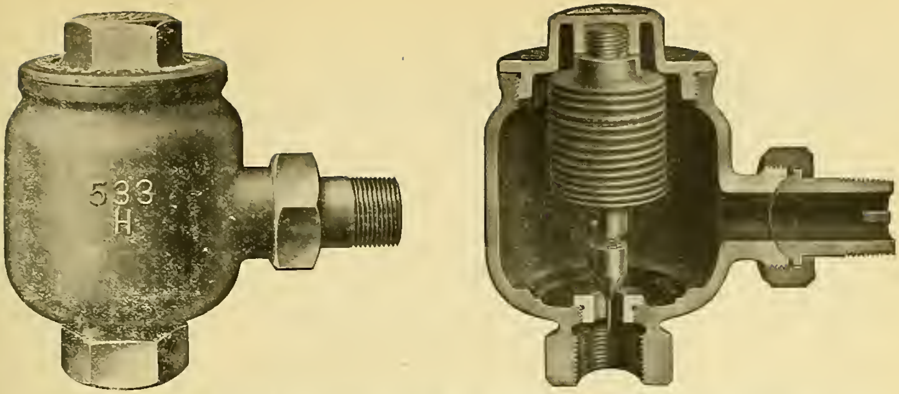


Fig. 23-4. No. 533 Model H Webster Sylphon Trap. Size of pipe connections, ¾-inch.
No. 534 Has same size body with 1-inch pipe connections to meet unusual specifications.
No. 544 Similar, but larger throughout for 1-inch pipe connections and greater duty.
No. 545 The largest in proportions and capacity. For 1¼-inch pipe connections.

charged at slightly below the temperature of the steam at the existing pressure.

This means that every radiator in use will be thoroughly efficient in heating, as there will be no "pocketing" of air or "bottling up" of water within the radiator.

As the valve is full open when cold, the radiator will be fully drained when steam is turned off, and the vacuum condition existing in the return line will extend within the radiator, assisting circulation when steam is again turned on.

Operation: As the steam first flows into the cool radiator, it expels the contained air and initial condensation through the wide-open trap. As the radiator warms up from inflow of steam, the bellows commences to expand, but remains partially open as long as the air and water in the trap are at a lower temperature than that of the steam. The moment the air is entirely expelled from trap body, and replaced with steam, the valve closes. It opens again when water and air at a temperature slightly less than that of the steam accumulate in the trap. Then, as the water and air escape and are replaced in the trap body by steam, the trap again closes, thus completing its cycle.

Model H
Angle

Model G
Straightway offset

Model R
Right corner

Model L
Left corner

Fig. 23-5. Bodies of Webster Series S Sylphon Traps.

Table 23-1. Models and Dimensions of Sylphon Traps

For convenience in making pipe connections, Webster Series 5 Sylphon Traps of the smaller sizes are made with four types of bodies as shown. Model H or angle is the one most used.

Size	Trap Nos. & Model	A	B	C	D
1/2"	512H	3"	1 5/8"	1 1/8"	4 1/8"
1/2"	522H	3 3/8"	1 7/8"	1 3/16"	5 1/4"
3/4"	523H	3 3/8"	2"	1 3/16"	5 1/4"
3/4"	533H	4 1/16"	2 3/8"	1 3/4"	5 3/8"
1"	534H	4 1/8"	2 5/8"	1 3/4"	5 13/16"
1"	544H	4 5/16"	2 9/16"	2"	6 13/16"
1 1/4"	545H	4 1/2"	2 9/16"	2"	6 13/16"

Size	Trap Nos. and Model	A	B	C	D	E
1/2"	512G, 512R or 512L	3"	1 1/2"	1"	4 1/4"	1 1/2"
1/2"	522G, 522R or 522L	3 3/8"	1 5/8"	1 1/4"	5 3/8"	1 5/8"
3/4"	523G, 523R or 523L	3 3/8"	1 13/16"	1 1/4"	5 9/16"	1 7/8"
3/4"	533G	4 7/16"	2 7/16"	1 3/4"	5 3/4"	Not
1"	534G	4 1/8"	2 7/16"	1 3/4"	6"	made

For ratings, see Chapter 13, page 00.

The Webster No. 7 Trap

Webster No. 7 Traps also realize all of the requirements for thoroughly satisfactory operation as radiator traps. They are applied at the outlets of steam radiators and coils, at drip points on steam supply lines and risers and at the outlets of blast sections on fan coils and provide continuous

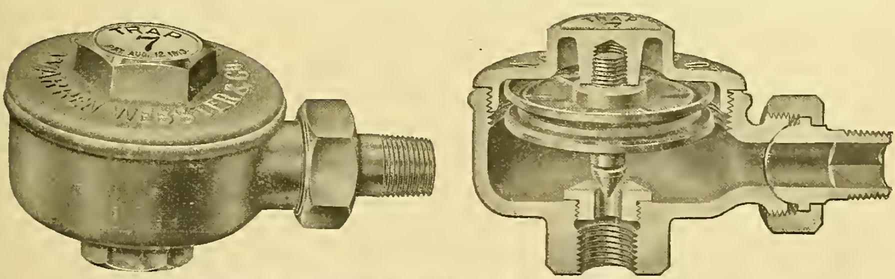


Fig. 23-10. Exterior and interior of No. 722 Webster Trap.

free and thorough removal of entrained air and water of condensation, without permitting any live steam to escape to waste in the return lines.

The inlet of the trap is attached to the radiator, coil or supply line by means of the union connection, and the outlet is piped into the return line.

The diaphragms, which form the active part of the operating member, are built of four successive phosphor-bronze plates instead of the usual two and for that reason there is greater diaphragm movement and the valve has greater lift than usually found in traps of similar types.

The expansion and contraction of the diaphragm member is produced by differences in volume and pressure of a hermetically-sealed fluid charge in response to changes in temperature. Even a very slight temperature change produces a powerful force to actuate the conical valve piece, which in closing, fits tightly on a sharp-edged seat.

No part of the valve mechanism is impaired by normal quantities of the scale and dirt that collect in steam-heating systems.

Table 23-2. Models and Dimensions of Webster Series 7 Traps

For convenience in making pipe connections, Webster Series 7 Traps are made with four types of bodies as shown. Model H or angle is the one most used.

Size	Trap No.	A	B	C	D	E
½″	712H	3¼″	$1\frac{7}{16}$″	$1\frac{19}{32}$″	$2\frac{15}{16}$″	
½″	722H	3½″	$1\frac{7}{16}$″	1⅞″	$3\frac{3}{16}$″	
¾″	723H	3½″	$1\frac{9}{16}$″	1⅞″	$3\frac{5}{16}$″	
¾″	733H	4¾″	1⅞″	2¼″	$4\frac{9}{16}$″	
1″	744H	4¾″	2″	2½″	$4\frac{9}{16}$″	
1¼″	745H	4¾″	2″	2½″	$4\frac{9}{16}$″	
½″	712G, 712R, 712L	3¼″	2⅛″	¾″	$3\frac{1}{16}$″	2⅛″
½″	722G, 722R, 722L	3½″	2¼″	¾″	3½″	2¼″

For ratings see Chapter 13, page 00.

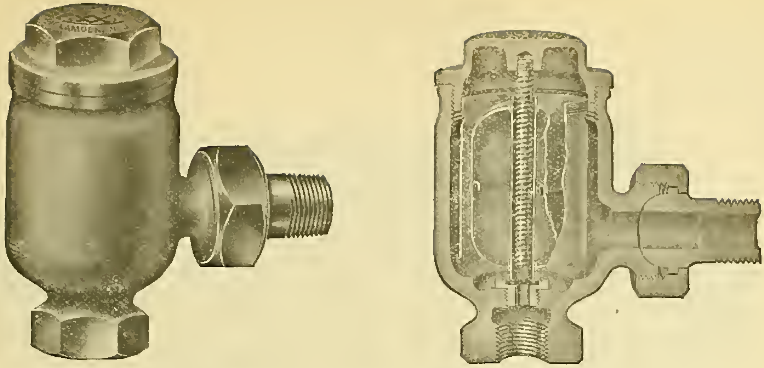


Fig. 23-15. No. 522 Water-seal Trap

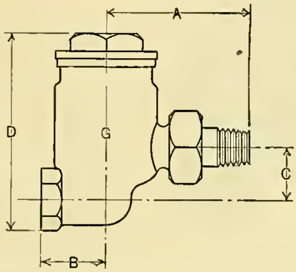
The Webster Water-seal Trap

This is an older design of automatic balanced valve, requiring no adjustment. It is suitable for blower apparatus and for basement main drips and for draining coils in factory installations.

It passes the maximum quantity of condensation, hot or cold, and at the same time relieves the heating unit of air, without waste of uncondensed steam.

Table 23-3. Models and Dimensions

For convenience in making pipe connections, Webster Water-seal Traps are made with four types of bodies, which have the same characteristics as the four body types for Webster Sylphon Traps as shown on page 23-4



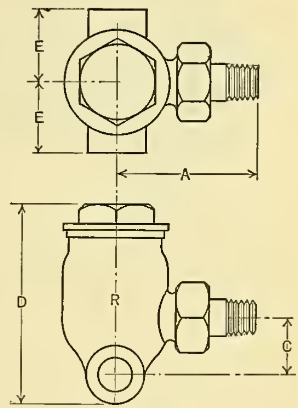


Fig. 23-16

MODELS G R and L

Size	A	B	C	D	E
½″-512	3″	1½″	1″	4¼″	1½″
½″-522	3⅜″	1⅝″	1¼″	5⅜″	1⅝″
¾″-523	3⅜″	1 13/16″	1¼″	5 9/16″	1⅞″
¾″-533	4 1/16″	2 7/16″	1¾″	5¾″	Not
1″-534	4⅛″	2 7/16″	1¾″	6″	Made

MODEL H

Size	A	B	C	D
½″-522	3⅜″	1⅞″	1 3/16″	4 9/16″
¾″-523	3⅜″	2″	1 3/16″	4⅝″
¾″-533	4 1/16″	2⅜″	1¾″	5⅜″
1″-534	4⅛″	2⅝″	1¾″	5 13/16″
1″-544	4 5/16″	2 9/16″	2″	6 13/16″
1¼″-545	4½″	2 9/16″	2″	6 13/16″

For ratings, see Chapter 13, Page 00

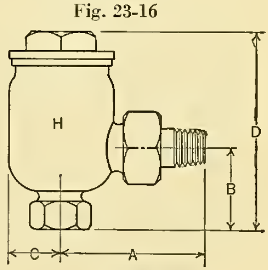
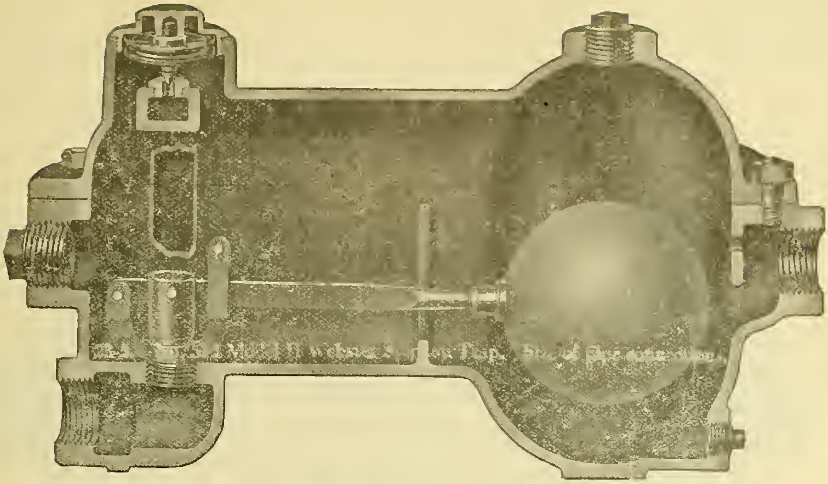


Fig. 23-17



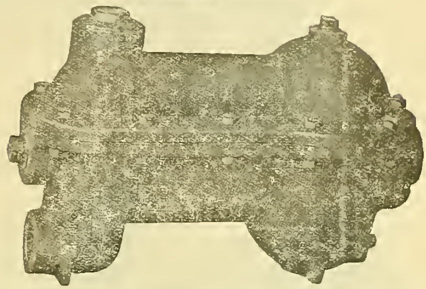


Fig. 23-18. Series 19T Webster Heavy-duty Trap with thermostatically-controlled air by-pass.

Series 19T with Thermostatically-controlled Air By-pass. For 15-lb. Maximum Operating Pressure:—The Webster Heavy-duty Trap handles unusually large quantities of condensation, and is for dripping main supply risers or main entering or leaving the building, for draining large sections of blower coils or pipe manifolds, for draining hot-water generators, etc.

Insofar as the discharge of condensation is concerned, this trap operates on the float principle and has a large water outlet to get the condensation away as quickly as possible from the unit to be drained.

Air is eliminated by means of a thermostatically actuated by-pass, as shown in Figure 23-18. The operating device, the valve piece and seat are the same as used in the Webster No. 7 Trap.

The body is of cast iron, as is also the cover, which is bolted on, easily removable and so designed that all interior parts are exposed for inspection upon its removal. The outlet is in the bottom of the body, and the inlet may be on either end, with the opposite opening plugged. It is recommended that wherever practical the inlet farthest away from the valve be used. An opening is provided at the bottom of the float chamber as a clean-out by-pass and for draining the trap when out of use.

The float has ample leverage to avoid sticking of the valve. The cone-pointed valve and square-edged seat prevent accumulation of dirt where it

might clog the port. The valve is water-sealed at all times, as the water level is always well above the seat. The float lever is kept within the vertical plane of action by guide flanges cast into the trap body.

For dimensions, see page 000

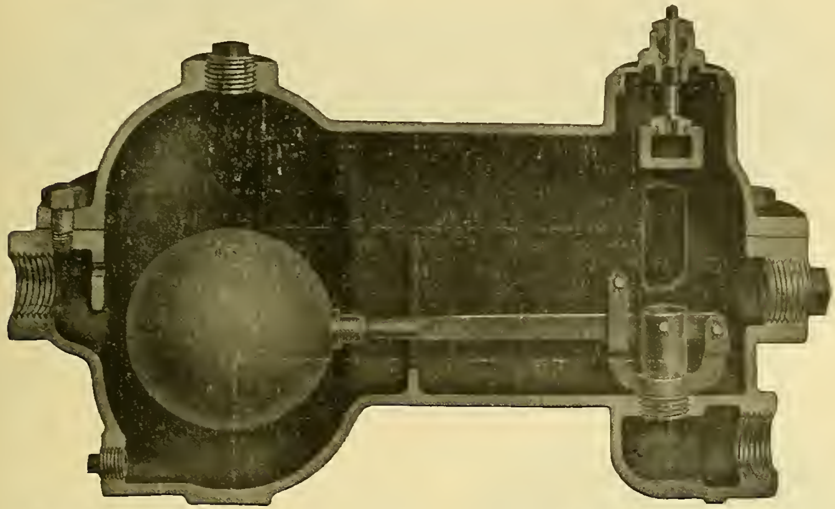
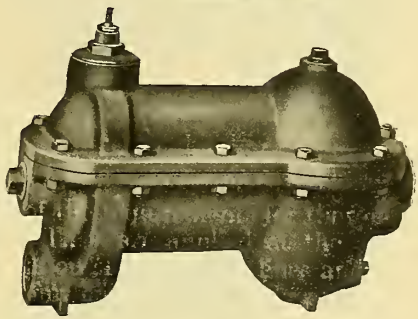


Fig. 23-19. Series 19-H Webster Heavy-duty Trap with hand-controlled air by-pass.



Series 19H with Hand-Controlled Air By-pass. For 15 lb. Maximum Operating Pressure:—This type of Webster Heavy-duty Trap is essentially the same in construction and is applied for exactly the same operating conditions as the Series 19 T Trap before described, but is designed to meet a requirement for manual adjustment of the air vent.

As shown in Figure 23-19, the air port is controlled by means of a conical valve closing upon a sharp-edged seat. The amount of air leakage is controlled by the use of a wrench in backing the valve off its seat as much as necessary or advisable. In all other respects the construction is identical with that of the Series 19 T Trap.

For dimensions, see page 000.

High-Differential Type Series 20, for Working Pressures up to 50 lb. per sq. in.:—The Webster High-differential Heavy-duty Trap is recommended for steam pressures higher than 15 pounds and where large quantities of condensation may be discharged. It is particularly applicable to problems like or similar to those described in Chapter 16.

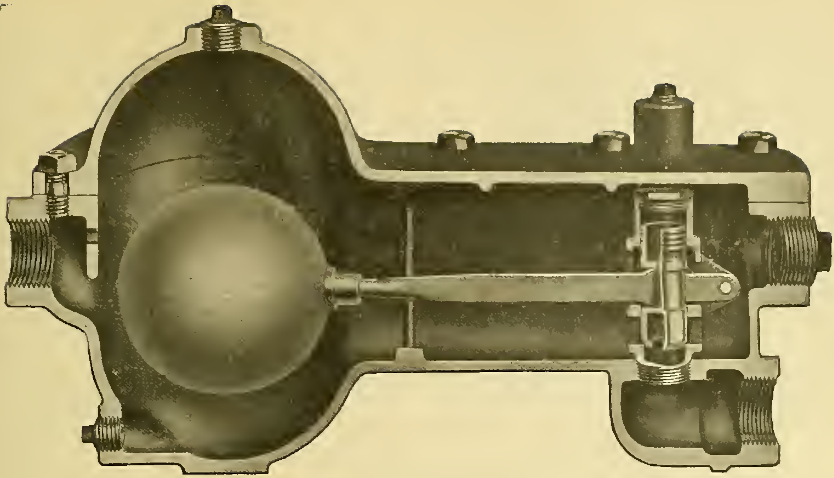
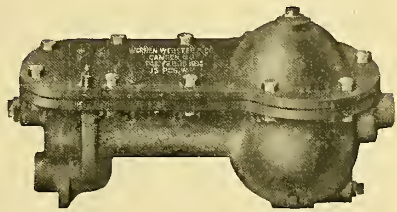


Fig. 23-20. Series 20 Webster High-differential Heavy-duty Trap for working pressures up to 50 lb. per sq. in.



The trap body is constructed of cast iron and has an easily removable cover of the same material. The valve is of the balanced type and operates against a monel-metal seat. The ball float is extra heavy to withstand the higher pressures.

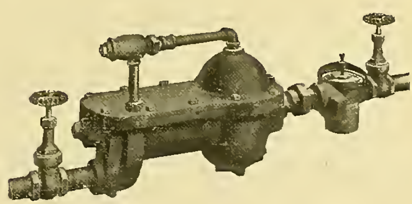


Fig. 23-21. Conventional arrangement of Series 20 Webster High-differential Heavy-Duty Trap and Webster Dirt Strainer (Inlet pipe may be connected to opposite end if desired.)

The Webster High-differential Heavy-duty Trap may be operated with a constant leakage through a hand-adjusted air vent, though the best practice calls for control of the air discharge by means of a thermostatically actuated valve in a by-pass of pipe and fittings as shown in Figure 23-21.

Table 23-4. Dimensions of Webster Heavy-duty Traps

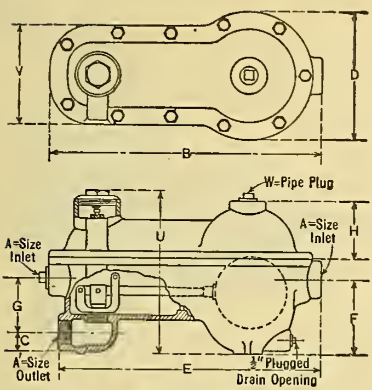


Fig. 23-23. Series 19H

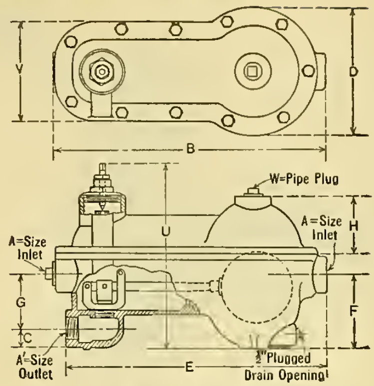


Fig. 23-22. Series 19T

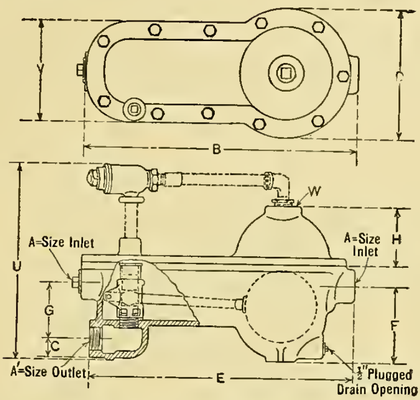


Fig. 23-24. High-differential

All dimensions in inches and subject to slight variation.

Number	A	A¹	B	C	D	E	F	G	H	U	V	W
Series 19T, with thermostatically controlled by-pass.												
0019-T	¾	¾	13¼	1	7½	12⅝	4⅛	3⅛	2½	9⅞	5⅝	½
019-T	¾	¾	15¾	1	8	15	4⅛	3⅛	2⅞	10⅛	6¼	¾
119-T	1¼	1¼	19⅛	1⅜	9	18⅜	5⅜	3⅞	4⅛	10¾	7	1
219-T	2	2	20⅝	1⅝	10½	19⅞	6⅛	4⅝	4⅜	12¾	8	1½
Series 19H, with hand controlled by-pass.												
0019-H	¾	¾	13¼	1	7½	12⅝	4⅛	3⅛	2½	10½	5⅝	½
019-H	¾	¾	15¾	1	8	15	4⅛	3⅛	2⅞	10⅞	6¼	¾
119-H	1¼	1¼	19⅛	1⅜	9	18⅜	5⅜	3⅞	4⅛	12⅞	7	1
219-H	2	2	20⅝	1⅝	10½	19⅞	6⅛	4⅝	4⅜	14⅜	8	1½
Series 20, High-differential type.												
020	¾	¾	15¾	1	8	15	4⅛	3⅛	2⅞	12¼	6¼	¾
120	1¼	1¼	19⅛	1⅜	9	18⅜	5⅜	3⅞	4⅛	13¾	7	1
220	2	2	20⅝	1⅝	10½	19⅞	6⅛	4⅝	4⅜	14⅞	8	1½

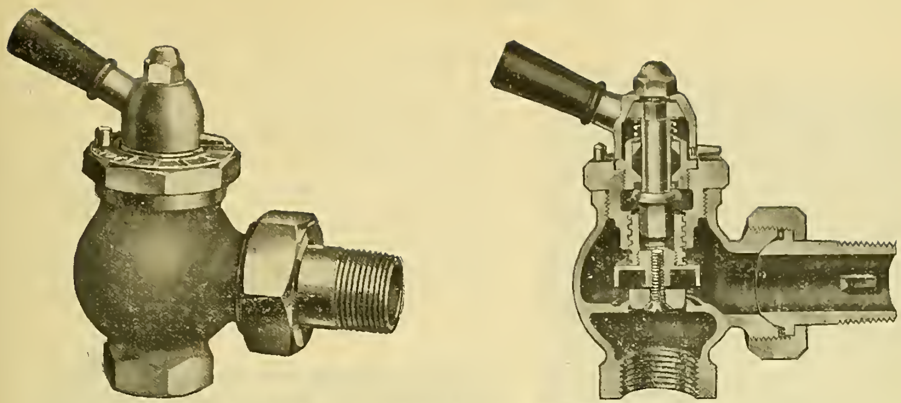


Fig. 23-25. The Webster Type W Modulation Valve.

The Webster Type W Modulation Valve

The Webster Type W Modulation Valve is a special-purpose radiator valve of the quick-opening, non-rising stem, straight-lift type, built for complete opening or closing with less than a single turn of the handle. Its manipulation is as simple and its control as effective as the movement that regulates light from a gas jet.

As the name implies, the principal function of the Webster Modulation Valve is to facilitate "Modulation" of temperature in each room according to the desires of the occupant, by varying the amount of steam admitted to the radiator or coil. A pointer attached to the handle traveling over a graduated dial indicates the amount of valve opening at all times.

With the valve full open, the discharge capacity through the ports is equal to that of the outlet connection of the valve.

Less than three-fourths of the valve lift and opening movement is required to produce modulation up to normal full heating requirement. The rest is in reserve to admit more steam during the heating-up period, as needed to compensate for the higher condensation rate caused by contact with the cold radiator and its surrounding air.

Construction Details: The modulation effect is produced by a patented limiting sleeve (see Figure 23-25) which varies admission of steam in progressive volume with the lift of the valve piece.

A Jenkins disc is used to insure tight closing. With the exception of this and the handle, all parts are of brass. The handle is of special composition and so formed that the hand of the operator does not come into contact with the heated surface of the valve body.

Application: The Webster Modulation Valve may be used on either hot-water type radiators (having connections from section to section at both top and bottom) or with steam type radiators (bottom connections only), although the former type is preferable.

Where the Webster Modulation Valve is used with the hot-water type of radiator, it should be placed at the top to bring the operating handle in

the most convenient location and to permit the steam to circulate across and downward. The air and condensation, being heavier, fall to the bottom in advance of the steam and give full efficiency to the part of the radiator heated.

Where the Webster Modulation Valve is used with a steam type radiator, or placed at the bottom of a hot-water type radiator, the response cannot be so quick with the Modulation Valve or with any other type of inlet valve for that matter, because some air is temporarily pocketed in the far sections. This air must find its way out through the return trap before full

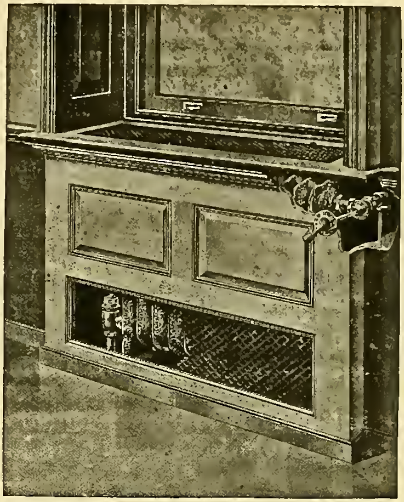


Fig. 23-26. Typical application of the extension stem principle.

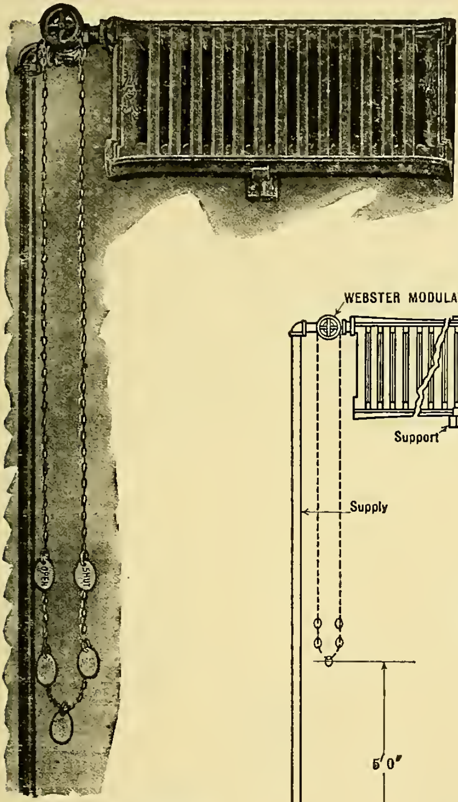


Fig. 23-27. Typical application of chain attachment to Webster Type W Modulation Valve.

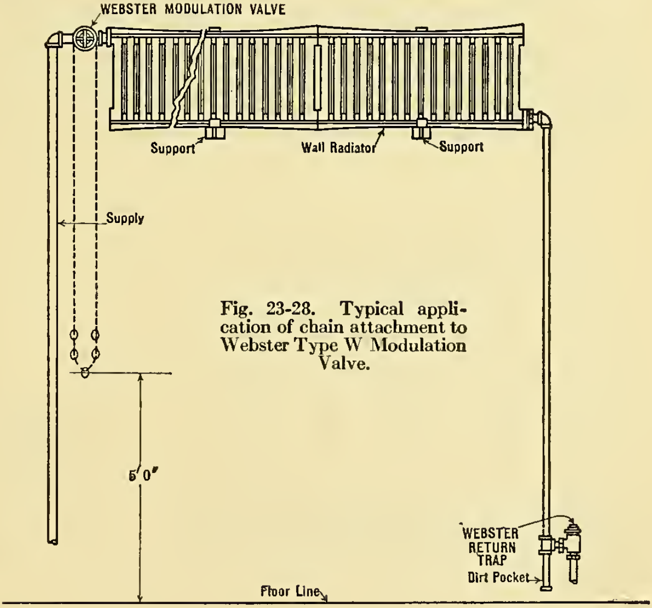


Fig. 23-28. Typical application of chain attachment to Webster Type W Modulation Valve.

efficiency is possible. As the inlet connection is always at the bottom of the steam type radiator, the Modulation Valve must be there also; a comparatively inconvenient position for operating.

If placed at the bottom of radiators, because other connections cannot be arranged, the inlet bushing should be eccentric and so located that the center line of the radiator inlet is above that of the radiator outlet. This is essential to prevent condensation from draining by gravity through the supply instead of the return connections, thus eliminating water-hammer.

Extension Stem: For attachment to radiators concealed in recesses or under window seats behind grilles, the Webster Modulation Valve is provided with an extension stem and a special dial that may be placed on the face, top or end of the grille or seat (see Figures 23-26).

The stem has a universal joint on each end, which permits operation of the valve from a point not directly in line with the valve stem, and at the same time provides enough play to avoid sticking or binding from any misalignment or shifting caused by expansion and contraction. This construction also avoids the difficulty of making very accurate stem connections.

The outside indicator dial, pointer and handle are similar to those used on top of the standard valve.

Chain Attachment: The Webster Modulation Valve to be applied to radiators or coils located in skylights, overhead, or on walls near the ceiling, can be fitted with a chain attachment for convenience in obtaining every advantage of the modulation feature (Figures 23-27 and 23-28).

The chain wheel is substituted for the handle of the standard type of Modulation Valve and the chain is made just long enough to permit easy grasp from the floor. Tags are attached to the lower portion of the chain in such positions that the end hanging at the bottom indicates the degree of valve opening.

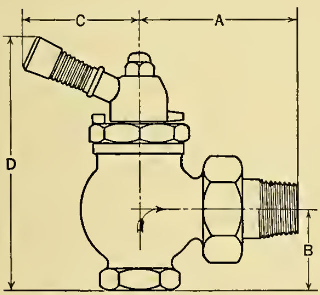


Fig. 23-29.

Table 23-5. Dimensions of Type W Modulation Valve.

Size	A	B	C	D
½	2⅞	1¼	2⅜	4⅜
¾	3⅛	1½	2⅜	4⅞
1	3⅜	1¾	2⅜	5⅝
1¼	3¾	2	2⅜	6⅛

All dimensions in inches and subject to slight variation.
For ratings, see Chapter 13, page 00.

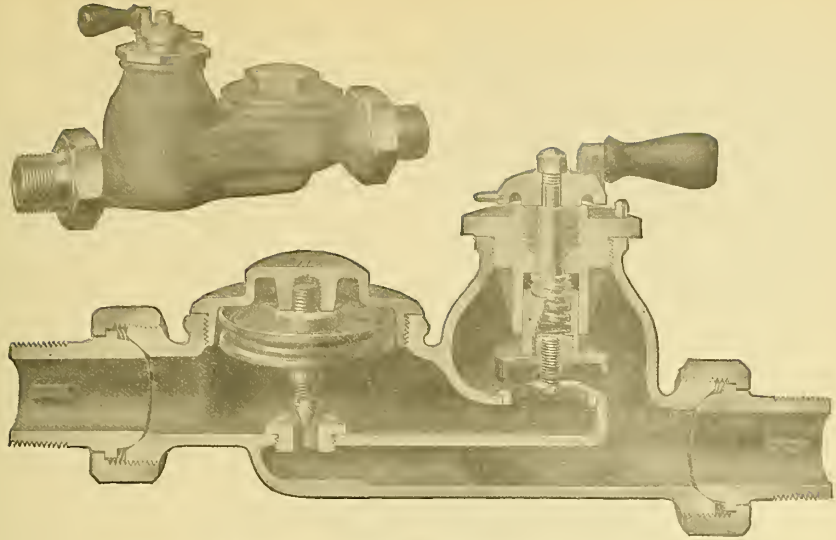


Fig. 23-30. The Webster Double-Service Valve.

The Webster Double-Service Valve

This is one of the latest developments of apparatus for simplifying piping connections in steam heating systems in certain types of construction.

Common practice in buildings of only one story and in some other instances calls for a steam supply line along the ceiling of the first floor to feed each radiator or coil through a short down-feed riser, which must be dripped into the return line. This multiplicity of unsightly connections is simplified by the use of Webster Double-service Valves, applied in the manner shown in Figure 23-31.

This valve performs "double service," as a supply valve for the radiator and as a trap for draining the riser.

The thermostatically controlled valve is open, when there is water or air in the riser, and permits the condensate to flow through a by-pass in the valve body into the radiator and thence to waste. Upon the presence of steam the thermostatic member expands, closes the valve, and thus prevents waste of steam.

Steam is admitted to the radiator in the amount desired, by means of the quick-opening valve, which is provided with a graduated dial and handle. This is not a Webster Modulation Valve, as the valve disc is designed for the special feature of quick opening without respect to modulating effect.

The valve body is best-quality cast iron, and all other parts except the valve disc and handle are brass. Nuts and nipples are provided at each connection to promote easy installation.

The thermostatic member, which is built up of four discs of phosphor bronze and filled with a volatile fluid, the conical valve piece and the sharp-

edged seat are of standard pattern as used in the Webster No. 7 Trap.

The inlet valve is provided with a ring seat and Jenkins disc, to insure tight closing. Its quick-opening feature is provided by a screw stem of such pitch that the valve will be completely opened with less than a complete turn of the handle.

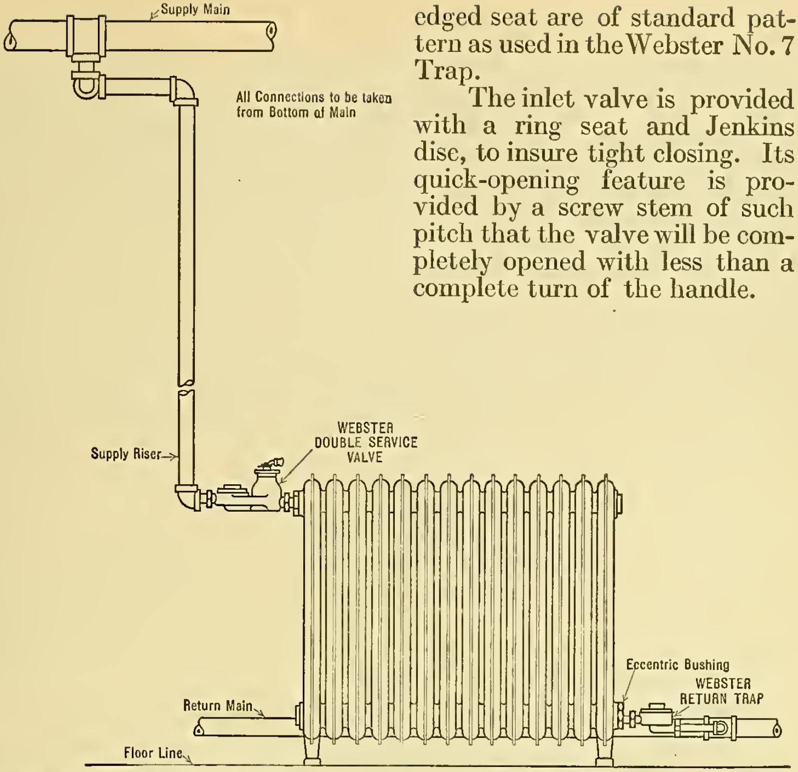


Fig. 23-31. Application of a Webster Double-service Valve to a standard cast-iron radiator.

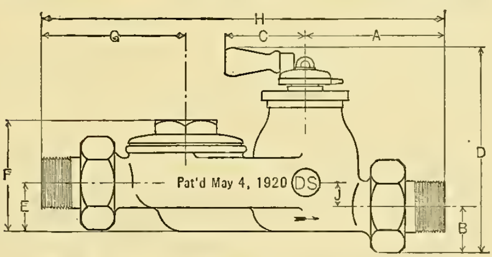


Fig. 23-32. The Webster Double-service valve

Table 23-6. Dimensions of Webster Double-service Valves in Inches.

Size	A	B	C	D	E	F	G	H	J
3/4	3 1/4	1	2 1/8	5 3/8	1 1/4	2 3/4	3 3/4	10	5/8
1	3 5/8	1 1/4	2 1/8	5 1/2	1 3/8	3	3 7/8	10 1/2	5/8
1 1/4	4	1 1/2	2 7/8	6 3/4	1 1/2	3 1/8	3 7/8	11 1/4	5/8
1 1/2	4 1/2	1 5/8	2 7/8	8	1 5/8	3 3/8	4	12 1/8	5/8

Webster Oil Separators of the standard types for removing cylinder oil, grease, etc., from currents of exhaust steam have steel multi-baffles, formed by a number of hooked plates interposed to the flow of the steam in a way to cause separation by impact, by change of direction and by adhesion. There is no unobstructed path through any Webster Oil Separator, yet the free area through which steam must pass is several times greater than the area of the inlet and outlet, thus minimizing the pressure loss due to friction.

The use of these separators protects boiler heating surfaces and interior surfaces of heating systems from the oil deposits that otherwise seriously impair heat transmission and often cause serious damage.

These separators may also be used for such special purposes as removing moisture or oil from compressed air and other gases.

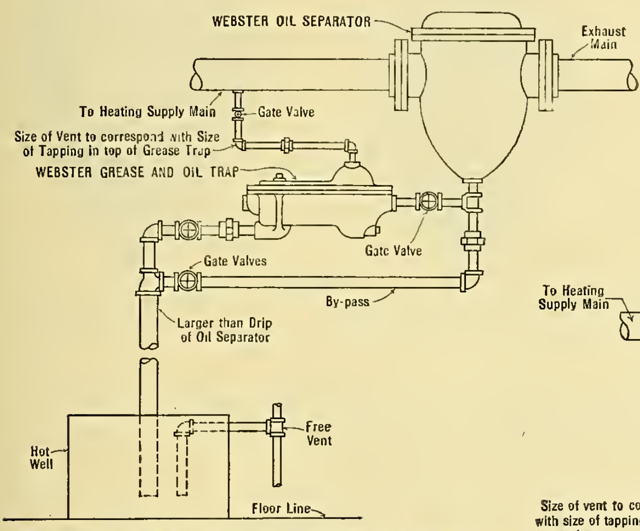


Fig. 23-33. Method of connecting a Webster Grease Trap to a Webster Oil Separator, where a partial vacuum may at times be carried on the heating main.

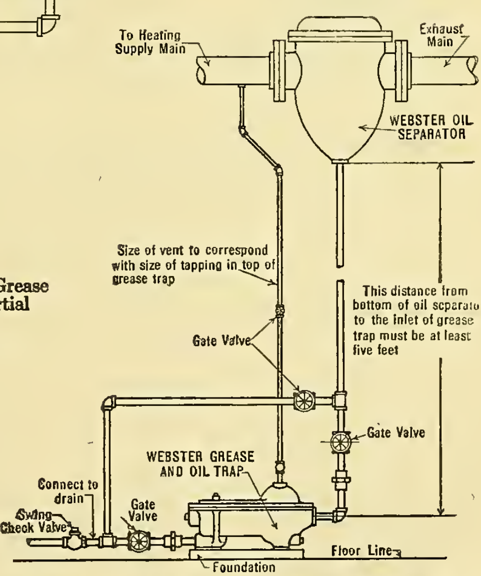


Fig. 23-34. Typical method of draining Webster Oil Separator through a Webster Grease Trap.

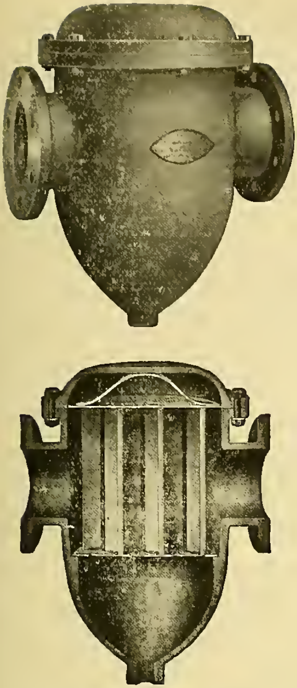


Fig. 23-35. Standard Horizontal. Sizes—2 to 6-inch.

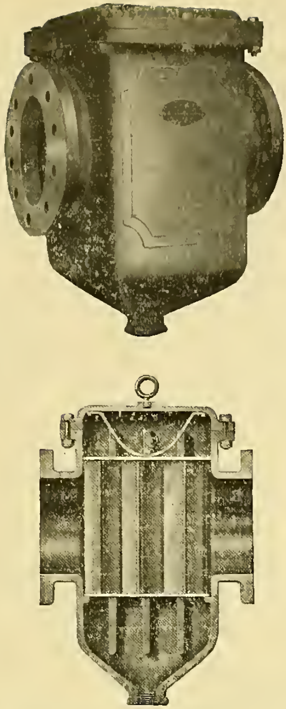


Fig. 23-36. Standard Horizontal. Sizes—8 to 16-inch.

Fig. 23-37. Vertical, for either ascending or descending currents. Sizes—3 to 16-inch.

Table 23-7. Maximum Ratings of Oil Separators in Pounds per Minute at Average Gauge Pressures Based Upon a Pipe Velocity. of 6000 Feet per Minute.

	PRESSURE LB. PER SQUARE INCH			
Size	0	5	10	15
2	5.2	6.7	8.4	10.
3	11.4	15.	18.6	22.
4	19.8	26.	32.	38.
5	31.	40.6	50.2	59.7
6	45.	59.	73.	86.5
8	78.	102.	126.	150.
10	123.	160.	200.	235.
12	176.	231.	285.	339.
14	222.	292.	361.	427.
16	294.	385.	475.	565.
18	375.	492.	608.	720.
20	452.	595.	735.	870.
22	550.	725.	900.	1060.
24	660.	870.	1070.	1270.

For lower velocities, the pounds carried will be proportional as the lower velocity is to 6000.

That Webster Separators are efficient in all their standard and special forms is indicated by absolute satisfaction in over 15,000 installations.

The material ordinarily used in the shells is close-grained cast iron, but special shell of semi-steel, cast steel or other material can be furnished at extra cost.

Table 23-8. Dimensions of Webster Oil Separators.

All dimensions in inches. Weights are approximate and do not include companion flanges, which are furnished only as extras and at extra cost.

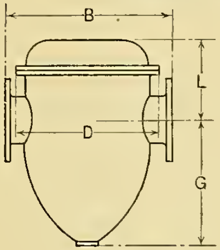


Fig. 23-38.

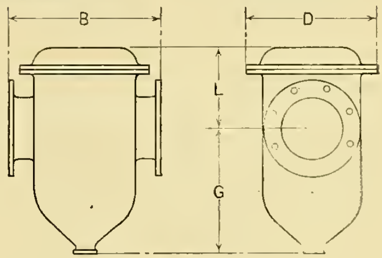


Fig. 23-39.

HORIZONTAL STANDARD TYPE

Fig.	Dimensions					Flanges			
	Size	B	D	G	L	Drip	Out. Diam.	Bolt Circle	No. & Sizes of Bolts
23-00	2	10¾	10	8½	6	¾	6	4¾	4–⅝
	3	15¼	13	11¼	6¾	¾	7½	6	4–⅝
	4	17⅞	15	13¼	8½	1	9	7½	8–⅝
	5	18⅝	15⅜	14⅝	8⅝	1	10	8½	8–¾
	6	19¾	16	14½	9⅝	1	11	9½	8–¾
23-00	8	21⅛	17⅝	18	10½	1¼	13½	11¾	8–¾
	10	22⅛	19¼	18½	12⅝	1½	16	14¼	12–⅞
	12	24¼	21⅝	19	13	2	19	17	12–⅞
	14	27½	25½	22	15¾	2	21	18¾	12–1
	16	34⅛	27¾	23½	18½	2½	23½	21¼	16–1

VERTICAL STANDARD TYPE

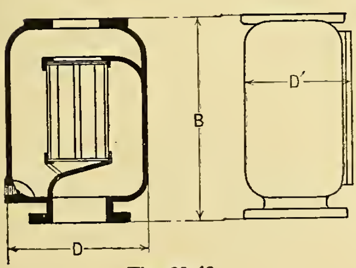


Fig. 23-40.

Dimensions				Flanges			
Size	B	D	D	Drip	Out. Diam.	Bolt Circle	No. & Size of Bolts
3	13⅜	9⅞	9½	¾	7½	6	4–⅝
4	17½	11½	10½	¾	9	7½	8–⅝
5	20¼	13½	11½	1	10	8½	8–¾
6	24	14¾	12¾	1	11	9½	8–¾
8	26¼	17¼	17½	1	13½	11¾	8–¾
10	32⅛	22	23	1	16	14¼	12–⅞
12	33½	25¼	25	1	19	17	12–⅞
14	41	28½	29¾	1½	21	18¾	12– 1
16	48¼	33¾	31½	1½	23½	21¼	16– 1

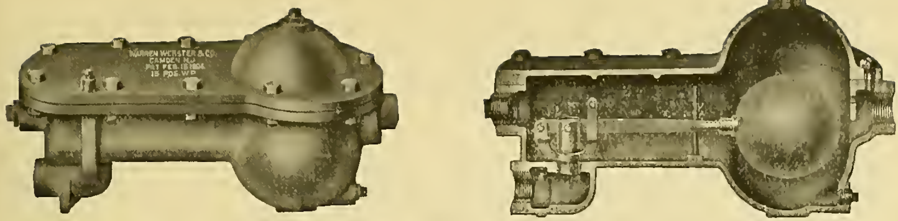


Fig. 23-41. The Webster Grease and Oil Trap.

Webster Grease and Oil Traps

The Webster Grease Trap is for use in draining oil separators on exhaust steam lines or on feed-water heaters, or for removing from the course of the steam any accumulations of oily drips at other points in the low-pressure steam mains or branches. It will operate with equal efficiency under any pressure from that of atmosphere to plus 15 pounds per square inch. It is not designed for use under high vacuum conditions.

As shown in the accompanying sectional illustration (Figure 23-41) the valve mechanism is simple, and the discharge orifice is designed to give the full area of the inlet opening.

The ball float and valve chamber are easily reached for quick cleaning without disturbing pipe connections.

Properly installed, the Webster Grease Trap should be provided with a by-pass in the piping around it; a check valve should be in the line beyond the outlet and by-pass, and an equalizing or vent pipe should be run from the top of the trap to the exhaust main beyond the Oil Separator.

Ratings for Webster Grease Traps: Because the mixture to be discharged is likely to be more or less viscous and sluggish in movement when it is cool it is impossible to rate Grease Traps on a condensation basis. The size of Grease Trap to be selected in any case is the same as the size of drip of the Oil Separator which it is to drain.

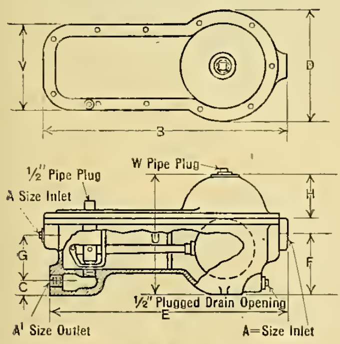


Fig. 23-42.

Table 23-9. Dimensions of Webster Grease and Oil Traps

Number	A	A^1	B	C	D	E	F	G	H	U	V	W
0016	¾	¾	13	1	7	12⅝	4	3	2⅝	7⅝	5⅛	½
016	¾	¾	15¾	1	8	15	4¼	3⅛	2⅞	8⅛	6¼	¾
116	1¼	1¼	19⅛	1⅜	9	18⅜	5⅜	3⅞	4	10¾	7	1
216	2	2	20⅝	1⅝	10½	19⅞	6⅛	4½	4⅜	12¼	8	1½

All dimensions in inches and subject to slight variation.

Where the Webster Oil Separator is installed in connection with a condensing engine, a special method of separator drainage becomes necessary.

An excellent arrangement for this purpose is shown, the apparatus consisting of a Standard Webster Horizontal Separator and a receiving tank of extra heavy wrought steel, made especially for and equipped with suitable fixtures for vacuum service.

The drain from the oil separator is so connected that the drips fall freely by gravity into the tank below it. An equalizing pipe is connected from the tank to the steam line leading to the condenser, thus equalizing the pressure between the two.

The whistle is attached to the high-pressure steam connection which is used for blowing down the tank. It is operated through a series of levers, by a counter-balanced open sink pan (performing the functions of a float) placed within the tank. As soon as the drip accumulates to a point where the tank should be emptied, the pan rises and causes the whistle to give an alarm to the engineer, who closes the valve in the drain pipe from the separator and the valve in the equalizing pipe, and opens the steaming-out valve and the tank drain. The contents will then pass from the tank into the receptacle provided for such oily drips, or to waste.

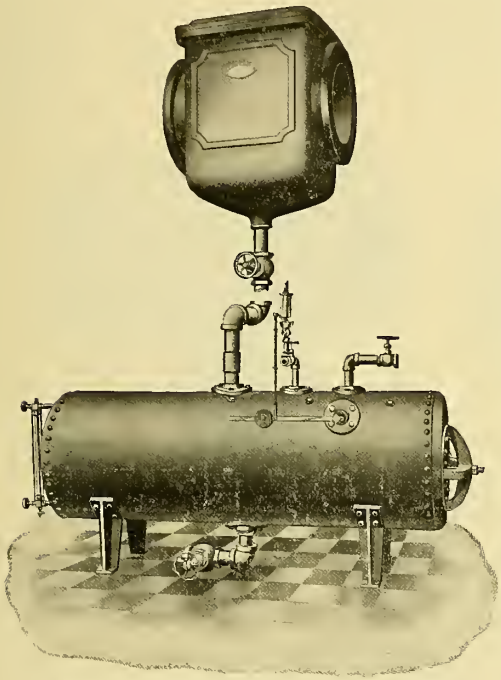


Fig. 23-13. Vacuum Oil Separator Draining System with tank and automatic whistle alarm

The whistle alarm, together with the sink pan and regulating gear operating it and the gauge glass for the tank, are provided as standard fixtures with the tank, but hand valves and connecting piping shown are not furnished.

Built in sizes from 8 to 24 inches, inclusive, and in larger sizes upon special order.

Webster Low-Pressure Receiver Oil Separators

These separators, acting as eliminators of oil and condensation and as receivers or mufflers, are used chiefly in exhaust steam lines between reciprocating engines and low or mixed-pressure turbines, or as receivers for the intermittent exhaust from groups of steam hammers.

They are of riveted steel construction, with cast-iron nozzles, and, in common with all the Webster Oil Separators, are equipped with hooked steel multibaffles.

The illustration shows one of the many forms of the Webster Low-pressure Receiver Oil Separator. The inlet and outlet nozzles may be located to conform with any direction of flow of steam. The axis of the shell may be either horizontal or vertical.

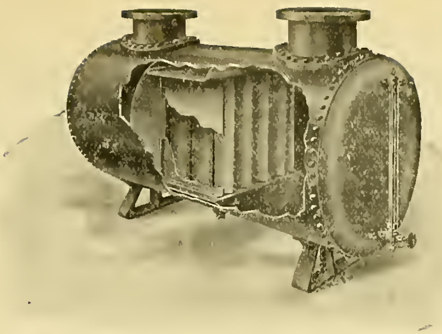


Fig. 23-44. The Webster Low-pressure Receiver Oil Separator.

Inquiries regarding the Webster Low-pressure Receiver Oil Separators should be accompanied by a sketch showing the proposed location of and space available for the separator, the sizes and locations of inlet and outlet nozzles and the direction of flow. The inquiry should state the maximum amount of steam to be purified.

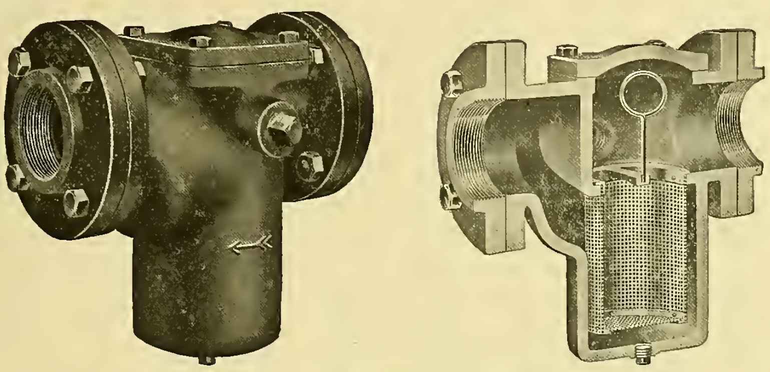


Fig. 23-45. The Webster Suction Strainer.

The Webster Suction Strainer

The Webster Suction Strainer is used to exclude from the cylinder of the vacuum pump the dirt and scale brought down with the condensation from a vacuum heating system. The use of this strainer prevents scoring of the pump-cylinder lining, valves and piston rods and the serious efficiency losses and repair bills that would follow such scoring. The strainer is provided with a tapping for the introduction of cold make-up water when same is desired and when specially ordered a spray nozzle is provided to insure thorough mixture of cold water and vapor in return. Another tapping is provided for a connection to the vacuum gauge and a third plugged outlet is for draining the body when the strainer is not in use. The shell and removable cover are of cast iron with composition gasket in the joint. Com-

panion flanges, drilled low pressure standard, are provided for inlet and outlet connections.

The basket is of perforated brass, and has at its top rim a casting in which is fastened a handle for lifting out the strainer. The perforations are .043 inch in diameter and of sufficient number to provide a total area twice that of the entering pipe.

The Webster Suction Strainer is to be placed in horizontal piping only, and should be set so that the axis of the body will be vertical. Water flows to it in the direction of the arrow (see exterior view), and its course through the strainer is evident from the sectional view on the previous page.

During the cleaning process it is customary, if the system must be maintained in operation, to use either the relay pump or the ejector, if there is one, and if not, to temporarily run the returns by gravity to the sewer or waste, closing the stop-valve in the main return. The entire operation occupies but a few minutes.

Table 23-10. Dimensions of Webster Suction Strainer

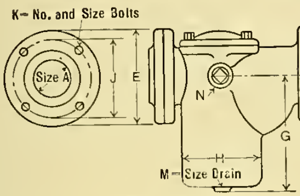
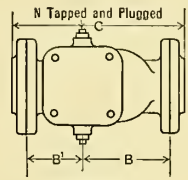


Fig. 33-46.

For maximum working pressure of 15 pounds



Top View.

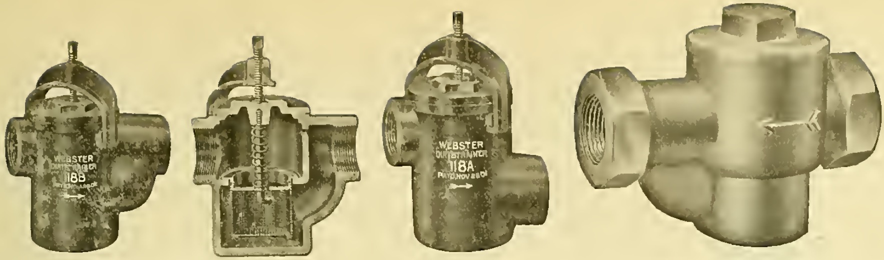
All dimensions in inches and subject to slight variations. Companion flanges furnished up to 12 inches.

Size A	B	B1	C	E	G	H	J	K	M	N
2	5⅝	4⅜	12	6	6⅞	5¼	4¾	4–⅝x2	½	¾
3	6⅝	4⅞	13¾	7½	8⅞	5¾	6	4–⅝x2¼	½	¾
4	8 3/16	5 13/16	16⅜	9	10⅝	7⅛	7½	8–⅝x2¾	½	¾
5	9⅝	6⅜	18⅝	10	12⅛	8¼	8½	8–¾x2¾	½	¾
6	10 15/16	7 1/16	20⅞	11	13 11/16	9⅝	9½	8–¾x2¾	½	¾
7	12 9/16	9 7/16	25	12½	19¼	13	10¾	8–¾x3	¾	1
8	14⅛	9⅞	27¼	13½	21	14½	11¾	8–¾x3⅛	¾	1
10	17¼	11¼	32¼	16	24⅞	16¾	14¼	12–⅞x3½	¾	1
12	21	12⅞	38	19	29	20	17	12-⅞x3½	¾	1

Webster Dirt Strainers

Webster Dirt Strainers are used in steam heating systems to prevent dirt from entering radiator traps or traps on drip points, mains or blast coils. They provide convenient receptacles for retention and accumulation of pipe chips, rust, dirt, etc., where impurities can do no harm and where they are easily and quickly removed.

The use of these strainers greatly lessens the amount of attention required to keep the system in thoroughly efficient operation and eliminates the incentive for neglect that is always to be expected with dirt pockets composed of pipe fittings, which cost nearly as much to make and are never as good.



Class B (Straightway) Class A (Offset) Class C (High pressure)

Fig. 23-47. Webster Dirt Strainers.

Three models are made: Class A with offset and Classes B and C with straightway pipe connections. All have cast-iron shell and cover, the latter made easily removable by means of a yoke and screw, in classes A and B and threads and hexogon top nut in Class C.

The basket is made from sheet brass perforated with .043-inch diameter holes. The total free area through the basket is one and a half times the area of the entering pipe. The sides of the basket are reinforced with strips which are continued upward to form a bale handle. This handle not only serves to make the basket easily removable but acts as a spring against the cover to hold the basket in place.

The range of types and sizes offers a selection for any service conditions.

Dimensions in inches and subject to slight variations.

CLASS A

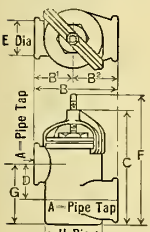


Fig. 23-48

CLASS C

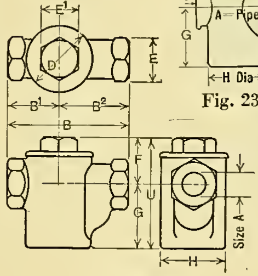


Fig. 23-49.

CLASS B

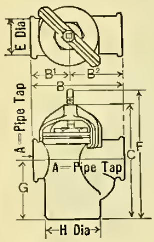


Fig. 23-50

Table 23-11. Dimensions of Webster Dirt Strainers, Classes A and B

Class A.—Offset (Fig. 23-48)

No.	Size A	B	B¹	B²	C	D	E	F	G	H
018-A	½ or ¾	3½	1¾	1¾	5 5/16	1¾	1⅝	6	2¾	2½
118-A	1 or 1¼	4¾	2¼	2½	6⅝	2	2⅛	7 5/16	3½	3¼
218-A	1½ or 2	6	2¾	3¼	8 11/16	3	3	9 3/16	4¾	4¼

Class B.—Straightway Fig. 23-50

No.	Size A	B	B¹	B²	C	E	F	G	H
018-B	½ or ¾	4¼	1¾	2½	5 5/16	1⅝	6	2¾	2½
118-B	1 or 1¼	5½	2¼	3¼	6⅝	2⅛	7 5/16	3½	3¼
218-B	1½ or 2	7¼	2¾	4½	8 11/16	3	9 3/16	4¾	4¼

Blass C.—For working pressures up to 100 lb. per sq. in. (Fig. 23-49)

Size A	B	B¹	B²	D	E	E¹	F	G	H	U
½	3 13/16	1 19/32	2 7/32	2⅛	1⅜	1⅛	1 15/32	2 1/16	2 1/16	3 17/32
¾	4 9/16	1 13/16	2¾	2⅜	1⅝	1⅛	1 11/16	2 1/16	2½	3¾
1	5 9/16	2 5/16	3¼	2⅞	2	1⅛	1 15/16	2 7/16	3 1/16	4⅜

The Webster Vacuum-pump Governor.

The vacumu pump of a vacuum heating system should be as nearly automatic in operation as possible.

The Webster Vacuum-pump Governor automatically controls the admission of steam to the pump cylinder or cylinders in proportion to the degree of vacuum required.

When only part of the heating load is "on," just enough steam is admitted into the pump to produce the degree of vacuum required. When the need is greater, the supply of steam is automatically increased.

The Webster Vacuum-pump Governor can be adjusted to control the vacuum to any predetermined degree, and may be readjusted when necessary.

It is remarkably sensitive through a wide range of adjustment.

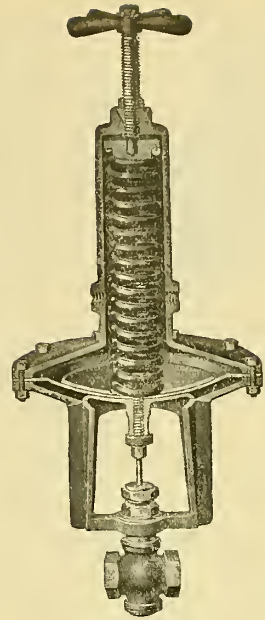


Fig. 23-51.

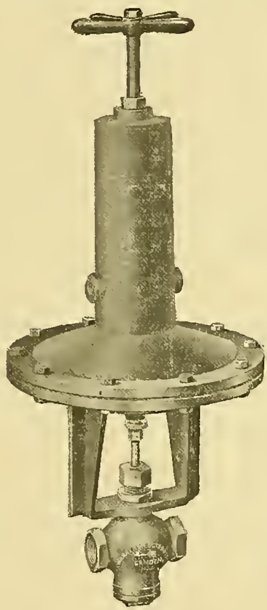


Fig. 23-52.

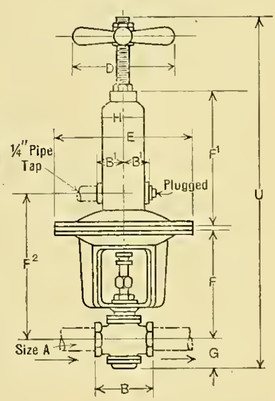


Fig. 23-53.

Table 23-12. Dimensions of Webster Vacuum-Pump Governors.

Size A	B	B¹	D	E	F	F¹	F²	G	H	U
¾	2⅞	1¾	5	9⅞	7½	10¼	10⅝	1 9/16	2⅞	23 11/16
1	3⅜	1¾	5	9⅞	7⅞	10¼	11	2	2⅞	24½
1¼	4	1¾	5	9⅞	8	10¼	11⅛	2	2⅞	24⅝
1½	4½	1¾	5	9⅞	8 9/16	10¼	11 11/16	2 5/16	2⅞	25½
2	5½	1¾	5	9⅞	8⅞	10¼	12	2 9/16	2⅞	26 1/16
2½	6⅜	1¾	5	9⅞	10¾	10¼	13⅞	3¼	2⅞	28½
3	7⅛	1¾	5	9⅞	11⅛	10¼	14¼	3⅞	2⅞	29½
3½	8	1¾	5	9⅞	11½	10¼	14⅝	4	2⅞	30

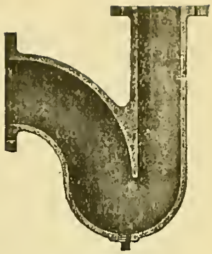


Fig. 23-34.
Webster Lift Fitting

Webster Lift Fittings are special devices used in pairs at points in Vacuum Heating Systems where condensation is to be lifted to a higher level. The condensation is lifted vertically to a higher level in "slugs" on the air-lift principle; the slugs being obtained by the use of a comparatively small diameter vertical return with its lower end submerged in the well below the level of the horizontal return which it drains. The lower Lift Fitting allows the condensation to accumulate in the well which is formed in the fittings until it seals the vertical passage, thus causing a slight reduction of the vacuum on the inlet side and forcing the water from the well through the vertical lift pipe to the higher level. The upper Lift Fitting allows the condensation to flow into the horizontal return without flowing back into the lifting line.

Lifts of six feet or over should be made in steps rather than all in one rise. The same idea applies to "drag lifts" when the condensation is to be lifted through a long, upwardly inclined return pipe.

Webster Lift Fittings are a big improvement upon and should be substituted for the home-made fittings which in the past have had to be made from combinations of ordinary tees or crosses and plugs, because nothing better was obtainable. Each Webster Fitting is a single casting, neat in appearance and correctly proportioned for capacity of well and for the area ratio of inlet to outlet. The use of these fittings eliminates all the guesswork and uncertainty about proper operation. They cost less than combinations of fittings when the labor cost as well as that of the fittings is considered.

Each fitting is provided with a clean-out plug for removing any accu-

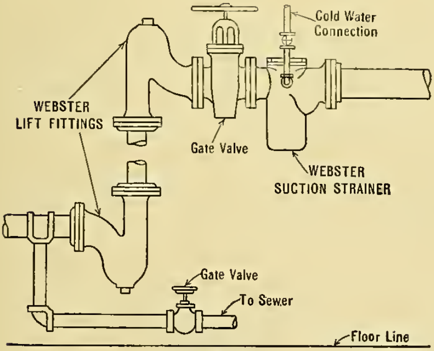


Fig. 23-35.
Typical application of Webster Lift Fittings

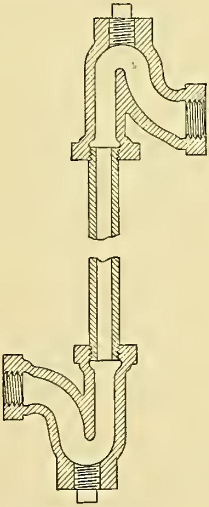


Fig. 23-56.
Long Screwed
Lift Connection

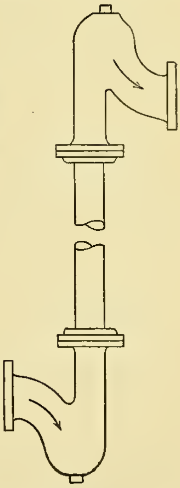


Fig. 23-57.
Long Flanged
Lift Connection

mulation of dirt or other foreig nmatter from the lift pocket. The larger sizes are flanged and finished and drilled to the low-pressure standard.

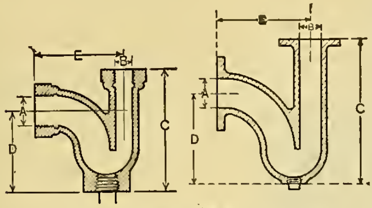
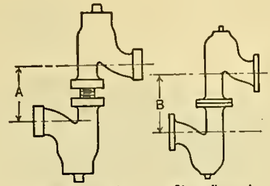


Fig. 3-58



Close screwed lift connection

Close flanged lift connection

Fig. 3-59

Table 23-13. Dimensions of Webster Lift Fittings in Inches—Series 20

Size		A	B	C	D	E	Drain
		Inlet	Outlet				
¾	Screwed	¾	½	3⅞	2⅝	2¾	½
1	"	1	¾	4⅜	3	3⅛	½
1¼	"	1¼	1	5¼	3½	3¾	¾
1½	"	1½	1	$6\frac{1}{16}$	$4\frac{1}{16}$	4¼	¾
2	"	2	1¼	6⅞	4⅝	4⅞	1
2½	"	2½	1½	8⅛	5⅜	5⅞	1
3	Flanged	3	2½	14¼	9	9⅜	1
4	"	4	3	17⅛	10⅝	11¼	1
5	"	5	3½	19¾	12½	$12\frac{15}{16}$	1
6	"	6	4	21¾	13⅞	$14\frac{5}{16}$	1
8	"	8	4½	25¼	16¼	17	1
10	"	10	6	31⅛	20⅛	20⅝	1
12	"	12	7	34½	22⅝	$23\frac{3}{16}$	1

Table 23-14. Minimum Distance Between Centers

¾-in. Screwed Fitting	A = 3⅛ in.
1-in. " "	A = 3¼ in.
1¼-in. " "	A = 4⅛ in.
1½-in. " "	A = 4¾ in.
2-in. " "	A = 5½ in.
2½-in. " "	A = 8 in.
3-in. Flanged Fitting	B = $10\frac{9}{16}$ in.
4-in. " "	B = $13\frac{1}{16}$ in.
5-in. " "	B = $14\frac{9}{16}$ in.
6-in. " "	B = $15\frac{13}{16}$ in.
8-in. " "	B = $18\frac{1}{16}$ in.
10-in. " "	B = $22\frac{1}{16}$ in.
12-in. " "	B = $23\frac{13}{16}$ in.

Webster Receiving Tanks, Plain Water-Control and Steam-Control Types

These tanks are used in connection with vacuum steam heating systems, to provide a place for storage of the condensation discharged by the vacuum pump and for liberation of the air that comes over with this condensation. Each type is designed for pressures not exceeding 15 pounds per square inch, for installation in horizontal position, and each type has proper receiving capacity and air-liberating surface.

The Plain Type receives the condensation and air through an end opening near the top. The air escapes through a vent in the top of the tank, and the water flows by gravity to the bottom outlet and to the feed-water heater or other point of disposal. If the rate of flow of returns to tank exceeds rate of discharge from tank, the excess overflows through an opening on the end near the top.

The Water-control and Steam-control types have regulating valves which are operated by sink pan and rigging similar to those used to regulate the water level in Webster Feed-water Heaters. These two types are also provided with perforated sections or baffles, to insure best operation of the sink pan.

The Water-control Type has its regulating valve arranged to automatically admit "make up" at all times when the returns from the heating system are temporarily insufficient to keep the water level in the tank at the pre-

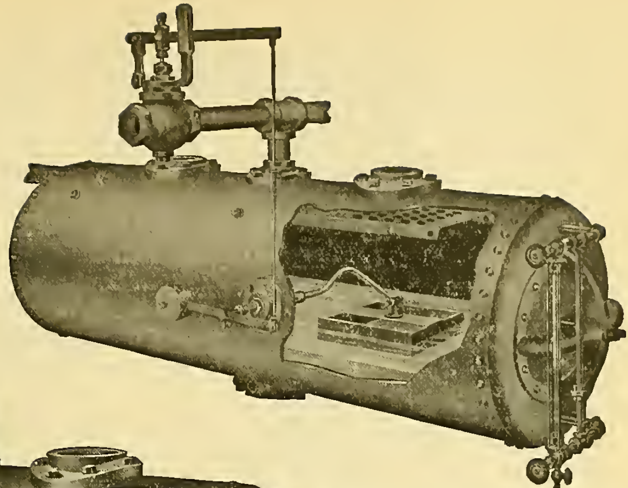
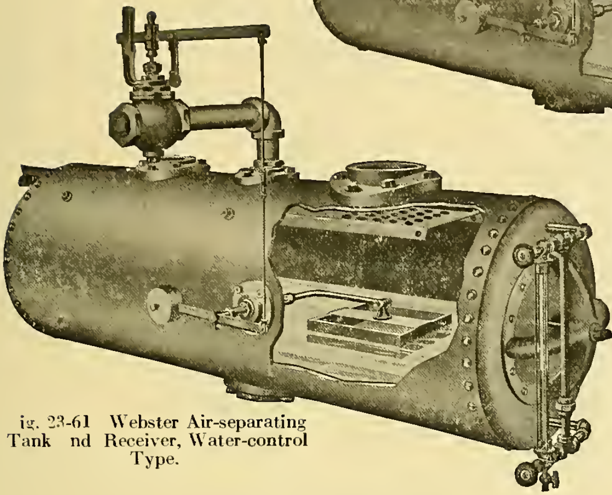


Fig. 23-69. Webster Air-Separating Tank and Receiver Steam-control Type.



ig. 23-61 Webster Air-separating Tank nd Receiver, Water-control Type.

determined point. The air is vented to atmosphere, the water flows by gravity to the heater or other place of disposal, and any excess of water overflows, as with the Plain Type.

The Steam-control Type, which is used where the boiler or boilers are to be fed in proportion to the returns reaching the receiving tank, has its regulating valve installed in the steam supply line to the boiler-feed pump. With water in the tank at or above the predetermined level, the boiler-feed pump is in operation, feeding the returns into the boiler, but when the tank level is below normal, the steam to the boiler-feed pump is shut off and the pump stopped until sufficient returns collect again. Make-up water, if necessary, may be introduced into the tank by hand. The venting of air to atmosphere, delivery of water by gravity flow and provision for overflow of excess water are the same as in the Plain Type.

All three types of Webster Receiving Tanks are made from riveted flange steel and have flat heads. The Water-control and Steam-control Types have removable manhole covers and gauge fittings in one end. Each tank is hand-made throughout from best obtainable materials. The sizes listed are standard. Larger sizes made only on special order.

Table 23-15. Dimensions of Webster Receiving Tanks

Note: Openings will be bushed to suit requirements. All dimensions in inches.

Plain Type

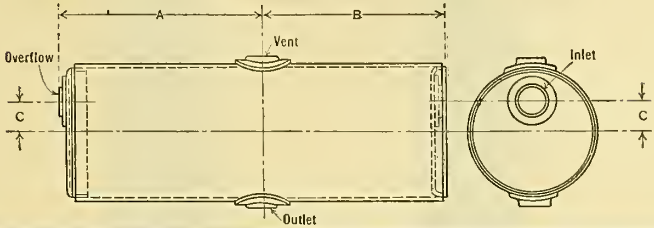


Fig. 23-62.

Size	Inlet	Outlet	Air Vent	Overflow	A	B	C
18 x 48	4	4	1½	4	25½	25	3½
24 x 72	5	5	2	5	38½	37	6
36 x 96	8	8	3	6	50¼	49	11½

Water-control Type

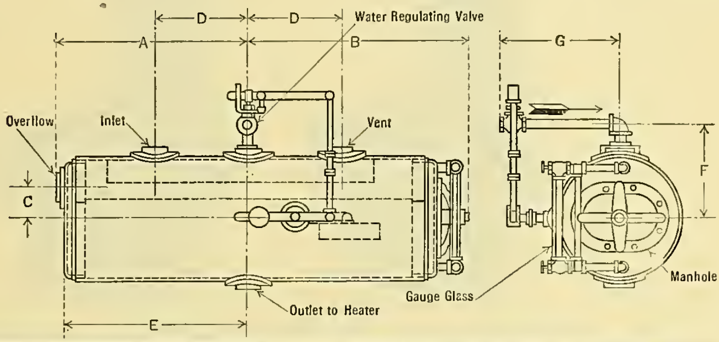


Fig. 23-63.

Size	Inlet	Outlet	Air Vent	Overflow	Reg. Valve	A	B	C	D	E	F	G
18 x 48	4	4	1½	4	1	25½	30½	3½	12	24¼	14	18¼
24 x 72	5	5	2	5	1½	37⅝	42½	6	18	36¼	18	23
36 x 96	8	8	3	6	2	50¼	54⅝	11½	18	48⅜	24	28¼

Steam-control Type

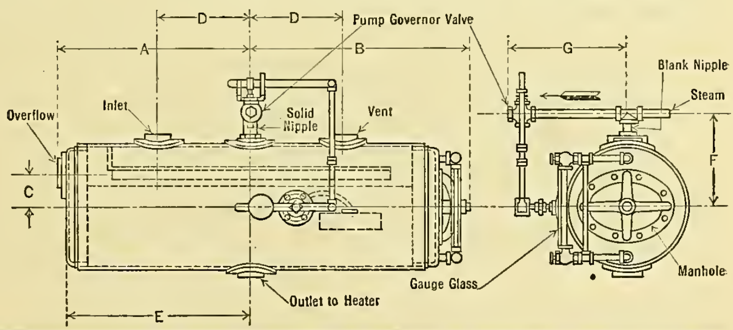


Fig. 23-64.

Size	Inlet	Outlet	Air Vent	Overflow	Gov. Valve	A	B	C	D	E	F	G
18 x 48	4	4	1½	4	1	25½	30½	3½	12	24¼	14	18¼
24 x 72	5	5	2	5	1½	37⅝	42½	6	18	36¼	18	23
36 x 96	8	8	3	6	2	50¼	54⅝	11½	18	48⅜	24	28¼

For ratings see Table 14-00, page 00.

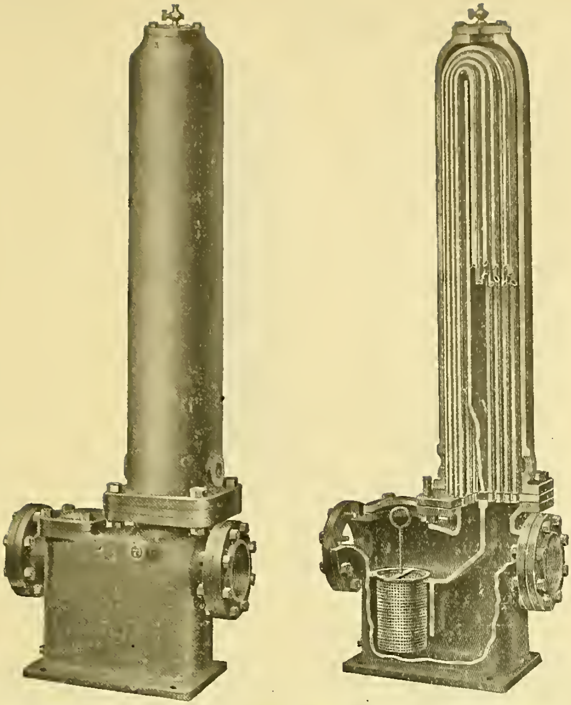


Fig. 23-65. The Webster Suction Strainer and Vapor Economizer.

The Webster Suction Strainer and Vapor Economizer

This special device, in addition to its function of protecting the vacuum pump, has a particular advantage in vacuum heating systems where some unusual operating condition results in the return of water to the vacuum pump at a high temperature.

Under such conditions, re-evaporation or transformation of water into steam vapor may occur, and the presence of this steam vapor adds to the duty of and may interfere with the proper operation of the pump.

If cold water is constantly required for making up the boiler-feed water it can be introduced in the standard Webster Suction Strainer, by the use of the Webster jet-head, without increasing the cost of plant operation. The special Webster Suction Strainer and Vapor Economizer is designed to meet conditions where no make-up water is required, and where the use of such water would entail waste.

The cold water is passed through a nest of copper coils and absorbs the heat of the steam vapor in the main return.

This water is not handled by the vacuum pump and does not mix with the condensation in the main return line, as the Economizer becomes merely an extension of the hot water piping system, under the available pressure.

Table 23-16. Dimensions of the Webster Suction Strainer and Vapor Economizer

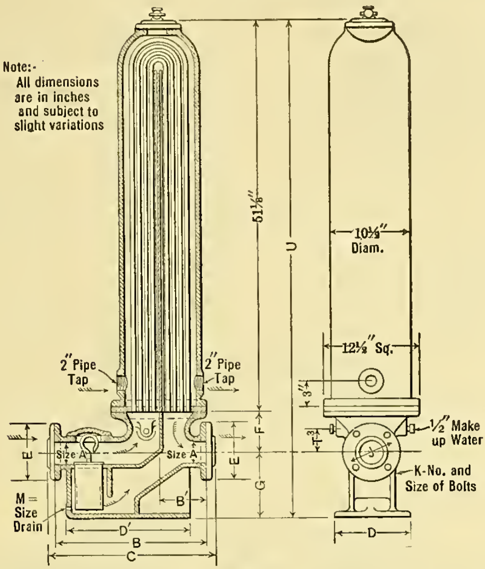


Fig. 23-66.

All dimensions in inches and subject to slight variation.

Size A	B	B¹	C	D	D¹	E	F	G	J	K	T³	U	M	Est.Wt
3	19⅝	6	22	10	15¾	7½	5½	8⅝	6	4–⅝ x 2¼	2¾	65¼	½	600 lb.
5	22¾	6¼	25½	12½	18⅞	10	6⅜	12	8½	8–¾ x 2¾	3½	69½	½	750 lb.
7	28¼	7¼	31⅜	14½	24¼	12½	7⅞	19	10¾	8–¾ x 3	4⅜	78	¾	900 lb.

Fig. 23-67. Webster Combination Gauges.

Gauges for Webster Systems

Webster Gauges are of the same high quality as all Webster apparatus and are furnished in various standard forms, and to suit special specifications.

The standard outfit furnished with Webster Vacuum Systems is a set

of two 5½-inch face, nickel-plated combination pressure and vacuum gauges, mounted on Monson, Me., slate board with Webster System name plate, thus identifying the system.

Single combination gauges are also furnished, both for Vacuum and Modulation Systems, in 5½-inch size.

Single gauges are also furnished with Webster Hylo Vacuum Sets, as elsewhere described. Larger gauges or slate or marble boards for three or four gauges can also be furnished when required.

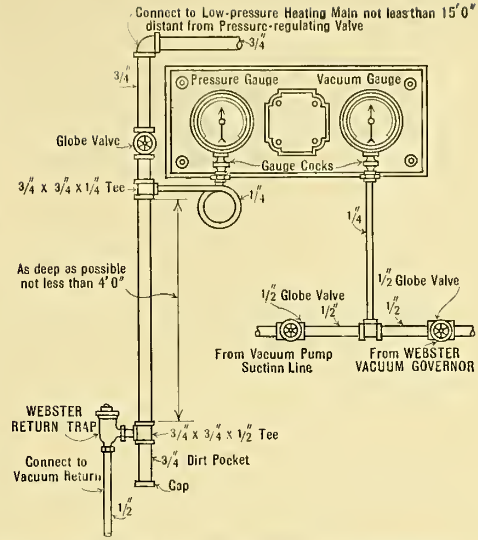


Fig. 23-68. Connections for Gauges, Webster Vacuum System.

The Webster Modulation Vent Trap

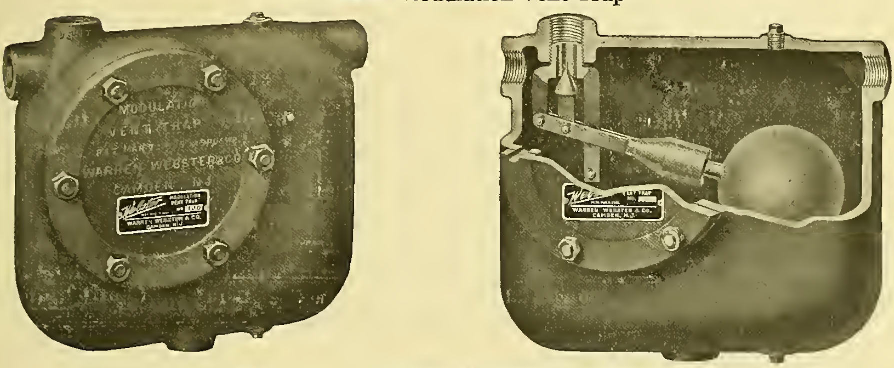


Fig. 23-69. The Webster Modulation Vent Trap.

This device is installed in the low point of the dry return line of the Webster Modulation System before the returns flow back into the boiler as feed water. It affords a simple, dependable method of venting the entrained air to atmosphere and of automatically insuring the return of the water to the boiler under fluctuating boiler pressures. The air vent is controlled by an internal float mechanism which is entirely free from mechanical troubles.

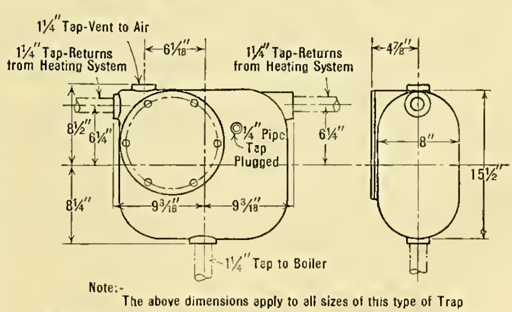


Fig. 23-70. Dimensions of Webster Modulation Vent Trap, Series 20.

Other means for returning water to the boiler are provided where structural features of the building or conditions of use are involved, but for the average building to which the Webster Modulation System is adaptable the Webster Modulation Vent Trap is used.

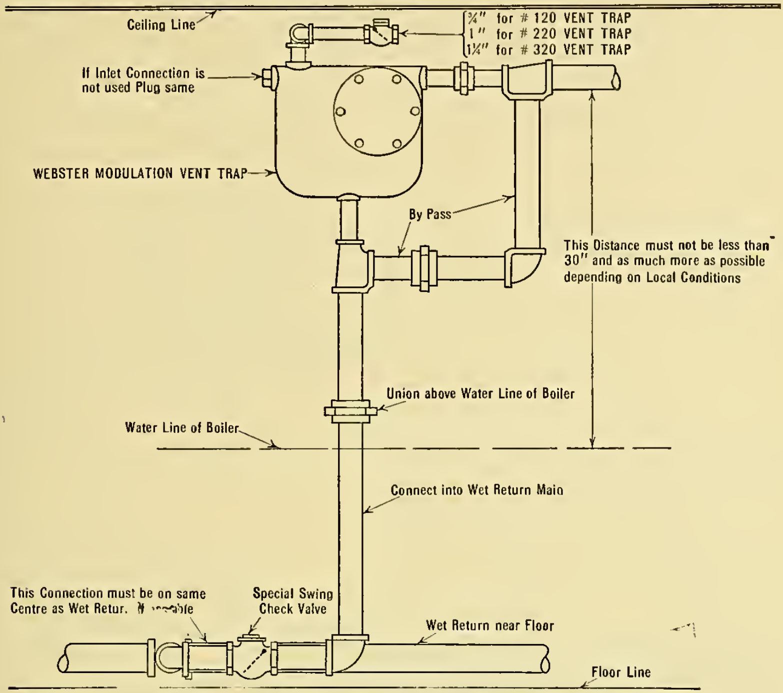


Fig. 23-71. Typical Installation of the Series 20 Webster Modulation Vent Trap.

Webster Hylo Vacuum-Control Sets

Each Webster Hylo Set consists of a Webster Hylo Vacuum Con troller, handling vapor and air only, a Webster Hylo Trap, handling water of condensation only, Webster Hylo Vacuum Gauges, and when needed, a Webster Lift Fitting.

The Webster Hylo Vacuum Controller regulates the vacuum from the low to the high vacuum through the action of the diaphragm and pilot valve. The vacuum differential, as fixed by the position of the weights on the diaphragm lever, may be anything from the high vacuum to almost nothing, as needed.

The Webster Hylo Trap permits condensation to flow from low to high vacuum without loss of differential. This trap is of ball-float type, with outlet water sealed.

The Webster Vacuum Gauges indicate the vacuum conditions upon both sides of the controller. Special arrangements of gauges and boards can be furnished where desired.

The Webster Lift Fitting operates on the "air-lift" principle, to assist in raising condensation back to level of the return pipe.

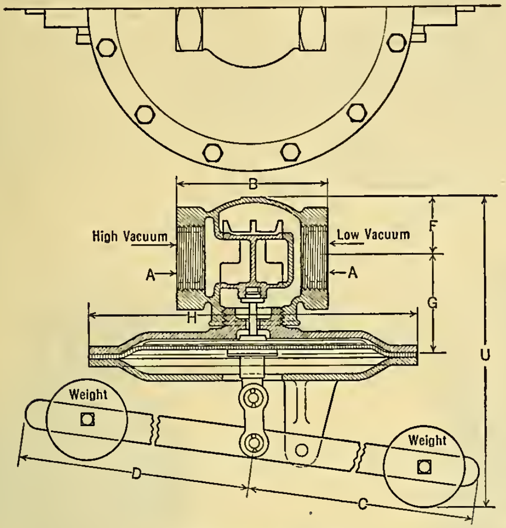


Fig. 23-72

Table 23-17. Dimensions of the Webster Hylo Controller

All dimensions in inches and subject to slight variations.

Size A	B	C	D	F	G	H	U
1	3½	12¾	11¼	1¼	2¼	10¾	9¼
1½	4¼	12¾	11¼	1¾	2⅞	10¾	10
2	5	12¾	11¼	2	3	10¾	10½

Fig. 23-73

Table 23-18. Dimensions of the Webster Hylo Trap for 15-lb. working pressure.

All dimensions in inches and subject to slight variation.

Number	A	A[1]	B	C	D	E	F	G	H	U	V	W
0016	¾	¾	13	1	7	12⅝	4	3	2½	7½	5⅛	½
016	¾	¾	15¾	1	8	15	4⅛	3⅛	2⅞	8⅛	6¼	¾
116	1¼	1¼	19⅛	1⅜	9	18⅜	5⅜	3⅞	4⅛	10¾	7	1
216	2	2	20⅝	1⅝	10½	19⅞	6⅛	4½	4⅜	12¼	8	1½

The ratings are the same as for the Webster Heavy-duty Trap, as given in Table 00-00.

The Webster Damper Regulator

The Webster Damper Regulator is used with the Webster Modulation System and automatically controls the opening of the draft door and check damper of the low-pressure, steam-heating boiler. It is extremely sensi-

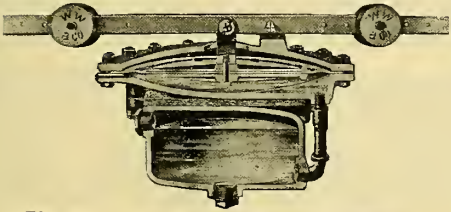


Fig. 23-74. The Webster Damper Regulator.

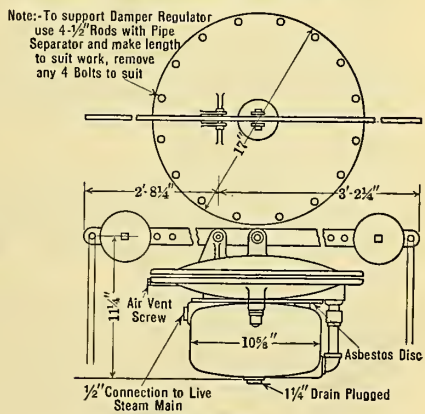


Fig. 23-75.
Dimensions of the Webster Damper Regulator.

Other means for returning water to the boiler are provided wher
structural features of the building or conditions of use are involved, but fo
the average building to which the Webster Modulation System is adaptabl
the Webster Modulation Vent Trap is used.

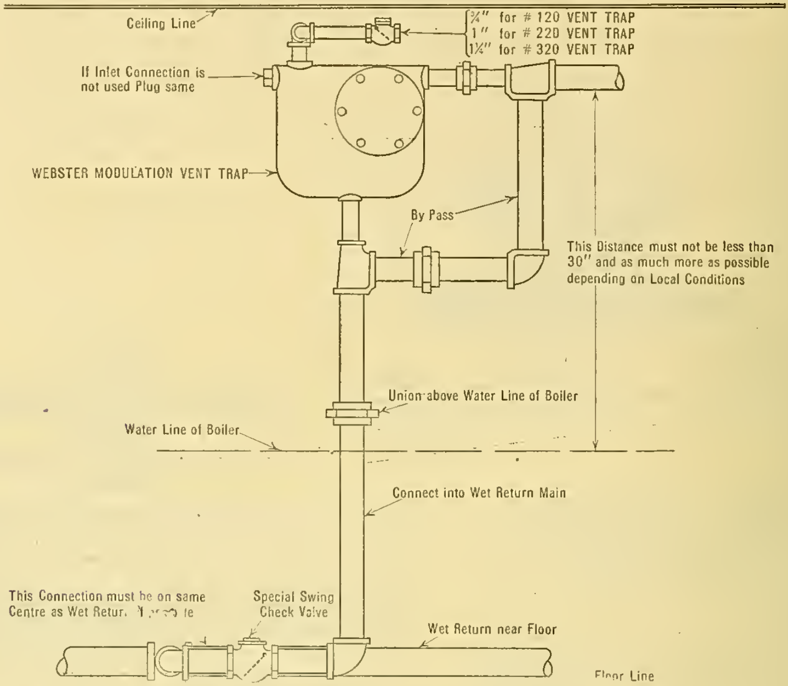


Fig. 23-71. Typical Installation of the Series 20 Webster Modulation Vent Trap.

Webster Hylo Vacuum-Control Sets

Each Webster Hylo Set consists of a Webster Hylo Vacuum Con-
troller, handling vapor and air only, a Webster Hylo Trap, handling wate
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The Webster Hylo Vacuum Controller regulates the vacuum from th
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the diaphragm lever, may be anything from the high vacuum to almos
nothing, as needed.

The Webster Hylo Trap permits condensation to flow from low t
high vacuum without loss of differential. This trap is of ball-float type, wit
outlet water sealed.

The Webster Vacuum Gauges indicate the vacuum conditions upo
both sides of the controller. Special arrangements of gauges and board
can be furnished where desired.

tive and accurate because of the ample diaphragm area and controls the fire to maintain the steam pressure always within a few ounces of that for which the regulator is set.

Table 23-19 Power Developed by Webster Damper Regulator

The following figures based upon tests with lever in mid-position afford a comparison with other damper regulators having much smaller diaphragms.

Pressure in lb. per sq. in.	½	1	2	3	4	5
Average pull at end of lever, lb.	4.125	8.25	16.5	24.75	33	41.25

Webster Conserving Valve

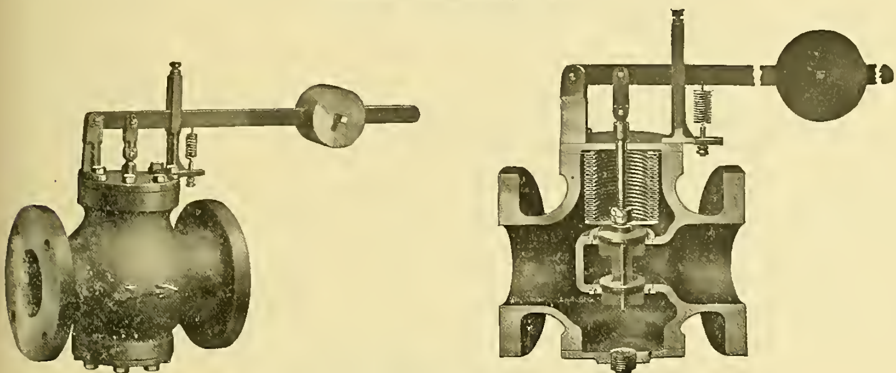


Fig. 23-76. The Webster Sylphon Conserving Valve

This valve is one of the modifications used in connection with the Webster Vacuum System when steam is furnished direct from low-pressure heating boilers, which are required to supply steam for other purposes than warming the building, at a constant pressure above that required for the heating system alone.

It also insures the constant operation of the low-pressure steam-driven vacuum pump.

It is placed in the main steam line from boiler, the steam connection to vacuum pump being taken from the inlet side of the conserving valve. The pressure for which the conserving valve is set must be built up on the inlet side, before the conserving valve will open and allow steam to enter the low-pressure heating main.

In consequence, the vacuum pump will automatically start into operation before steam is admitted into the low-pressure heating main. The partial vacuum created in the return mains and radiators assures quick circulation as soon as the conserving valve automatically opens and permits the steam to flow into the main.

Inversely, when steam is cut off the heating system the pump will continue to operate until the condensation is thoroughly drained, assuring the return of all of the condensation to the boiler. With the type of boiler used with the heating systems of this design, this is a very important matter.

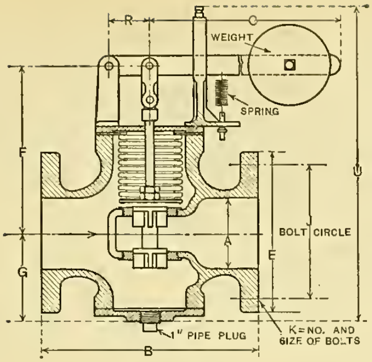


Fig. 23-77

Table 23-20. Dimensions of Webster Conserving Valves
All dimensions in inches and subject to slight variations

Size A	B	C	E	F	G	J	K	R	U
4	12	20	9	9⅜	4⅞	7½	8—⅝	2¼	$16\frac{15}{16}$
5	12	20	10	9⅝	5¼	8½	8—¾	2¼	$17\frac{13}{16}$
6	13	31¼	11	10¼	$6\frac{1}{16}$	9½	8—¾	2¾	$19\frac{5}{16}$
8	13¼	31¼	13½	11¼	$7\frac{3}{16}$	11¾	8—¾	2¾	$21\frac{15}{16}$
10	15	$36\frac{9}{16}$	16	12½	8⅜	14¼	12—⅞	$3\frac{7}{16}$	$25\frac{15}{16}$

The Webster High-Pressure Sylphon Trap

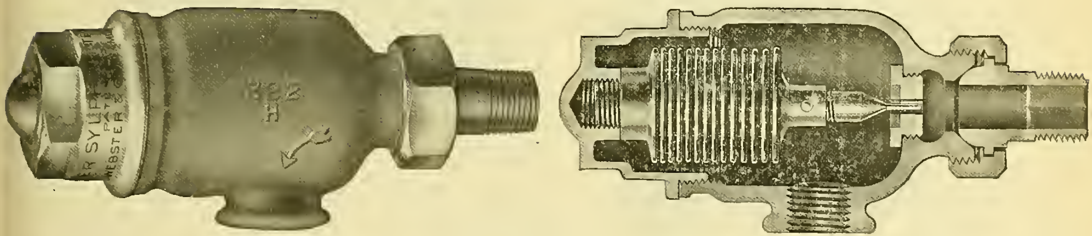


Fig. 23-78. The Webster High-Pressure Sylphon Trap

This trap is in many respects like the standard Webster Sylphon Trap described on page 00. The body construction is the same except that the position of inlet and outlet opening and the spud connection of the inlet are reversed.

As the trap must operate at comparatively high steam pressure with resulting high temperature, the thermostatic member or bellows is located outboard of the valve. The sylphon bellows, surrounded in this position with the cooler vapor from the discharged condensate, is extremely sensitive to the much higher temperature of the steam, and consequently acts quickly and positively to close the valve against steam passage through the trap.

In consequence also of the higher pressure, the valve piece and the seat are constructed of monel metal, which successfully resists wire-drawing and its accompanying wear.

The Webster High-pressure Sylphon Trap is made in three sizes and for two pressure ranges—Class 2 for pressures up to 50 pounds per square inch, and Class 3 for pressures to 100 pounds per square inch.

Application diagrams for this device are shown on page 00 of Chapter 19.

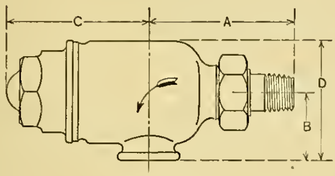


Fig. 23-79

Table 23-21 Dimensions of Webster High-pressure Sylphon Traps

SIZE	A	B	C	D
½″—822	3⅜″	1⅝″	3¼″	2¾″
¾ —833	$4\frac{1}{16}$	2⅛	2¾	3⅞
1 —844	$4\frac{5}{16}$	2½	3¾	4½

The Webster Low-pressure Boiler Feeder—Series 14

In connection with heating boilers fed from hydro-pneumatic tanks, and under certain other conditions, a Webster Boiler Feeder is necessary. This device is shown in the diagram on page 00, as part of a Webster Hydro-pneumatic System.

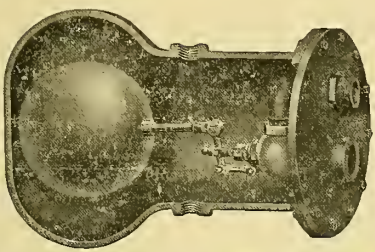


Fig. 23.80. Webster Low-pressure Boiler Feeder

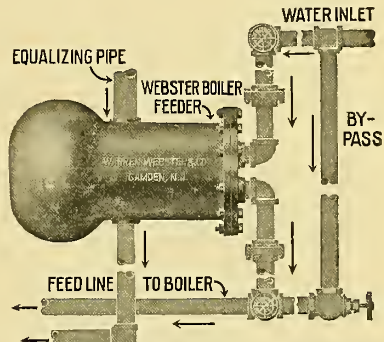


Fig. 23-81. Conventional arrangement of Webster Low-pressure Boiler Feeder

When the water level in the boiler lowers, the ball float opens the feed valve and allows the water to discharge directly to boiler.

The valve is of the double-balanced type with large orifice area, because of the low differential between the tank pressure and the boiler pressure. The ball float is large enough to give the power required to move the valve lever without excessive difference of water level.

An important point in the construction of the boiler feeder is that the valve and gear are within the casing. There are no outside glands to keep tight and any leakage which occurs is within the body of the device and hence into the boiler.

The working parts are easily accessible, but seldom need attention.

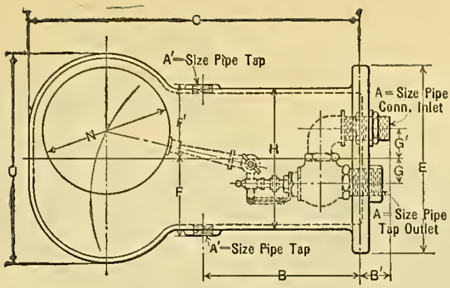


Fig. 23-82

Table 23-22. Dimensions of Series 14 Webster Low-pressure Boiler Feeder

Dimensions in inches and subject to slight variation

Number	A	A¹	B	B¹	C	E	F	G	G¹	H	N	U	Estimated Weight
114	½	1	12½	2	25⅜	14½	6¼	1¼	1⅛	11⅜	10	15⅝	217
	¾	1	12½	2	25⅜	14½	6¼	1½	1⅜	11⅜	10	15⅝	220
	1	1	12½	2	25⅜	14½	6¼	1½	1⅝	11⅜	10	15⅝	222
	1¼	1	12½	2¼	25⅜	14½	6¼	1½	2	11⅜	10	15⅝	225
214	1½	2	15	2¼	31½	16½	7¼	2¼	2⅜	13⅜	12	19⅞	305
	2	2	15	2¼	31½	16½	7¼	2¼	2⅞	13⅜	12	19⅞	310
314	2½	2½	19	3¼	36¾	18½	8	3	3	15	12	20½	450
	3	2½	19	3¼	36¾	18½	8	3	3¾	15	12	20½	460

Webster Hydro-Pneumatic Tanks

Single and Double-control Types

Webster Hydro-pneumatic Tanks are used in place of open-vent tanks for receiving returns in steam heating systems where sufficient head room to produce the necessary static head is not available for the installation of a plain receiving tank.

The general design is the same as that of Webster Steam-control and Webster Water-control Receiving Tanks, except that in the Single-control Hydro-pneumatic Tanks the sink pan and rigging control the escape of air through the vent pipe and in the Double-control type this feature is supplemented by an additional sink pan rigged to control a water valve in the tank discharge.

In both Single and Double-control types the air is permitted to escape freely until the tank is half filled with condensation, when the vent closes and the remaining air is confined. The air vent is open whenever the condensation flows by gravity against the resistance in the outlet connection. When the necessary head is greater than that due to the tank being half full of condensation, the air vent is closed. Further accumulation of air and water creates additional pressure until this added to the gravity head overcomes the resistance and condensation flows through outlet until water line reaches middle of tank. Then the air vent opens to permit escape of

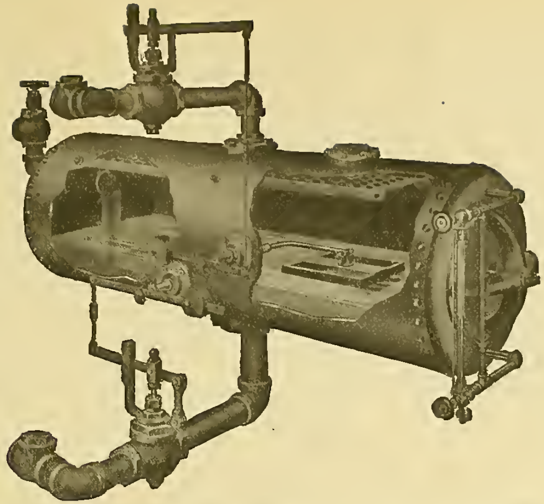


Fig. 23-83. Webster Hydro-pneumatic Tank, with Double Control.

air. When the tank has no gravity head to the heater or boiler the necessary head to overcome the resistance in the outlet is by confined pressure only.

The Double-control Hydro-pneumatic Tank in addition has its water-control valve arranged to close just before the water level reaches the bottom of the tank. The Double-control type serves to prevent the admission of air into the system through the discharge from the tank when the pressure in the open feed-water heater or boiler may be less than that of the atmosphere.

Both Single and Double-control Tanks are used under pressure greater than the atmosphere and must be provided with means for preventing excessive pressure due to obstruction of overflow. For this purpose a water-relief valve is provided, which should be piped to an open funnel to facilitate observation and correction of unnecessary waste.

Both Single and Double-control types of tanks are made of riveted flange steel plate, have flat heads and are for installation in horizontal position. A perforated inside baffle running along the top serves to distribute the water and this makes certain that the sink pans are kept filled with water.

Manholes and covers and gauge glass fittings are regular equipment with both types of tanks.

The sizes listed are standard, but others can be made to order.

Table 23-00. Dimensions of Webster Hydro-pneumatic Tanks

Openings will be bushed to suit requirements. All dimensions in inches.

Single-control Type.

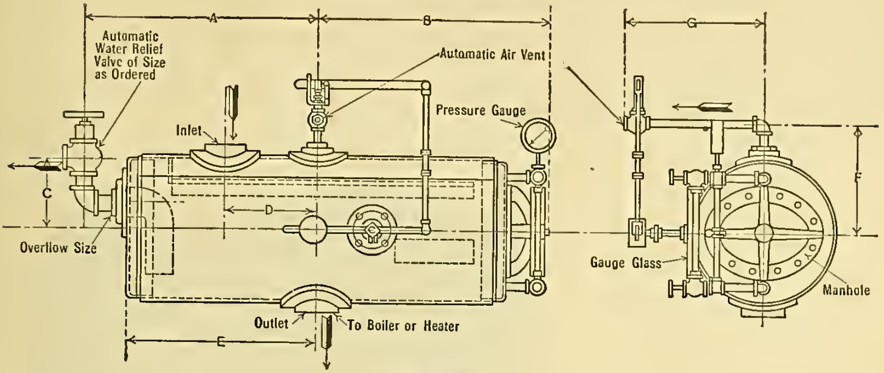


Fig. 23-84.

Size	Inlet	Outlet	Vent Valve	Overflow	A	B	C	D	E	F	G
18 x 48	4	4	¾	4	26¼	30½	3½	12	24¼	14	18
24 x 72	5	5	1¼	5	38⅜	42½	6	18	36¼	18	22¾
36 x 96	(2)8	8	1½	6	50¼	54⅝	11½	18	48⅜	24	28¼

Double-control Type.

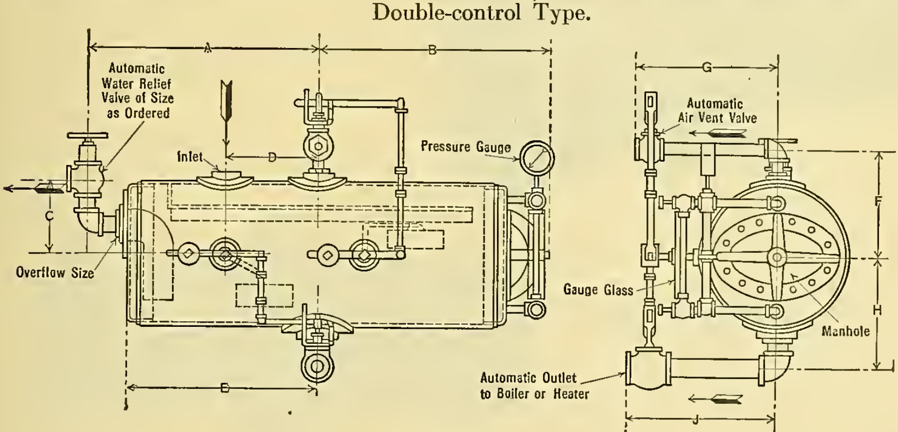


Fig. 23-85.

Size	Inlet	Outlet	Vent Valve	Overflow	A	B	C	D	E	F	G	H	J
18 x 48	4	4	¾	4	29¾	30½	10	12	24¼	14	18	20	19¼
24 x 72	5	5	1¼	5	43¾	42½	13	18	36¼	18	22¾	22	25¾
36 x 96	(2)8	8	1½	6	58	54⅝	18	18	48⅜	24	28¼	31⅜	35

For ratings, see Table 14-00, page 00.

Webster Expansion Joints are constructed with cast-iron bodies and brass-slip sleeves and in both single and double slip types. The single types may be specially equipped with ball-and-socket joints, which are of value for preserving the proper alignment of the steam piping.

The body of the Webster Expansion Joint is provided with anchors made integral with the body castings for rigid connection to a foundation or a bracket. Service connections are provided for greatest convenience in tapping the steam main for branch piping.

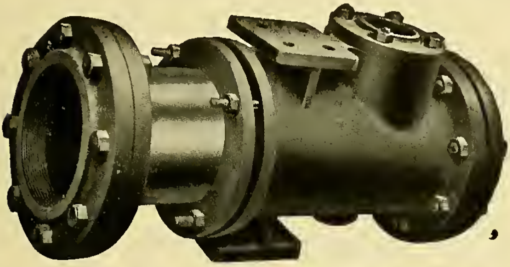


Fig. 23-86. Class D (at left).
Webster Expansion Joint.

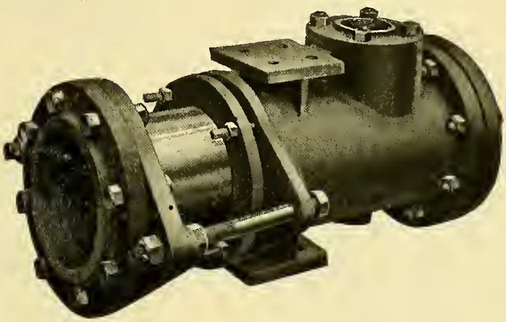


Fig. 23-87. Class DH (at right).
Webster Expansion Joint.

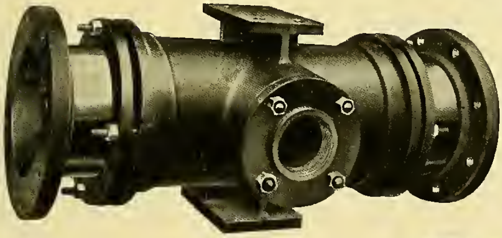


Fig. 23-88. Class G (at left).
Webster Expansion Joint.

Fig. 23-89. Class GH (at right).
Webster Expansion Joint.

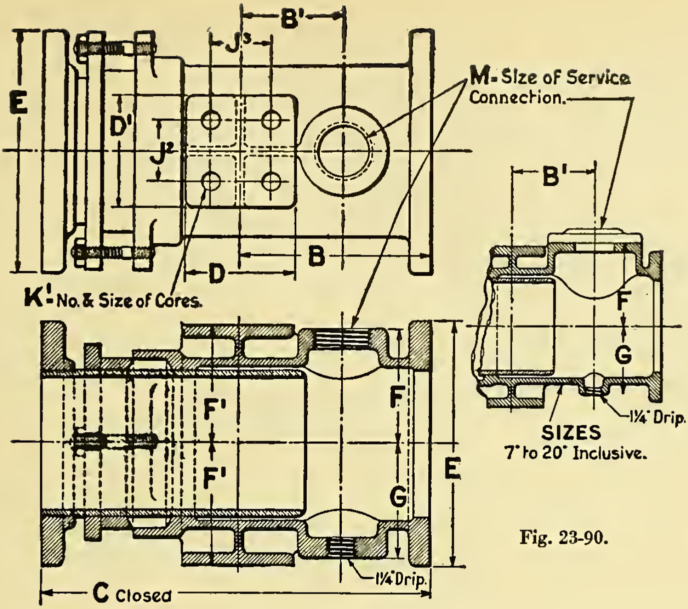


Fig. 23-90.

Table 23-24. Class D Webster Expansion Joints for Low-Pressure Steam

Maximum working pressure, 15 lb. per sq. in.

This joint has single slip and maximum traverse of 5 inches and is made with a close-grained cast-iron body and brass tubing or cast sleeve.

Standard equipment includes service connections, anchor plates and gland packing. Companion flanges are furnished only when specially ordered. Flanges are drilled low-pressure standard unless specially ordered otherwise.

Dimensions (in inches)

Size	B	B^1	C	D	D^1	E	F	F^1	G	J^2	J^3	K^1	M
2	6	3⅞	13¼	3	3	6	2½	3	2½	1¾	1¾	2– ¾	1¼
2½	6⅝	4⅛	14½	3	3	7	3¼	3½	3¼	1¾	1¾	2– ¾	2
3	6⅝	4⅛	14½	3	3	7½	2⅞	3½	2⅞	1¾	1¾	2– ¾	2
3½	7½	4½	16	4	4	8½	3½	3¾	3½	2	2	4– ⅞	2
4	7½	4½	16	5	5	9	4	4¼	4	2½	2½	4– ⅞	2½
5	8⅜	4⅝	17⅛	5	5	10	4½	5⅛	4½	2½	2½	4– ⅞	2½
6	8¾	4⅞	17¼	6	6	11	5	5⅜	5	3	3	4– ⅞	2½
7	12½	7	20⅞	6	6	12½	6⅝	6½	5¼	3	3	4– ⅞	3
8	13½	7½	21⅞	6	8	13½	7¼	7¼	6	5	3	4– ⅞	3½
10	14½	7¾	23½	6	8	16	8⅝	8½	7	5	3	4–1	4
12	15⅞	8¾	25⅞	7	8	19	9¾	9⅝	8¼	5	3½	4–1	5
14	17½	9¾	28⅛	8	8	21	10¾	10⅝	8¾	5	4	4–1⅛	6
16	17¾	9¾	28½	8	8	23½	12	11⅞	10	5	4	4–1⅛	6
18	18	9¾	28¾	8	12	25	13¼	13⅛	11	9	4	4–1⅛	6
20	18	9¾	30⅝	8	12	27½	14¼	14⅛	12	9	4	4–1¼	6

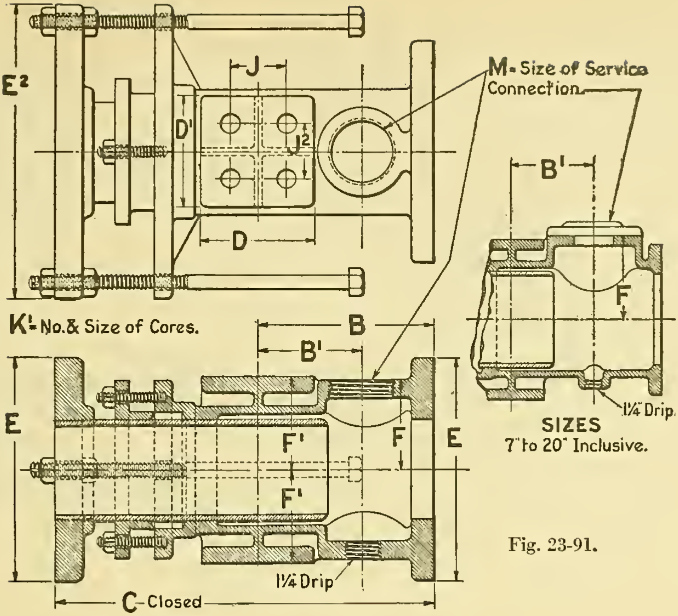


Fig. 23-91.

Table 23. Class DH Webster Expansion Joints for High-Pressure Steam

Maximum working pressure, 125 lb. per sq. in.

This joint has single slip and maximum traverse of 5 inches and is made with a close-grained cast-iron body and brass tubing or cast sleeve.

Standard equipment includes service connections, anchor plates, limit bolts and gland packing. Companion flanges are furnished only when specially ordered. Flanges are drilled low-pressure standard unless specially ordered otherwise.

Dimensions (in inches)

Size	B	B^1	C	D	D^1	E	E^2	F	F^1	J^2	J^3	K^1	M
2	6	3⅞	13¼	3	3	6	8¾	2½	3	1¾	1¾	2– ¾	1¼
2½	6⅝	4⅛	14½	3	3	7	10½	3¼	3½	1¾	1¾	2– ¾	2
3	6⅝	4⅛	14½	3	3	7½	11	2⅞	3½	1¾	1¾	2– ¾	2
3½	7½	4½	16	4	4	8½	13	3½	3¾	2	2	4– ⅞	2
4	7½	4½	16	5	5	9	12	4	4¼	2½	2½	4– ⅞	2½
5	8⅜	4⅝	17⅛	5	5	10	14¼	4½	5⅛	2½	2½	4– ⅞	2½
6	8¾	4⅞	17¼	6	6	11	15¼	5	5⅜	3	3	4– ⅞	2½
7	10½	7	20⅞	6	6	12½	17¼	6⅝	6½	3	3	4– ⅞	3
8	13½	7½	22⅛	6	8	13½	18½	7¼	7¼	5	3	4– ⅞	3½
10	14½	7¾	23½	6	8	16	21¾	8⅝	8½	5	3	4–1	4
12	15⅞	8¾	25⅞	7	8	19	24¾	9¾	9⅝	5	3½	4–1	5

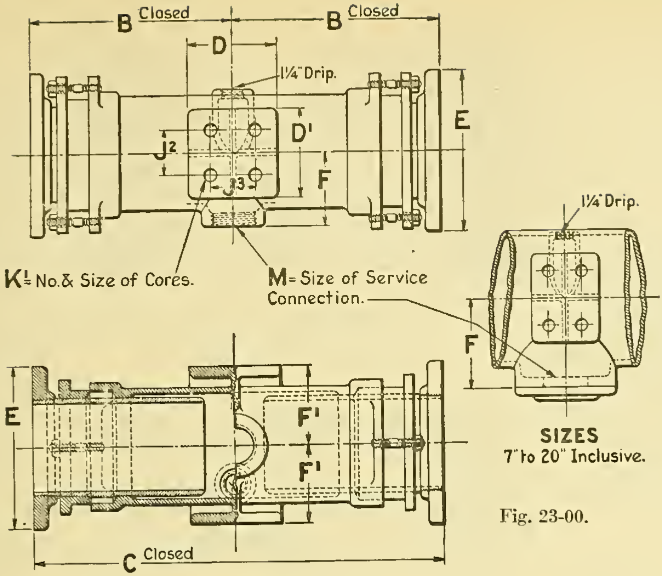


Fig. 23-00.

Table 23-26. Class G Webster Expansion Joints for Low-Pressure Steam

Maximum working pressure, 15 lb. per sq. in.

This joint has double slip and maximum traverse of 10 inches and is made with a close-grained cast-iron body and brass tubing or cast sleeve.

Standard equipment includes service connections, anchor plates and gland packing. Companion flanges are furnished only when specially ordered. Flanges are drilled low-pressure standard unless ordered otherwise.

Dimensions (in inches)

Size	B	C	D	D^1	E	F	F^1	J^2	J^3	K^1	M
2	11⅛	22⅛	3	3	6	2½	3	1¾	1¾	2– ¾	1¼
2½	11⅝	23¼	3	3	7	3¼	3½	1¾	1¾	2– ¾	2
3	12⅝	25¼	3	3	7½	2⅞	3½	1¾	1¾	2– ¾	2
3½	12⅝	25⅛	4	4	8½	3¼	3¾	2	2	4– ⅞	2
4	13⅜	26¾	5	5	9	3½	4¼	2½	2½	4– ⅞	2½
5	13¾	27½	5	5	10	4½	5⅛	2½	2½	4– ⅞	2½
6	13⅞	27¾	6	6	11	5	5⅜	3	3	4– ⅞	2½
7	14⅛	28⅛	6	6	12½	6⅝	6½	3	3	4– ⅞	3
8	15⅞	31¾	6	8	13½	6¾	7¼	5	3	4– ⅞	3½
10	17	33⅞	6	8	16	8	8½	5	3	4–1	4
12	18⅛	36¼	7	8	19	9	9⅝	5	3½	4–1	5
14	18⅞	37¾	8	8	21	10½	10⅝	5	4	4–1⅛	6
16	19½	38⅞	8	8	23½	12	11⅞	5	4	4–1⅛	6
18	20⅛	40⅛	8	12	25	13¼	13⅛	9	4	4–1⅛	6
20	22	44	8	12	27½	14	14⅛	9	4	4–1¼	6

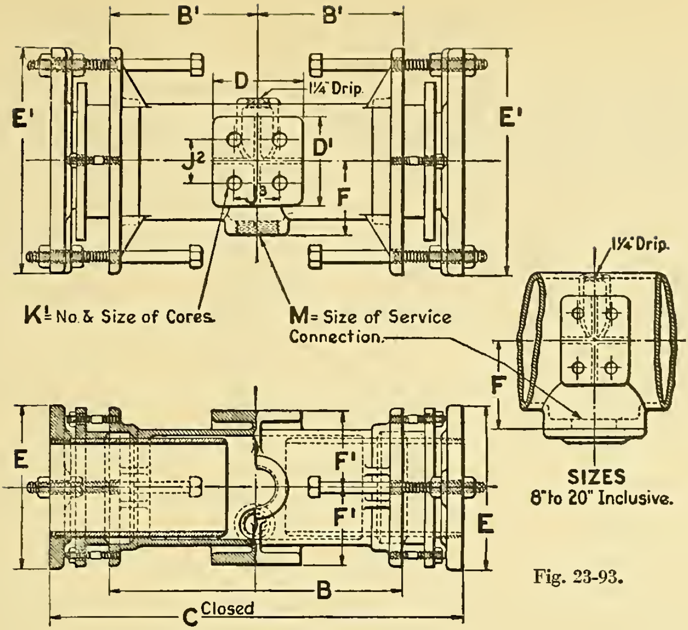


Fig. 23-93.

Table 23-27 Class GH Webster Expansion Joints for High-Pressure Steam

Maximum working pressure, 125 lb. per sq. in.

This joint has double slip and maximum traverse of 10 inches and is made with a close-grained cast-iron body and brass tubing or cast sleeve.

Standard equipment includes service connections, anchor plates, limit bolts, and gland packing. Companion flanges are furnished only when specially ordered. Flanges are drilled low-pressure standard unless specially ordered otherwise.

Dimensions (in inches)

Size	B	B	C	D	D¹	E	E¹	F	F¹	J²	J³	K¹	M
2	16⅛	8⅛	22⅛	3	3	6	8¾	2½	3	1¾	1¾	2– ¾	1¼
2½	16½	8¼	23¼	3	3	7	10½	3¼	3½	1¾	1¾	2– ¾	2
3	18½	9¼	25¼	3	3	7½	11	2⅞	3½	1¾	1¾	2– ¾	2
3½	18	9	25⅛	4	4	9	13	3¼	3¾	2	2	4– ⅞	2
4	19	9½	26¾	5	5	9	12	3½	4¼	2½	2½	4– ⅞	2½
5	19½	9¾	27½	5	5	10	14¼	4½	5⅛	2½	2½	4– ⅞	2½
6	19¾	9⅞	27¾	6	6	11	15¼	5	5⅜	3	3	4– ⅞	2½
8	23	11½	31¾	6	8	13½	18½	6¾	7¼	5	3	4– ⅞	3½
10	24¼	12⅛	33⅞	6	8	16	21¾	8	8½	5	3	4–1	4
12	25¾	12⅞	36¼	7	8	19	24¾	9	9⅝	5	3½	4–1	5

Table 23-28. Distance Between Anchor Points and Webster Expansion Joints for Various Steam Pressure Conditions

The following table is recommended as a guide in the design of steam piping for determination of the proper points of installation of Webster Expansion Joints. In such design the *maximum pressure* which the pipe line must sustain during acceptance tests or other special conditions must be selected as the "Gauge Pressure."

Gauge Pressure	Temperature Difference Above Zero	Expansion Inches Per 100 Feet	Safe Maximum Distance in Feet Between Anchors for Single-slip Expansion Joints*
0	212	1.53	260
5.3	227	1.64	245
10.3	240	1.73	225
15.3	250	1.8	220
20.3	259	1.87	215
25.3	267	1.93	210
30.3	274	1.98	202
40.3	286	2.06	195
50.3	297	2.14	190
75.3	320	2.31	175
100.3	337	2.43	166
125.3	352	2.54	160

*For Double-slip Joints, the safe distance from the joint to an anchor in each direction may be the distance specified for a Single Joint, provided the body of the Double Joint itself is securely anchored.

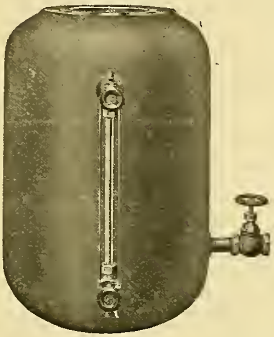
Webster Steam Separators of the standard types for removing moisture from live steam, have cast-iron corrugated baffles against which the steam impinges, causing a sudden change in direction of flow and consequently freeing the steam of the entrained moisture.

The port openings in every Webster Steam Separator are of such size as to minimize loss of steam pressure from unnecessary friction.

These separators may also be used for special purposes, as removing moisture from compressed air, assuring operation of steam whistles by removing moisture from their steam supplies. etc,

The material ordinarily used in the shells is close-grained cast iron, but special shells of semi-steel, cast steel or other material can be furnished at extra cost.

Webster Standard Steam Separators



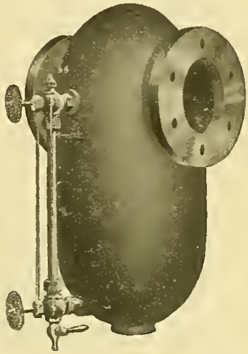


Fig. 23-94. Class B. Vertical. For maximum working pressure of 150 pounds per square inch. Sizes—2 to 12-inch.

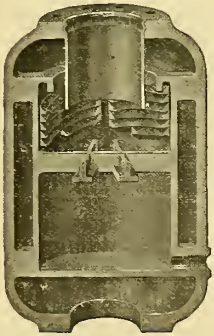


Fig. 23-95. Class BH. Vertical. For maximum working pressure of 200 pounds per square inch. Sizes—2 to 12-inch.

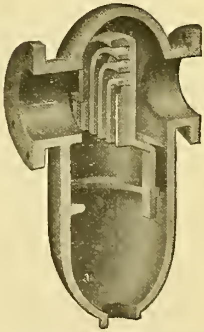


Fig. 23-96. Class C. Horizontal. For maximum working pressure of 150 pounds per square inch. Sizes—2 to 12-inch.

Table 23-29. Dimensions of Webster Standard Steam Separators

All dimensions in inches. Weights are approximate and do not include companion flanges, which are furnished only as extras and at extra cost.

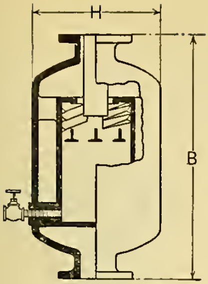


Fig. 23-97. Class B.

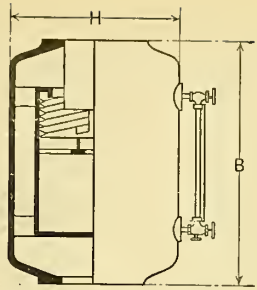


Fig. 23-98. Class BH.

CLASS B				CLASS BH				Flanges		
Size	B	H	Drip	Size	B	H	Drip	Out. Diam.	Bolt Circle	No. & Size of Bolts
2	21	10⅛	½	2	16¼	10¼	½	6½	5	4– ⅝
3	24½	12¾	¾	3	19½	13	¾	8¼	6⅝	8– ¾
4	28	15	1	4	23½	15½	1	10	7⅞	8– ¾
5	31½	17⅝	1	5	25	18⅛	1	11	9¼	8– ¾
6	35	20	1	6	28	20⅝	1	12½	10⅝	12– ¾
7	42	20⅛	1	7	33	20¾	1	14	11⅞	12– ⅞
8	43	23	1¼	8	35	23⅝	1¼	15	13	12– ⅞
10	51	27½	1¼	10	41½	27½	1¼	17¼	15¼	16–1
12	60	33⅝	1¼	12	50½	33⅝	1¼	20½	17¾	16–1⅛

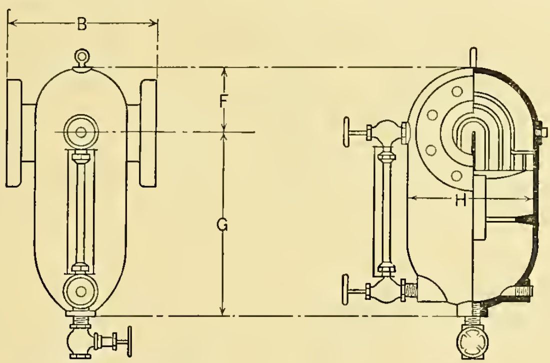


Fig. 23-99. Class C.

DIMENSIONS						FLANGES		
Size	B	F	G	H	Drip	Out. Diam.	Bolt Circle	No. & Size of Bolts
2	9½	4	12½	7½	½	6½	5	4– ⅝
3	11½	5⅜	14¼	10¼	¾	8¼	6⅝	8– ¾
4	13⅛	5⅞	16	11¼	1	10	7⅞	8– ¾
5	14⅞	7⅜	17¾	14¼	1	11	9¼	8– ¾
6	16⅝	8⅛	19½	15¾	1	12½	10⅝	12– ¾
8	20¼	10⅜	23¼	20¼	1	15	13	12– ⅞
10	24⅜	12⅜	26¾	24¼	1¼	17¼	15¼	16–1
12	27½	14¾	30	29	1½	20½	17¾	16–1⅛

Table 23-29. Dimensions of Webster Standard Steam Separators (Continued)

All dimensions in inches. Weights are approximate and do not include companion flanges, which are furnished only as extras and at extra cost.

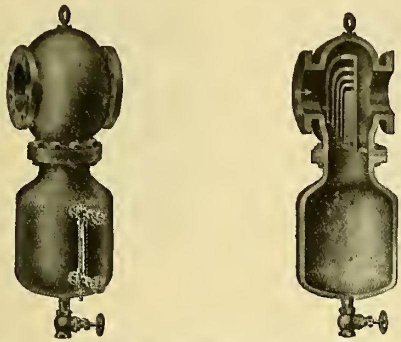


Fig. 23-100. Class F and FH Horizontal Types. Maximum working pressure, Class F, 150 pounds, and Class FH, 200 pounds per square inch. Sizes, 2 to 12-inch.

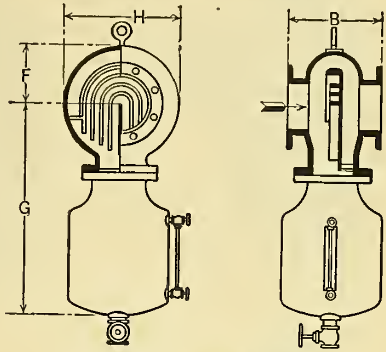


Fig. 23-101. Class F and FH

	DIMENSIONS					FLANGES		
	CLASSES F AND FH							
Size	B	H	F	G	Drip	Out. Diam.	Bolt Circle	No. & Size of Bolts
2	8¾	7½	3⅞	19⅝	½	6½	5	4– ⅝
3	9¾	10	5⅛	21⅝	¾	8¼	6⅝	8– ¾
4	10½	11⅛	5⅝	23	1	10	7⅞	8– ¾
5	11¾	14	7⅛	26	1	11	9¼	8– ¾
6	13	15½	7⅞	28¾	1	12½	10⅝	12– ¾
8	15½	20	10⅛	32	1¼	15	13	12– ⅞
10	18¾	24¼	12¼	43½	1¼	17¼	15¼	16–1
12	20½	29	14¾	46	1¼	20½	17¾	16–1⅛

Table 23-30 Advantages of Using Steam Separators

- Protection
  - *From* Water, Scale, Grit
    - *Due to* Priming, Foaming — *In* Boilers
    - Bends, Pockets, Radiation — *In* Piping
- Economy
  - *In* Operation
    - *Because of* Less Lubrication, Lower Water Rate, Less Wear and Tear
  - *In* Installation
    - *Because of* Smaller Pipe Sizes Required

For Engines, Turbines, Pumps, Air Compressors

Special Types of Webster Steam Separators

Working pressures up to 150 pounds per square inch.

Fig. 23-102.
Class L—Angle Type
With Horizontal Outlet.

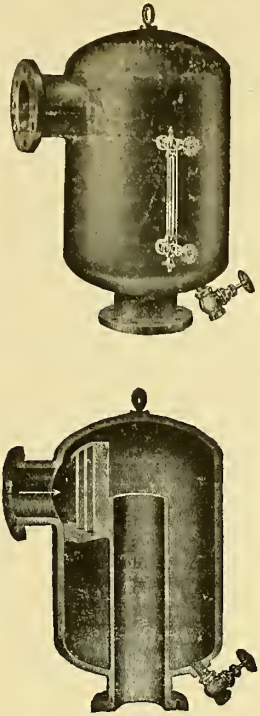


Fig. 23-103.
Class M—Angle Type.
With Bottom Outlet.

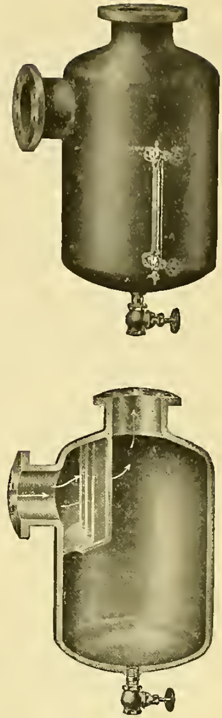


Fig. 23-104.
Class N—Angle Type.
With Top Outlet.

Table 23-31. Ratings of Webster Steam Separators

Pounds per minute at average gauge pressures. Based upon a pipe velocity of 6000 feet per minute.

Size	GAUGE PRESSURES 100 Lb. per Sq. Inch	125 Lb. per Sq. Inch	150 Lb. per Sq. Inch	200 Lb. per Sq. Inch
2	35.	43.3	51.6	66.6
3	78.3	96.7	112.	141.
4	140.	167.	196.	250.
5	215.	258.	300.	391.
6	317.	383.	450.	583.
7	433.	516.	600.	783.
8	550.	660.	800.	1000.
10	883.	1083.	1250.	1580.
12	1250.	1533.	1800.	2333.

CHAPTER XXIV

Specifications for Webster Systems

THE following specifications cover typical Webster Systems only in a general way, and are subject to many variations. It is advised that wherever practical a Webster Heating Engineer be called into consultation during the preparation of plans and specifications for Webster Systems.

Specifications for the Webster Vacuum System of Steam Heating

(This specification is drawn for a system of the usual up-feed type. For the variations known as the Webster Conserving System and the Webster Hylo System, revised special clauses will be furnished by Warren Webster & Company on application.)

General: (Here specify the general requirements of the contract, such as intent of drawings and specifications; verification of measurements; co-operation with other contractors; foreman; ordinances; permits; protection of work and buildings; rights reserved; extra work; return of specifications and drawings; payments, etc.)

Cutting of Floors and Walls: The [building] [heating] contractor will cut all holes in floors and walls and provide trenches for piping which may be necessary for this work, and at completion make all repairs to floors and walls so cut.

Scope of Work: This specification is intended to cover a 2-pipe low-pressure heating system known as the Webster Vacuum System of Steam Heating.

It is intended to supply radiation for heating the building to a temperature of .... degrees fahr. when the outside temperature is zero or a corresponding equivalent difference in temperature, with doors and windows reasonably tight.

Special Apparatus: The contractor is to secure from Warren Webster & Company the specified Webster Vacuum System Apparatus, which he is to erect and connect as part of this contract.

Standard Apparatus: In addition to the special apparatus, this contractor is to furnish all other material and labor necessary for the complete work as shown on plans or called for in specifications.

Radiation: All pipe coils must be made up of standard weight mild-steel pipe and best gray-cast iron fittings and manifolds. All radiators must be of the .... pattern equal in every respect to that manufactured by .... and must be of the heights and columns shown on plans. They must be of the [steam] [hot-water] type.

(*Note:* If of hot-water pattern, specify that the radiators "shall be connected with the supply at the top and the return at the diagonally opposite lower corner." If of steam type, specify that they "shall be provided with eccentric bushings and connected so that the bottom of the return connection will be lower than the bottom of the supply connection." Where Webster Modulation Valves are to be used, hot-water type radiators should be specified.)

Contractors supplying new radiation ordered for this work shall, if they be called upon to do so, demonstrate to the satisfaction of the owners or their authorized representative, that the radiation furnished contains in each section of the different types supplied the amount of prime heating surface mentioned in the lists published by the manufacturers of the respective types. This must be demonstrated by actual measurement and the development of the exposed surface of the sections.

The heating contractor is to instruct the manufacturer of the radiation that he requires same to be thoroughly pickled and cleaned before shipment and that the outlets are to be plugged with loose wooden plugs. The manufacturer must issue his certificate to the contractor showing that these radiators have been so cleaned. These radiators are to be kept plugged until same are connected up to the different pipe lines.

Air-valve tappings are to be omitted and the outlets plugged.

Radiators must be tapped or bushed for sizes of supplies and returns as shown on plans.

Coil Hangers: Overhead radiators are to be hung in special pipe hangers and in no case shall these coil hangers be more than 10 feet apart.

Wall coils are to be hung on cast or wrought-iron plates spaced as directed by their manufacturer, screwed to 1½-inch strap-iron brackets bent to shape, and securely fastened to the walls with 2 expansion bolts each. Brackets must be spaced not over 10-foot centers. Wall radiators must be hung as directed by manufacturers.

Straps shall be painted 2 coats of lead and oil paint of colors as directed by owners before radiators are set in place. Owners must be given opportunity to paint walls or ceilings before radiators are set.

Return Traps: The return end of every radiator, pipe coil or other form of heating unit must be provided with a Webster Return Trap (of the type selected). The size of the trap shall be governed by the amount of condensation from the radiation unit as called for on plans. The connections of Webster Return Traps must be made to the approval of Warren Webster & Company, who will provide the contractor with service details showing approved forms of connection.

Supply Valves (*Alternate for Webster Modulation Valves*): Each radiation unit must be provided with a Webster Modulation Valve connected to the top supply tapping.

(*Alternate for Ordinary Supply Valves*): Each radiation unit must be provided with a gate or angle pattern valve equal in every respect to that manufactured by............

The sizes of supply valves, the radiator tappings and the sizes of horizontal branches from risers to radiators must be as shown on the plans, or as hereafter described in this specification.

Pipe: All low-pressure pipe must be standard full-weight, mild-steel equal to that manufactured by .... All high-pressure piping must be extra heavy. All screwed piping must be fitted with occasional flanged unions.

Straighten all pipe, ream all burrs and remove all dirt before erecting pipe or fittings. Have all runs plumb and parallel with building. Provide Webster Expansion Joints of the types and sizes and at the points shown on plans. Support all pipes securely and in such manner as to permit unobstructed movement between anchorages for expansion and contraction.

So far as possible, all horizontal runs must be graded in the direction of steam flow.

Fittings: All fittings shall be best gray iron, straight and true and free from blowholes or other defects; equal to those manufactured by .... Fittings for low pressure shall be standard weight; those for high pressure shall be extra heavy.

Valves: All check, gate and globe valves must be equal to those manufactured by ....

Heat Mains: From the low-pressure side of pressure-reducing valve run a pipe to connect into the exhaust steam main where shown on plans. (Here should follow a description of the course of the steam main and its branches.)

Horizontal runs must grade not less than 1 inch in 25 feet.

Live Steam Connection: Connect a .. inch line from outlet in live steam main (where indicated on plans) to the heating main through the pressure-regulating valve. This valve shall be .. inch size and equal to that manufactured by ...., and shall be set to reduce the steam pressure from ... to (1 lb. per sq. in. or less).

Provide a 3-valve by-pass as shown, the valve in front of the reducing valve to be of the globe pattern. Run a "control pipe" as shown. Place a low-pressure gauge and a ¾-inch pop alarm valve set at 10 lb. pressure in the heat main about 10 feet from the discharge of pressure-reducing valve.

Risers: A system of supply and return risers is to be run as shown on plans. Risers are to be run [exposed] [concealed] and are to be of sizes marked on plans. All radiator branches must grade back to risers or mains with as much grade as possible, in no case less than 1 inch in 5 feet. All connections are to be made with ample provision for expansion and contraction *and particular care is to be taken that branches are run without pockets.*

Return Piping: All return risers and branches are to connect into return mains. Horizontal return piping must be graded toward the vacuum pump not less than 1 inch in 40 feet.

Dirt Traps: The bottom of all supply connections taken from the heating main must

be dripped into the vacuum return by means of a cooling leg, a gate valve, a Webster Dirt Strainer and a Webster Return Trap of size shown on plans.

Note: In large installations it is advisable to run a separate gravity drip line and connect drip of each riser or drip point of main through ¾ in. line with gate valve to this line. The discharge of this gravity drip line to be to the feed-water heater through loop seal or to the vacuum return through heavy-duty trap.

Dirt Strainers: Provide and connect Webster Dirt Strainers of the sizes specified and at the points indicated on the plans.

Lift Fittings: Where lifts occur in the vacuum return lines they are each to be provided with a pair of Webster Lift Fittings of the sizes called for on plans and connected according to special service detail furnished by the manufacturer.

Boilers: (Here specify the make, size and type of boiler or boilers required; also the equipment required for the complete boiler plant, including smoke breeching, damper regulator, gauges, feed pump, injector and any other necessary accessories.)

Vacuum Pumps: (Here specify the make, size, type and number of pumps required "to be furnished upon (concrete or other material) foundations to be provided by this contractor." Detail specifications of pumps should describe either the electric-driven type (Nash, etc.) or the steam-driven type (Blake-Knowles, Burnham, Marsh, etc.). For steam-driven pump, specify "simplex, double-acting type, brass lining, and fitted for hot-water service" and that "each pump shall be provided with a Detroit double-connected lubricator.)

Each pump shall have ample capacity for handling the products of condensation from the entire heating system.

In the suction of the vacuum pump, which must be connected to the returns from the heating system, a Webster Suction Strainer must be installed. This connection must be provided with by-pass to sewer or drain.

The discharge from steam-driven vacuum pump must be connected to the proper tapping in the receiving tank.

All connections must be properly valved and made complete.

Suction Strainer: In the suction pipe to the vacuum pump, place a Webster Suction Strainer. This strainer must be connected to accord with special service detail furnished by the manufacturer.

Vacuum Governor: In the steam connection to vacuum pump below the lubricator there must be placed a . . . inch Webster Vacuum-pump Governor with 3-valve by-pass. Same must be connected by means of ½-inch vacuum line to the suction strainer and also to the vacuum gauge on board. Each branch must be provided with a globe valve.

Gauges: Furnish and erect at convenient position two 5½-inch compound gauges mounted on a slate board. Connect one gauge to equalizing line between heat main and reducing valve, one gauge to a line connecting vacuum governor with vacuum return at suction strainer. All gauge piping to be ½-inch and all branches valved.

Air-separating Tank: Furnish a Webster Air-separating Tank . . . in. in diameter by . . . in. long.

Erect the separating tank as high above the heater as possible, as shown on plans, and to it make connections from discharge of vacuum pumps and to feed-water heater through long loop seal.

From top outlet on tank make a 1¼-in. vent connection to atmosphere.

Feed-water Heater: Furnish and erect on foundation one Webster Feed-water Heater of sufficient capacity for heating the required feed water to within 5 deg. of the temperature of the steam entering same.

The drip from oil separator is to connect to waste line through a Webster Grease Trap with 3-valve by-pass.

The contractor is to make all necessary steam, water and drain connections as shown or called for.

Steam Separator: Furnish and connect Webster Steam Separator of approved type to steam lines as shown or called for.

The drip from bottom of each separator is to be connected into a high-pressure trap of approved make. Each trap is to be provided with a 3-valve by-pass. The discharge lines from these traps are to be connected into the feed-water heater.

Covering: After all piping and apparatus has been tested and made tight to the

approval of the architect or his representative, the following covering is to be applied. (Here specify necessary covering for boilers, heater, separator, and all (specify which) piping, valves and fittings.)

Painting and Bronzing: All radiators, coils and exposed piping throughout the building, after being tested, are to be painted or bronzed 2 coats as follows:

All radiators, coil and exposed piping are to be painted 1 coat of sizing and then bronzed or painted; color as selected by architect or owner.

All exposed parts of boiler and heater to be painted 2 coats of black asphaltum paint.

Tests: The system shall be tested under 15 lb. steam pressure.

Washout caps will be furnished for Webster Traps. The interiors of traps are to be removed, when testing and washing out system, running all condensation to waste.

Inspection: This job is to be inspected by a representative of Warren Webster & Company before acceptance.

Guarantee: The contractor must agree to make good at his own expense any defects in labor or material furnished by him for this work which may develop within 1 year from the completion of this contract.

The entire system when completed is to be tested in the presence of the architect or his representative, and made tight without caulking. The contractor will be held liable for any damage to the building or its contents due to leaks or other defects in his work which may develop during the period of installation and test.

Specifications for the Webster Moderation System of Steam Heating

(This specification is drawn for a large residence. It is, of course, subject to modifications and variations for other kinds of buildings, for other sources of steam than house boiler, etc., for which revised typical specification clauses will be furnished by Warren Webster & Company on request.)

General: (Here specify the general requirements of the contract such as intent of drawings and specifications; verification of measurements; co-operation with other contractors; foreman; ordinances; permits; protection of work and buildings; rights reserved; extra work; return of specifications and drawings; payments, etc.)

Cutting of Floors and Walls: The [building] [heating] contractor will cut all holes in floors and walls and provide trenches for piping which may be necessary for this work, and at completion make all repairs to floors and walls so cut.

Scope of Work: This specification is intended to cover a 2-pipe open-return heating system known as the Webster Modulation System of Steam Heating.

It is intended that sufficient radiation shall be supplied for heating the building to a temperature of ... degrees fahr. when the outside temperature is ... degrees fahr. or a corresponding equivalent difference in temperature, based upon all doors and windows being fitted reasonably tight to prevent excessive infiltration of cold air.

Special Apparatus: The contractor is to secure from Warren Webster & Company the specified Webster Modulation System Apparatus which he is to erect and connect as part of this contract.

Standard Apparatus: In addition to the special apparatus, this contractor is to furnish all other material and labor necessary for the complete work as shown on plans or called for in specifications.

Boilers: (Here specify the make, size and type of boiler or boilers required, specifying also the equipment required for the complete boiler plants, including smoke breeching and other necessary accessories.)

Note: Boilers and auxiliary equipment must be installed in accordance with Warren Webster & Company's standard.

Damper Regulator: Furnish one Webster Damper Regulator for each boiler; to be connected in accordance with the manufacturer's standard details.

Gauges: A special compound gauge for Webster Modulation System is to be installed for each boiler. This gauge will be furnished by the manufacturers of the system.

Radiators: All radiators throughout the building shall be of .... or equal approved make; all radiators to be of the hot-water type with supply tapping at top and

return tapping eccentric at diagonally opposite lower corner. Radiators to be of the height and columns and to contain the surface indicated on plans. In no case is radiation to project above window sill. In connecting all radiators, the inlet end shall be placed next to feed risers, if possible.

The indirect stacks are to be .... (make and type) cast-iron radiation, to be of the size and contain the number of sections as called for on plans.

The heating contractor is to instruct the manufacturer of the radiation that same is to be thoroughly pickled and cleaned before shipment and that the outlets are to be plugged with loose wooden plugs. The manufacturer must issue his certificate to the contractor showing that these radiators have been so cleaned. These radiators are to be kept plugged until they are installed and connected.

Air valve tappings are to be omitted and the outlets plugged.

Radiators must be tapped or bushed for sizes of supplies and returns as shown on plans.

Hangers: Hangers for indirect stacks are to be strong wrought-iron or pipe supports.

Enclosures for Radiators: The enclosures and grilles for enclosed radiators will be furnished by ....

Return Traps: The return end of every radiator, pipe-coil or other form of heating unit must be provided with a Webster Return Trap (of the type selected). The size of the trap for each radiation unit shall be as shown on plan or called for in specification. The connections of Webster Return Traps must be made to the approval of Warren Webster & Company, who will provide the contractor with service details showing approved forms of connection.

Supply Valves: Each radiation unit must be provided with a Webster Modulation Valve connected to the top supply tapping.

The sizes of supply valves, the radiator tappings and the sizes of horizontal branches from risers to radiators must be as shown on plans.

Each overhead radiator must be provided with a Webster Modulation Valve with chain attachment.

Provide a Webster Modulation Extended-stem Valve for each radiator behind a grille.

Pipe: All pipe must be standard full-weight, mild-steel equal to that manufactured by .... All screwed piping must be fitted with occasional flanged unions.

Straighten all pipe, ream all burrs and remove all dirt before erecting pipe or fittings. Have all runs plumb and parallel with building. Allowance for expansion and contraction must be provided. Support all pipes securely and in such manner as to permit unobstructed movement between anchorages for expansion and contraction.

So far as possible, all horizontal runs must be graded in the direction of steam flow; where this is not possible, the pipe lines shall be materially increased in size as shown on plans.

Fittings: All fittings shall be best gray-iron, straight and true and free from holes or other defects; equal to those manufactured by .... Fittings shall be standard-weight.

Valves: All gate valves must be equal to those manufactured by .... All check valves must be Nelson No. 85, of balanced type with vertical seat.

Fresh-air Inlets: Fresh-air inlets for indirect heating are to be taken from openings provided in walls. Another contractor will provide heavy copper wire screens having ½-in. mesh, and louvers over the mouth of each inlet.

Sheet Metal Work: The ducts supplying fresh air to the indirect stacks, the indirect stack casings and the hot-air flue from indirect stacks to registers are to be made of galvanized iron. They are to be properly braced and locked tight to prevent air leakage. An adjustable lock quadrant hand damper is to be provided in cold-air connection to each indirect stack.

The metal used for all ducts and flues is to conform to the following gauges:

Ducts that have one dimension over 48 in., ... gauge.

Ducts that have one dimension from 30 to 48 in., ... gauge.

Ducts that have one dimension from 12 to 30 in., ... gauge.

Ducts that have one dimension smaller than 12 in., ... gauge.

The indirect stack casings are to be made of ... gauge iron and are to be built neatly

around stacks and provided with cleanout doors in bottom or side.

REGISTERS: The registers for the outlets of hot-air flues from indirect stacks will be furnished by ....; their installation is included within this contract.

STEAM PIPING: From the steam outlets on boiler rise and connect to a steam header over boiler. From top of header take branches as shown. The steam lines are to be run close to ceiling of cellar with a grade of 1 in. in 25 ft. The branches for risers are to be taken from top of mains. Steam header and main are to be dripped to wet drip line where shown.

RISERS: A system of supply and return risers is to be run as shown. Risers are to be run [exposed] [concealed], and are to be of sizes marked on plans. Unless otherwise noted on plans, branches to radiators above first floor are to be run concealed in floor construction and branches to first floor radiators are to be run overhead in cellar as close to ceiling as possible. All radiator branches are to grade back to risers or mains with as much grade as possible, in no case less than 1 in. in 5 ft. All connections are to be made with ample provision for expansion and contraction and *particular care is to be taken that branches are run without pockets.*

RETURN PIPING: All return risers and returns from first floor radiators are to connect into overhead return mains. The return mains are to start as high as possible and grade toward the Webster Modulation Vent Trap 1 in. in 25 ft. *The vent trap (or traps) to be located where shown and at least 30 inches above the water line and as much higher as possible.* The vent trap is to be vented through check valve as marked on plans. Make a ... in. city water supply connection to boiler with cock, also a ... in. drain to waste through gate valve from the return header of boiler as directed. Check valves are to be installed where shown.

A wet drip line is to be run on wall near floor as shown, and connected to boiler. To this line connect drips of mains, indirect radiators and lines from vent trap as shown.

COVERING: After all piping and apparatus has been tested and made tight to the approval of the architect, the following covering is to be applied. (Here specify necessary covering for boilers, and all steam, return and drip piping, valves and fittings.)

PAINTING AND BRONZING: All radiators, coi's and exposed piping throughout the building, after being tested, are to be painted or bronzed 2 coats as follows: All radiators, coils and exposed piping throughout the building are to be painted 1 coat of sizing and then bronzed or painted; color as selected by architect or owner.

All exposed parts of boiler to be painted 2 coats of black asphaltum paint.

Radiators or ducts which are visible through grilles or registers are to be painted 2 coats of dull black.

TESTS: The system shall be tested under 10-lb. steam pressure. The entire system shall be thoroughly washed out before final test, wasting condensation to sewer or other point of disposal.

INSPECTION: This work is to be inspected by a representative of Warren Webster & Company before acceptance.

GUARANTEE: The contractor must agree to make good at his own expense any defects in labor or material furnished by him for this work which may develop within 1 year from the completion of this contract.

The entire system when completed is to be tested in the presence of the architect or his representative, and made tight without caulking. The contractor will be held liable for any damage to the building or its contents due to leaks or other defects in his work which may develop during the period of installation and test.

CHAPTER XXV

Webster Sylphon Trap Attachments

I. For "Sylphonizing" Webster Traps of Earlier Types

STEAM heating, like almost every other science, has developed progressively through experience.

Being pioneers in this field Warren Webster & Co. have had ample incentive and opportunity for experimental research and development, and have constantly improved their product and methods, discarding and abandoning earlier types of apparatus as improved forms were adopted.

The Webster Sylphon Trap (shown and described on page 00) is now generally recognized by leading architects and engineers to be the most satisfactory type of device for return line systems.

It is in its tenth year of success and the total number in use is rapidly approaching the million mark.

Owners of buildings and plants in which old-style Webster Valves are in use will be vitally interested in knowing that such valves can be readily converted into Webster Sylphon Traps by means of the Webster Sylphon Attachments described in this chapter. The conversion necessary to bring the heating system thoroughly up to date can be made at a very moderate cost. No breaking or touching of pipe connections is involved, as the old valve bodies are utilized.

The advantages to be derived from the "change over" will be evident from the description of the Webster Sylphon Trap, on page 00, which description will equally fit the earlier Webster Valves after they are converted by means of Webster Sylphon Attachments.

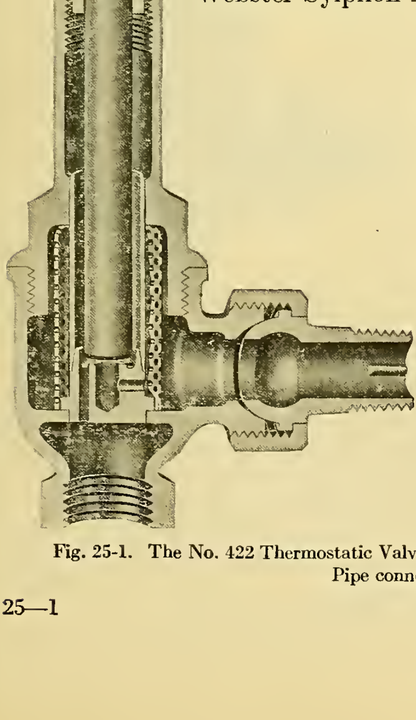
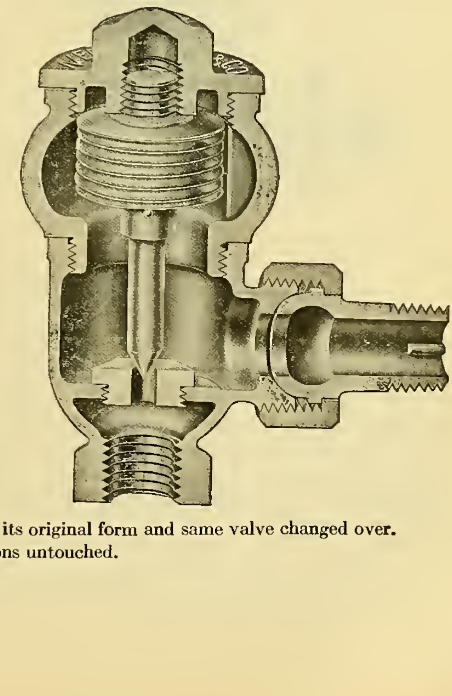




Fig. 25-1. The No. 422 Thermostatic Valve in its original form and same valve changed over. Pipe connections untouched.

CONVERSION OF NO. 422 WEBSTER THERMOSTATIC VALVES: The method of changing over by means of the S-A-13 Webster Sylphon Attachment is indicated by the illustrations.

It is only necessary to remove the old bonnets and interior parts, tapping the body for the insertion of a new solid brass seat by means of a tapping tool. The Webster Sylphon Trap Attachment may then be inserted and the old valve has become a new Webster Sylphon Trap equal in performance to the standard Webster Sylphon Traps which are furnished to thousands of new customers each year.

For conversion of Multiple-unit Thermostatic Valves, see page 00. The entire change may be made in less than five minutes.

CONVERSION OF WEBSTER MOTOR VALVES: This is accomplished practically the same as with the No. 422 Webster Thermostatic Valve (which see for description), except that a slightly different Sylphon Attachment is used.

The illustrations show the No. 522 M Sylphon Attachments for ½-inch motor-valves of the disc-port type. The No. 533 M Attachment for ¾-inch motor-valves is of exactly the same construction. These same Sylphon Attachments may be applied to the '03 motor-valves of the pin-port type where this special type of valve is to be changed over.

For conversion of Multiple-unit Motor Valves, see page 00.

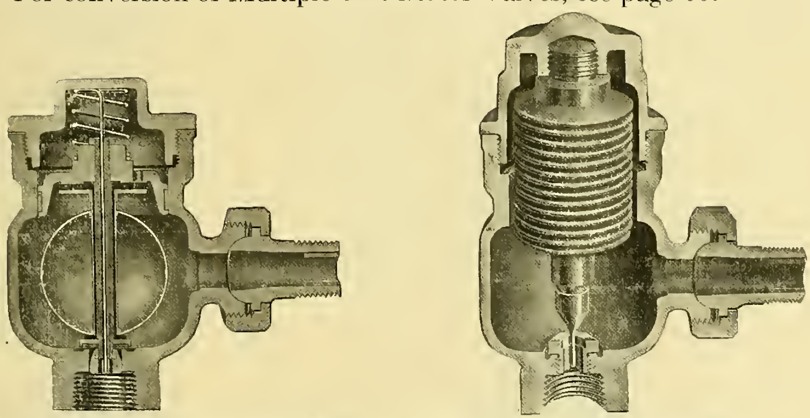
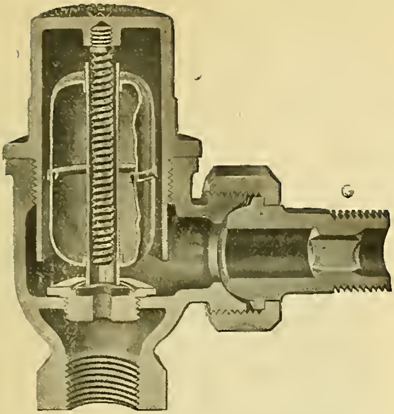


Fig. 25-2. ½-Inch Webster Motor-Valve, Disc-Port Type, in its original form and same valve changed over. Pipe connections untouched.

CONVERSION OF NO. 422 WEBSTER WATER-SEAL MOTORS: The method of changing over, as illustrated, involves the use of the same attachment as for changing over the Webster Thermostatic Valve as just described. In the case of the Water-seal Motor, however, the operation is simplified through the old body being already tapped for the valve seat.

It is only necessary to remove the old bonnets and interior parts, and insert the new solid brass seat. The Webster Sylphon Trap Attachment may then be inserted and the old valve becomes a new Webster Sylphon Trap.

For conversion of Multiple-unit Motor Valves, see page 00.



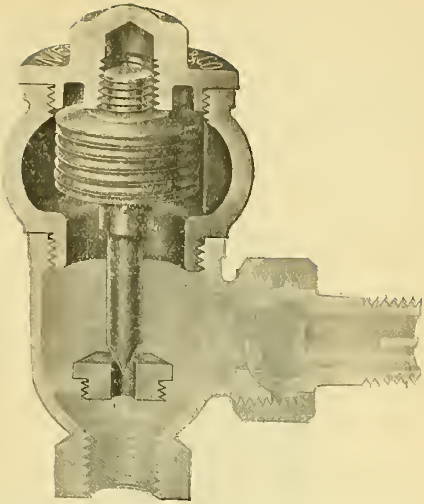


Fig. 25-3. The No. 422 Water-seal Motor in its original form and same motor changed over.

Pipe connections untouched

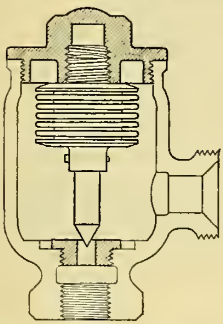


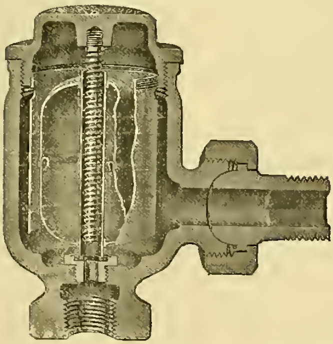
Fig. 25-4. No. 5-C-15 Sylphon Attachment for 522 or 523 Water-seal Trap where the Discharge Rating is low.

Conversion of No. 522 Water-seal Traps: The change-over in this instance requires only removal of the old bonnets and interior parts, and inserting the new Webster Sylphon Trap Attachments.

The entire change may be made in less than five minutes.

For conversion of Multiple-unit Water-seal Traps, see page 00.

Similar Webster Sylphon Attachments can be furnished for all the other sizes of Webster Water-seal Traps as follows:



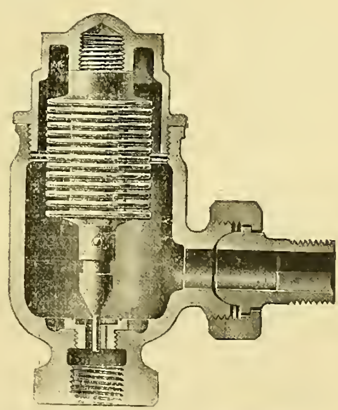


Fig. 25-4. No. 522 Webster Water-seal Trap in its original form and same trap changed over, using 12-C-15 Sylphon Attachment for higher discharge rating.

¾-in.—523	Webster	Water-seal	Trap	
¾-in.—533	"	"	"	"
1-in.—534	"	"	"	"
1-in.—544	"	"	"	"
1¼-in.—545	"	"	"	"

The No. 522 and No. 523 take the same Sylphon Attachment. Another attachment applies equally for No. 533 and No. 534. No. 544 and No. 545 each have an individual attachment.

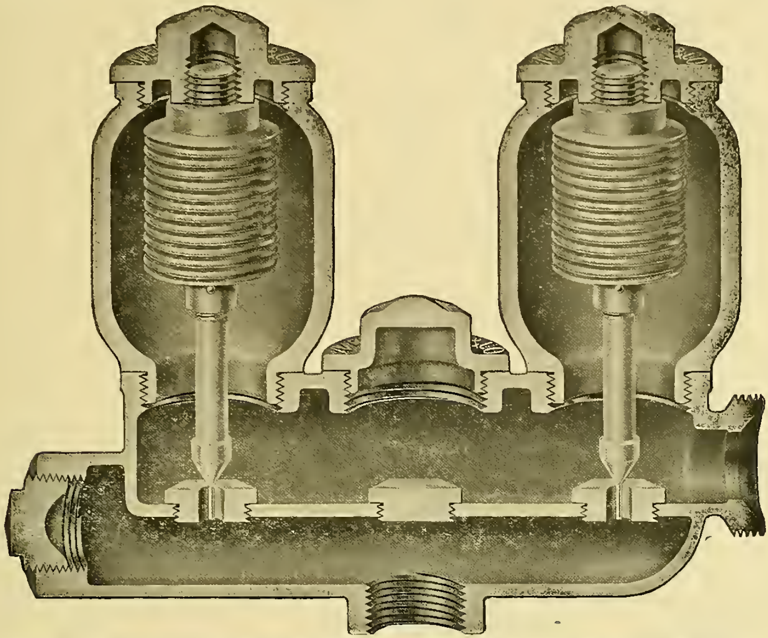


Fig. 25-6. Multiple-unit Thermostatic Valve changed over. Pipe connections untouched. Note how intervening openings are blanked out by new cap and solid seat.

Conversion of Multiple-unit Webster Valves of Earlier Types: On units of radiation beyond the capacity of a single valve it was the practice in the past to recommend and use a Multiple-unit Valve, made up of a special body having multiple openings to receive two or more bonnets similar in all respects to those used in the standard single-unit valve.

For changing these Multiple-unit Webster Valves by means of Sylphon Attachments, the use of No. 12-A-15 Sylphon Attachments is recommended for the alternate openings in the valve body, the intervening outlets being plugged as shown in Fig. 26-6.

As these Multiple Valves were made up to 6-unit, it is necessary to state whether the attachment is desired for 2-unit, 3-unit, etc., so that the proper number of attachments and solid seats and blanking-out caps will be furnished.

The Multiple-unit Valve, when changed will have capacity equal to

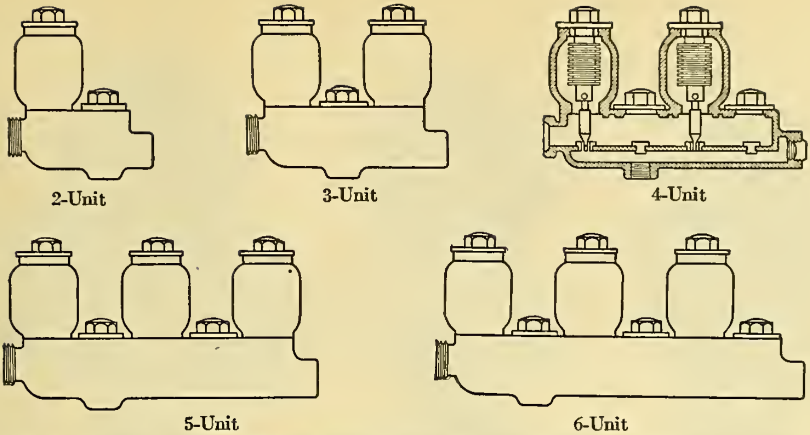


Fig. 25-7. Showing the method of applying Webster Sylphon Attachments and blanking seats and caps to the different multi-unit bodies

(and possibly in excess of) the requirements for which the valve was originally installed.

II. For "Sylphonizing" Radiator Outlet Valves of Other Makes

A GREAT demand has developed for Webster Sylphon Attachments, not only in connection with early types of Webster Valves, but for other makes of valves and traps, and in the converting of old gravity systems in which the ordinary type of hand-wheel shut-off valve was employed.

To meet the requirements of a wide variety of sizes and types of valve and trap bodies the Attachments described in the following pages have been designed. The principle is the same with each attachment. The variation is only in the work of application.

With the instructions furnished and the tools loaned for the purpose, the work of Websterizing, by means of these attachments, is so simple that it can be done in a few minutes for each radiator, and so cleanly that there is no disturbance or damage to surroundings or furnishings.

The use of these Webster Sylphon Attachments, properly applied throughout the building, will often effect the same advantages as extensive changes in piping and at a small fraction of the cost. And further, the whole work of change-over can be done without interrupting the operation of the system as a whole.

The Series 18 Webster Sylphon Attachments are made in two general forms:

Class A in which the attachment parts are fitted in an extension body which screws into the old trap or valve body; and Class C in which the attachment parts are fitted into a special brass cap which is threaded to fit the old valve or trap body.

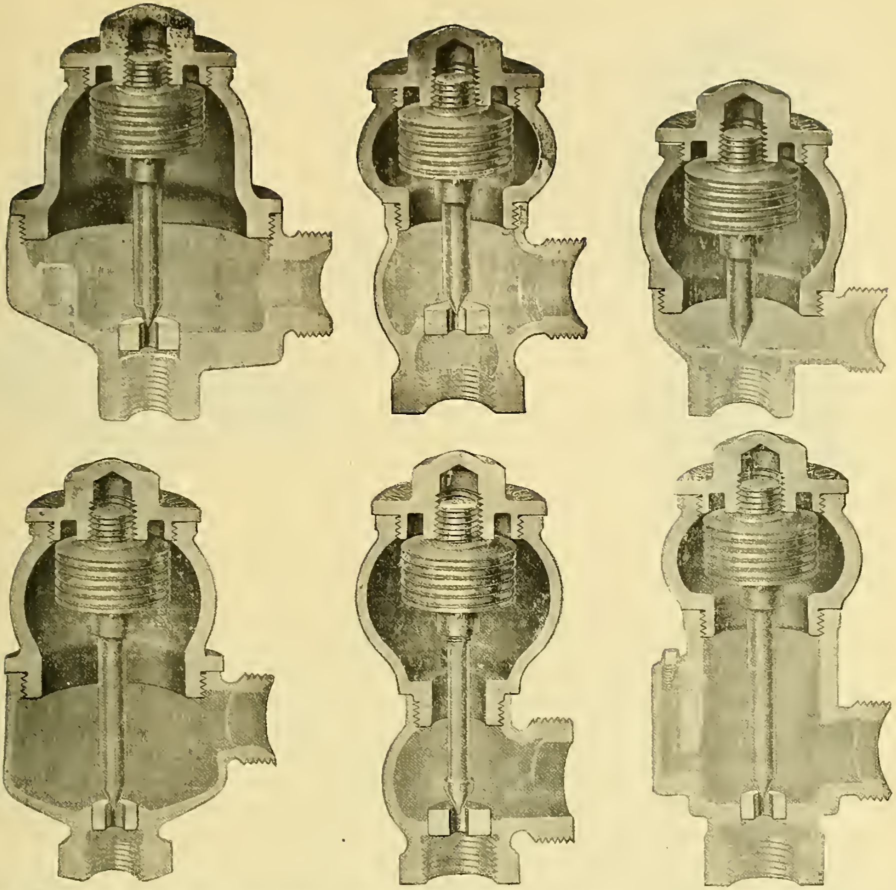


Fig. 25-8. 5-A Extension Attachments (Five-fold Sylphon bellows) applied to valve bodies of various makes.

The Class A Extension Attachments are made with extension bodies to receive 5-fold Sylphon Bellows (symbol 5-A) and to receive 12-fold Sylphon Bellows (symbol 12-A).

The extension bodies of both the 5-A and 12-A classes are made with a threaded opening at the top to receive a standard cap but of varying diameters of the lower part of the body so that the lower end may be threaded to fit the thread of the old body.

The illustrations show the full series of Extension Attachments from 5-A-12 to 5-A-27 inclusive. The 12-A Extension Attachments are similarly made in sizes 12-A-12 to 12-A-27 inclusive, although the application of only two of this type are shown.

The capacity required as indicated by tipe of radiator determines whether a 5-A or 12-A Extension Attachment should be used.

It will be noted that the valve stem attached to the Sylphon Bellows

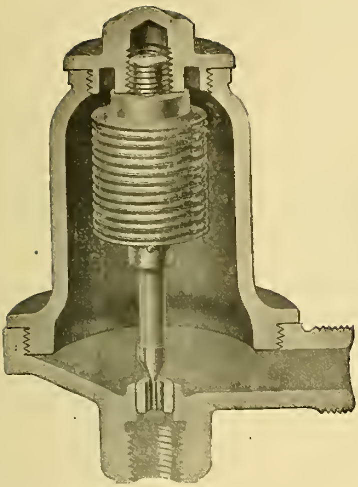
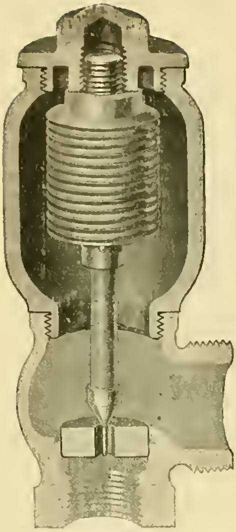


Fig. 25-9. Typical Class A Sylphon Attachments having extension bodies.
12-A Extension Attachments (Twelve-fold Sylphon bellows). Push-fit seats are installed for correct final adjustment.



varies in length with the type of valve body, but is of similar design in all cases.

The seat requires a little explanation. It is impractical to use a threaded seat, as a constant distance must be maintained from body face to seat face and this cannot be done with a threaded seat because of the variations in the distance mentioned which will occur in bodies of same make and size.

The seat is made to push-fit in the body opening which is previously prepared by reaming if necessary to the desirable diameter. Final attachment to gauge depth to meet any variation in the depth of the valve body is made by means of a push-in tool which we loan for the purpose.

In the case of ordinary globe or angle valve bodies and in various makes of float traps where preparation in this respect was not previously provided the push-fit seat described above provides means to obtain the correct final adjustment without difficulty.

The valve stem is a solid brass rod with a conical taper or seating and is of varying length as determined: (1) By the gauge depth of the old body

5-A

12-A

Fig. 25-10. Typical Extension bodies.

from bonnet face to seat. (2) By the diameter of orifice in the seat; and (3) By the rating of the radiation unit to which the valve is connected. Where necessary to provide greater vapor space through the neck of the valve, the rod is turned down to smaller diameter at such points.

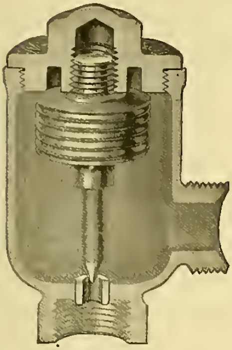
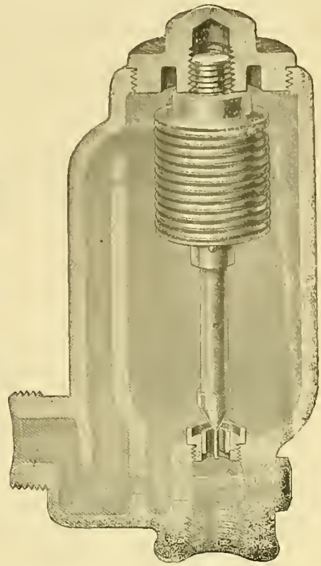


Fig. 25-11. Typical Class C Sylphon Cap Attachments placed entirely within the old bodies and push-fit seats installed for correct final adjustment.

At the left is a 5-C Attachment (Five-fold Sylphon bellows).

At the right is a 12-C Attachment (Twelve-fold Sylphon bellows).

Note this special case of a new screwed-in seat with a pushed-in ferrule for insuring accurate adjustment.



The Class C Cap Sylphon Attachments are designed for those forms of old valve and trap bodies in which the expanding member (Sylphon Bellows) and conical valve piece may be placed entirely within the old body without the use of an extension body.

With this class of attachment it is necessary to provide a special cap, threaded to fit the existing body, but the design has been standardized so that few patterns need be used to meet a wide variety of bodies.

The Class C Cap Attachments, like the Extension Attachments, are made to receive either the 5-fold or 12-fold Sylphon Bellows to which the symbols 5-C and 12-C are given.

The illustrations on page 00 show the application of Class C Cap Attachments to two different shapes of valve bodies.

The description given previously in reference to the valve stem and seat for the Extension Attachments, applies equally to the Cap Attachments.

CHAPTER XXVI

Fuel Saving by Preheating Boiler-Feed Water

WHERE exhaust steam is available and would otherwise be wasted, a considerable saving of fuel may be effected by utilizing a direct-contact (open) feed-water heater to transfer heat from the exhaust steam to the cold feed water.

The saving amounts to approximately one per cent of fuel for each 11 degrees increase in the feed-water temperature. This is the figure taken for ordinary calculations.

A more accurate method of computing this saving takes into consideration the total heat in the steam generated in the boiler, as well as the final and initial temperatures of the feed water.

This formula is

$$\text{Total saving in per cent} = \frac{100\ (t_1 - t_2)}{H + 32 - t_2}$$

in which

H = total heat above 32 deg. fahr. per pound of steam at the boiler pressure.

t_1 = temperature of water after heating.

Table 26-1. Percentage of Total Heat of Steam Saved per Degree Increase in Feed-water Temperature for Various Pressures of Saturated Steam

Initial Temp. Deg. Fahr.	Gauge Pressure in Boiler—Pounds Per Sq. In.										
	0	10	25	50	75	100	125	150	175	200	225
	Value of H										
	1150.4	1160.2	1169.2	1178.4	1184.3	1188.8	1192.2	1195.0	1197.3	1199.2	1200.9
	Per Cent Saved Per Degree Increase in Temperature										
32	.0869	.0862	.0855	.0849	.0844	.0841	.0839	.0837	.0835	.0834	.0833
40	.0875	.0868	.0861	.0854	.0850	.0847	.0844	.0843	.0841	.0840	.0839
50	.0883	.0875	.0869	.0862	.0857	.0854	.0852	.0850	.0848	.0847	.0846
60	.0891	.0883	.0876	.0869	.0865	.0862	.0859	.0857	.0855	.0854	.0853
70	.0899	.0891	.0884	.0877	.0872	.0869	.0866	.0864	.0863	.0861	.0860
80	.0907	.0899	.0892	.0884	.0880	.0877	.0874	.0872	.0870	.0869	.0867
90	.0915	.0907	.0900	.0892	.0888	.0884	.0882	.0879	.0878	.0876	.0875
100	.0924	.0916	.0908	.0900	.0896	.0892	.0889	.0887	.0886	.0884	.0883
110	.0932	.0924	.0916	.0909	.0904	.0900	.0897	.0895	.0893	.0892	.0891
120	.0941	.0933	.0925	.0917	.0912	.0909	.0906	.0903	.0902	.0900	.0899
130	.0950	.0941	.0934	.0925	.0921	.0917	.0914	.0912	.0910	.0908	.0907
140	.0959	.0950	.0942	.0934	.0929	.0925	.0922	.0920	.0918	.0916	.0915
150	.0968	.0959	.0951	.0943	.0938	.0935	.0931	.0929	.0927	.0925	.0924
160	.0978	.0969	.0960	.0952	.0947	.0943	.0940	.0937	.0935	.0934	.0932
170	.0987	.0978	.0970	.0961	.0956	.0952	.0948	.0946	.0944	.0942	.0941
180	.0997	.0988	.0979	.0970	.0965	.0961	.0957	.0955	.0953	.0951	.0950
190	1.008	.0998	.0989	.0980	.0974	.0970	.0967	.0964	.0962	.0960	.0959
200	1.018	1.008	.0999	.0990	.0984	.0980	.0976	.0974	.0972	.0970	.0968
210	1.028	1.018	1.009	.0999	.0994	.0990	.0986	.0983	.0981	.0979	.0978
220	1.039	1.029	1.019	1.010	1.001	.0999	.0996	.0993	.0991	.0989	.0987

t_2 = temperature of water before heating.

Example: Assume a boiler pressure of 140 lb. per sq. in. absolute, and initial and final temperatures of 40 deg. fahr. and 210 deg. fahr. respectively. The total saving according to this formula is 14.359 per cent, where by the "one per cent for each 11-degree increase" rule, the saving for the same conditions figures 15.45 per cent.

For convenience the results as figured from the more accurate formula have been reduced in Table 26-1, to a basis of per cent of saving per degree increase of temperature.

WEBSTER FEED-WATER HEATERS: Webster Feed-water Heaters, for obtaining the fuel savings just mentioned and other benefits not so easily measured, are made in the following types:

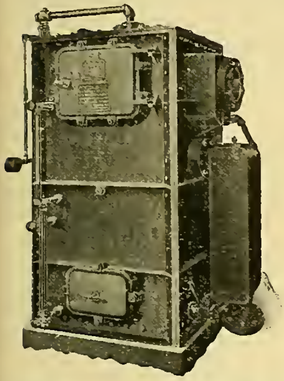


Fig. 26-1. Series 100 Class B Webster Feed-water Heater.

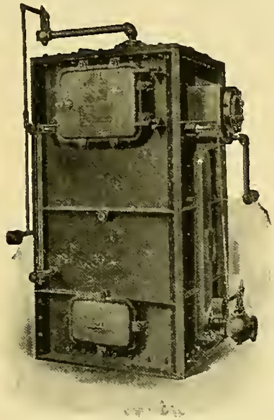


Fig. 26-2. Series 200 Class EB and Series 300 Class EBH Webster Feed-water Heater. Standard Type. Smaller sizes.

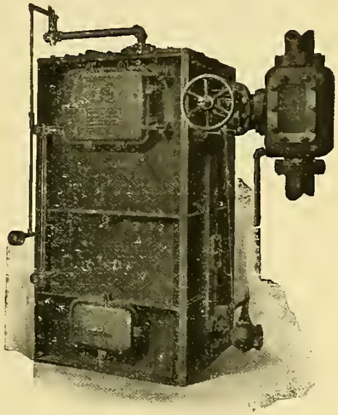


Fig. 26-3. Series 400 Class EBP and Series 500 Class EBPH Webster Feed-water Heater. Preference Cut-out Type.

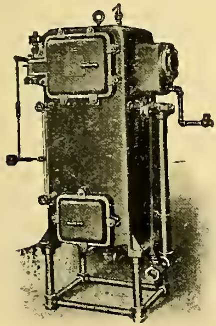


Fig. 26-4. Series 800 Class EF Webster Feed-water Heater, Standard Type.

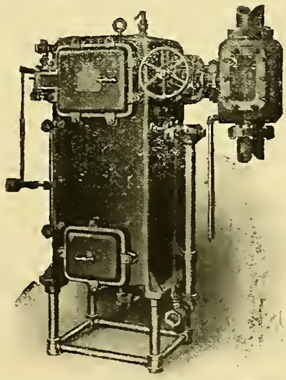


Fig. 26-5. Series 900 Class EFP Webster Feed-water Heater. Preference Cut-out Type.

Series 100, Class B, with overflow seal: The standard type for utilizing exhaust steam at atmospheric pressure and for a maximum steam pressure of ½ pound per square inch. May be operated on either induction or thoroughfare principle.

Series 200, Class EB: The standard type for use in connection with exhaust steam systems under pressures not exceeding 5 pounds per square inch. Best operated on induction principle.

Series 300, Class EBH: Same as Series 200, Class EB, but suitable for pressures up to 10 pounds per square inch maximum. Tested to 15 pounds per square inch.

Series 400, Class EBP: Same as Series 200, Class EB, but with independent oil separator large enough to purify all exhaust. Specially designed for use with exhaust steam heating or drying systems under pressures not exceeding 5 pounds per square inch.

Series 500, Class EBPH: Same as Series 400, Class EBP, but suitable

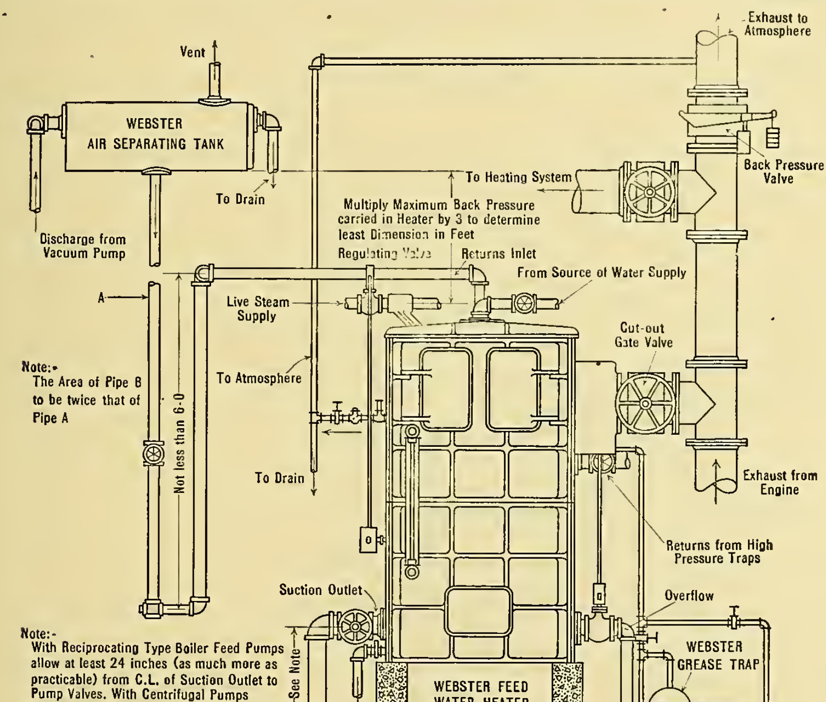


Fig. 26-6. Webster Feed-water Heater installation in connection with a Vacuum Heating System. Water inlet automatically controlled. The heater shown is of the standard type. Any other type of Webster Heater would be connected in the same way.

for pressures up to 10 pounds per square inch maximum. Tested to 15 pounds per square inch.

Series 800, Class EF: This type is for smaller capacities, 50 to 350 h.p., and is similar to Series 200, Class EB, except that the shell is a one-piece casting and is supported by a framework made from pipe and fittings. It is suitable for working pressures up to 10 pounds per square inch.

Series 900, Class EFP: Same as Series 800, Class EF, but including the large size oil separator and the cut-out valve.

WEBSTER FEED-WATER HEATERS, STANDARD TYPE: The heater shell is rectangular and made up of close-grained cast-iron plates. The exposed parts of the shell are protected by best quality heat-resisting graphite paint and all outside brass work is brightly finished.

The heater is easily cleaned, as the interior is accessible without disturbing any of the pipe connections. The large hinged doors may be quickly opened, and the trays withdrawn. The lower chamber, containing the filter, is accessible through the filter doors. Where the doors are bolted to

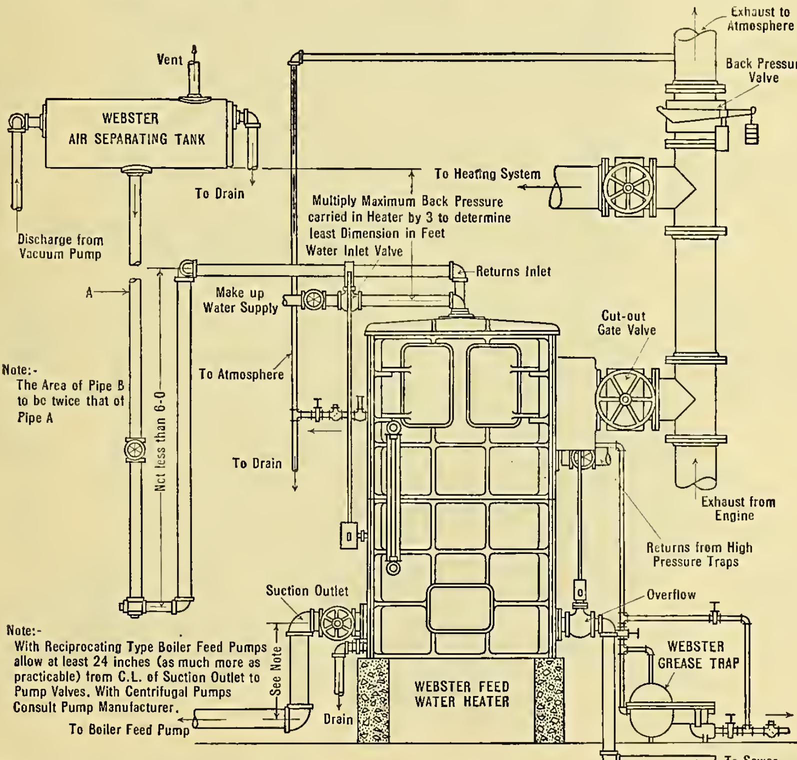


Fig. 26-7. Webster Feed-water Heater installation in connection with a Vacuum Heating System. Water inlet manually controlled. The heater shown is of the standard type. Any other type of Webster Heater would be connected in the same way.

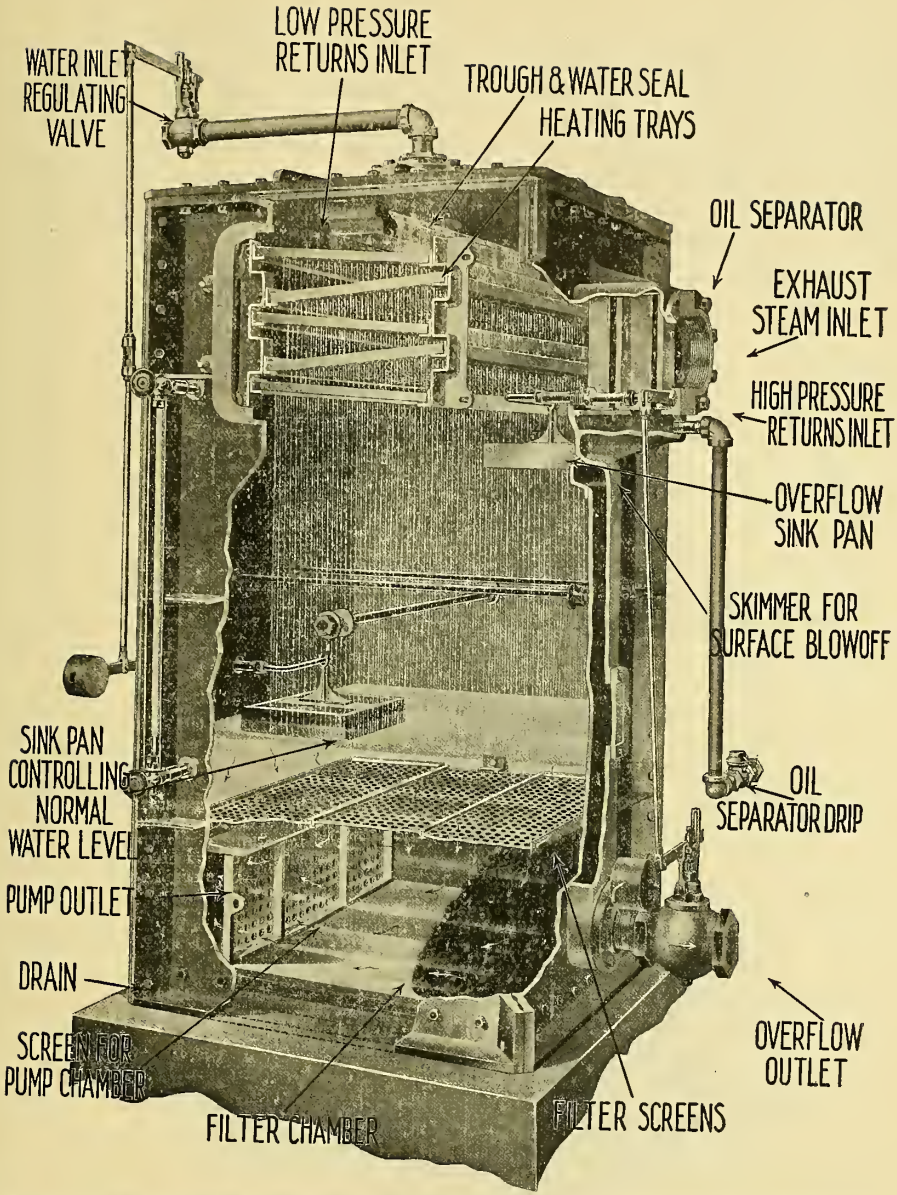


Fig. 26-8. Series 200 Class EB and Series 300 Class EBH Webster Feed-water Heaters, Standard Type.

the heater body, the shell is suitably reinforced, the faces being machined to insure tight joints.

The water supply to the heater is controlled automatically, the regulating valve being operated by a series of levers connected to an open copper sink pan (performing the functions of a float), placed within the heater shell.

Any dangerous excess of water automatically passes out of the heater when the water reaches the overflow level. Except in the case of the 100 Series, the excess water is automatically passed out through a valve actuated in the same manner as the cold water supply-valve, that is, by another open sink pan placed within the heating chamber. This valve is normally

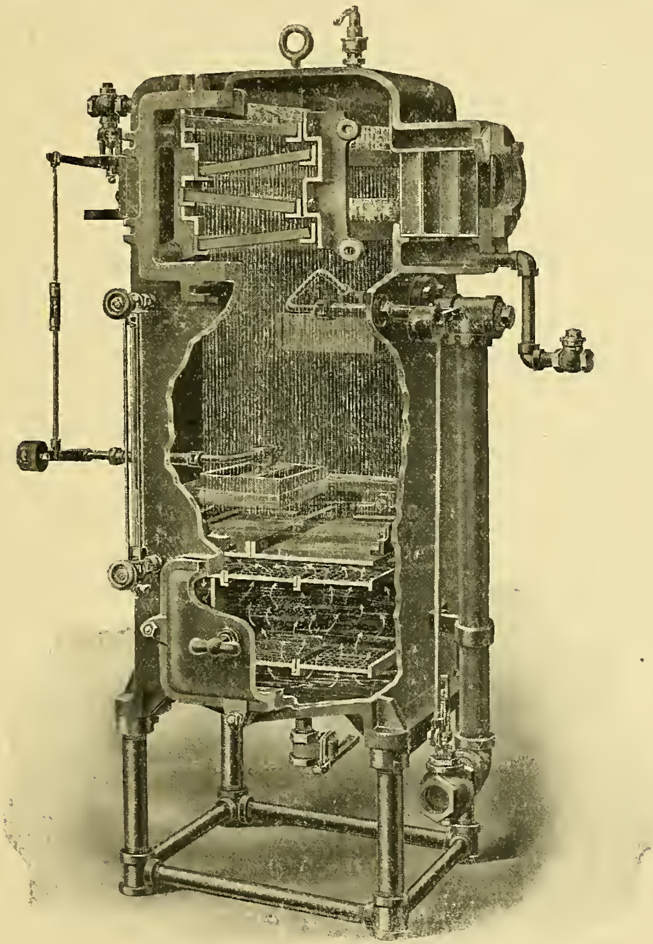


Fig. 26-9. Series 800 Class EF Webster Feed-water Heater Standard Type.

closed, preventing loss of steam.

The Webster Oil Separator which forms a part of each heater is well known and extensively used as an independent unit for removing oil from exhaust steam mains, hence its use in the Webster Feed-water Heater.

The feed water, entering the heater through the automatically controlled valve inlet, passes into the water-sealed distributing trough, which has two wide, extended lips. The water, overflowing from this trough in even sheets, is distributed over a series of oppositely inclined finely perforated metal trays, arranged one above the other as shown in the illustration on page 00. The water in its downward course falls from one tray to the other, part of it passing through the tray perforations and the balance falling from the lower edge of the tray to the tray immediately below.

This method of water travel provides the necessary surface contact for the steam and water so that the highest possible temperature is imparted to the water, causing a liberation of gases and precipitation of solids. Ample space is provided to insure uniform distribution of the steam around the trays.

By reason of the large storage chamber it is possible to utilize the heater as a "receiver" for condensation from heating systems, dry kilns, heating apparatus, etc. Between the level at which the cold water supply-valve is closed and the overflow level there is ample space for the accumulation and storage of such returns.

The filter is located in the lower compartment of the heater. In this settling chamber, opportunity is given for the precipitation and filtration of the particles of sediment and impurities and for frequent drainage through a quick-opening drain valve.

The filter bed is commonly composed of coke or other suitable material, which is contained between the perforated division screens already mentioned. This material can be renewed whenever necessary.

The large doors at the front allow ready access for charging and cleaning.

The Webster Preference Cut-Out Heater: This type, as may be noted from the illustrations, combines a Webster Heater and a large oil separator with a cut-out gate valve intervening. The oil separator has sufficient capacity to remove the oil from the exhaust steam delivered from the engines, pumps and other sources. This arrangement is therefore especially desirable where exhaust steam is to be utilized in heating or drying systems, cooking kettles or other industrial processes.

A Webster Grease Trap is used in draining the separator. Steam from the engines and auxiliaries should be combined in a common exhaust pipe before reaching the heater. This exhaust pipe may enter the separator horizontally or vertically, the latter condition being usual with the exhaust steam current upward.

Upon reaching the preference oil separator the steam flows horizontally through the baffles, which are of the standard Webster design (see Figure 00–00), comprising a number of hooked steel plates interposed in the course of the steam, causing separation by contact, by change of direction and by adhesion. The ports through which the steam is guided and the free

area through the baffles are especially designed to prevent any considerable loss of pressure.

After passing through the baffles, the steam may pass to the heater, or to the outlet into the heating system or other apparatus using exhaust steam, or to the atmosphere.

Particularly valuable advantages of the Webster Preference Cut-Out Heater are:

1. The considerable saving in piping connections and additional apparatus which may be accomplished by its use as compared with the Standard Heater.

2. The cut-out valve used in the Webster Preference Cut-Out Heater

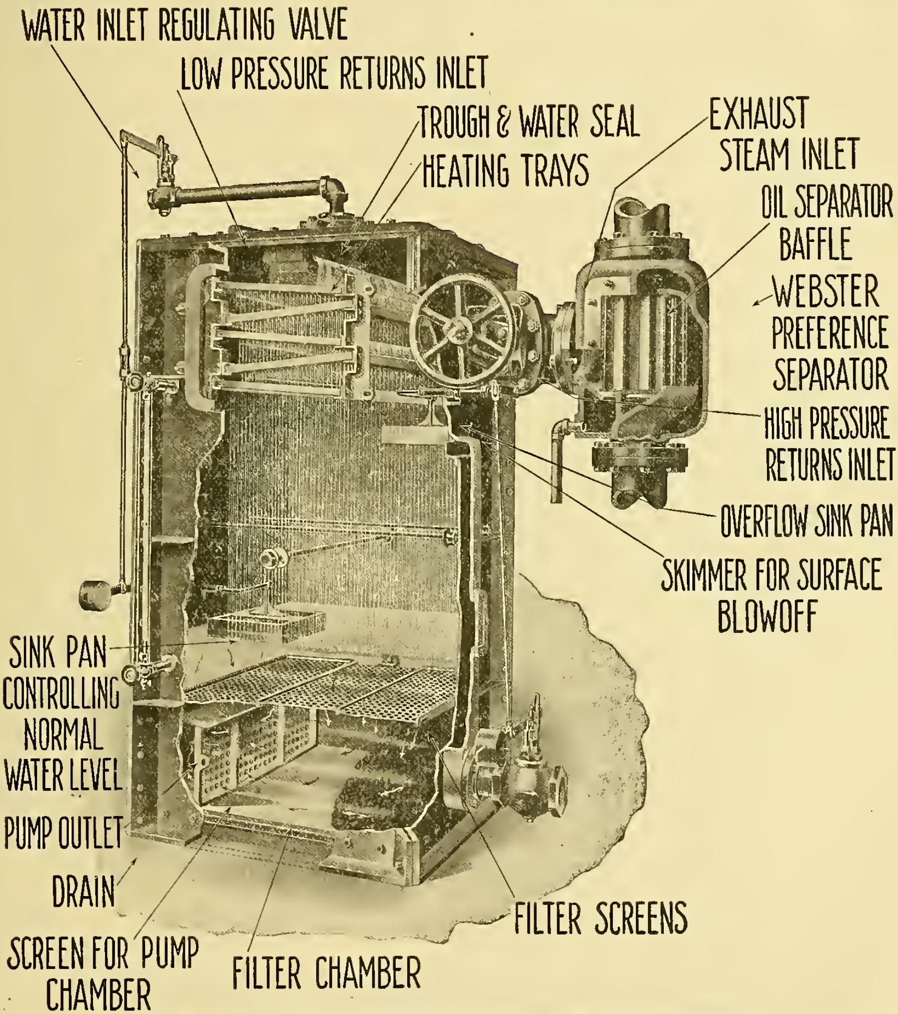


Fig. 26-10. Series 400 Class EBP and Series 500 Class EBPH Preference Cut-out Type Webster Feed-water Heaters.

is most reliable for its purpose. When the heater is cut out for internal inspection or cleaning, the course of the exhaust steam through the oil separator is such that no steam is in contact with the side wall of the heater. Steam passes through the separator and on to atmosphere or the heating system without warming up the heater body to a degree that would endanger or discomfort the man who may have to enter. A thorough clean-out is possible at any time without having to wait until the whole plant is shut down.

3. The grease and oil trap too is not integral with the overflow of the heater, so that if its outlet becomes temporarily deranged, oil cannot get back into the heater through the overflow opening.

Fig. 26-11. Series 900 Class EFP Preference Cut-out Type Webster Feed-water Heater.

Table 26-2. Dimensions of Series 200, Class EB, Webster Feed-water Heaters

For working pressure up to 5 pounds per square inch

SPECIFICATIONS

No.	CAPACITY*		Drawing No.	HEATING TRAYS		CUBIC CONTENTS		Filter	Wkg. Pres.	WEIGHTS, LBS.	
	Horsepower	Lb. Min.		Area Sq. In.	Material	Total Cu. Ft.	Water Cu. Ft.			Shipping	Max.
203	to 400	No. of Lbs. Per Min. = ½ of Rated Horsepower	9247	12.5	American Ingot Iron or Copper	24.25	14.7	Downward Flow	5 Pounds Per Sq. Inch	2600	3600
205	425 to 650		9203	16.5		40.75	25.5			3700	5400
207	675 to 900		9250	24.0		60.2	40.0			4700	7300
210	925 to 1350		9254	33.0		81.6	52.0			5700	9000
215	1375 to 1850		9252	51.6		121.3	80.0			8000	13100
220	1875 to 2400		9257	63.8		151.6	104.0			9000	15700
225	2425 to 3000		9256	82.0		180.0	128.1			10300	18400
230	3100 to 4000		22457	95.7		214.0	133.5			13000	21300
235	4100 to 5500		13377	121.5		242.0	140.0			15000	23600
250	5600 to 7500		13626	160.1		318.5	179.0			20000	31400
285	7600 to 9500		22196	201.5		400.	222.0			22000	36000
299	9600 to 12000		18779	243.0		485.3	268.0			25000	41700

* One rated horsepower-capacity for heating 30 pounds of water per hour from 40 degrees fahr. to a temperature within 5 degrees of the steam temperature.

No.	CONNECTIONS								
	①	②	③	④	⑤	⑥	⑦	⑧	⑨
203	6	1	4	2½	¾	1½	2½	2½	1
205	8	1½	4	3	1	1½	3	2½	1
207	8	2	5	3½	1	2	4	3	1
210	10	2	5	3½	1	2	5	3	1
215	12	2½	6	4	1	2½	5	3	1¼
220	14	2½	6	4	1¼	3	5	5	1½
225	16	3	8	5	1¼	4	5	5	1½
230	18	3	8	6	1¼	4	2–5	5	1½
235	20	4	10	8	1¼	5	2–3	5	1½
250	22	5	10	8	1¼	5	2–3	5	1½
285	24	5	12	8	2–1½	5	2–4	6	2
299	28	6	12	10	2–1½	5	2–6	8	2

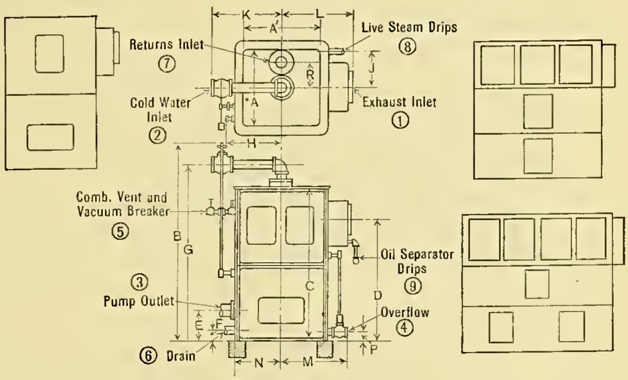


Fig. 26-12.

No.	TRAYS		FOUNDATION		OVERALL	DIMENSIONS																
	No.	Size	Lg.	Wd.	Hgt.	A	A'	B	C	D	E	F	G	H	J	K	L	M	N	P	R
203	5	15 x24	35	35	80¾	26	26	80¾	66	54⅝	6¾	4⅜	73¼	18½	9	22	22	25¼	16	5¾	7⅞
205	5	15½x30⅝	41	41	88	32	32	88	72	57⅝	7¼	5¼	79¾	21½	11½	27	25½	28¾	19⅛	7¼	8¾
207	6	16 x36	45	45	101⅝	36	36	101⅝	84	69¼	7¼	6	93¼	25	13¾	28½	27½	34	21⅛	9⅛	11
210	12	10 x40⅜	51	51	101⅝	42	42	101⅝	84	67¼	7⅞	6	93¼	28	15½	31½	31½	37	24⅝	8	10½
215	12	13½x46	57	57	115⅛	48	48	115⅛	96	77⅛	8½	7	104¾	33¼	16	36	36¾	41½	27⅝	8¼	13¾
220	12	16¾x47	69	57	115½	48	60	115½	96	81⅝	8½	7	105½	40	18	41½	45	47½	33⅝	10¾	13¾
225	24	17½x28	69	66	117¾	57	60	117¾	96	82⅜	9	7½	106½	43	19¾	42	42⅝	46¼	33⅝	10¾	16
230	18	16¾x47	93	57	115¼	48	84	115¼	96	77	9	7½	105⅛	52	20	57	55	53⅞	45⅝	11¼	12½
235	24	15½x47	105	57	120⅝	48	96	120⅝	96	77	12½	9¼	105⅞	61¼	23¾	64	61	63⅛	51⅝	12	13¾
250	48	15½x31	105	72	122⅞	63	96	122⅞	96	75	11½	9¼	107⅜	61	25⅞	65	67¼	63⅛	51⅝	12	20
285	48	15½x39	105	89	122⅜	80	96	122⅜	96	75	....	9¼	107⅜	61¼	27	65	67¼	63⅛	....	12	16
299	48	15½x47	105	105	124⅜	96	96	124⅜	96	75	....	8½	107⅜	61¼	40	65	67¼	67⅛	....	12	36½

All sizes and dimensions in inches.

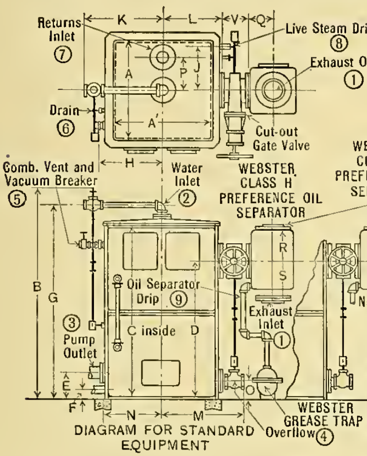
NOTE: The above data (except weights) applies also to extra heavy 300 series Class EBH Heaters for working pressures up to 10 pounds per square inch.

Table 26-3. Dimensions of 400 Series Class EBP Webster Feed-water Heaters

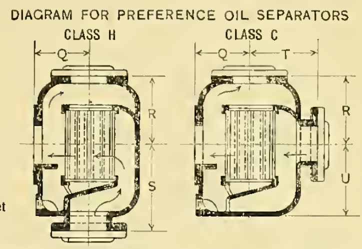
For working pressure up to 5 pounds per square inch
SPECIFICATIONS

No.	Capacity* Horsepower	Capacity* Lb. Min.	Drawing No.	Heating Trays Area Sq. In.	Heating Trays Material	Cubic Contents Total Cu. Ft.	Cubic Contents Water Cu. Ft.	Filter	Wkg. Press	Weights Lbs. Shipping	Weights Lbs. Max.
403	to 400	No. of lbs. per Min. = ½ of rated Horsepower	13166	12.5	American Ingot Iron or Copper	24.25	14.7	Downward Flow	5 Pounds Per Square Inch	3500	4500
405	425 to 650		13188	16.5		40.75	25.5			4950	6650
407	675 to 900		13167	24.0		60.20	40.0			6700	9300
410	925 to 1350		13165	33.0		81.60	52.0			8050	11350
415	1375 to 1850		13171G	51.6		121.30	80.0			10800	15900

* One rated horsepower = Capacity for heating 30 pounds per hour from 40 deg. Fahr. to a temperature within 5 degrees of the steam temperature.



Note:- The Table of Dimensions below refers to Heaters with Standard Equipment. Separators smaller or larger than Standard will be furnished if desired. The table at right shows sizes of all Preference Oil Separators which can be used with this type Heater



Size	Capacity Lbs. Steam per Min.	Dimensions Q	R	S	T	U	Size Drip
6	46	8	11	13	9⅝	10¾	1
8	80	9¼	12½	14	11	11⅝	1
10	125	11¾	14⅜	17⅞	14¾	14	1
12	175	13¼	15½	18	15⅝	15¼	1
14	255	15	18	23	19⅛	18⅛	1½
16	335	17¾	21	27¼	22¼	22	1½
18	375	19⅜	24	29⅛	24½	23⅝	1½
20	475	20¼	25⅜	30¾	25¼	24¾	1½

Fig. 26-13.

Size No.	Standard Equipment Size Sep'r.	Size Valve	Size Trap	Connections ①	②	③	④	⑤	⑥	⑦	⑧	⑨	Trays No.	Trays Size	Foundation L'gth	Foundation Width	Overall Height
403	10	6	1	10	1	4	2½	¾	1½	2½	2½	1	5	15 x24	35	35	80¾
405	12	8	1	12	1½	4	3	1	1½	3	2½	1	5	15½x30⅝	41	41	88
407	16	8	1½	16	2	5	3½	1	2	4	3	1	6	16 x36	45	45	101⅝
410	18	10	1½	18	2	5	3½	1	2	5	3	1	12	10 x40⅜	51	51	101⅝
415	20	12	1½	20	2½	6	4	1	2½	5	3	1¼	12	13½x46	57	57	115⅛

Size No.	Dimensions A	A'	B	C	D	E	F	G	H	J	K	L	M	N	O	P	V
403	26	26	80¾	66	54⅝	6¾	4⅜	73¼	18½	9	22	17½	25¼	16	5¾	7⅞	10½
405	32	32	88	72	57⅝	7¼	5¼	79¾	21½	11½	27	20½	28¾	19⅛	7¼	8¾	11½
407	36	36	101⅝	84	69¼	7¼	6	93¼	25	13¾	28½	22½	34	21⅛	9⅛	11	11½
410	42	42	101⅝	84	67¼	7⅞	6	93¼	28	15½	31½	25½	37	24⅝	8	10½	13
415	48	48	115⅛	96	77⅛	8½	7	104¾	33¼	16	36	28½	41½	27⅝	8¼	13¾	14

NOTE: The dimensions and data above—except weights—may be used also for the 500 Series Class EBPH Extra-heavy pattern Webster Feed-water Heaters.

Table 26-4. Dimensions of Series 800, Class EF, Webster Feed-water Heaters

For working pressure up to 10 pounds per square inch

SPECIFICATIONS

No.	CAPACITY *		Drawing No.	HEATING TRAYS		CUBIC CONTENTS		Filter	Wkg. Press	WEIGHTS, LBS.	
	Horsepower	Lb. Min.		Area Sq. In.	Material	Total Cu. Ft.	Water Cu. Ft.			Shipping	Max.
800	to 75	No. of Pds. Per Min = 1/2 of Rated Horsepower	17045	4.5	American Ingot Iron or Copper	7.1	4.2	Upward Flow	10 Pds. Per Square Inch	1125	1400
801	80 to 125		16660	5.0		9.8	5.9			1450	1850
801½	130 to 175		16661	5.6		11.6	7.3			1700	2200
802	180 to 260		16662	9.0		16.4	11.08			2200	2900
803	265 to 400		16663	13.5		22.9	14.2			2450	3350

* One rated horsepower = Capacity for heating 30 pounds of water per hour from 40 degrees fahr. to a temperature within 5 degrees of the steam temperature.

No.	CONNECTIONS								
	①	②	③	④	⑤	⑥	⑦	⑧	⑨
800	3	1	2	1½	¾	1¼	1½	1¼	¾
801	4	1	2½	2	¾	1¼	1½	1½	¾
801½	4	1	3	2	¾	1¼	1½	1½	¾
802	5	1	3	2	¾	1¼	2	1½	¾
803	6	1	4	2½	¾	1½	2½	2	1

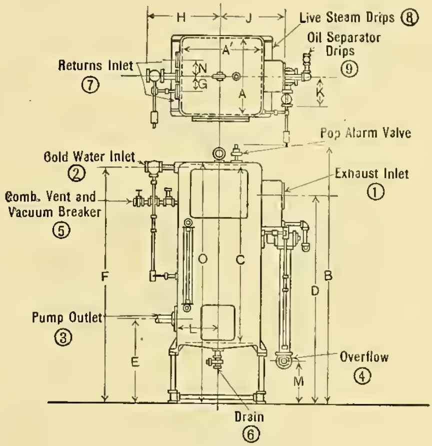


Fig. 26-14.

No.	TRAYS		WATERLINE			FILTER			DIMENSIONS															
	No.	Size	O'er	Pow	Rec.	Th.	Ar.	Cu. Ft.	A	A'	B	C	D	E	F	G	H	J	K	L	M	N	O
800	4	10x16	39⅛	35½	32⅝	6	2.0	.9	16	18	62	43¼	48	20½	55¼	4⅞	18½	14	7⅛	9¾	10⅛	3¼	57
801	4	10x18	45	38¾	35⅛	6	2.5	1.2	18	20	68½	47¼	54⅜	23	61¾	4⅞	19½	17⅛	9⅜	10¾	11⅜	3⅞	63½
801½	4	10x20	47¾	42½	35⅛	6	2.8	1.4	20	20	72½	51	58½	23	65	4⅞	19½	17⅛	9⅜	10¾	11⅜	4⅞	67¼
802	4	14x23	55½	49¼	36⅝	6	3.6	1.8	22¾	22¾	79	58⅛	63⅝	24½	71⅝	5	21	19¼	9½	12¼	12⅞	5	74
803	5	15x26	56½	50⅛	40¼	6	4.6	2.4	25¾	25¾	84½	61	68	27⅛	77	5½	22½	21⅞	10	13¾	13⅞	5½	79⅜

All sizes and dimensions in inches.

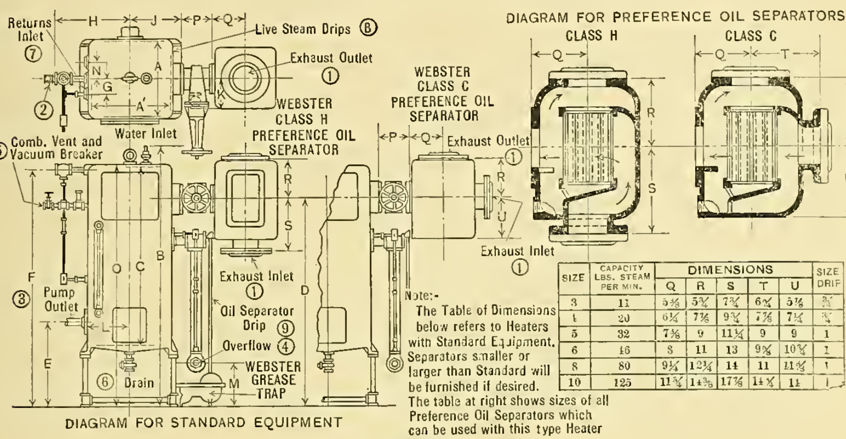
Table 26-5. Dimensions of 900 Series Class EFP Webster Feed-water Heaters

For working Pressure up to 10 pounds per square inch

SPECIFICATIONS

No.	CAPACITY* Horsepower	CAPACITY* Lb. Min.	Drawing No.	HEATING TRAYS Area Sq. In.	HEATING TRAYS Material	CUBIC CONTENTS Total Cu. Ft.	CUBIC CONTENTS Water Cu. Ft.	Filter	Wkg. Press	WEIGHTS, LBS. Shipping	WEIGHTS, LBS. Max.
900	to 75	No. of Lbs. Per Min. ½ of Rated Horsepower	17198	4.5	American Ingot Iron or Copper	7.1	4.2	Upward Flow	10 Pounds Per Sq. In.	1675	1925
901	80 to 125		16837	5.0		9.8	5.9			1780	2140
901½	130 to 175		16724	5.6		11.6	7.3			2200	2600
902	180 to 260		17203	9.0		16.4	11.08			2700	3425
903	265 to 400		17205	13.5		22.9	14.2			3200	4100

* One rated horsepower = Capacity for heating 30 pounds per hour from 40 degrees fahr. to a temperature within 5 degrees of the steam temperature.



SIZE	CAPACITY LBS. STEAM PER MIN.	DIMENSIONS Q	R	S	T	U	SIZE DRIP
3	11	5⅜	5¾	7¾	6¾	5⅛	¾
4	20	6¼	7⅞	9¾	7⅞	7¼	¾
5	32	7⅝	9	11¼	9	9	1
6	46	8	11	13	9⅝	10¾	1
8	80	9¼	12¼	14	11	11¾	1
10	125	11¾	14⅜	17⅞	14¼	14	1

Fig. 26-15.

Size No.	STANDARD EQUIPMENT Size Sep'r	Size Valve	Size Trap	CONNECTIONS ①	②	③	④	⑤	⑥	⑦	⑧	⑨	TRAYS No.	TRAYS Size	FOUNDATION L'gth	FOUNDATION Width	Overall Heigth
900	4	3	¾	4	1	2	1½	¾	1¼	1½	1¼	¾	4	10x16	26	23	62
901	6	4	1	6	1	2½	2	¾	1¼	1½	1½	1	4	10x18	28	25	68½
901½	8	4	1	8	1	3	2	¾	1¼	1½	1½	1	4	10x20	28	27	72½
902	8	5	1	8	1	3	2	¾	1¼	2	1½	1	4	14x23	30	30	79
903	10	6	1	10	1	4	2½	¾	1½	2½	2	1	5	15x26	33	33	84½

Size No.	DIMENSIONS A	A'	B	C	D	E	F	G	H	J	K	L	M	N	O	P
900	16	18	62	43¼	48	20½	55¼	3¼	18½	14	7⅛	9¾	10⅛	3¼	57	8
901	18	20	68½	47¼	54⅜	23	61¾	3⅞	19½	13	9⅜	10¾	11⅜	3⅞	63½	9
901½	20	20	72½	51	58⅛	23	65½	4⅞	19½	13¼	9⅜	10¾	11⅜	4⅞	67¼	9
902	22¾	22¾	79	56½	63⅝	24½	71⅝	5	21	13¼	9½	12¼	12⅞	5	74	10
903	25¾	25¾	84½	61	68	27⅛	77	5½	22½	16¼	10	13¾	13⅞	5½	79⅜	10½

All sizes and dimensions in inches.

THE WEBSTER-LEA HEATER METER

THIS apparatus is a practical combination of the thoroughly efficient Webster Feed-water Heater of the rectangular cast-iron type, with the Lea V-Notch Recording Meter so arranged that either unit may be operated with equal efficiency, either in combination with or independently of the other.

Fig. 26-16. Typical Webster-Lea Heater Meter.

Besides heating the boiler feed water to the boiling point, this apparatus indicates the actual amount of boiler evaporation. Its continuous meter

Fig. 26-17. Typical chart from a Webster-Lea Heater Meter.

records show up careless or improper firing methods, leakage, condensation due to poor installation, inferior coal and in other words, act as a check upon the general efficiency of the entire boiler plant.

The charts (Fig. 26–17) can be integrated by means of a standard planimeter, such as used for engine indicator charts, and an integrating attachment giving the total flow for any period is supplied. The readings from the integrating attachment indicate approximately quantities of water which have passed over the weir.

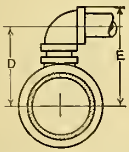
Where it is desired to have a record of the feed water temperature on the same chart with the meter record, a special attachment can be fitted to any standard instrument. The meter chart and drum are made wider to provide 2½ inches for temperature calibrations. This space has twenty-five equal divisions calibrated in any 50 or 100 deg. interval specified. For example, the range may be 175-225 deg. or 150-250 deg. or 100-150 deg., etc.

CHAPTER XXVII

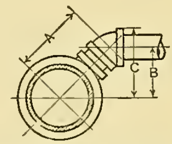
Miscellaneous Useful Information

THE tables in the following pages cover many subjects on which the Heating Engineer mnst have readily available data. They have been selected after careful consideration and will be found reliable and sufficiently accurate in every respect to meet the requirements of good practice

Table 27-00. Space Required for Branch Connections



From Compilation of
F. D. B. Ingalls, M. E.,
in Model Boiler Manual



Minimum Height of Connections off Pipe Mains

Mains	Branches	A	B	C	D	E	Branches	Mains
2	1	3 3/8	2 3/8	3 13/32	3 31/32	5	1	2
2	1 1/4	3 11/16	2 5/8	3 7/8	4 7/16	5 11/16	1 1/4	2
2	1 1/2	4	2 27/32	4 7/32	4 13/16	6 3/16	1 1/2	2
2 1/2	1	3 3/4	2 21/32	3 11/16	4 11/32	5 3/8	1	2 1/2
2 1/2	1 1/4	4 1/16	2 7/8	4 1/8	4 13/16	6 1/16	1 1/4	2 1/2
2 1/2	1 1/2	4 3/8	3 3/32	4 15/32	5 3/16	6 9/16	1 1/2	2 1/2
2 1/2	2	4 7/8	3 7/16	5 1/8	5 7/8	7 9/16	2	2 1/2
3	1	4 1/16	2 7/8	3 29/32	4 21/32	5 11/16	1	3
3	1 1/4	4 3/8	3 3/32	4 11/32	5 1/8	6 3/8	1 1/4	3
3	1 1/2	4 11/16	3 5/16	4 11/16	5 1/2	6 7/8	1 1/2	3
3	2	5 3/16	3 11/16	5 3/8	6 3/16	7 7/8	2	3
3	2 1/2	5 9/16	3 15/16	6	6 13/16	8 7/8	2 1/2	3
3 1/2	1	4 11/32	3 1/16	4 3/32	4 15/16	5 31/32	1	3 1/2
3 1/2	1 1/4	4 21/32	3 5/16	4 9/16	5 13/32	6 21/32	1 1/4	3 1/2
3 1/2	1 1/2	4 31/32	3 17/32	4 29/32	5 25/32	7 5/32	1 1/2	3 1/2
3 1/2	2	5 15/32	3 7/8	5 9/16	6 15/32	8 5/32	2	3 1/2
3 1/2	2 1/2	5 27/32	4 1/8	6 3/16	7 3/32	9 5/32	2 1/2	3 1/2
4	1	4 11/16	3 5/16	4 11/32	5 9/32	6 5/16	1	4
4	1 1/4	5	3 17/32	4 25/32	5 3/4	7	1 1/4	4
4	1 1/2	5 5/16	3 3/4	5 1/8	6 1/8	7 1/2	1 1/2	4
4	2	5 13/16	4 1/8	5 13/16	6 13/16	8 1/2	2	4
4	2 1/2	6 3/16	4 3/8	6 7/16	7 7/16	9 1/2	2 1/2	4
5	1 1/4	5 17/32	3 29/32	5 5/32	6 9/32	7 17/32	1 1/4	5
5	1 1/2	5 27/32	4 1/8	5 1/2	6 21/32	8 1/32	1 1/2	5
5	2	6 11/32	4 1/2	6 3/16	7 11/32	9 1/32	2	5
5	2 1/2	6 23/32	4 3/4	6 13/16	7 31/32	10 1/32	2 1/2	5
6	1 1/4	6 3/16	4 3/8	5 5/8	6 5/16	8 3/16	1 1/4	6
6	1 1/2	6 1/2	4 5/8	6	7 5/16	8 11/16	1 1/2	6
6	2	7	4 31/32	6 21/32	8	9 11/16	2	6
6	2 1/2	7 3/8	5 7/32	7 9/32	8 5/8	10 11/16	2 1/2	6
8	2	8 1/4	5 27/32	7 17/32	9 1/4	10 15/16	2	8
8	2 1/2	8 5/8	6 1/8	8 3/16	9 7/8	11 15/16	2 1/2	8
8	3	9	6 3/8	8 3/4	10 7/16	12 13/16	3	8

Table 27-00. Dimensions of Standard Wrought-Iron Pipe *

Black and Galvanized for Temperatures up to 450 deg.

1¼ and smaller proved to 300 pounds per square inch by hydraulic pressure.
1½ and larger proved to 500 pounds per square inch by hydraulic pressure.

Nominal Diameter	Actual Outside Diameter	Actual Inside Diameter	Inside Circumference	Outside Circumference	Length of Pipe per Sq. Ft. of Inside Surface	Length of Pipe per Sq. Ft. of Outside Surface	Inside Area	Outside Area	Length of Pipe Containing One Cubic Foot	Weight per Ft.
In.	In.	In.	In.	In.	Ft.	Ft.	In.	In.	Ft.	Lb.
⅛	0.405	0.270	0.848	1.272	14.15	9.44	0.0572	0.129	2500.	0.243
¼	0.54	0.361	1.144	1.696	10.50	7.075	0.1041	0.229	1385.	0.422
⅜	0.675	0.494	1.552	2.121	7.67	5.657	0.1916	0.358	751.5	0.561
½	0.84	0.623	1.957	2.652	6.13	4.502	0.3048	0.554	472.4	0.845
¾	1.05	0.824	2.589	3.299	4.635	3.637	0.5333	0.866	270.	1.126
1	1.315	1.048	3.292	4.134	3.679	2.903	0.8627	1.357	166.9	1.670
1¼	1.66	1.380	4.335	5.215	2.768	2.301	1.496	2.164	96.25	2.258
1½	1.90	1.611	5.061	5.969	2.371	2.01	2.038	2.835	70.65	2.694
2	2.375	2.067	6.494	7.461	1.848	1.611	3.355	4.430	42.36	3.600
2½	2.875	2.468	7.754	9.032	1.547	1.328	4.783	6.491	30.11	5.773
3	3.50	3.067	9.636	10.996	1.245	1.091	7.388	9.621	19.49	7.547
3½	4.00	3.548	11.146	12.566	1.077	0.955	9.887	12.566	14.56	9.055
4	4.50	4.026	12.648	14.137	0.949	0.849	12.730	15.904	11.31	10.66
4½	5.00	4.508	14.153	15.708	0.848	0.765	15.939	19.635	9.03	12.34
5	5.563	5.045	15.849	17.475	0.757	0.629	19.990	24.299	7.20	14.50
6	6.625	6.065	19.054	20.813	0.63	0.577	28.889	34.471	4.98	18.767
7	7.625	7.023	22.063	23.954	0.544	0.595	38.737	45.663	3.72	23.27
8	8.625	7.982	25.076	27.096	0.478	0.444	50.039	58.426	2.88	28.177
9	9.625	9.001	28.277	30.433	0.425	0.394	63.633	73.715	2.26	33.70
10	10.75	10.019	31.475	33.772	0.381	0.355	78.838	90.762	1.80	40.06
11	12.00	11.25	35.343	37.699	0.340	0.318	98.942	113.097	1.455	45.95
12	12.75	12.000	38.264	40.840	0.313	0.293	116.535	132.732	1.235	48.98
14	14.00	13.25	41.268	43.982	0.290	0.273	134.582	153.938	1.069	53.92
15	15.00	14.25	44.271	47.124	0.271	0.254	155.968	176.715	.923	57.89
16	16.00	15.25	47.274	50.265	0.254	0.238	177.867	201.062	.809	61.77
18	18.00	17.25	53.281	56.548	0.225	0.212	225.907	254.469	.638	69.66
20	20.00	19.25	59.288	62.832	0.202	0.191	279.720	314.160	.515	77.57

* Walworth Manufacturing Company.

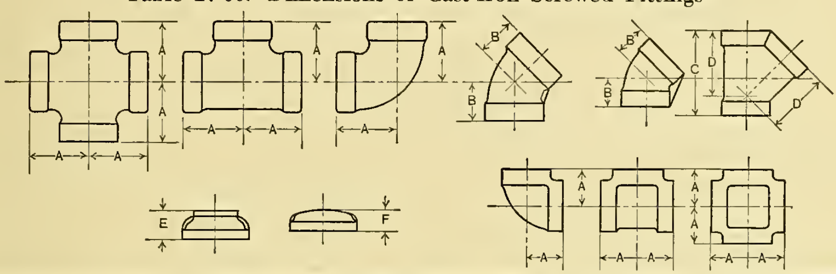
Table 27-00. Dimensions of Black and Galvanized Wrought-Iron Pipe—Extra Strong and Double Extra Strong

Size	EXTRA STRONG				DOUBLE EXTRA STRONG			
	Diameters		Thickness	Weight per Foot Plain Ends	Diameters		Thickness	Weight per Foot Plain Ends
	External	Internal			External	Internal		
⅛	.405	.215	.095	.314				
¼	.540	.302	.119	.535				
⅜	.675	.423	.126	.738				
½	.840	.546	.147	1.087	.840	.252	.294	1.714
¾	1.050	.742	.154	1.473	1.050	.434	.308	2.440
1	1.315	.957	.179	2.171	1.315	.599	.358	3.659
1¼	1.660	1.278	.191	2.996	1.660	.896	.382	5.214
1½	1.900	1.500	.200	3.631	1.900	1.100	.400	6.408
2	2.375	1.939	.218	5.022	2.375	1.503	.436	9.029
2½	2.875	2.323	.276	7.661	2.875	1.771	.552	13.695
3	3.500	2.900	.300	10.252	3.500	2.300	.600	18.583
3½	4.000	3.364	.318	12.505	4.000	2.728	.636	22.850

Table 27-00. Dimensions of Black and Galvanized Wrought-Iron Pipe—Extra Strong and Double Extra Strong—*Continued*

Size	Extra Strong: Diameters External	Extra Strong: Diameters Internal	Extra Strong: Thickness	Extra Strong: Weight per Foot Plain Ends	Double Extra Strong: Diameters External	Double Extra Strong: Diameters Internal	Double Extra Strong: Thickness	Double Extra Strong: Weight per Foot Plain Ends
4	4.500	3.826	.337	14.983	4.500	3.152	.674	27.541
4 1/2	5.000	4.290	.355	17.611	5.000	3.580	.710	32.530
5	5.563	4.813	.375	20.778	5.563	4.063	.750	38.552
6	6.625	5.761	.432	28.573	6.625	4.897	.864	53.160
7	7.625	6.625	.500	38.048	7.625	5.875	.875	63.079
8	8.625	7.625	.500	43.388	8.625	6.875	.875	72.424
9	9.625	8.625	.500	48.728				
10	10.750	9.750	.500	54.735				
11	11.750	10.750	.500	60.075				
12	12.750	11.750	.500	65.415				
13	14.000	13.000	.500	72.091				
14	15.000	14.000	.500	77.431				
15	16.000	15.000	.500	82.771				

Table 27-00. Dimensions of Cast-Iron Screwed Fittings*



Size, Inches	Standard A Inches	Standard B Inches	Extra Heavy A Inches	Extra Heavy B Inches	Standard and Extra Heavy C Inches	Standard and Extra Heavy D Inches	Standard and Extra Heavy E Inches	Standard and Extra Heavy F Inches
1/4	13/16	3/4	....	...	....	....	...	...
3/8	15/16	13/16	....	...	....	....	...	...
1/2	1 1/8	7/8	....	...	2 1/2	1 7/8	...	...
3/4	1 5/16	1	....	...	3	2 1/4	...	...
1	1 7/16	1 1/8	....	...	3 1/2	2 3/4	...	...
1 1/4	1 3/4	1 5/16	2	1 3/8	4 1/4	3 1/4	...	...
1 1/2	1 15/16	1 7/16	2 1/4	1 1/2	4 7/8	3 13/16	...	...
2	2 1/4	1 11/16	2 9/16	1 5/8	5 3/4	4 1/2	...	...
2 1/2	2 11/16	1 15/16	3	1 15/16	6 1/4	5 3/16	...	...
3	3 1/8	2 3/16	3 1/2	2 1/4	7 7/8	6 1/8	2 15/16	...
3 1/2	3 7/16	2 3/8	4 1/8	2 1/2	8 7/8	6 7/8	3 1/8	...
4	3 3/4	2 5/8	4 11/16	2 9/16	9 3/4	7 5/8	3 3/8	2 1/16
4 1/2	4 1/16	2 13/16	5 1/8	2 3/4	11 5/8	9 1/4	3 5/8	2 3/16
5	4 7/16	3 1/16	5 1/2	3	11 5/8	9 1/4	3 7/8	2 3/8
6	5 1/8	3 7/16	6 1/8	3 5/16	13 7/16	10 3/4	4 3/8	2 5/8
7	5 13/16	3 7/8	7 1/4	3 3/4	15 1/4	12 1/4	4 13/16	2 7/8
8	6 1/2	4 1/4	8 1/8	4	16 15/16	13 5/8	5 1/4	3 1/8
9	7 3/16	4 11/16	9 1/8	4 3/4	20 11/16	16 3/4	5 11/16	3 3/8
10	7 7/8	5 3/16	11 3/8	4 7/8	20 11/16	16 3/4	6 3/16	3 5/8
12	9 1/4	6	13 3/8	5 1/2	24 1/8	19 5/8	7 1/8	4 1/4

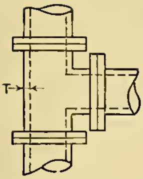
NOTE—The above dimensions are subject to a slight variation.

*Crane Co.

Table 27-00. Rules for Standard Weight Flanged Fittings

American 1915 Standard 125-lb. Working Pressure

Shell Thickness in Inches

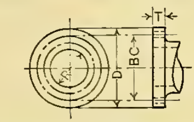


Size Fitting, Inches	Shell Thickness	Size Fitting, Inches	Shell Thickness	Size Fitting, Inches	Shell Thickness
2	7/16	5	½	12	⅞
2½	7/16	6	9/16	14	15/16
3	½	7	⅝	15	15/16
3½	½	8	11/16	16	15/16
4	½	9	¾	18	1
4½	½	10	13/16	20	1 1/16

1. Standard reducing elbows carry same dimensions center-to-face as regular elbows of largest straight size.
2. Standard tees, crosses and laterals, reducing on run only, carry same dimensions face-to-face as largest straight size.
3. Where long-turn fittings are specified, it has reference only to elbows which are made in two center-to-face dimensions and to be known as "elbows" and "long-turn elbows," the latter being used only when so specified.
4. All standard weight fittings must be guaranteed for 125-lb. working pressure, and each must have mark cast on indicating maker and guaranteed working steam pressure.
5. Standard weight fittings and flanges to be plain faced, and bolt holes to be ⅛ in. larger in diameter than bolts; bolt holes to straddle center lines.
6. Size of all fittings scheduled indicates inside diameter of ports.
7. Square head bolts with hexagonal nuts are generally recommended for use.
8. Double-branch elbows, side-outlet elbows and side-outlet tees, whether straight or reducing, carry same dimensions center-to-face and face-to-face as regular tees and elbows.
9. Bull-head tees or tees increasing on outlet, will have same center-to-face and face-to-face dimensions as a straight fitting of the size of the outlet.
10. Tees, crosses and laterals 16-in. and smaller, reducing on the outlet, use the same dimensions as straight sizes of the larger port.

Table 27-00. Standard Flanges and Bolts

1915 Standard, 125-lb. Working Pressure



PIPE	FLANGE		BOLTS		BOLT HOLES	
Size P	Diam. D	Thickness T	No.	Size Diam.	Bolt Circle B. C.	Size Diam.
1	4	7/16	4	7/16	3	9/16
1¼	4½	½	4	7/16	3⅜	9/16
1½	5	9/16	4	½	3⅞	⅝
2	6	⅝	4	⅝	4¾	¾
2½	7	11/16	4	⅝	5½	¾
3	7½	¾	4	⅝	6	¾
3½	8½	13/16	4	⅝	7	¾
4	9	15/16	8	⅝	7½	¾
4½	9¼	15/16	8	¾	7¾	⅞
5	10	15/16	8	¾	8½	⅞
6	11	1	8	¾	9½	⅞
7	12½	1 1/16	8	¾	10¾	⅞
8	13½	1⅛	8	¾	11¾	⅞
9	15	1⅛	12	¾	13¼	⅞
10	16	1 3/16	12	⅞	14¼	1
12	19	1¼	12	⅞	17	1
14	21	1⅜	12	1	18¾	1⅛
15	22¼	1⅜	16	1	20	1⅛
16	23½	1 7/16	16	1	21¼	1⅛
18	25	1 9/16	16	1⅛	22¾	1¼
20	27½	1 11/16	20	1⅛	25	1¼
22	29½	1 13/16	20	1¼	27¼	1⅜
24	32	1⅞	20	1¼	29½	1⅜
26	34¼	2	24	1¼	31¾	1⅜
28	36½	2 1/16	28	1¼	34	1⅜
30	38¾	2⅛	28	1⅜	36	1½
32	41¾	2¼	28	1½	38½	1⅝
34	43¾	2 5/16	32	1½	40½	1⅝
36	46	2⅜	32	1½	42¾	1⅝
38	48¾	2⅜	32	1⅝	45¼	1¾
40	50¾	2½	36	1⅝	47¼	1¾

Sizes 18-in. and larger, reducing on the outlet, are made in two lengths, depending on the size of the outlet, as given in the table of dimensions.

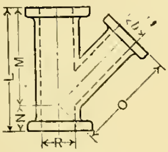
11. For fittings reducing on the run only, a long-body pattern will be used. Y's are special and made to suit connections. Double-branch elbows are not made reducing on the run.

12. Steel flanges, fittings and valves are recommended for super-heated steam.

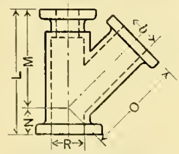
13. If flanged fittings for lower working pressure than 125 lb. are made, they shall conform in all dimensions, except thickness of shell, to this standard and shall have the guaranteed working pressure cast on each fitting. Flanges for these fittings must be standard dimensions.

Table 27-00. Standard Flanged Reducing Laterals

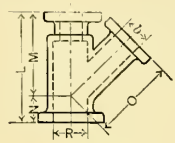
1915 Standard, 125-lb. Working Pressure



Reducing Lateral



Reducing-on-Run Lateral

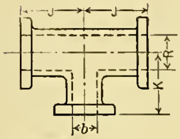


Reducing-on-Run and Branch Lateral

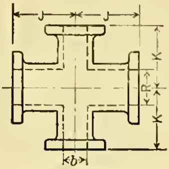
SIZE		DIMENSIONS, INCHES				FLANGES	
Run	Branch b	L	M	N	O	Diam.	Thickness
1	—	—	—	—	—	4	7/16
1¼	1¼ or less	8	6¼	1¾	6¼	4½	½
1½	1½ " "	9	7	2	7	5	9/16
2	2 " "	10½	8	2½	8	6	⅝
2½	2½ " "	12	9½	2½	9½	7	11/16
3	3 " "	13	10	3	10	7½	¾
3½	3½ " "	14½	11½	3	11½	8½	13/16
4	4 " "	15	12	3	12	9	15/16
4½	4½ " "	15½	12½	3	12½	9¼	15/16
5	5 " "	17	13½	3½	13½	10	15/16
6	6 " "	18	14½	3½	14½	11	1
7	7 " "	20½	16½	4	16½	12½	1 1/16
8	8 " "	22	17½	4½	17½	13½	1⅛
9	9 " "	24	19½	4½	19½	15	1⅛
10	10 " "	25½	20½	5	20½	16	1 3/16
12	12 " "	30	24½	5½	24½	19	1¼
14	14 " "	33	27	6	27	21	1⅜
15	15 " "	34½	28½	6	28½	22¼	1⅜
16	16 " "	36½	30	6½	30	23½	1 7/16
18	9 " "	26	25	1	27½	25	1 9/16
18	18 to 10 inc.	39	32	7	32	25	1 9/16
20	10 and less	28	27	1	29½	27½	1 11/16
20	20 to 12 inc.	43	35	8	35	27½	1 11/16
22	10 and less	29	28½	½	31½.	29½	1 13/16
22	22 to 12 inc.	46	37½	8½	37½	29½	1 13/16
24	12 and less	32	31½	½	34½	32	1⅞
24	24 to 14 inc.	49½	40½	9	40½	32	1⅞
26	12 and less	35	35	0	38	34¼	2
26	26 to 14 inc.	53	44	9	44	34¼	2
28	14 and less	37	37	0	40	36½	2 1/16
28	28 to 15 inc.	56	46½	9½	46½	36½	2 1/16
30	15 and less	39	39	0	42	38¾	2⅛
30	30 to 16 inc.	59	49	10	49	38¾	2⅛

Table 27-00. Standard Flanged Bull-Head Reducing Tees and Crosses

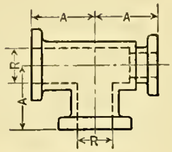
1915 Standard, 125-lb. Working Pressure



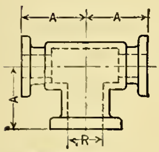
Reducing Tee



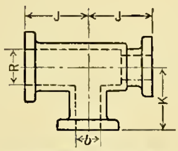
Reducing Cross



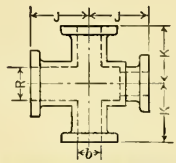
Reducing-on-Run Tee



Bullhead Tee



Reducing-on-Run and Branc Tee Cross



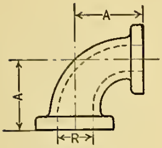
Reducing-on-Run Branch

SIZE		DIMENSIONS, INCHES					FLANGES	
R	Branch "B"	A & J	K	Branch "b"	J	K	Diam.	Thickness
1	—	—	—				4	$\frac{7}{16}$
1¼	1 or less	3¾	3¾				4½	½
1½	1¼ " "	4	4				5	$\frac{9}{16}$
2	1½ " "	4½	4½				6	⅝
2½	2 " "	5	5				7	$\frac{11}{16}$
3	2½ " "	5½	5½				7½	¾
3½	3 " "	6	6				8½	$\frac{13}{16}$
4	3½ " "	6½	6½				9	$\frac{15}{16}$
4½	4 " "	7	7				9¼	$\frac{15}{16}$
5	4½ " "	7½	7½				10	$\frac{15}{16}$
6	5 " "	8	8				11	1
7	6 " "	8½	8½				12½	$1\frac{1}{16}$
8	7 " "	9	9				13½	1⅛
9	8 " "	10	10				15	1⅛
10	9 " "	11	11				16	$1\frac{3}{16}$
12	10 " "	12	12				19	1¼
14	12 " "	14	14				21	1⅜
15	14 " "	14½	14½				22¼	1⅜
16	15 " "	15	15				23½	$1\frac{7}{16}$
18	18 to 14 inc.	16½	16½	12 or less	13	15½	25	$1\frac{9}{16}$
20	20 to 15 inc.	18	18	14 " "	14	17	27½	$1\frac{11}{16}$
22	22 to 16 inc.	20	20	15 " "	14	18	29½	$1\frac{13}{16}$
24	24 to 18 inc.	22	22	16 " "	15	19	32	1⅞
26	26 to 20 inc.	23	23	18 " "	16	20	34¼	2
28	28 to 20 inc.	24	24	18 " "	16	21	36½	$2\frac{1}{16}$
30	30 to 22 inc.	25	25	20 " "	18	23	38¾	2⅛
32	32 to 22 inc.	26	26	20 " "	18	24	41¾	2¼
34	34 to 24 inc.	27	27	22 " "	19	25	43¾	$2\frac{5}{16}$
36	36 to 26 inc.	28	28	24 " "	20	26	46	2⅜
38	38 to 26 inc.	29	29	24 " "	20	28	48¾	2⅜
40	40 to 28 inc.	30	30	26 " "	22	29	50¾	2½

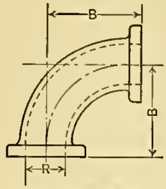
Note—A reduction in size *on the run* does not affect the dimensions but branch outlets of small size such as are listed below will reduce the dimensions of fittings 18 in. or over in size.

Table 27-00. Standard Flanged Elbows, Crosses, Laterals and Reducers

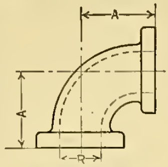
1915 Standard, 125-lb. Working Pressure



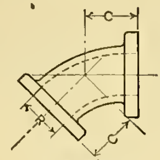
Elbow



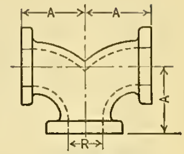
Long-Turn Elbow



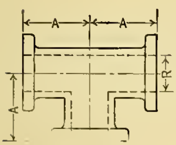
Reducing Elbow



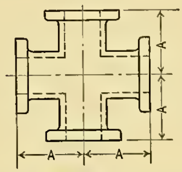
45° Elbow



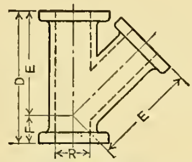
Double-Branch Elbow



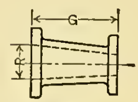
Straight Tee



Straight Cross



Straight Lateral



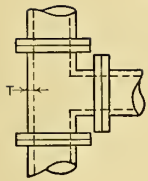
Reducer

SIZE	DIMENSIONS, INCHES						FLANGE	
Run	A	B	C	D	E	G	Diam.	Thickness
1	3½	5	1¾	7½	5¾	—	4	7/16
1¼	3¾	5½	2	8	6¼	—	4½	½
1½	4	6	2¼	9	7	—	5	9/16
2	4½	6½	2½	10½	8	—	6	⅝
2½	5	7	3	12	9½	—	7	11/16
3	5½	7¾	3	13	10	6	7½	¾
3½	6	8½	3½	14½	11½	6½	8½	13/16
4	6½	9	4	15	12	7	9	15/16
4½	7	9½	4	15½	12½	7½	9¼	15/16
5	7½	10¼	4½	17	13½	8	10	15/16
6	8	11½	5	18	14½	9	11	1
7	8½	12¾	5½	20½	16½	10	12½	1 1/16
8	9	14	5½	22	17½	11	13½	1⅛
9	10	15¼	6	24	19½	11½	15	1⅛
10	11	16½	6½	25½	20½	12	16	1 3/16
12	12	19	7½	30	24½	14	19	1¼
14	14	21½	7½	33	27	16	21	1⅜
15	14½	22¾	8	34½	28½	17	22¼	1⅜
16	15	24	8	36½	30	18	23½	1 7/16
18	16½	26½	8½	39	32	19	25	1 9/16
20	18	29	9½	43	35	20	27½	1 11/16
22	20	31½	10	46	37½	22	29½	1 13/16
24	22	34	11	49½	40½	24	32	1⅞
26	23	36½	13	53	44	26	34¼	2
28	24	39	14	56	46	28	36½	2 1/16
30	25	41½	15	59	49	30	38¾	2⅛
32	26	44	16	—	—	32	41¾	2¼
34	27	46½	17	—	—	34	43¾	2 5/16
36	28	49	18	—	—	36	46	2⅜
38	29	51½	19	—	—	38	48¾	2⅜
40	30	54	20	—	—	40	50¾	2½

Table 27-00. Rules for Extra-Heavy Flanged Fittings

American 1915 Standard 250-lb. Working Pressure

Shell Thickness in Inches



Size Fitting, Inches	Shell Thickness	Size Fitting, Inches	Shell Thickness	Size Fitting, Inches	Shell Thickness
2	⅝	5	¾	12	1⅛
2½	⅝	6	$\frac{13}{16}$	14	$1\frac{3}{16}$
3	⅝	7	⅞	15	1¼
3½	⅝	8	$\frac{15}{16}$	16	$1\frac{5}{16}$
4	⅝	9	1	18	$1\frac{7}{16}$
4½	$\frac{11}{16}$	10	$1\frac{1}{16}$	20	1½

1. Extra heavy reducing elbows carry same dimensions center to face as regular elbows of largest straight size.

2. Extra heavy tees, crosses and laterals, reducing on run only, carry same dimensions face-to-face as largest straight size.

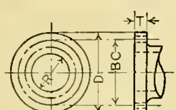
3. Where long turn fittings are specified, it has reference only to elbows which are made in two center-to-face dimensions and to be known as elbows and long-turn elbows, the latter being used only when so specified.

4. Extra heavy fittings must be guaranteed for 250-lb. working pressure, and each fitting must have some mark cast on it indicating the maker and guaranteed working steam pressure.

5. All extra heavy fittings and flanges to have a raised surface $\frac{1}{16}$ in. high inside of bolt holes for gaskets. Thickness of flanges and center-to-face dimensions of fittings include this raised surface. Bolt holes to be ⅛ in. larger in diameter than bolts. Bolt holes to straddle center lines.

Table 27-00. Extra-Heavy Pipe Flanges and Bolts

1915 Standard, 250-lb. Working Pressure



PIPE	FLANGE		BOLTS		BOLT HOLES	
Size P	Diam. D	Thickness T	No.	Size	Bolt Circle B. C.	Bolt Hole
1	4½	$\frac{11}{16}$	4	½	3¼	⅝
1¼	5	¾	4	½	3¾	⅝
1½	6	$\frac{13}{16}$	4	⅝	4½	¾
2	6½	⅞	4	⅝	5	¾
2½	7½	1	4	¾	5⅞	⅞
3	8¼	1⅛	8	¾	6⅝	⅞
3½	9	$1\frac{3}{16}$	8	¾	7¼	⅞
4	10	1¼	8	¾	7⅞	⅞
4½	10½	$1\frac{5}{16}$	8	¾	8½	⅞
5	11	1⅜	8	¾	9¼	⅞
6	12½	$1\frac{7}{16}$	12	¾	10⅝	⅞
7	14	1½	12	⅞	11⅞	1
8	15	1⅝	12	⅞	13	1
9	16¼	1¾	12	1	14	1⅛
10	17½	1⅞	16	1	15¼	1⅛
12	20½	2	16	1⅛	17¾	1¼
14	23	2⅛	20	1⅛	20¼	1¼
15	24½	$2\frac{3}{16}$	20	1¼	21½	1⅜
16	25½	2¼	20	1¼	22½	1⅜
18	28½	2⅜	24	1¼	24¾	1⅜
20	30½	2½	24	1⅜	27	1½
22	33	2⅝	24	1½	29¼	1⅝
24	36	2¾	24	1⅝	32	1¾
26	38¼	$2\frac{13}{16}$	28	1⅝	34½	1¾
28	40¾	$2\frac{15}{16}$	28	1⅝	37	1¾
30	43	3	28	1¾	39¼	1⅞
32	45¼	3⅛	28	1⅞	41½	2
34	47½	3¼	28	1⅞	43½	2
36	50	3⅜	32	1⅞	46	2
38	52¼	$3\frac{7}{16}$	32	1⅞	48	2
40	54½	$3\frac{9}{16}$	36	1⅞	50¼	2

6. Size of all fittings scheduled indicates inside diameter of ports.

7. Square head bolts with hexagonal nuts are generally recommended for use.

8. Double branch elbows, side outlet elbows and side outlet tees, whether straight or reducing sizes, carry same dimensions center-to-face and face-to-face as regular tees and elbows.

9. Bull head tees or tees increasing on outlet, will have same center-to-face and face-to-face dimensions as a straight fitting of the size of the outlet.

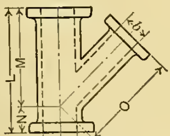
10. Tees, crosses and laterals 16-in. and smaller, reducing on the outlet, use the same dimensions as straight size of the larger port. Sizes 18 in. and larger, reducing on the outlet, are made in two lengths, depending on the size of the outlet as given in the table of dimensions.

11. For fittings reducing on the run only a long body pattern will be used. Y's are special and made to suit connections. Double branch elbows are not made reducing on the run.

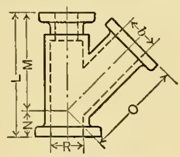
12. Steel flanges, fittings and valves are recommended for super-heated steam.

Table 27-00. Extra-Heavy Flanged Reducing Laterals

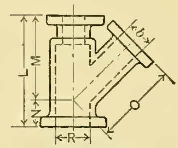
1915 Standard, 250-lb. Working Pressure



Reducing Lateral



Reducing-on-Run Lateral

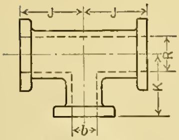


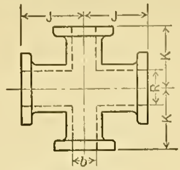
Reducing-on-Run and Branch Lateral

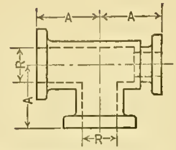
SIZE		DIMENSIONS, INCHES				FLANGES	
Run	Branch	L	M	N	O	Diam.	Thickness
1	—	—	—	—	—	4½	11/16
1¼	1¼ and less	9½	7¼	2¼	7¼	5	¾
1½	1½ " "	11	8½	2½	8½	6	13/16
2	2 " "	11½	9	2½	9	6½	⅞
2½	2½ " "	13	10½	2½	10½	7½	1
3	3 " "	14	11	3	11	8¼	1⅛
3½	3½ " "	15½	12½	3	12½	9	1 3/16
4	4 " "	16½	13½	3	13½	10	1¼
4½	4½ " "	18	14½	3½	14½	10½	1 5/16
5	5 " "	18½	15	3½	15	11	1⅜
6	6 " "	21½	17½	4	17½	12½	1 7/16
7	7 " "	23½	19	4½	19	14	1½
8	8 " "	25½	20½	5	20½	15	1⅝
9	9 " "	27½	22½	5	22½	16¼	1¾
10	10 " "	29½	24	5½	24	17½	1⅞
12	12 " "	33½	27½	6	27½	20½	2
14	14 " "	37½	31	6½	31	23	2⅛
15	15 " "	39½	33	6½	33	24½	2 3/16
16	16 " "	42	34½	7½	34½	25½	2¼
18	9 " "	34	31	3	32½	28	2⅜
18	16 to 10 inc.	45½	37½	8	37½	28	2⅜
20	10 and less	37	34	3	36	30½	2½
20	18 to 12 inc.	49	40½	8½	40½	30½	2½
22	10 and less	40	37	3	39	33	2⅝
22	20 to 12 inc.	53	43½	9½	43½	33	2⅝
24	12 and less	44	41	3	43	36	2¾
24	22 to 14 inc.	57½	47½	10	47½	36	2¾

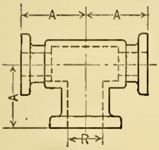
Table 27-00. Extra-Heavy Flanged Bull-Head Reducing Tees and Crosses

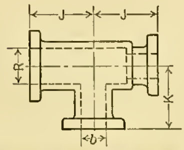
1915 Standard, 250-lb. Working Pressure

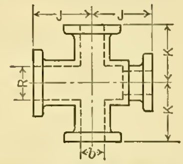

Reducing Tee


Reducing Cross


Reducing-on-Run Tee


Bullhead Tee


Reducing-on-Run and Branch Tee

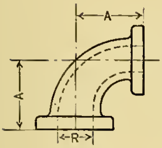

Reducing-on-Run aud Branch Cross

SIZE Run	Branch	DIMENSIONS, INCHES J	K	Branch	J	K	FLANGES Diam.	Thickness
1	—	—	—				4½	11/16
1¼	1 or less	4¼	4¼				5	¾
1½	1¼ " "	4½	4½				6	13/16
2	1½ " "	5	5				6½	⅞
2½	2 " "	5½	5½				7½	1
3	2½ " "	6	6				8¼	1⅛
3½	3 " "	6½	6½				9	1 3/16
4	3½ " "	7	7				10	1¼
4½	4 " "	7½	7½				10½	1 5/16
5	4½ " "	8	8				11	1⅜
6	5 " "	8½	8½				12½	1 7/16
7	6 " "	9	9				14	1½
8	7 " "	10	10				15	1⅝
9	8 " "	10½	10½				16¼	1¾
10	9 " "	11½	11½				17½	1⅞
12	10 " "	13	13				20½	2
14	12 " "	15	15				23	2⅛
15	14 " "	15½	15½				24½	2 3/16
16	15 " "	16½	16½	Branch	J	K	25½	2¼
18	18 to 14 inc.	18	18	12 or less	14	17	28	2⅜
20	20 to 15 inc.	19½	19½	14 " "	15½	18½	30½	2½
22	22 to 16 inc.	20½	20½	15 " "	16½	20	33	2⅝
24	24 to 18 inc.	22½	22½	16 " "	17	21½	36	2¾
26	26 to 20 inc.	24	24	18 " "	19	23	38¼	2 13/16
28	28 to 20 inc.	26	26	18 " "	19	24	40¾	2 15/16
30	30 to 22 inc.	27½	27½	20 " "	20½	25½	43	3
32	32 to 22 inc.	29	29	20 " "	20½	26½	45¼	3⅛
34	34 to 24 inc.	30½	30½	22 " "	22	28	47½	3¼
36	36 to 26 inc.	32½	32½	24 " "	23½	29½	50	3⅜
38	38 to 26 inc.	34	34	24 " "	23½	30½	52¼	3 7/16
40	40 to 28 inc.	35½	35½	26 " "	25	31½	54½	3 9/16

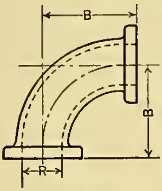
Note—A reduction in size *on the run* does not affect the dimensions but branch outlets of smaller size than those listed below will reduce the dimensions of fittings 18 in. or over in size.

Table 27-00. Extra-Heavy Flanged Elbows, Crosses, Laterals and Reducers

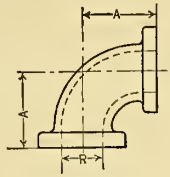
1915 Standard, 250-lb. Working Pressure



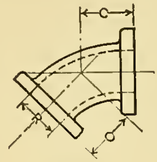
Elbow



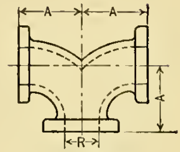
Long-Turn Elbow



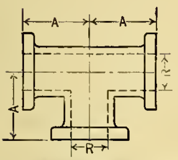
Reducing Elbow



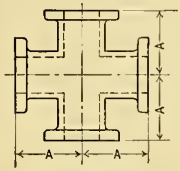
45° Elbow



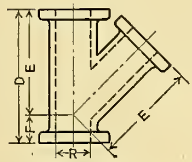
Double-Branch Elbow



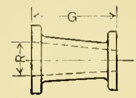
Straight Tee



Straight Cross



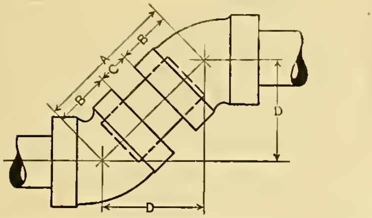
Straight Lateral



Reducer

SIZE Run	A	B	C	D	E	G	FLANGE Diam.	FLANGE Thickness
1	4	5	2	8½	6½	—	4½	11/16
1¼	4¼	5½	2½	9½	7¼	—	5	¾
1½	4½	6	2¾	11	8½	—	6	13/16
2	5	6½	3	11½	9	—	6½	⅞
2½	5½	7	3½	13	10½	—	7½	1
3	6	7¾	3½	14	11	6	8¼	1⅛
3½	6½	8½	4	15½	12½	6½	9	1 3/16
4	7	9	4½	16½	13½	7	10	1¼
4½	7½	9½	4½	18	14½	7½	10½	1 5/16
5	8	10¼	5	18½	15	8	11	1⅜
6	8½	11½	5½	21½	17½	9	12½	1 7/16
7	9	12¾	6	23½	19	10	14	1½
8	10	14	6	25½	20½	11	15	1⅝
9	10½	15¼	6½	27½	22½	11½	16¼	1¾
10	11½	16½	7	29½	24	12	17½	1⅞
12	13	19	8	33½	27½	14	20½	2
14	15	21½	8½	37½	31	16	23	2⅛
15	15½	22¾	9	39½	33	17	24½	2 3/16
16	16½	24	9½	42	34½	18	25½	2¼
18	18	26½	10	45½	37½	19	28	2⅜
20	19½	29	10½	49	40½	20	30½	2½
22	20½	31½	11	53	43½	22	33	2⅝
24	22½	34	12	57½	47½	24	36	2¾
26	24	36½	13	—	—	26	38¼	2 13/16
28	26	39	14	—	—	28	40¾	2 15/16
30	27½	41½	15	—	—	30	43	3
32	29	44	16	—	—	32	45¼	3⅛
34	30½	46½	17	—	—	34	47½	3¼
36	32½	49	18	—	—	36	50	3⅜
38	34	51½	19	—	—	38	52¼	3 7/16
40	35½	54	20	—	—	40	54½	3 9/16

Table 27-00. 45-Degree Offset Connections



Pipe Size	Centre to Centre A	Centre to Face B	Face to Face of 45's C	Offset D
½	2¼	⅞	½	1 19/32
¾	2½	1	½	1 25/32
1	2¾	1⅛	½	1 31/32
1¼	3⅛	1 5/16	½	2 7/32
1½	3⅜	1 7/16	½	2 13/32
2	3⅞	1 11/16	½	2¾
2½	4⅝	1 15/16	¾	3 9/32
3	5⅛	2 3/16	¾	3⅝
3½	5½	2⅜	¾	3 29/32
4	6	2⅝	¾	4¼
4½	6⅜	2 13/16	¾	4½
5	7⅛	3 1/16	1	5 1/16
6	7⅞	3 7/16	1	5 19/32
7	8¾	3⅞	1	6 7/32
8	10	4½	1	7 1/16

NOTE: The Offset "D" is equal to the distance "A" divided by 1.411.

Table 27-00. Conversion of Mercury and Vapor Pressures

Inches of Mercury to Pounds per Square Inch

Tenths	0	1	2	3	4	5	6	7	8	9
Inches	Lb. Sq. In	Lb. Sq. In.	Lb. Sq. In.	Lb. Sq. In.	Lb. Sq. In.	Lb. Sq. In.	Lb. Sq. In.	Lb. Sq. In.	Lb. Sq. In.	Lb. Sq. In.
0	0.	0.49	0.98	1.47	1.96	2.46	2.95	3.44	3.93	4.42
10	4.91	5.40	5.89	6.39	6.88	7.37	7.86	8.35	8.84	9.33
20	9.82	10.32	10.81	11.30	11.79	12.28	12.77	13.26	13.75	14.24
30	14.74	15.2	15.7	16.2	16.7	17.2	17.7	18.2	18.7	19.1
40	19.6	20.1	20.6	21.1	21.6	22.1	22.6	23.1	23.6	24.1
50	24.6	25.1	25.5	26.0	26.5	27.0	27.5	28.0	28.5	29.0
60	29.5	30.0	30.5	30.9	31.4	31.9	32.4	32.9	33.4	33.9
70	34.4	34.9	35.4	35.9	36.3	36.8	37.3	37.8	38.3	38.8
80	39.3	39.8	40.3	40.8	41.3	41.8	42.2	42.7	43.2	43.7
90	44.2	44.7	45.2	45.7	46.2	46.7	47.2	47.6	48.1	48.6
100	49.1	49.6	50.1	50.6	51.1	51.6	52.1	52.6	53.0	53.5

Pounds per Square Inch to Inches of Mercury

Pounds	In. Hg.	In. Hg.	In. Hg.	In. Hg.	In. Hg.	In. Hg.	In. Hg.	In. Hg.	In. Hg.	In. Hg.
0	0.	2.0352	4.0704	6.1056	8.1408	10.1760	12.2112	14.2464	16.2816	18.3168
10	20.352	22.3872	24.4224	26.4576	28.4928	30.528	32.5632	34.5984	36.6336	38.6688
20	40.704	42.7392	44.7744	46.8096	48.8448	50.8809	52.9152	54.9504	56.9856	59.0208
30	61.056	63.0912	65.1264	67.1616	69.1968	71.2320	73.2672	75.3024	77.3376	79.3728
40	81.408	83.4432	85.4784	87.5136	89.5488	91.5840	93.6192	95.6544	97.6896	99.7148
50	101.76	103.795	105.830	107.865	109.900	111.936	113.971	116.006	118.041	120.077
60	122.11	124.145	126.180	128.215	130.250	132.286	134.321	136.356	138.391	140.427
70	142.46	144.495	146.530	148.565	150.600	152.636	154.671	156.706	158.741	160.777
80	162.81	164.945	166.880	168.915	170.950	172.986	175.021	177.056	179.091	181.127
90	183.16	185.195	187.230	189.265	191.300	193.335	195.371	197.406	199.441	201.476
100	203.53	205.565	207.600	209.635	211.670	213.706	215.741	217.776	219.811	221.846

Table 27-00. Dimensions of Tubular Boilers *

Diameter of Boiler	Number of Tubes	Diameter of Tubes	Area Through Tubes in Square Feet	Square Feet Heating Surface in Heads	Square Feet Heating Surface in One Linear Foot Shell and Tubes	Horsepower in One Linear Foot Shell and Tubes	Length of Tubes	Horsepower	Rate of Evaporation 9 lb. Steam per lb. of Coal				Rate of Evaporation 8 lb. Steam per lb. of Coal			
									Size of Grate	Size of Uptake	Area Smoke-pipe Sq. In.	Coal per Sq. Ft. of Grate per Hr.	Size of Grate	Size of Uptake	Area Smoke-pipe Sq. In.	Coal per Sq. Ft. of Grate per Hr.
30	28	2½	0.78	2.8	20.7	1.37	6	8.5	......	......	....	....	24x36	10x14	140	6
							7	9.9	......	......	....	....	24x36	10x14	140	7
							8	11.2	......	......	....	....	24x36	10x14	140	8
							9	12.6	Use fig	ures in	last	four	24x42	10x14	140	8
							10	14.0	colum	ns for	30-i	nch	24x42	10x14	140	8½
36	34	2½	0.97	4.0	25.0	1.67	8	13.6	and 36	-inch b	oiler	s	30x36	10x16	160	8½
							9	15.3	......	......	....	....	30x42	10x18	180	8½
							10	16.9	......	......	....	....	30x42	10x18	180	9
							11	18.6	......	......	....	....	30x48	10x20	200	9
							12	20.9	......	......	....	....	30x48	10x20	200	9½
42	34	3	1.44	5.3	30.2	2.0	9	18.5	36x36	10x20	180	9	36x42	10x20	200	8½
							10	20.5	36x36	10x20	180	9½	36x42	10x20	200	9
							11	22.5	36x36	10x20	180	10½	36x48	10x25	250	9
							12	24.5	36x36	10x20	180	11½	36x48	10x25	250	9½
							13	26.5	36x42	10x22	200	11	36x48	10x25	250	10½
							14	28.5	36x42	10x22	200	12	36x54	10x28	280	10½
48	44	3	1.86	7.2	38.4	2.56	10	30.4	42x36	10x22	200	11	42x48	10x28	280	9
							11	33.2	42x36	10x22	200	12	42x48	10x28	280	10
							12	35.7	42x42	10x25	220	11½	42x54	10x32	320	10
							13	38.3	42x42	10x25	220	12	42x54	10x32	320	10½
							14	40.8	42x42	10x25	220	12½	42x60	10x36	360	10½
							15	43.4	42x48	10x28	250	12	42x60	10x36	360	11
							16	45.9	42x48	10x28	250	12½	42x60	10x36	360	11
54	54	3	2.28	9.3	46.4	3.10	11	34.6	48x42	10x28	250	9½	48x54	10x38	380	8½
							12	37.7	48x42	10x28	250	10	48x54	10x38	380	9
							13	40.8	48x42	10x28	250	11	48x54	10x38	380	10
							14	43.9	48x42	10x28	250	12	48x54	10x38	380	10½
							15	47.0	48x48	10x33	290	11½	48x60	10x40	400	10
							16	50.1	48x54	10x38	320	12	48x60	10x40	400	11
	46	3½	2.67	8.8	46.3	3.10	17	53.0	48x54	10x38	320	12½	48x60	10x40	400	11½
60	72	3	3.04	11.2	59.6	4.0	12	48.4	54x48	10x38	320	10	54x60	12x40	460	9½
							13	52.4	54x48	10x38	320	10½	54x60	12x40	460	10
							14	56.4	54x48	10x38	320	12	54x60	12x40	400	11
							15	60.4	54x54	12x35	370	12	54x66	12x42	500	10½
							16	64.4	54x54	12x35	370	12½	54x66	12x42	500	11½
	64	3½	3.71	10.5	62.4	4.16	17	71.4	54x60	12x40	400	12½	54x72	12x48	550	11½
							18	75.6	54x60	12x40	400	13½	54x72	12x48	550	12
66	90	3	3.80	13.5	74.2	4.95	14	70.1	60x54	12x40	400	12	60x66	12x48	500	11
							15	75.0	60x60	12x44	450	12	60x72	12x52	620	11
							16	80.0	60x60	12x44	450	12½	60x72	12x52	620	12
	78	3½	4.52	12.6	75.2	5.10	17	86.0	60x60	12x44	450	13	66x72	12x56	670	11½
							18	91.1	60x60	12x44	450	13½	66x72	12x56	670	12
							19	96.2	60x66	12x48	500	13	66x72	12x56	670	12½
	62	4	4.32	12.4	69.2	4.60	20	93.1	60x66	12x48	500	13	66x72	12x56	670	12
72	114	3	4.81	15.7	92.5	6.16	14	87.4	66x60	12x48	500	12	66x72	12x56	670	11½
							15	93.6	66x60	12x48	500	13	66x72	12x56	670	12
							16	99.7	66x60	12x48	500	14	72x72	12x62	740	12
	98	3½	5.75	15.0	93.0	6.20	17	106.4	66x66	12x52	540	13½	72x72	12x62	740	13
							18	112.6	66x66	12x52	540	14	72x72	12x62	790	13
							19	118.8	66x66	12x52	540	14½	72x72	12x62	790	14
	72	4	5.02	15.0	79.7	5.31	20	107.3	66x60	12x48	500	14½	72x72	12x62	740	13

* Hubbard's Steam Power Plants, Second Edition.

Table 27-00. Properties of Saturated Steam

Pressure, Lb. Absolute	Temperature, Deg. Fahr.	Specific Volume, Cu. Ft. per Lb.	Heat of the Liquid, B.t.u.	Latent Heat of Evap., B.t.u.	Total Heat of Steam, B.t.u.	Pressure, Lb. Absolute
1	101.83	333.0	69.8	1034.6	1104.4	1
2	126.15	173.5	94.0	1021.0	1115.0	2
3	141.52	118.5	109.4	1012.3	1121.6	3
4	153.01	90.5	120.9	1005.7	1126.5	4
5	162.28	73.33	130.1	1000.3	1130.5	5
6	170.06	61.89	137.9	995.8	1133.7	6
7	176.85	53.56	144.7	991.8	1136.5	7
8	182.86	47.27	150.8	988.2	1139.0	8
9	188.27	42.36	156.2	985.0	1141.1	9
10	193.22	38.38	161.1	982.0	1143.1	10
11	197.75	35.10	165.7	979.2	1144.9	11
12	201.96	32.36	169.9	976.6	1146.5	12
13	205.87	30.03	173.8	974.2	1148.0	13
14	209.55	28.02	177.5	971.9	1149.4	14
15	213.0	26.27	181.0	969.7	1150.7	15
16	216.3	24.79	184.4	967.6	1152.0	16
17	219.4	23.38	187.5	965.6	1153.1	17
18	222.4	22.16	190.5	963.7	1154.2	18
19	225.2	21.07	193.4	961.8	1155.2	19
20	228.0	20.08	196.1	960.0	1156.2	20
22	233.1	18.37	201.3	956.7	1158.0	22
24	237.8	16.93	206.1	953.5	1159.6	24
26	242.2	15.72	210.6	950.6	1161.2	26
28	246.4	14.67	214.8	947.8	1162.6	28
30	250.3	13.74	218.8	945.1	1163.9	30
32	254.1	12.93	222.6	942.5	1165.1	32
34	257.6	12.22	226.2	940.1	1166.3	34
36	261.0	11.58	229.6	937.7	1167.3	36
38	264.2	11.01	232.9	935.5	1168.4	38
40	267.3	10.49	236.1	933.3	1169.4	40
42	270.2	10.02	239.1	931.2	1170.3	42
44	273.1	9.59	242.0	929.2	1171.2	44
46	275.8	9.20	244.8	927.2	1172.0	46
48	278.5	8.84	247.5	925.3	1172.8	48
50	281.0	8.51	250.1	923.5	1173.6	50
52	283.5	8.20	252.6	921.7	1174.3	52
54	285.9	7.91	255.1	919.9	1175.0	54
56	288.2	7.65	257.5	918.2	1175.7	56
58	290.5	7.40	259.8	916.5	1176.4	58
60	292.7	7.17	262.1	914.9	1177.0	60
62	294.9	6.95	264.3	913.3	1177.6	62
64	297.0	6.75	266.4	911.8	1178.2	64
66	299.0	6.56	268.5	910.2	1178.8	66
68	301.0	6.38	270.6	908.7	1179.3	68
70	302.9	6.20	272.6	907.2	1179.8	70
72	304.8	6.04	274.5	905.8	1180.4	72
74	306.7	5.89	276.5	904.4	1180.9	74
76	308.5	5.74	278.3	903.0	1181.4	76
78	310.3	5.60	280.2	901.7	1181.8	78
80	312.0	5.47	282.0	900.3	1182.3	80

Table 27-00. Properties of Saturated Steam—*Continued*

Pressure, Lb. Absolute	Temperature, Deg. Fahr.	Specific Volume, Cu. Ft. per Lb.	Heat of the Liquid, B.t.u.	Latent Heat of Evap., B.t.u.	Total Heat of Steam, B.t.u.	Pressure, Lb. Absolute
82	313.8	5.34	283.8	899.0	1182.8	82
84	315.4	5.22	285.5	897.7	1183.2	84
86	317.1	5.10	287.2	896.4	1183.6	86
88	318.7	5.00	288.9	895.2	1184.0	88
90	320.3	4.89	290.5	893.9	1184.4	90
92	321.8	4.79	292.1	892.7	1184.8	92
94	323.4	4.69	293.7	891.5	1185.2	94
96	324.9	4.60	295.3	890.3	1185.6	96
98	326.4	4.51	296.8	889.2	1186.0	98
100	327.8	4.429	298.3	888.0	1186.3	100
105	331.4	4.230	302.0	885.2	1187.2	105
110	334.8	4.047	305.5	882.5	1188.0	110
115	338.1	3.880	309.0	879.8	1188.8	115
120	341.3	3.726	312.3	877.2	1189.6	120
125	344.4	3.583	315.5	874.7	1190.3	125
130	347.4	3.452	318.6	872.3	1191.0	130
135	350.3	3.331	321.7	869.9	1191.6	135
140	353.1	3.219	324.6	867.6	1192.2	140
145	355.8	3.112	327.4	865.4	1192.8	145
150	358.5	3.012	330.2	863.2	1193.4	150
155	361.0	2.920	332.9	861.0	1194.0	155
160	363.6	2.834	335.6	858.8	1194.5	160
165	366.0	2.753	338.2	856.8	1195.0	165
170	368.5	2.675	340.7	854.7	1195.4	170
175	370.8	2.602	343.2	852.7	1195.9	175
180	373.1	2.533	345.6	850.8	1196.4	180
185	375.4	2.468	348.0	848.8	1196.8	185
190	377.6	2.406	350.4	846.9	1197.3	190
195	379.8	2.346	352.7	845.0	1197.7	195
200	381.9	2.290	354.9	843.2	1198.1	200
205	384.0	2.237	357.1	841.4	1198.5	205
210	386.0	2.187	359.2	839.6	1198.8	210
215	388.0	2.138	361.4	837.9	1199.2	215
220	389.9	2.091	363.4	836.2	1199.6	220
225	391.9	2.046	365.5	834.4	1199.9	225
230	393.8	2.004	367.5	832.8	1200.2	230
235	395.6	1.964	369.4	831.1	1200.6	235
240	397.4	1.924	371.4	829.5	1200.9	240
245	399.3	1.887	373.3	827.9	1201.2	245
250	401.1	1.850	375.2	826.3	1201.5	250

Table 27-00. Horsepower of an Engine

a = Area of the piston in square inches. p = Mean effective pressure of the steam on the piston per square inch. v = Velocity of piston per minute.

$$\text{Then hp.} = \frac{a \times p \times v}{33{,}000}$$

The mean pressure in the cylinder when cutting off at

¼ stroke = boiler pressure multiplied by .597
⅓ " = " " " " .670
⅜ " = " " " " .743
½ " = " " " " .847
⅝ stroke = boiler pressure multiplied by .919
⅔ " = " " " " .937
¾ " = " " " " .966
⅞ " = " " " " .992

Table 27-00. Properties of Air

Temperature, Deg. Fahr.	Vol. of Dry Air with Unity at 32 Deg. Fahr.	Cubic Feet per Lb. of Air	Weight per Cu. Ft. of Dry Air	Elastic Force of Vapor In. of Mercury	Feet of Vapor from 1 Lb. of Water	B.t.u. Absorbed per Cu. Ft. of Air per Deg. Fahr.		Cu. Ft. of Air Raised 1 Deg. Fahr. by 1 B.t.u.	
						Dry Air	Sat. Air	Dry Air	Sat. Air
Zero	0.935	11.58	0.0864	0.044	......	0.02056	0.02054	48.5	48.7
12	0.960	11.87	0.0842	0.074	......	0.02004	0.02006	50.1	50.0
22	0.980	12.14	0.0824	0.118	......	0.01961	0.01963	51.1	51.0
32	1.000	12.40	0.0807	0.181	3289	0.01921	0.01924	52.0	51.8
42	1.020	12.64	0.0791	0.267	2252	0.01882	0.01884	53.2	52.8
52	1.041	12.88	0.0776	0.388	1595	0.01847	0.01848	54.0	53.8
60	1.057	12.39	0.0764	0.522	1227	0.01818	0.01822	55.0	54.6
62	1.061	13.13	0.0761	0.556	1135	0.01811	0.01812	55.2	54.7
70	1.078	13.34	0.0750	0.754	882	0.01777	0.01794	56.3	55.5
72	1.082	13.39	0.0747	0.785	819	0.01767	0.01790	56.5	55.8
82	1.102	13.64	0.0733	1.092	600	0.01744	0.01770	57.2	56.5
92	1.122	13.90	0.0720	1.501	444	0.01710	0.01751	58.5	57.1
100	1.139	13.95	0.0710	1.929	356	0.01690	0.01735	59.1	57.8
102	1.143	14.14	0.0707	2.036	334	0.01682	0.01731	59.5	57.8
112	1.163	14.40	0.0694	2.731	253	0.01651	0.01711	60.6	58.5
122	1.184	14.65	0.0682	3.621	194	0.01623	0.01691	61.7	59.1
132	1.204	14.90	0.0671	4.752	151	0.01596	0.01670	62.5	59.9
142	1.224	15.15	0.0660	6.165	118	0.01571	0.01652	63.7	60.6
152	1.245	15.40	0.0649	7.930	93.3	0.01544	0.01634	65.0	61.5
162	1.265	15.65	0.0638	10.099	74.5	0.01518	0.01616	66.2	62.4
172	1.285	15.90	0.0628	12.758	59.2	0.01494	0.01598	67.1	63.3
182	1.306	16.17	0.0618	15.960	48.6	0.01471	0.01580	68.0	64.2
192	1.326	16.42	0.0609	19.828	39.8	0.01449	....	68.9	....
202	1.347	16.67	0.0600	24.450	32.7	0.01426	....	69.5	....
212	1.367	16.92	0.0591	29.921	27.1	0.01406	....	71.4	....

Table 27-00. Volume and Weight of Air at Atmospheric Pressure at Temperatures Between 212 and 850 Deg. Fahr.

Temperature, Degrees Fahrenheit	Volume, One Pound in Cubic Feet	Weight One Cubic Foot in Pounds	Temperature, Degrees Fahrenheit	Volume, One Pound in Cubic Feet	Weight One Cubic Foot in Pounds	Weight One Cubic Foot in Pounds	Volume, One Pound in Cubic Feet	Temperature, Degrees Fahrenheit
212	16.925	.059084	320	19.647	.050898	550	25.444	.039302
220	17.127	.058388	340	20.151	.049625	575	26.074	.038352
230	17.379	.057541	360	20.655	.048414	600	26.704	.037448
240	17.631	.056718	380	21.159	.047261	650	27.964	.035760
250	17.883	.055919	400	21.663	.046162	700	29.224	.034219
260	18.135	.055142	425	22.293	.044857	750	30.484	.032804
270	18.387	.054386	450	22.923	.043624	800	31.744	.031502
280	18.639	.053651	475	23.554	.042456	850	33.004	.030299
290	18.891	.052935	500	24.184	.041350			
300	19.143	.052238	525	24.814	.040300			

Table 27-00. Heat Units Per Pound and Weight Per Cubic Foot of Water Between 32 Degrees Fahrenheit and 340 Degrees Fahrenheit*

Temperature, Degrees Fahr	Heat Units per Pound	Weight per Cubic Foot	Temperature, Degrees Fahr.	Heat Units per Pound	Weight per Cubic Foot	Temperature, Degrees Fahr	Heat Units per Pound	Weight per Cubic Foot	Temperature, Degrees Fahr.	Heat Units per Pound	Weight per Cubic Foot	Temperature, Degrees Fahr	Heat Units per Pound	Weight per Cubic Foot	Temperature, Degrees Fahr.	Heat Units per Pound	Weight per Cubic Foot
32	0.00	62.42	70	38.06	62.30	108	75.95	61.90	146	113.86	61.27	184	151.89	60.49	222	190.1	59.58
33	1.01	62.42	71	39.06	62.30	109	76.94	61.88	147	114.86	61.25	185	152.89	60.47	223	191.1	59.55
34	2.01	62.42	72	40.05	62.29	110	77.94	61.86	148	115.86	61.24	186	153.89	60.45	224	192.1	59.53
35	3.02	62.43	73	41.05	62.28	111	78.94	61.85	149	116.86	61.22	187	154.90	60.42	225	193.1	59.50
36	4.03	62.43	74	42.05	62.27	112	79.93	61.83	150	117.86	61.20	188	155.90	60.40	226	194.1	59.48
37	5.04	62.43	75	43.05	62.26	113	80.93	61.82	151	118.86	61.18	189	156.90	60.38	227	195.2	59.45
38	6.04	62.43	76	44.04	62.26	114	81.93	61.80	152	119.86	61.16	190	157.91	60.36	228	196.2	59.42
39	7.05	62.43	77	45.04	62.25	115	82.92	61.79	153	120.86	61.14	191	158.91	60.33	229	197.2	59.40
40	8.05	62.43	78	46.04	62.24	116	83.92	61.77	154	121.86	61.12	192	159.91	60.31	230	198.2	59.37
41	9.05	62.43	79	47.04	62.23	117	84.92	61.75	155	122.86	61.10	193	160.91	60.29	231	199.2	59.34
42	10.06	62.43	80	48.03	62.22	118	85.92	61.74	156	123.86	61.08	194	161.92	60.27	232	200.2	59.32
43	11.06	62.43	81	49.03	62.21	119	86.91	61.72	157	124.86	61.06	195	162.92	60.24	233	201.2	59.29
44	12.06	62.43	82	50.03	62.20	120	87.91	61.71	158	125.86	61.04	196	163.92	60.22	234	202.2	59.27
45	13.07	62.42	83	51.02	62.19	121	88.91	61.69	159	126.86	61.02	197	164.93	60.19	235	203.2	59.24
46	14.07	62.42	84	52.02	62.18	122	89.91	61.68	160	127.86	61.00	198	165.93	60.17	236	204.2	59.21
47	15.07	62.42	85	53.02	62.17	123	90.90	61.66	161	128.86	60.98	199	166.94	60.15	237	205.3	59.19
48	16.07	62.42	86	54.01	62.16	124	91.90	61.65	162	129.86	60.96	200	167.94	60.12	238	206.3	59.16
49	17.08	62.42	87	55.01	62.15	125	92.90	61.63	163	130.86	60.94	201	168.94	60.10	239	207.3	59.14
50	18.08	62.42	88	56.01	62.14	126	93.90	61.61	164	131.86	60.92	202	169.95	60.07	240	208.3	59.11
51	19.08	62.41	89	57.00	62.13	127	94.89	61.59	165	132.86	60.90	203	170.95	60.05	241	209.3	59.08
52	20.08	62.41	90	58.00	62.12	128	95.89	61.58	166	133.86	60.88	204	171.96	60.02	242	210.3	59.05
53	21.08	62.41	91	59.00	62.11	129	96.89	61.56	167	134.86	60.86	205	172.96	60.00	243	211.4	59.03
54	22.08	62.40	92	60.00	62.09	130	97.89	61.55	168	135.86	60.84	206	173.97	59.98	244	212.4	59.00
55	23.08	62.40	93	60.99	62.08	131	98.89	61.53	169	136.86	60.82	207	174.97	59.95	245	213.4	58.97
56	24.08	62.39	94	61.99	62.07	132	99.88	61.52	170	137.87	60.80	208	175.98	59.93	246	214.4	58.94
57	25.08	62.39	95	62.99	62.06	133	100.88	61.50	171	138.87	60.78	209	176.98	59.90	247	215.4	58.91
58	26.08	62.38	96	63.98	62.05	134	101.88	61.49	172	139.87	60.76	210	177.99	59.88	248	216.4	58.89
59	27.08	62.37	97	64.98	62.04	135	102.88	61.47	173	140.87	60.73	211	178.99	59.85	249	217.4	58.86
60	28.08	62.37	98	65.98	62.03	136	103.88	61.45	174	141.87	60.71	212	180.00	59.83	250	218.5	58.83
61	29.08	62.36	99	66.97	62.02	137	104.87	61.43	175	142.87	60.69	213	181.0	59.80	260	228.6	58.55
62	30.08	62.36	100	67.97	62.00	138	105.87	61.41	176	143.87	60.67	214	182.0	59.78	270	238.8	58.26
63	31.07	62.35	101	68.97	61.99	139	106.87	61.40	177	144.88	60.65	215	183.0	59.75	280	249.0	57.96
64	32.07	62.35	102	69.96	61.98	140	107.87	61.38	178	145.88	60.62	216	184.0	59.73	290	259.3	57.65
65	33.07	62.34	103	70.96	61.97	141	108.87	61.36	179	146.88	60.60	217	185.0	59.70	300	269.6	57.33
66	34.07	62.33	104	71.96	61.95	142	109.87	61.34	180	147.88	60.58	218	186.1	59.68	310	279.9	57.00
67	35.07	62.33	105	72.95	61.94	143	110.87	61.33	181	148.88	60.56	219	187.1	59.65	320	290.2	56.66
68	36.07	62.32	106	73.95	61.93	144	111.87	61.31	182	149.89	60.53	220	188.1	59.63	330	300.6	56.30
69	37.06	62.31	107	74.95	61.91	145	112.86	61.29	183	150.89	60.51	221	189.1	59.60	340	311.0	55.94

*Steam, Babcock & Wilcox Co.

Table 27-00. Weight of Water at Temperatures Used in Physical Calculations

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

Table 27-00. Volume and Weight of Distilled Water at Various Temperatures *

Temperature, Deg. Fahr.	Relative Volume Water at 39.2 Deg.=1	Weight in Lb. per Cubic Foot	Temperature, Deg. Fahr.	Relative Volume, Water at 39.2 Deg.=1	Weight in Lb. per Cubic Foot	Temperature, Deg. Fahr.	Relative Volume, Water at 39.2 Deg.=1	Weight in Lb. per Cubic Foot	Temperature, Deg. Fahr.	Relative Volume, Water at 39.2 Deg. =1	Weight in Lb. per Cubic Foot
32	1.000176	62.42	160	1.02337	61.00	290	1.0830	57.65	430	1.197	52.2
39.2	1.000000	62.43	170	1.02682	60.80	300	1.0890	57.33	440	1.208	51.7
40	1.000004	62.43	180	1.03047	60.58	310	1.0953	57.00	450	1.220	51.2
50	1.00027	62.42	190	1.03431	60.36	320	1.1019	56.66	460	1.232	50.7
60	1.00096	62.37	200	1.03835	60.12	330	1.1088	56.30	470	1.244	50.2
70	1.00201	62.30	210	1.04256	59.88	340	1.1160	55.91	480	1.256	49.7
80	1.00338	62.22	212	1.04343	59.83	350	1.1235	55.57	490	1.269	49.2
90	1.00504	62.11	220	1.0469	59.63	360	1.1313	55.18	500	1.283	48.7
100	1.00698	62.00	230	1.0515	59.37	370	1.1396	54.78	510	1.297	48.1
110	1.00915	61.86	240	1.0562	59.11	380	1.1483	54.36	520	1.312	47.6
120	1.01157	61.71	250	1.0611	58.83	390	1.1573	53.94	530	1.329	47.0
130	1.01420	61.55	260	1.0662	58.55	400	1.167	53.5	540	1.35	46.3
140	1.01705	61.38	270	1.0715	58.26	410	1.177	53.0	550	1.37	45.6
150	1.02011	61.20	280	1.0771	57.96	420	1.187	52.6	560	1.39	44.9

* Marks and Davis. (Steam, Babcock & Wilcox Co.)

Table 27-00. Boiling Point of Water at Various Altitudes

Boiling Point, Degrees Fahrenheit	Altitude Above Sea Level, Feet	Atmospheric Pressure, Pounds per Square Inch	Barometer Reduced to 32 Degrees, Inches	Boiling Point, Degrees Fahrenheit	Altitude Above Sea Level, Feet	Atmospheric Pressure, Pounds per Square Inch	Barometer Reduced to 32 Degrees, Inches
184	15221	8.20	16.70	199	6843	11.29	22.99
185	14649	8.38	17.06	200	6304	11.52	23.47
186	14075	8.57	17.45	201	5764	11.76	23.95
187	13498	8.76	17.83	202	5225	12.01	24.45
188	12934	8.95	18.22	203	4697	12.26	24.96
189	12367	9.14	18.61	204	4169	12.51	25.48
190	11799	9.34	19.02	205	3642	12.77	26.00
191	11243	9.54	19.43	206	3115	13.03	26.53
192	10685	9.74	19.85	207	2589	13.30	27.08
193	10127	9.95	20.27	208	2063	13.57	27.63
194	9579	10.17	20.71	209	1539	13.85	28.19
195	9031	10.39	21.15	210	1025	14.13	28.76
196	8481	10.61	21.60	211	512	14.41	29.33
197	7932	10.83	22.05	212	Sea Level	14.70	29.92
198	7381	11.06	22.52				

Table 27-00. Friction of Water in Pipes

Giving Velocity in Feet per Second, Friction Head in Feet and Friction Loss in Pounds per Square Inch for Each 100 Feet of Pipe Discharging a Given Quantity of Water in Gallons per Minute. (Weisbach Formula.)

Gallons per Minute	Vel. Ft. per Second	Friction Head in Feet	Friction Loss Lb. per Sq. In.	Vel. Ft. per Second	Friction Head in Feet	Friction Loss Lb. per Sq. In.	Vel. Ft. per Second	Friction Head in Feet	Friction Loss Lb. per Sq. In.	Vel. Ft. per Second	Friction Head in Feet	Friction Loss Lb. per Sq. In.	Vel. Ft. per Second	Friction Head in Feet	Friction Loss Lb. per Sq. In.	Vel. Ft. per Second	Friction Head in Feet	Friction Loss Lb. per. Sq. In.
	¾″ Pipe			1″ Pipe			1¼″ Pipe			1½″ Pipe			2″ Pipe			2½″ Pipe		
5	3.64	7.59	3.3	2.04	1.93	0.84	1.30	0.71	0.31	0.91	0.27	0.12	0.49	0.092	0.04	0.244	0.046	0.02
10	7.28	29.90	13.0	4.08	10.26	3.16	2.60	2.41	1.05	1.82	1.08	0.47	0.98	0.277	0.12	0.656	0.092	0.04
15	10.92	66.01	28.7	6.12	16.05	6.98	3.90	5.47	2.38	2.73	2.23	0.97	1.47	0.577	0.25	0.985	0.185	0.08
20	14.56	115.92	50.4	8.16	28.29	12.30	5.20	9.36	4.07	3.64	3.81	1.66	2.04	0.97	0.42	1.315	0.323	0.14
25	18.20	180.00	78.00	10.20	43.70	19.00	6.50	14.72	6.4	4.55	5.02	2.62	2.60	1.43	0.62	1.645	0.485	0.21
30				12.24	63.25	27.50	7.80	21.04	9.15	5.46	8.62	3.75	3.03	2.09	0.91	1.97	0.693	0.30
35				14.28	85.10	37.00	9.10	28.52	12.4	6.37	11.61	5.05	3.54	2.76	1.22	2.29	0.92	0.40
40				16.32	110.40	48.00	10.40	37.03	16.10	7.28	14.99	6.52	4.05	3.68	1.60	2.62	1.19	0.53
45							11.70	46.46	20.2	8.19	18.74	8.15	4.56	4.60	1.99	2.95	1.49	0.66
50							13.00	57.27	24.9	9.10	23.00	10.00	5.10	5.61	2.44	3.30	1.86	0.81
60							15.6	85.50	37.0	10.92	32.95	14.25	6.12	8.88	3.50	3.95	2.70	1.17
70							18.2	114.0	49.3	12.74	44.60	19.30	7.14	11.09	4.80	4.60	3.46	1.50
75							19.5	129.0	56.1	13.65	51.52	22.4	7.70	12.23	5.32	4.93	4.14	1.80
80										14.56	58.45	25.3	8.16	14.55	6.30	5.26	4.62	2.00
90										16.38	81.50	35.25	9.18	18.02	7.80	5.91	5.96	2.58
100										18.20	89.70	39.0	10.2	21.75	9.46	6.50	7.36	3.20
125													12.80	34.27	14.9	8.13	11.24	4.89
150													15.3	48.76	21.2	9.80	16.10	7.00
175																11.43	21.75	9.46
185																12.08	24.50	10.61
200																13.06	28.68	12.47

Gallons per Minute	Vel. Ft. per Second	Friction Head in Feet	Friction Loss Lb. per Sq. In.	Vel. Ft. per Second	Friction Head in Feet	Friction Loss Lb. per Sq. In.	Vel. Ft. per Second	Friction Head in Feet	Friction Loss Lb. per Sq. In.	Vel. Ft. per Second	Friction Head in Feet	Friction Loss Lb. per Sq. In.	Vel. Ft. per Second	Friction Head in Feet	Friction Loss Lb. per Sq. In.	Vel. Ft. per Second	Friction Head in Feet	Friction Loss Lb. per. Sq. In.
	3″ Pipe			3½″ Pipe			4″ Pipe			5″ Pipe			6″ Pipe			7″ Pipe		
10	0.448	0.046	0.02															
15	0.672	0.092	0.04	0.498	0.046	0.02												
20	0.896	0.138	0.06	0.664	0.069	0.03												
25	1.12	0.231	0.10	0.83	0.092	0.04												
30	1.345	0.30	0.13	0.996	0.138	0.06												
35	1.569	0.393	0.17	1.163	0.208	0.09												
40	1.790	0.53	0.23	1.329	0.254	0.11	1.04	0.138	0.06									
45	2.016	0.647	0.28	1.494	0.323	0.14	1.17	0.1615	0.07									
50	2.24	0.80	0.35	1.66	0.393	0.17	1.30	0.208	0.09									
60	2.688	1.155	0.50	1.992	0.555	0.24	1.56	0.30	0.13	0.88	0.1156	0.05						
70	3.136	1.385	0.60	2.324	0.879	0.38	1.82	0.439	0.19	1.04	0.162	0.07						
75	3.360	1.70	0.75	2.490	0.913	0.395	1.95	0.485	0.21	1.20	0.174	0.075						
80	3.584	2.08	0.90	2.656	0.948	0.41	2.08	0.580	0.23	1.28	0.185	0.08						
90	4.032	2.54	1.10	2.988	1.247	0.56	2.34	0.60	0.26	1.44	0.208	0.09						
100	4.480	3.01	1.31	3.320	1.478	0.64	2.60	0.763	0.33	1.60	0.277	0.12	1.14	0.115	0.05			
125	5.60	4.57	1.99	4.15	2.219	0.96	3.25	1.13	0.49	2.00	0.393	0.17	1.42	0.161	0.07			
150	5.80	6.55	2.85	4.98	3.12	1.35	3.80	1.59	0.69	2.40	0.578	0.25	1.71	0.231	0.10	1.20	0.093	0.04
175	7.92	8.85	3.85	5.81	4.208	1.82	4.45	2.146	0.93	2.80	0.785	0.34	2.00	0.302	0.13	1.38	0.115	0.05
185	8.34	9.94	4.30	6.14	4.62	2.00	4.70	2.484	1.075	2.96	0.84	0.36	2.11	0.36	0.156	1.55	0.13	0.056
200	9.04	11.54	5.02	6.64	5.50	2.38	5.1	2.82	1.22	3.20	0.972	0.42	2.28	0.39	0.17	1.70	0.162	0.07
250	11.28	17.84	7.76	8.30	8.55	3.70	6.4	4.37	1.89	4.00	1.50	0.65	2.80	0.60	0.26	2.10	0.277	0.12
265	12.40	20.09	8.72	8.80	9.60	4.15	6.79	6.45	2.09	4.24	1.69	0.73	3.03	0.70	0.303	2.23	0.31	0.134
300	13.52	25.76	11.20	9.96	11.63	5.04	7.60	6.15	2.66	4.80	2.15	0.93	3.40	0.85	0.37	2.40	0.393	0.17

Hot Water Averages 8 Lb. per Gallon

Table 27-00. Pressures Corresponding to Given Heads of Water in Feet

Water at maximum density. Temperature, 39.2 deg. fahr.
h=head in feet. P=pressure in lb. per sq. inch=.443 h.

H	P	H	P	H	P	H	P	H	P	H	P	H	P
1	.433	16	6.928	31	13.42	46	19.92	61	26.41	76	32.91	91	39.40
2	.866	17	7.361	32	13.86	47	20.35	62	26.85	77	33.34	92	39.84
3	1.299	18	7.794	33	14.29	48	20.78	63	27.28	78	33.77	93	40.27
4	1.732	19	8.227	34	14.72	49	21.22	64	27.71	79	34.21	94	40.70
5	2.165	20	8.660	35	15.15	50	21.65	65	28.14	80	34.64	95	41.13
6	2.598	21	9.09	36	15.59	51	22.08	66	28.58	81	35.07	96	41.57
7	3.031	22	9.53	37	16.02	52	22.52	67	29.01	82	35.51	97	42.00
8	3.464	23	9.96	38	16.45	53	22.95	68	29.44	83	35.94	98	42.43
9	3.897	24	10.39	39	16.89	54	23.38	69	29.88	84	36.37	99	42.87
10	4.330	25	10.82	40	17.32	55	23.81	70	30.31	85	36.80	100	43.30
11	4.763	26	11.26	41	17.75	56	24.25	71	30.74	86	37.24		
12	5.196	27	11.69	42	18.19	57	24.68	72	31.18	87	37.67		
13	5.629	28	12.12	43	18.62	58	25.11	73	31.61	88	38.10		
14	6.062	29	12.56	44	19.05	59	25.55	74	32.04	89	38.54		
15	6.495	30	12.99	45	19.48	60	25.98	75	32.47	90	38.97		

Table 27-00. Pressure, in Ounces Per Square Inch Corresponding to Various Heads of Water, in Inches*

Head in Inches	Decimal parts of an Inch .0	.1	.2	.3	.4	.5	.6	.7	.8	.9
0	....	.06	.12	.17	.23	.29	.35	.40	.46	.52
1	.58	.63	.69	.75	.81	.87	.93	.98	1.04	1.09
2	1.16	1.21	1.27	1.33	1.39	1.44	1.50	1.56	1.62	1.67
3	1.73	1.79	1.85	1.91	1.96	2.02	2.08	2.14	2.19	2.25
4	2.31	2.37	2.42	2.48	2.54	2.60	2.66	2.72	2.77	2.83
5	2.89	2.94	3.00	3.06	3.12	3.18	3.24	3.29	3.35	3.41
6	3.47	3.52	3.58	3.64	3.70	3.75	3.81	3.87	3.92	3.98
7	4.04	4.10	4.16	4.22	4.28	4.33	4.39	4.45	4.50	4.56
8	4.62	4.67	4.73	4.79	4.85	4.91	4.97	5.03	5.08	5.14
9	5.20	5.26	5.31	5.37	5.42	5.48	5.54	5.60	5.66	5.72

*Suplee's M. E. Reference Book

Table 27-00. Expansion of Wrought-Iron Pipe on the Application of Heat *

Temp. Air When Pipe is Fitted Deg. Fahr.	Increase in Length in Inches per Foot When Heated to 160	180	200	212	220	228	240	274
0	.0128	.0144	.016	.017	.0176	.0182	.0192	.0219
32	.0102	.0118	.0134	.0144	.015	.0157	.0166	.0194
50	.0088	.0104	.012	.013	.0136	.0142	.0152	.0179
70	.0072	.0088	.0104	.0114	.012	.0126	.0136	.0163

Co:— .0000067 per deg.

* Holland Heating Manual.

Table 27-00. Expansion and Weight of Water from 32 to 500 Deg. Fahr.

Temperature	Relative Volume by Expansion	Weight of One Cubic Foot	Weight of One Gallon	Temperature	Relative Volume by Expansion	Weight of One Cubic Foot	Weight of One Gallon	Temperature	Relative Volume by Expansion	Weight of One Cubic Foot	Weight of One Gallon
Deg. Fahr.		Pounds	Pounds	Deg. Fahr.		Pounds	Pounds	Deg. Fahr.		Pounds	Pounds
32	1.00000	62.418	10.0101	90	1.00459	62.133	9.964	170	1.02690	60.783	9.748
35	.99993	62.422	10.0103	95	1.00554	62.074	9.955	175	1.02906	60.665	9.728
39.1	.99989	62.425	10.0112	100	1.00639	62.022	9.947	180	1.03100	60.548	9.711
40	.99989	62.425	10.0112	105	1.00739	61.960	9.937	185	1.03300	60.430	9.691
45	.99993	62.422	10.0103	110	1.00889	61.868	9.922	190	1.03500	60.314	9.672
46	1.00000	62.418	10.0101	115	1.00989	61.807	9.913	195	1.03700	60.198	9.654
50	1.00015	62.409	10.0087	120	1.01139	61.715	9.897	200	1.03889	60.081	9.634
52.3	1.00029	62.400	10.0072	125	1.01239	61.654	9.887	205	1.0414	59.98	9.611
55	1.00038	62.394	10.0063	130	1.01390	61.563	9.873	210	1.0434	59.82	9.594
60	1.00074	62.372	10.0053	135	1.01539	61.472	9.859	212	1.0466	59.64	9.565
62	1.00101	62.355	10.0000	140	1.01690	61.381	9.844	250	1.06243	58.75	8.422
65	1.00119	62.344	9.9982	145	1.01839	61.291	9.829	300	1.09563	56.97	9.136
70	1.00160	62.313	9.9933	150	1.01989	61.201	9.815	400	1.15056	54.25	8.700
75	1.00239	62.275	9.9871	155	1.02164	61.096	9.799	500	1.22005	51.16	8.204
80	1.00299	62.232	9.980	160	1.02340	60.991	9.781				
85	1.00379	62.182	9.972	165	1.02589	60.843	9.757				

Table 27-00. Contents of Round Tanks in U. S. Gallons, for Each Foot in Depth

Diameter Ft.	Diameter In.	Gallons, 1 Foot in Depth	Diameter Ft.	Diameter In.	Gallons, 1 Foot in Depth	Diameter Ft.	Diameter In.	Gallons, 1 Foot in Depth	Diameter Ft.	Diameter In.	Gallons, 1 Foot in Depth
1	0	5.8735	7	0	287.8032	15	0	1321.5454	23	0	3107.1001
1	3	9.1766	7	3	308.7270	15	3	1365.9634	23	3	3175.0122
1	6	13.2150	7	6	330.3859	15	6	1407.5165	23	6	3243.6595
1	9	17.9870	7	9	352.7665	15	9	1457.0032	23	9	3313.0403
2	0	23.4940	8	0	375.9062	16	0	1503.6250	24	0	3383.1563
2	3	29.7340	8	3	399.7666	16	3	1550.9797	24	3	3454.0051
2	6	36.7092	8	6	424.3625	16	6	1599.0696	24	6	3525.5929
2	9	44.4179	8	9	449.2118	16	9	1647.8930	24	9	3597.9068
3	0	52.8618	11	0	710.6977	17	0	1697.4516	25	0	3670.9596
3	3	62.0386	11	3	743.3686	17	3	1747.7431	25	3	3744.7452
3	6	73.1504	11	6	776.7746	17	6	1798.7698	25	6	3819.2657
3	9	82.5959	11	9	810.9143	17	9	1850.5301	25	9	3894.5203
4	3	93.9754	12	0	848.1890	18	0	1903.0254	26	0	3970.5098
4	6	103.0300	12	3	881.3966	18	3	1956.2537	26	3	4047.2322
4	9	118.9386	12	6	917.7395	18	6	2010.2171	26	6	4124.6898
4	8	132.5209	12	9	954.8159	18	9	2064.9140	26	9	4202.9610
5	0	146.8384	13	0	992.6274	21	0	2590.2290	27	0	4281.8072
5	3	161.8886	13	3	1031.1719	21	3	2652.2532	27	3	4361.4664
5	6	177.6740	13	6	1070.4514	21	6	2715.0413	27	6	4441.8607
5	9	194.1913	13	9	1108.0645	21	9	2778.5486	27	9	4522.9886
6	0	211.4472	14	0	1151.2129	22	0	2842.7910	28	0	4604.8517
6	3	229.4342	14	3	1192.6940	22	3	2907.7664	28	3	4686.4876
6	6	218.1564	14	6	1234.9104	22	6	2973.4889	28	6	4770.7787
6	9	267.6122	14	9	1277.8615	22	9	3039.9209	28	9	4854.8434

Table 27-00. Cost of Water at Stated Rates per 1000 Gallons

Number of Cubic Feet	Cost per 1000 Gallons							
	5 Cents	6 Cents	8 Cents	10 Cents	15 Cents	20 Cents	25 Cents	30 Cents
20	$0.007	$0.009	$0.012	$0.015	$0.021	$0.030	$0.037	$0.045
40	0.015	0.018	0.024	0.030	0.045	0.060	0.075	0.090
60	0.022	0.027	0.036	0.045	0.066	0.090	0.112	0.135
80	0.030	0.036	0.048	0.060	0.090	0.120	0.150	0.180
100	0.037	0.049	0.060	0.075	0.111	0.150	0.187	0.224
200	0.075	0.090	0.120	0.150	0.225	0.299	0.374	0.449
300	0.112	0.135	0.180	0.224	0.336	0.449	0.561	0.673
400	0.150	0.180	0.239	0.299	0.450	0.598	0.748	0.898
500	0.188	0.224	0.299	0.374	0.564	0.748	0.935	1.122
600	0.224	0.269	0.359	0.449	0.448	0.898	1.122	1.346
700	0.262	0.314	0.419	0.524	0.786	1.047	1.309	1.571
800	0.299	0.350	0.479	0.598	0.897	1.197	1.496	1.795
900	0.337	0.404	0.539	0.673	1.011	1.346	1.683	2.020
1,000	0.374	0.449	0.598	0.748	1.122	1.496	1.870	2.244
2,000	0.748	0.898	1.197	1.496	2.244	2.992	3.740	4.488
3,000	1.122	1.346	1.795	2.244	3.366	4.488	5.610	6.732
4,000	1.496	1.795	2.393	2.992	4.488	5.984	7.480	8.976
5,000	1.870	2.244	2.992	3.740	5.610	7.480	9.350	11.220
6,000	2.244	2.692	3.590	4.488	6.732	8.976	11.220	13.464
7,000	2.618	3.141	4.189	5.236	7.854	10.472	13.090	15.708
8,000	2.992	3.590	4.787	5.984	8.976	11.968	14.961	17.953
9,000	3.366	4.039	5.385	6.732	10.098	13.464	16.831	20.197
10,000	3.74	4.488	5.984	7.480	11.122	14.961	18.701	22.441
20,000	7.48	8.976	11.968	14.961	22.443	29.992	37.402	44.882
30,000	11.22	13.46	17.95	22.44	33.664	44.88	56.10	67.32
40,000	14.96	17.95	23.94	29.92	44.885	59.84	74.80	89.77
50,000	18.70	22.44	29.92	37.40	56.103	74.80	93.50	112.20
60,000	22.44	26.92	35.90	44.88	67.323	89.76	112.20	134.64
70,000	26.18	31.41	41.89	52.36	78.543	104.72	130.90	157.08
80,000	29.92	35.90	47.87	59.84	89.766	119.68	149.61	179.53
90,000	33.66	40.39	53.85	67.32	100.986	134.64	168.31	201.97
100,000	37 40	44.88	59.84	74.80	111.22	149.61	187.01	224.41
200,000	74.81	89.76	119.68	149.61	224.43	299.22	374.02	448.82
300,000	112.20	134.64	179.53	224.41	336.64	448.83	561.03	673.24
400,000	149.61	179.53	239.37	299.22	448.85	598.44	748.05	897.66
500,000	187 01	224.41	299.22	374.02	561.03	748.05	935.06	1122.07
600,000	224.41	269.29	359.06	448.83	673.23	897.66	1122.07	1346.49
700,000	261.81	314.18	418.90	523.63	785.43	1047.27	1309.08	1570.88
800,000	299.22	359.06	478.75	598.44	897.66	1196.88	1496.10	1795.32
900,000	336.62	403.94	538.59	673.24	1009.86	1346.49	1683.11	2019.73
1,000,000	374.02	448.83	598.44	748.05	1122.06	1498.10	1870.12	2244.15

Table 27-00. Water Conversion Factors *

U. S. gallons	× 8.33	=pounds.	Cubic feet of water (39.1°)	× 62.425	=pounds.
U. S. gallons	× 0.13368	=cubic ft.	Cubic feet of water (39.1°)	× 7.48	=U.S. gal.
U. S. gallons	× 231.00000	=cubic in.	Cubic feet of water (39.1°)	× 0.028	=tons.
U. S. gallons	× 3.78	=liters.	Pounds of water	× 27.72	=cubic in.
Cubic inches of water (39.1°)	× 0.036024	=pounds.	Pounds of water	× 0.01602	=cubic ft.
Cubic inches of water (39.1°)	× 0.004329	=U.S. gal.	Pounds of water	× 0.12	=U.S. gal.
Cubic inches of water (39.1°)	× 0.576384	=ounces.			

* American Machinist Hand Book.

Table 27-00. Classification of Coals*

1 cubic foot of anthracite coal weighs 55 to 66 lb.
1 " " " bituminous " " 50 to 55 lb.
1 " " " semi-bituminous coal weighs 48 to 53 lb.

Name of Coal	Percentages of Combustible		B.t.u. per Pound of Combustible
	Fixed Carbon	Volatile Matter	
Anthracite	97.0 to 92.5	3.0 to 7.5	14,600 to 14,800
Semi-Anthracite	92.5 to 87.5	7.5 to 12.5	14,700 to 15,500
Semi-Bituminous	87.5 to 75.0	12.5 to 25.0	15,500 to 16,000
Bituminous, East	75.0 to 60.0	25.0 to 40.0	14,800 to 15,300
" West	65.0 to 50.0	35.0 to 50.0	13,500 to 14,800
Lignite	50.0 and under	50.0 and over	11,000 to 13,500

* Harding and Willard.

Table 27-00. Names and Sizes of Bituminous or "Soft" Coal*

For "Domestic" soft coals there are no uniform names and sizes, but they are marketed in the various states under about these classes:

"Screenings" usually smallest sizes.
"Duff" goes through ⅛-inch screen.
"No. 3 Nut" goes through 1¼-in. screen, over ¾-inch screen.
"No. 2 Nut" goes through 2-inch screen, over 1¼-inch screen.
"No. 1 Domestic Nut" goes through 3-inch screen, over 1½- or 2-inch screen.
"No. 4 Washed" goes through ¾-inch screen, over ¼-inch screen.
"No. 3 Washed Chestnut" goes through 1¼-inch screen, over ¾-inch screen.
"No. 2 Washed Stove" goes through 2-inch screen, over 1¼-inch screen.
"No. 1 Washed Egg" goes through 3-inch screen, over 2-inch screen.
"No. 3 Roller Screened Nut" goes through 1½-inch screen, over 1-inch screen.
"No. 2 Roller Screened Nut" goes through 2-inch screen, over 1½-inch screen.
"No. 1 Roller Screened Nut" goes through 3½-inch screen, over 2-inch screen.
"Egg" goes through 6-inch, over 3-inch screen.
"Lump" or "Block" goes through 6-inch screen, or over.
"Run-of-Mine" in fine and large lumps.
Pocahontas Smokeless: generally sized as: "Nut," "Egg," "Lump," and "Mine-Run."

* Harding and Willard.

Table 27-00. Heat Values of Bituminous Coals*

From selected free-burning and caking soft fuels taken from *U. S. Geological Survey* Bulletin No. 332, and *U. S. Bureau of Mines* Bulletin No. 23

State	Test No.	Kind of Fuel	County	B.t.u. per Lb. Dry Coal
Alabama	375	Soft—caking	Bibb	13,671
Alabama	484	Soft—free burning	Jefferson	14,447
Arkansas	293	Soft—caking	Sebastian	13,705
Arkansas	308	Semi-anthracite—caking	Johnson	14,125
Arkansas	340	Lignite	Quachita	9,549
Georgia	481	Soft—free burning	Chattooga	12,865
Illinois	418	Soft—free burning	Williamson	12,920
Illinois	511	Soft briquettes	St. Clair	13,271
Illinois	509	Soft—caking	Saline	13,621
Indiana	428	Soft—free burning	Greene	13,099
Indiana	435	Soft—caking	Pike	13,545
Indiana	464	Soft briquettes	Parke	11,930
Indian Territory	437	Soft—free burning		13,932
Indian Territory	449	Semi-anthracite		14,682
Kansas	311	Soft—free burning	Linn	12,343
Kentucky	434	Soft—free burning	Union	14,026

* Harding and Willard.

Table 27-00. Heat Values of Bituminous Coals*—*Continued*

From selected free-burning and caking soft fuels taken from *U. S. Geological Survey* Bulletin No. 332, and *U. S. Bureau of Mines* Bulletin No. 23

State	Test No.	Kind of Fuel	County	B.t.u. per Lb. Dry Coal
Maryland	490	Soft—free burning	Allegany	14,515
Maryland	518	Soft briquettes	Allegany	14,717
Missouri	319	Soft—caking	Randolph	11,747
Montana	477	Lignite—free burning	Carbon	11,628
New Mexico	392	Soft—caking	Colfax	13,059
New Mexico	387	Soft—free burning	Colfax	12,721
Ohio	483	Soft—free burning	Belmont	13,381
Pennsylvania	473	Soft—caking	Indiana	14,240
Pennsylvania	499	Soft—free burning	Cambria	14,119
Pennsylvania	514	Soft briquettes	Westmoreland	14,382
Tennessee	409	Soft briquettes	Claiborne	14,092
Tennessee	368	Soft—free burning	Campbell	14,008
Tennessee	363	Soft—caking	Grundy	13,257
Texas	291	Lignite—free burning	Wood	11,131
Utah	404	Soft—free burning	Summit	12,586
Virginia	482	Anthracite—free burning	Montgomery	12,679
Virginia	507	Soft—caking	Tazewell	14,177
Washington	290	Subbit—free burning	King	11,772
Washington	359	Soft—free burning	Kittitas	12,996
West Virginia	305	Soft—free burning	Marion	13,964
West Virginia	439	Soft—caking	Kanawha	13,995
Wyoming	399	Soft—free burning	Carbon	12,222
Wyoming	400	Subbit—free burning	Unita	12,488

NOTE—These values were obtained at the *St. Louis Testing Plant* from 139 samples of coal. The heating values of the various coals were established by "actually burning one gram of the air-dried coal in oxygen in a Mahler-bomb calorimeter." These values in B.t.u. give the theoretical maximum thermal value of soft coals.

*Harding & Willard.

Table 27-00. Names and Sizes of Anthracite or "Hard" Coal*

Names of Sizes	Will Pass Through	Will Not Pass Through
Buckwheat No. 1 / " No. 2	½-in. mesh	¼-in. mesh
or Rice	¼-in. mesh	⅛-in. mesh
Pea	¾-in. mesh	½-in. mesh
Chestnut, or Nut	1¼-in. mesh	¾-in. mesh
Stove or Range	1¾-in. mesh	1¼-in. mesh
Egg—in the East	2½-in. mesh	1¾-in. mesh
Large Egg—Chicago	4 -in. mesh	2¾-in. mesh
Small Egg—Chicago	2¾-in. mesh	2 -in. mesh
Broken, or Grate	4 -in. mesh	2½-in. mesh

*Harding and Willard.

Table 27-00. Composition and Heat Values of Anthracite Coals*

Locality	Fixed Carbon	Volatile	Moisture	Ash	Sulphur	B.t.u. per Lb. of Dry Coal
Anthracite						
Pennsylvania	78.60	....	....	14.80	0.40	.....
Buckwheat	81.32	3.84	3.88	10.96	0.67	12,200
Wilkesbarre	76.94	6.42	1.34	15.30	....	11,801
Scranton	79.23	3.73	3.33	13.70	....	12,149
Scranton	84.46	5.37	0.97	9.20	....	12,294
Cross Creek	89.19	1.96	3.62	5.23	....	13,723
Lehigh Valley	75.20	7.36	1.44	16.00	....	12,423
Lykens Valley	76.94	6.21	....	....	....	15,300
Lykens Valley	81.00	5.00	....	....	....	15,300
Wharton	86.40	3.08	3.71	6.22	0.58	15,000
Buck Mt	82.66	3.95	3.04	9.88	0.46	15,070
Beaver Meadow	88.94	2.38	1.50	7.11	0.01	.....
Lackawanna	87.74	3.91	2.12	6.35	0.12	.....
Rhode Island	85.00	....	....	7.00	0.90	.....
Arkansas	74.49	14.73	1.52	9.26	....	13,217
Semi-Anthracite						
Pennsylvania, Loyalsock	83.34	8.10	1.30	6.23	1.03	15,400
Bernice	82.52	3.56	0.96	3.27	0.24	15,050
Bernice	89.39	8.56	0.97	9.34	1.04	15,475
Wilkes-Barre	88.90	7.68	....	3.49	....	14,199
Lycoming Creek	71.53	13.84	0.67	13.96	0.03	.....
Virginia, Natural Coke	75.08	12.44	1.12	11.38	0.47	.....
Arkansas	74.06	14.93	1.35	9.66	....	.....
Indian Territory	73.21	13.65	5.11	8.03	1.18	13,662
Maryland, Easby	83.60	16.40	....	....	....	11,207

*Harding & Willard

Table 27-00. Weight of Materials

Dry Woods

Material	Weight in Lb. of One Cu. Ft.	Material	Weight in Lb. of One Cu. Ft.	Material	Weight in Lb. of One Cu. Ft.
Ash	43–53	Fir, Spruce	30–44	Oak—American red	54
Beech	43–53	Greenheart	70	" English	48–58
Birch	40–46	Hornbeam	47	Pine—red	30–44
Boxwood	57–83	Larch	31–37	" white	27–34
Cork	15	Lignum-vitae	83	" yellow	29–41
Ebony	70–83	Mahogany—Honduras	35	Teak	41–55
Elm	34–45	" Spanish	53		

Stones, Earth, Etc.

Material	Weight in Lb. of One Cu. Ft.	Material	Weight in Lb. of One Cu. Ft.	Material	Weight in Lb. of One Cu. Ft.
Asphaltum	64–112	Glass—flint	187	Mud—dry and close	80–110
Brick—common	100–125	" plate	169	" wet and fluid	104–120
" fire	137–150	Granite	164–175	Sand—dry	88–110
Cement—Portland	80– 90	Gravel	90–125	" wet	118–129
Clay	120	Grindstone	134	Sandstone	130–170
Concrete	120–140	Lime—quick	52	Victoria stone (crushed granite. Portland cement, silica)	144
Earth	77–120	Limestone and marbles	150–179		
Glass—crown	156	Mortar—hardened	88–118		

Table 27-00. Weight of Materials—*Continued*

Metals and Alloys

MATERIAL		Specific Gravity	Weight in Lb. of One Cu. Ft.	Weight in Lb. of One Cu. In.	Cu. In. in One Lb.
Aluminum—cast		2.569	160	.093	10.80
" wrought		2.681	167	.097	10.35
" bronze		7.787	485	.281	3.56
Antimony		6.712	418	.242	4.13
Arsenic		5.748	358	.207	4.83
Bismuth		9.827	612	.354	2.82
Brass—cast	from	7.868	490	.284	3.53
	to	8.430	525	.304	3.29
	average	8.109	505	.292	3.42
" Muntz—metal		8.221	512	.296	3.37
" naval (rolled)		8.510	530	.307	3.26
" sheet		8.462	527	.305	3.28
" wire		8.558	533	.308	3.24
Bronze (gun-metal)	from	8.478	528	.306	3.27
	to	8.863	552	.319	3.13
	average	8.735	544	.315	3.18
Copper—cast		8.622	537	.311	3.22
" hammered		8.927	556	.322	3.11
" sheet		8.815	549	.318	3.15
" wire		8.895	554	.321	3.12
Gold (pure)		19.316	1203	.696	1.44
" standard 22 carat fine		17.502	1090	.631	1.59
(Gold 11—Copper 1)					
Iron—cast	from	6.904	430	.249	4.02
	to	7.386	499	.266	3.76
	average	7.209	464	.260	3.85
Iron—wrought	from	7.547	470	.272	3.56
	to	7.803	486	.281	3.68
	average	7.707	480	.278	3.60
Lead—cast		11.368	708	.410	2.44
" sheet		11.432	712	.412	2.43
Manganese		8.012	499	.289	3.46
Nickel—cast		8.285	516	.299	3.35
" rolled		8.687	541	.313	3.19
Platinum		21.516	1340	.775	1.29
Silver		10.517	655	.379	2.64
Steel	from	7.820	487	.282	3.55
	to	7.916	493	.285	3.51
	average	7.868	490	.284	3.53
Tin		7.418	462	.267	3.74
White Metal (Babbitt's)		7.322	456	.264	3.79
Zinc—cast		6.872	428	.248	4.04
" sheet		7.209	449	.260	3.85

Table 27-00. Specific Heat and Densities of Building Materials†

Building Materials	Specific Heat	Building Materials	Specific Heat	Densities	Lb. per 1 Cu. Ft.
Brickwork	0.1950	Oakwood	0.5700	Stonework	160
Masonry	.2159	Birch	.4800	Wood	40
Plaster	.2000	Glass	.1977	Slate	170
Pinewood	.4670			Plaster	90

† Harding and Willard.

Table 27-00. Specific Heats of Various Substances

Solids

	Temperature,* Degrees Fahrenheit	Specific Heat		Temperature,* Degrees Fahrenheit	Specific Heat
Copper	59–460	0.0951	Glass (normal ther. 16^{III})	66–212	0.1988
Gold	32–212	.0316	Lead	59	.0299
Wrought iron	59–212	.1152	Platinum	32–212	.0323
Cast Iron	68–212	.1189	Silver	32–212	.0559
Steel (soft)	68–208	.1175	Tin	105–64	.0518
Steel (hard)	68–208	.1165	Ice	......	.5040
Zinc	32–212	.0935	Sulphur (newly fused)	......	.2025
Brass (yellow)	32	.0883			

Liquids

	Temperature,* Degrees Fahrenheit	Specific Heat		Temperature,* Degrees Fahrenheit	Specific Heat
Water	59	1.0000	Sulphur (melted)	246–297	0.2350
Alcohol	32	0.5475	Tin (melted)	......	.637
	176	.7694	Sea-water (sp.gr. 1.0043)	64	.980
Mercury	32	.3346	Sea-water (sp.gr. 1.0463)	64	.903
Benzol	50	.4066	Oil of turpentine	32	.411
	122	.4502	Petroleum	64–210	.498
Glycerine	59–102	......	Sulphuric acid	68–133	.3363
Lead (melted)	to 360	.0410	Olive oil	......	.309

Gases

	Temperature,* Degrees Fahrenheit	Specific Heat at Constant Pressure	Specific Heat at Constant Volume		Temperature,* Degrees Fahrenheit	Specific Heat at Constant Pressure	Specific Heat at Constant Volume
Air	32–392	0.2375	0.1693	Carbon monoxide	41–208	0.2425	0.1728
Oxygen	55–405	.2175	.1553	Carbon dioxide	52–417	.2169	.1535
Nitrogen	32–392	.2438	.1729	Methane	64–406	.5929	.4505
Hydrogen	54–388	3.4090	2.4141	Blast-Fur. gas (approx.)		.2277	......
				Flue gas (approx.)	......	.2400	......

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

Table 27-00. Tensile Strength of Materials

Average Value in Pounds per Square Inch

Antimony	1053
Aluminum—castings	15000
" sheet	24000
" bars	28000
Brass—yellow	26880
Bronze—cast	34000
" delta metal—cast	44800
" " " rolled	67200
" gun metal	32000
" phosphor	40000
" manganese	62720
" Tobin	78500
Copper—cast	22400
" sheet	30240
" wire	40000
Cast Steel	80000

Gold	20384
Iron—cast	25000
" "	18000
" wrought	45000
Lead—cast	1800
" rolled sheet	3320
Platinum wire	53000
"Puddled" Semi-steel	35000 to 42000
Silver—cast	40000
Steel—cast	60000 to 80000
" forgings	60000 to 95000
Tin—cast	3360
Zinc—cast	3360
" sheet	15680

Woods

Ash	11000 to 17000
Beech	11500 to 18000
Cedar	10300 to 11400
Chestnut	10500
Elm	13000 to 13489
Hemlock	8700
Hickory	12800 to 18000
Locust	20500 to 24800
Maple	10500 to 10584
Oak—white	10253 to 19500
Pine—white	10000 to 12000
" yellow	12600 to 19200
Spruce	10000 to 19500
Walnut-black	9286 to 16000

Table 27-00. Lineal Expansion of Solids at Ordinary Temperatures

(Tabular Values Represent Increase per Foot per 100 Degrees Increase in Temperature, Fahrenheit or Centigrade)

Substance	Temperature Conditions* Degrees Fahrenheit	Coefficient per 100 Degrees Fahrenheit	Coefficient per 100 Degrees Centigrade
Brass (cast)	32 to 212	.001042	.001875
Brass (wire)	32 to 212	.001072	.001930
Copper	32 to 212	.000926	.001666
Glass (English flint)	32 to 212	.000451	.000812
Glass (French flint)	32 to 212	.000484	.000872
Gold	32 to 212	.000816	.001470
Granite (average)	32 to 212	.000482	.000868
Iron (cast)	104	.000589	.001061
Iron (soft forged)	0 to 212	.000634	.001141
Iron (wire)	32 to 212	.000800	.001440
Lead	32 to 212	.001505	.002709
Mercury	32 to 212	.009984†	.017971†
Platinum	104	.000499	.000899
Limestone	32 to 212	.000139	.000251
Silver	104	.001067	.001921
Steel (Bessemer rolled, hard)	0 to 212	.00056	.00101
Steel (Bessemer rolled, soft)	0 to 212	.00063	.00117
Steel (cast, French)	104	.000734	.001322
Steel (cast annealed, English)	104	.000608	.001095

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

Table 27-00. Decimal Equivalents of Fractions of an Inch

Fractions				Decimals	Fractions				Decimals	Fractions				Decimals
1/64	..	..	..	.015625	23/64	..	..	..	.359375	45/64	..	..	..	.703125
..	1/32	..	..	.03125	..	..	..	3/8	.375	..	23/32	..	..	.71875
3/64	..	..	..	.046875	25/64	..	..	..	.390625	47/64	..	..	..	.734375
..	..	1/16	..	.0625	..	13/32	..	..	.40625	..	..	..	3/4	.75
5/64	..	..	..	.078125	27/64	..	..	..	.421875	49/64	..	..	..	.765625
..	3/32	..	..	.09375	..	..	7/16	..	.4375	..	25/32	..	..	.78125
7/64	..	..	..	.109375	29/64	..	..	..	.453125	51/64	..	..	..	.796875
..	..	..	1/8	.125	..	15/32	..	..	.46875	..	..	13/16	..	.8125
9/64	..	..	..	.140625	31/64	..	..	..	.484375	53/64	..	..	..	.828125
..	5/32	..	..	.15625	..	..	..	1/2	.5	..	27/32	..	..	.84375
11/64	..	..	..	.171875	33/64	..	..	..	.515625	55/64	..	..	..	.859375
..	..	3/16	..	.1875	..	17/32	..	..	.53125	..	..	..	7/8	.875
13/64	..	..	..	.203125	35/64	..	..	..	.546875	57/64	..	..	..	.890625
..	7/32	..	..	.21875	..	..	9/16	..	.5625	..	29/32	..	..	.90625
15/64	..	..	..	.234375	37/64	..	..	..	.578125	59/64	..	..	..	.921875
..	..	..	1/4	.25	..	19/32	..	..	.59375	..	..	15/16	..	.9375
17/64	..	..	..	.265625	39/64	..	..	..	.609375	61/64	..	..	..	.953125
..	9/32	..	..	.28125	..	..	..	5/8	.625	..	31/32	..	..	.96875
19/64	..	..	..	.296875	41/64	..	..	..	.640625	63/64	..	..	..	.984375
..	..	5/16	..	.3125	..	21/32	..	..	.65625	..	..	..	1	1.00
21/64	..	..	..	.328125	43/64	..	..	..	.671875					
..	11/32	..	..	.34375	..	..	11/16	..	.6875					

Table 27-00. Decimals of a Foot for Inches and Fractions of an Inch

Inch	0″	1″	2″	3″	4″	5″	6″	7″	8″	9″	10″	11″
0	0	.0833	.1667	.2500	.3333	.4167	.5000	.5833	.6667	.7500	.8333	.9167
1/32	.0026	.0859	.1693	.2526	.3359	.4193	.5026	.5859	.6693	.7526	.8359	.9193
1/16	.0052	.0885	.1719	.2552	.3385	.4219	.5052	.5885	.6719	.7552	.8385	.9219
3/32	.0078	.0911	.1745	.2578	.3411	.4245	.5078	.5911	.6745	.7578	.8411	.9245
1/8	.0104	.0937	.1771	.2604	.3437	.4271	.5104	.5937	.6771	.7604	.8437	.9271
5/32	.0130	.0964	.1797	.2630	.3464	.4297	.5130	.5964	.6797	.7630	.8464	.9297
3/16	.0156	.0990	.1823	.2656	.3490	.4323	.5156	.5990	.6823	.7656	.8490	.9323
7/32	.0182	.1016	.1849	.2682	.3516	.4349	.5182	.6016	.6849	.7682	.8516	.9349
1/4	.0208	.1042	.1875	.2708	.3542	.4375	.5208	.6042	.6875	.7708	.8542	.9375
9/32	.0234	.1068	.1901	.2734	.3568	.4401	.5234	.6068	.6901	.7734	.8568	.9401
5/16	.0260	.1094	.1927	.2760	.3594	.4427	.5260	.6094	.6927	.7760	.8594	.9427
11/32	.0286	.1120	.1953	.2786	.3620	.4453	.5286	.6120	.6953	.7786	.8620	.9453
3/8	.0312	.1146	.1979	.2812	.3646	.4479	.5312	.6146	.6979	.7812	.8646	.9479
13/32	.0339	.1172	.2005	.2839	.3672	.4505	.5339	.6172	.7005	.7839	.8672	.9505
7/16	.0365	.1198	.2031	.2865	.3698	.4531	.5365	.6198	.7031	.7865	.8698	.9531
15/32	.0391	.1224	.2057	.2891	.3724	.4557	.5391	.6224	.7057	.7891	.8724	.9557
1/2	.0417	.1250	.2083	.2917	.3750	.4583	.5417	.6250	.7083	.7917	.8750	.9583
17/32	.0443	.1276	.2109	.2943	.3776	.4609	.5443	.6276	.7109	.7943	.8776	.9609
9/16	.0469	.1302	.2135	.2969	.3802	.4635	.5469	.6302	.7135	.7969	.8802	.9635
19/32	.0495	.1328	.2161	.2995	.3828	.4661	.5495	.6328	.7161	.7995	.8828	.9661
5/8	.0521	.1354	.2188	.3021	.3854	.4688	.5521	.6354	.7188	.8021	.8854	.9688
21/32	.0547	.1380	.2214	.3047	.3880	.4714	.5547	.6380	.7214	.8047	.8880	.9714
11/16	.0573	.1406	.2240	.3073	.3906	.4740	.5573	.6406	.7240	.8073	.8906	.9740
23/32	.0599	.1432	.2266	.3099	.3932	.4766	.5599	.6432	.7266	.8099	.8932	.9766
3/4	.0625	.1458	.2292	.3125	.3958	.4792	.5625	.6458	.7292	.8125	.8958	.9792
25/32	.0651	.1484	.2318	.3151	.3984	.4818	.5651	.6484	.7318	.8151	.8984	.9818
13/16	.0677	.1510	.2344	.3177	.4010	.4844	.5677	.6510	.7344	.8177	.9010	.9844
27/32	.0703	.1536	.2370	.3203	.4036	.4870	.5703	.6536	.7370	.8203	.9036	.9870
7/8	.0729	.1562	.2396	.3229	.4062	.4896	.5729	.6562	.7396	.8229	.9062	.9896
29/32	.0755	.1589	.2422	.3255	.4089	.4922	.5755	.6589	.7422	.8255	.9089	.9922
15/16	.0781	.1615	.2448	.3281	.4115	.4948	.5781	.6615	.7448	.8281	.9115	.9948
31/32	.0807	.1641	.2474	.3307	.4141	.4974	.5807	.6641	.7474	.8307	.9141	.9974
1	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1.0000

Table 27-00. Decimals of a Foot Equivalent to Inches and Fractions of an Inch

Inches	0″	1/8″	1/4″	3/8″	1/2″	5/8″	3/4″	7/8″
0	0	.01042	.02083	.03125	.04166	.05208	.06250	.07292
1	.0833	.0937	.1042	.1146	.1250	.1354	.1459	.1563
2	.1667	.1771	.1875	.1979	.2083	.2188	.2292	.2396
3	.2500	.2604	.2708	.2813	.2917	.3021	.3125	.3229
4	.3333	.3437	.3542	.3646	.3750	3854	.3958	.4063
5	.4167	.4271	.4375	.4479	.4583	.4688	.4792	.4896
6	.5000	.5104	.5208	.5313	.5417	.5521	.5625	.5729
7	.5833	.5937	.6042	.6146	.6250	.6354	.6459	.6563
8	.6667	.6771	.6875	.6979	.7083	.7188	.7292	.7396
9	.7500	.7604	.7708	.7813	.7917	.8021	.8125	.8229
10	.8333	.8437	.8542	.8646	.8750	.8854	.8958	.9063
11	.9167	.9271	.9375	.9479	.9583	.9688	.9792	.9896

Table 27-00. Mensuration of Surfaces and Volumes

Area of rectangle = length × breadth.
Area of triangle = base × ½ perpendicular height.
Diameter of circle = radius × 2.
Circumference of circle = diameter × 3.1416.
Area of circle = square of diameter × .7854.

$$\text{Area of sector of circle} = \frac{\text{area of circle} \times \text{number of degrees in arc.}}{360}$$

Area of surface of cylinder = circumference × length + area of two ends.
To find the diameter of circle having given area: Divide the area by .7854, and extract the square root.
To find the volume of a cylinder: Multiply the area of the section in square inches by the length in inches = the volume in cubic inches. Cubic inches divided by 1728 = volume in cubic feet.
Surface of a sphere = square of diameter × 3.1416.
Solidity of a sphere = cube of diameter × .5236.
Side of an inscribed cube = radius of a sphere × 1.1547.
Area of the base of a pyramid or cone, whether round square or triangular, multiplied by one-third of its height = the solidity.
Diam. × .8862 = side of an equal square.
Diam. × .7071 = side of an inscribed square.
Radius × 6.2832 = circumference.
Circumference = 3.5446 × $\sqrt{\text{Area of circle.}}$
Diameter = 1.1283 × $\sqrt{\text{Area of circle.}}$
Length of arc = No. of degrees × .017453 radius.
Degrees in arc whose length equals radius = 57° 2958′.
Length of an arc of 1° = radius × .017543.
Length of an arc of 1 Min. = radius × .0002909.
Length of an arc of 1 Sec. = radius × .0000048.

π	=	Proportion of circumference to diameter = 3.1415926.
π^2	=	9.8696044.
$\sqrt{\pi}$	=	1.7724538.
Log. π	=	0.49715.
$1/\pi$	=	0.31831.
1/360	=	.002778.
$360/\pi$	=	114.59.

Table 27-00. Electrical Units

Volt—The unit of electrical motive force. Force required to send one ampere of current through one ohm of resistance.
Ohm—Unit of resistance. The resistance offered to the passage of one ampere, when impelled by one volt.
Ampere—Unit of current. The current which one volt can send through a resistance of one ohm.
Coulomb—Unit of quantity. Quantity of current which, impelled by one volt, would pass through one ohm in one second.
Farad—Unit of capacity. A conductor or condenser which will hold one coulomb under the pressure of one volt.
Joule—Unit of work. The work done by one watt in one second.
Watt—The unit of electrical energy, and is the product of ampere and volt. That is, one ampere of current flowing under a pressure of one volt gives one watt of energy.
One electrical horsepower is equal to 746 watts.
One Kilowatt is equal to 1,000 watts.
To find the watts consumed in a given electrical circuit, such as a pump motor, multiply the volts by the amperes.
To find the volts, divide the watts by the amperes.
To find the amperes, divide the watts by the volts.
To find the electrical horsepower required by a motor, divide the watts of the motor by 746.
To find the mechanical horsepower necessary to generate the required electrical horsepower, divide the latter by the efficiency of the generator.
To find the amperes of a given circuit, of which the volts and ohms resistance are known, divide the volts by the ohms.
To find the volts, when the amperes and watts are known, multiply the amperes by the ohms.
To find the resistance in ohms, when the volts and amperes are known, divide the volts by the amperes.

Table 27-00. Circumferences and Areas of Circles Advancing by Eighths

Diam.	Circum.	Area
1/64	.04909	.00019
1/32	.09818	.00077
3/64	.14726	.00173
1/16	.19635	.00307
3/32	.29452	.00690
1/8	.39270	.01227
5/32	.49087	.01917
3/16	.58905	.02761
7/32	.68722	.03758
1/4	.78540	.04909
9/32	.88357	.06213
5/16	.98175	.07670
11/32	1.0799	.09281
3/8	1.1781	.11045
13/32	1.2763	.12962
7/16	1.3744	.15033
15/32	1.4726	.17257
1/2	1.5708	.19635
17/32	1.6690	.22166
9/16	1.7671	.24850
19/32	1.8653	.27688
5/8	1.9635	.30680
21/32	2.0617	.33824
11/16	2.1598	.37122
23/32	2.2580	.40574
3/4	2.3562	.44179
25/32	2.4544	.4 937
13/16	2.5525	.51849
27/32	2.6507	.55914
7/8	2.7489	.60132
29/32	2.8471	.64504
15/16	2.9452	.69029
31/32	3.0434	.73708
1.	3.1416	.7854
1/16	3.3379	.8866
1/8	3.5343	.9940
3/16	3.7306	1.1075
1/4	3.9270	1.2272
5/16	4.1233	1.3530
3/8	4.3197	1.4849
7/16	4.5160	1.6230
1/2	4.7124	1.7671
9/16	4.9087	1.9175
5/8	5.1051	2.0739
11/16	5.3014	2.2365
3/4	5.4978	2.4053
13/16	5.6941	2.5802
7/8	5.8905	2.7612
15/16	6.0868	2.9483
2.	6.2832	3.1416
1/16	6.4795	3.3410
1/8	6.6759	3.5466
3/16	6.8722	3.7583
1/4	7.0686	3.9761
5/16	7.2649	4.2000
3/8	7.4613	4.4301
7/16	7.6576	4.6664
1/2	7.8540	4.9087
9/16	8.0503	5.1572
5/8	8.2467	5.4119

Diam.	Circum.	Area
2 11/16	8.4430	5.6727
3/4	8.6394	5.9396
13/16	8.8357	6.2126
7/8	9.0321	6.4918
15/16	9.2284	6.7771
3.	9.4248	7.0686
1/16	9.6211	7.3662
1/8	9.8175	7.6699
3/16	10.014	7.9798
1/4	10.210	8.2958
5/16	10.407	8.6179
3/8	10.603	8.9462
7/16	10.799	9.2806
1/2	10.996	9.6211
9/16	11.192	9.9678
5/8	11.388	10.321
11/16	11.585	10.680
3/4	11.781	11.045
13/16	11.977	11.416
7/8	12.174	11.793
15/16	12.370	12.177
4.	12.566	12.566
1/16	12.763	12.962
1/8	12.959	13.364
3/16	13.155	13.772
1/4	13.352	14.186
5/16	13.548	14.607
3/8	13.744	15.033
7/16	13.941	15.466
1/2	14.137	15.904
9/16	14.334	16.349
5/8	14.530	16.800
11/16	14.726	17.257
3/4	14.923	17.721
13/16	15.119	18.190
7/8	15.315	18.665
15/16	15.512	19.147
5.	15.708	19.635
1/16	15.904	20.129
1/8	16.101	20.629
3/16	16.297	21.135
1/4	16.493	21.648
5/16	16.690	22.166
3/8	16.886	22.691
7/16	17.082	23.221
1/2	17.279	23.758
9/16	17.475	24.301
5/8	17.671	24.850
11/16	17.868	25.406
3/4	18.064	25.967
13/16	18.261	26.535
7/8	18.457	27.109
15/16	18.653	27.688
6.	18.850	28.274
1/8	19.242	29.465
1/4	19.635	30.680
3/8	20.028	31.919
1/2	20.420	33.183
5/8	20.813	34.472
3/4	21.206	35.785
7/8	21.598	37.122

Diam.	Circum.	Area
7.	21.991	38.485
1/8	22.384	39.871
1/4	22.776	41.282
3/8	23.169	42.718
1/2	23.562	44.179
5/8	23.955	45.664
3/4	24.347	47.173
7/8	24.740	48.707
8.	25.133	50.265
1/8	25.525	51.849
1/4	25.918	53.456
3/8	26.311	55.088
1/2	26.704	56.745
5/8	27.096	58.426
3/4	27.489	60.132
7/8	27.882	61.862
9.	28.274	63.617
1/8	28.667	65.397
1/4	29.060	67.201
3/8	29.452	69.029
1/2	29.845	70.882
5/8	30.238	72.760
3/4	30.631	74.662
7/8	31.023	76.589
10.	31.416	78.540
1/8	31.809	80.516
1/4	32.201	82.516
3/8	32.594	84.541
1/2	32.987	86.590
5/8	33.379	88.664
3/4	33.772	90.763
7/8	34.165	92.886
11.	34.558	95.033
1/8	34.950	97.205
1/4	35.343	99.402
3/8	35.736	101.62
1/2	36.128	103.87
5/8	36.521	106.14
3/4	36.914	108.43
7/8	37.306	110.75
12.	37.699	113.10
1/8	38.092	115.47
1/4	38.485	117.86
3/8	38.877	120.28
1/2	39.270	122.72
5/8	39.663	125.19
3/4	40.055	127.68
7/8	40.448	130.19
13.	40.841	132.73
1/8	41.233	135.30
1/4	41.626	137.89
3/8	42.019	140.50
1/2	42.412	143.14
5/8	42.804	145.80
3/4	43.197	148.49
7/8	43.590	151.20
14.	43.982	153.94
1/8	44.375	156.70

Diam.	Circum.	Area
14 1/4	44.768	159.48
3/8	45.160	162.30
1/2	45.553	165.13
5/8	45.946	167.99
3/4	46.338	170.87
7/8	46.731	173.78
15.	47.124	176.71
1/8	47.517	179.67
1/4	47.909	182.65
3/8	48.302	185.66
1/2	48.695	188.69
5/8	49.087	191.75
3/4	49.480	194.83
7/8	49.873	197.93
16.	50.265	201.06
1/8	50.658	204.22
1/4	51.051	207.39
3/8	51.444	210.60
1/2	51.836	213.82
5/8	52.229	217.08
3/4	52.622	220.35
7/8	53.014	223.65
17.	53.407	226.98
1/8	53.800	230.33
1/4	54.192	233.71
3/8	54.585	237.10
1/2	54.978	240.53
5/8	55.371	243.98
3/4	55.763	247.45
7/8	56.156	250.95
18.	56.549	254.47
1/8	56.941	258.02
1/4	57.334	261.59
3/8	57.727	265.18
1/2	58.119	268.80
5/8	58.512	272.45
3/4	58.905	276.12
7/8	59.298	279.81
19.	59.690	283.53
1/8	60.083	287.27
1/4	60.476	291.04
3/8	60.868	294.83
1/2	61.261	298.65
5/8	61.654	302.49
3/4	62.046	306.35
7/8	62.439	310.24
20.	62.832	314.16
1/8	63.225	318.10
1/4	63.617	322.06
3/8	64.010	326.05
1/2	64.403	330.06
5/8	64.795	334.10
3/4	65.188	338.16
7/8	65.581	342.25
21.	65.973	346.36
1/8	66.366	350.50
1/4	66.759	354.66
3/8	67.152	358.84

Table 27-00. Circumferences and Areas of Circles Advancing by Eighths—*Continued*

Diam.	Circum.	Area	Diam.	Circum.	Area	Diam.	Circum.	Area	Diam.	Circum.	Area
21 ½	67.544	363.05	28 ¾	90.321	649.18	36.	113.097	1017.9	43 ¼	135.874	1469.1
⅝	67.937	367.28	⅞	90.713	654.84	⅛	113.490	1025.0	⅜	136.267	1477.6
¾	68.330	371.54				¼	113.883	1032.1	½	136.659	1486.2
⅞	68.722	375.83	29.	91.106	660.52	⅜	114.275	1039.2	⅝	137.052	1494.7
			⅛	91.499	665.23	½	114.668	1046.3	¾	137.445	1503.3
22.	69.115	380.13	¼	91.892	671.96	⅝	115.061	1053.5	⅞	137.837	1511.9
⅛	69.508	384.46	⅜	92.284	677.71	¾	115.454	1060.7			
¼	69.900	388.82	½	92.677	683.49	⅞	115.846	1068.0	44.	138.230	1520.5
⅜	70.293	393.20	⅝	93.070	689.30				⅛	138.623	1529.2
½	70.686	397.61	¾	93.462	695.13	37.	116.239	1075.2	¼	139.015	1537.9
⅝	71.079	402.04	⅞	93.855	700.98	⅛	116.632	1082.5	⅜	139.408	1546.6
¾	71.471	406.49				¼	117.024	1089.8	½	139.801	1555.3
⅞	71.864	410.97	30.	94.248	706.86	⅜	117.417	1097.1	⅝	140.194	1564.0
			⅛	94.640	712.76	½	117.810	1104.5	¾	140.586	1572.8
23.	72.257	415.48	¼	95.033	718.69	⅝	118.202	1111.8	⅞	140.979	1581.6
⅛	72.649	420.00	⅜	95.426	724.64	¾	118.596	1119.2			
¼	73.042	424.56	½	95.819	730.62	⅞	118.988	1126.7	45.	141.372	1590.4
⅜	73.435	429.13	⅝	96.211	736.62				⅛	141.764	1599.3
½	73.827	433.74	¾	96.604	742.64	38.	119.381	1134.1	¼	142.157	1608.2
⅝	74.220	438.36	⅞	96.997	748.69	⅛	119.773	1141.2	⅜	142.550	1617.0
¾	74.613	443.01				¼	120.166	1149.2	½	142.942	1626.0
⅞	75.006	447.69	31.	97.389	754.77	⅜	120.559	1156.6	⅝	143.335	1634.9
			⅛	97.782	760.87	½	120.951	1164.2	¾	143.728	1643.9
24.	75.398	452.39	¼	98.175	766.99	⅝	121.344	1171.7	⅞	144.121	1652.9
⅛	75.791	457.11	⅜	98.567	773.14	¾	121.737	1179.3			
¼	76.184	461.86	½	98.960	779.31	⅞	122.129	1186.9	46.	144.513	1661.9
⅜	76.576	466.64	⅝	99.353	785.51				⅛	144.906	1670.9
½	76.969	471.44	¾	99.746	791.73	39.	122.522	1194.6	¼	145.299	1680.0
⅝	77.362	476.26	⅞	100.138	797.98	⅛	122.915	1202.3	⅜	145.691	1689.1
¾	77.754	481.11				¼	123.308	1210.0	½	146.084	1698.2
⅞	78.147	485.98	32.	100.531	804.25	⅜	123.700	1217.7	⅝	146.477	1707.4
			⅛	100.924	810.54	½	124.093	1225.4	¾	146.869	1716.5
25.	78.540	490.87	¼	101.316	816.86	⅝	124.486	1233.2	⅞	147.262	1725.7
⅛	78.933	495.79	⅜	101.709	823.21	¾	124.878	1241.0			
¼	79.325	500.74	½	102.102	829.58	⅞	125.271	1248.8	47.	147.655	1734.9
⅜	79.718	505.71	⅝	102.494	835.97				⅛	148.048	1744.2
½	80.111	510.71	¾	102.887	842.39	40.	125.664	1256.6	¼	148.440	1753.5
⅝	80.503	515.72	⅞	103.280	848.83	⅛	126.056	1264.5	⅜	148.833	1762.7
¾	80.896	520.77				¼	126.449	1272.4	½	149.226	1772.1
⅞	81.289	525.84	33.	103.673	855.30	⅜	126.842	1280.3	⅝	149.618	1781.4
			⅛	104.065	861.79	½	127.235	1288.2	¾	150.011	1790.8
26.	81.681	530.93	¼	104.458	868.31	⅝	127.627	1296.2	⅞	150.404	1800.1
⅛	82.074	536.05	⅜	104.851	874.85	¾	128.020	1304.2			
¼	82.467	541.19	½	105.243	881.41	⅞	128.413	1312.2	48.	150.796	1809.6
⅜	82.860	546.35	⅝	105.636	888.00				⅛	151.189	1819.0
½	83.252	551.55	¾	106.029	894.62	41.	128.805	1320.3	¼	151.582	1828.5
⅝	83.645	556.76	⅞	106.421	901.26	⅛	129.198	1328.3	⅜	151.975	1837.9
¾	84.038	562.00				¼	129.591	1336.4	½	152.367	1847.5
⅞	84.430	567.27	34.	106.814	907.92	⅜	129.983	1344.5	⅝	152.760	1857.0
			⅛	107.207	914.61	½	130.376	1352.7	¾	153.153	1866.5
27.	84.823	572.56	¼	107.600	921.32	⅝	130.769	1360.8	⅞	153.545	1876.1
⅛	85.216	577.87	⅜	107.992	928.06	¾	131.161	1369.0			
¼	85.608	583.21	½	108.385	934.82	⅞	131.554	1377.2	49.	153.938	1885.7
⅜	86.001	588.57	⅝	108.778	941.61				⅛	154.331	1895.4
½	86.394	593.96	¾	109.170	948.42	42.	131.947	1385.4	¼	154.723	1905.0
⅝	86.786	599.37	⅞	109.563	955.25	⅛	132.340	1393.7	⅜	155.116	1914.7
¾	87.179	604.81				¼	132.732	1402.0	½	155.509	1924.4
⅞	87.572	610.27	35.	109.956	962.11	⅜	133.125	1410.3	⅝	155.902	1934.2
			⅛	110.348	969.00	½	133.518	1418.6	¾	156.294	1943.9
28	87.965	615.75	¼	110.741	975.91	⅝	133.910	1427.0	⅞	156.687	1953.7
⅛	88.357	621.26	⅜	111.134	982.84	¾	134.303	1435.4			
¼	88.750	626.80	½	111.527	989.80	⅞	134.696	1443.8	50.	157.080	1963.5
⅜	89.143	632.36	⅝	111.919	996.78						
½	89.535	637.94	¾	112.312	1000.38	43.	135.088	1452.2			
⅝	89.928	643.55	⅞	112.705	1010.8	⅛	135.481	1460.7			

Table 27-00. Fractional Equivalents, Powers, Roots and Velocity Heads of Numbers

Number	Frac. Equiv	Square Root	Cube Root	Square	Cube	Velocity Head $\sqrt{2\,G \times No.}$	Number	Frac. Equiv	Square Root	Cube Root	Square	Cube	Velocity Head $\sqrt{2\,G \times No.}$
.01	..	.1	.2154	.0001	.000001	.802	.3281	21/64	.5728	.6897	.1077	.03533	4.594
.0156	1/64	.125	.25	.0002441	.000003815	1.003	.33	..	.5745	.6910	.1089	.03594	4.607
.02	..	.1414	.2714	.0004	.000008	1.131	.34	..	.5831	.6980	.1156	.03930	4.677
.03	..	.1732	.3107	.0009	.000027	1.389	.3438	11/32	.5863	.7005	.1182	.04062	4.702
.0313	1/32	.1768	.3150	.0009766	.00003052	1.418	.35	..	.5916	.7047	.1225	.04288	4.745
.04	..	.2	.3420	.0016	.000064	1.604	.3594	23/64	.5995	.7110	.1292	.04641	4.808
.0469	3/64	.2165	.3606	.002197	.000103	1.756	.36	..	.6	.7114	.1296	.04666	4.812
.05	..	.2236	.3684	.0025	.000125	1.793	.37	..	.6083	.7179	.1369	.05065	4.879
.06	..	.2449	.3915	.0036	.000216	1.965	.375	3/8	.6124	.7211	.1406	.05273	4.911
.0625	1/16	.25	.3968	.003906	.0002441	2.005	.38	..	.6164	.7243	.1444	.05487	4.944
.07	..	.2646	.4121	.0049	.000343	2.122	.39	..	.6245	.7306	.1521	.05932	5.009
.0781	5/64	.2795	.4275	.006104	.0004768	2.242	.3906	25/64	.625	.7310	.1526	.05960	5.013
.08	..	.2828	.4309	.0064	.000512	2.269	.4	..	.6325	.7368	.16	.64	5.072
.09	..	.3	.4481	.0081	.000729	2.406	.4063	13/32	.6374	.7406	.1650	.06705	5.112
.0938	3/32	.3062	.4543	.008789	.0008240	2.456	.41	..	.6403	.7429	.1681	.06892	5.135
.1	..	.3162	.4642	.01	.001	2.537	.42	..	.6481	.7489	.1764	.07409	5.198
.1094	7/64	.3307	.4782	.01196	.001308	2.653	.4219	27/64	.6495	.75	.1780	.07508	5.209
.11	..	.3317	.4791	.0121	.001331	2.660	.43	..	.6557	.7548	.1849	.07951	5.259
.12	..	.3464	.4932	.0144	.001728	2.778	.4375	7/16	.6614	.7591	.1914	.08374	5.305
.125	1/8	.3536	.5	.01562	.001953	2.836	.44	..	.6633	.7606	.1936	.08518	5.320
.13	..	.3606	.5066	.0169	.002197	2.892	.45	..	.6708	.7663	.2025	.09113	5.380
.14	..	.3742	.5193	.0196	.002744	3.001	.4531	29/64	.6732	.7681	.2053	.09304	5.399
.1406	9/64	.375	.5200	.01978	.002781	3.008	.46	..	.6782	.7719	.2116	.09734	5.440
.15	..	.3873	.5313	.0225	.003375	3.106	.4688	15/32	.6847	.7768	.2197	.1030	5.491
.1563	5/32	.3953	.5386	.02441	.003815	3.170	.47	..	.6856	.7775	.2209	.1038	5.498
.16	..	.4	.5429	.0256	.004096	3.208	.48	..	.6928	.7830	.2304	.1106	5.557
.17	..	.4123	.5540	.0289	.004913	3.307	.4844	31/64	.6960	.7853	.2346	.1136	5.582
.1719	11/64	.4146	.5560	.02954	.005077	3.325	.49	..	.7	.7884	.2401	.1176	5.614
.18	..	.4243	.5646	.0324	.005832	3.403	.5	1/2	.7071	.7937	.25	.125	5.671
.1875	3/16	.433	.5724	.03516	.006592	3.473	.51	..	.7141	.7990	.2601	.1327	5.728
.19	..	.4359	.5749	.0361	.006859	3.496	.5156	33/64	.7181	.8019	.2658	.1371	5.759
.20	..	.4472	.5848	.04	.008	3.587	.52	..	.7211	.8042	.2704	.1406	5.784
.2031	13/64	.4507	.5878	.04126	.008381	3.615	.53	..	.7280	.8093	.2809	.1489	5.839
.21	..	.4583	.5944	.0441	.009261	3.675	.5313	17/32	.7289	.8099	.2822	.1499	5.846
.2188	7/32	.4677	.6025	.04785	.01047	3.751	.54	..	.7349	.8143	.2916	.1575	5.894
.22	..	.4690	.6037	.0484	.01065	3.762	.5469	35/64	.7395	.8178	.2991	.1636	5.931
.23	..	.4796	.6127	.0529	.01217	3.846	.55	..	.7416	.8193	.3025	.1664	5.948
.2344	15/64	.4841	.6165	.05493	.01287	3.883	.56	..	.7483	.8243	.3136	.1756	6.002
.24	..	.4899	.6215	.0576	.01382	3.929	.5625	9/16	.75	.8255	.3164	.1780	6.015
.25	1/4	.5	.6300	.0625	.01563	4.010	.57	..	.7550	.8291	.3249	.1852	6.055
.26	..	.5099	.6383	.0676	.01758	4.090	.5781	37/64	.7603	.8330	.3342	.1932	6.098
.2656	17/64	.5154	.6428	.07056	.01874	4.131	.58	..	.7616	.8340	.3364	.1951	6.108
.27	..	.5196	.6463	.0729	.01968	4.167	.59	..	.7681	.8387	.3481	.2054	6.161
.28	..	.5292	.6542	.0784	.02195	4.244	.5938	19/32	.7706	.8405	.3525	.2093	6.180
.2813	9/32	.5303	.6552	.07910	.02225	4.253	.6	..	.7746	.8434	.36	.2160	6.212
.29	..	.5385	.6619	.0841	.02439	4.319	.6094	39/64	.7806	.8478	.3713	.2263	6.261
.2969	19/64	.5448	.6671	.08814	.02617	4.370	.61	..	.7810	.8481	.3721	.2270	6.264
.30	..	.5477	.6694	.09	.027	4.393	.62	..	.7874	.8527	.3844	.2383	6.315
.31	..	.5568	.6768	.0961	.02979	4.466	.625	5/8	.7906	.8550	.3906	.2441	6.341
.3125	5/16	.5590	.6786	.09766	.03052	4.483	.63	..	.7937	.8573	.3969	.2500	6.366
.32	..	.5657	.6840	.1024	.03277	4.537	.64	..	.8	.8618	.4096	.2621	6.416

Table 27-00. Fractional Equivalents, Powers, Roots and Velocity Heads of Numbers—*Continued*

Number	Frac. Equiv	Square Root	Cube Root	Square	Cube	Velocity Head $\sqrt{2\,G\,x\,No.}$	Number	Frac. Equiv	Square Root	Cube Root	Square	Cube	Velocity Head $\sqrt{2\,G\,x\,No.}$
.6406	41/64	.8004	.8621	.4104	.2629	6.419	.96	..	.9798	.9865	.9216	.8847	7.858
.65	..	.8062	.8662	.4225	.2746	6.466	.9688	31/32	.9843	.9895	.9385	.9091	7.894
.6563	21/32	.8101	.8690	.4307	.2826	6.497	.97	..	.9849	.9899	.9409	.9127	7.899
.66	..	.8124	.8707	.4356	.2875	6.516	.98	..	.9899	.9933	.9604	.9412	7.940
.67	..	.8185	.8750	.4489	.3008	6.565	.9844	63/64	.9922	.9918	.9690	.9538	7.957
.6719	43/64	.8197	.8759	.4514	.3033	6.574	.99	..	.9950	.9967	.9801	.9703	7.980
.68	..	.8246	.8794	.4624	.3144	6.614	1.	..	1.	1.	1.	1.	8.021
.6875	11/16	.8292	.8826	.4727	.3249	6.650	1.1	..	1.049	1.032	1.21	1.331	8.412
.69	..	.8307	.8837	.4761	.3285	6.662	1.2	..	1.095	1.063	1.44	1.728	8.786
.70	..	.8367	.8879	.49	.3430	6.710	1.3	..	1.14	1.091	1.69	2.197	9.145
.7031	45/64	.8395	.8892	.4944	.3476	6.725	1.4	..	1.183	1.119	1.96	2.744	9.490
.71	..	.8426	.8921	.5041	.3579	6.758	1.5	..	1.225	1.1145	2.25	3.375	9.823
.7188	23/32	.8478	.8958	.5166	.3713	6.799	1.6	..	1.265	1.170	2.56	4.096	10.14
.72	..	.8485	.8963	.5184	.3732	6.805	1.7	..	1.304	1.193	2.89	4.913	10.45
.73	..	.8544	.9004	.5329	.3890	6.853	1.8	..	1.342	1.216	3.24	5.832	10.76
.7344	47/64	.8570	.9022	.5393	.3961	6.873	1.9	..	1.378	1.239	3.61	6.859	11.06
.74	..	.8602	.9045	.5476	.4052	6.899	2.	..	1.414	1.260	4.	8.	11.34
.75	¾	.8660	.9086	.5625	.4219	6.946	2.1	..	1.449	1.281	4.41	9.261	11.62
.76	..	.8718	.9126	.5776	.4390	6.992	2.2	..	1.483	1.301	4.84	10.65	11.90
.7656	49/64	.875	.9148	.5862	.4488	7.018	2.3	..	1.517	1.320	5.29	12.17	12.16
.77	..	.8775	.9166	.5929	.4565	7.038	2.4	..	1.549	1.339	5.76	13.82	12.43
.78	..	.8832	.9205	.6084	.4746	7.083	2.5	..	1.581	1.357	6.25	15.63	12.68
.7813	25/32	.8839	.9210	.6104	.4768	7.089	2.6	..	1.612	1.375	6.76	17.58	12.93
.79	..	.8888	.9244	.6241	.4930	7.129	2.7	..	1.643	1.392	7.29	19.68	13.18
.7969	51/64	.8927	.9271	.6350	.5060	7.159	2.8	..	1.673	1.409	7.84	21.95	13.42
.8	..	.8944	.9283	.64	.5120	7.174	2.9	..	1.703	1.426	8.41	24.39	13.66
.81	..	.9	.9322	.6561	.5314	7.218	3.	..	1.732	1.442	9.	27.	13.89
.8125	13/16	.9014	.9331	.6602	.5364	7.229	3.1	..	1.761	1.458	9.61	29.79	14.12
.82	..	.9055	.9360	.6724	.5514	7.263	3.2	..	1.789	1.474	10.24	32.77	14.35
.8281	53/64	.9100	.9391	.6858	.5679	7.298	3.3	..	1.817	1.489	10.89	35.94	14.57
.83	..	.9110	.9398	.6889	.5718	7.307	3.4	..	1.844	1.504	11.56	39.30	14 79
.84	..	.9165	.9435	.7056	.5927	7.351	3.5	..	1.871	1.518	12.25	42.88	15.01
.8438	27/32	.9186	.9449	.7120	.6007	7.367	3.6	..	1.897	1.533	12.96	46.66	15.22
.85	..	.9219	.9473	.7225	.6141	7.394	3.7	..	1.924	1.547	13.69	50.65	15.43
.8594	55/64	.9270	.9507	.7385	.6347	7.435	3.8	..	1.949	1.560	14.44	54.87	15.64
.86	..	.9274	.9510	.7396	.6361	7.438	3.9	..	1.975	1.574	15.21	59.32	15.85
.87	..	.9327	.9546	.7569	.6585	7.481	4.	..	2.	1.587	16.	64.	16.04
.875	⅞	.9354	.9565	.7656	.6699	7.502	4.1	..	2.025	1.601	16.81	68.92	16.24
.88	..	.9381	.9583	.7744	.6815	7.524	4.2	..	2.049	1.613	17.64	74.09	16.44
.89	..	.9434	.9619	.7921	.7050	7.566	4.3	..	2.074	1.626	18.49	79.51	16.63
.8906	57/64	.9437	.9621	.7932	.7065	7.569	4.4	..	2.098	1.639	19.36	85.18	16.82
.9	..	.9487	.9655	.81	.7290	7.609	4.5	..	2.121	1.651	20.25	91.13	17.01
.9063	29/32	.9520	.9677	.8213	.7443	7.635	4.6	..	2.145	1.663	21.16	97.34	17.20
.91	..	.9539	.9691	.8281	.7536	7.651	4.7	..	2.168	1.675	22.09	103.8	17.39
.92	..	.9592	.9726	.8464	.7787	7.693	4.8	..	2.191	1.687	23.04	110.6	17.57
.9219	59/64	.9601	.9732	.8499	.7835	7.701	4.9	..	2.214	1.698	24.01	117.6	17.75
.93	..	.9644	.9761	.8649	.8044	7.734	5.	..	2.236	1.710	25.	125.	17.93
.9375	15/16	.9682	.9787	.8789	.8240	7.766							
.94	..	.9695	.9796	.8836	.8306	7.776							
.95	..	.9747	.9831	.9025	.8574	7.817							
.9531	61/64	.9763	.9840	.9084	.8659	7.830							

Table 27-00. Measures of Weight, Capacity and Area

Long Measure

12 inches = 1 foot.
3 feet = 1 yard
$5\frac{1}{2}$ yards = 1 rod.
4 rods = 1 chain.
10 chains = 1 furlong.
8 furlongs = 1 mile.

Square Measure

144 square inches = 1 square foot.
9 square feet = 1 square yard.
$30\frac{1}{4}$ square yards = 1 square rod.
160 square rods = 1 acre.
640 acres = 1 square mile.

Cubic Measure

1728 cubic inches = 1 cubic foot.
27 cubic feet = 1 cubic yard.
24.75 cubic feet = 1 perch.
128 cubic feet = 1 cord.

Liquid Measure

4 gills = 1 pint.
2 pints = 1 quart.
4 quarts = 1 gallon.
$31\frac{1}{2}$ gallons = 1 barrel.
2 barrels = 1 hogshead.

Avoirdupois Weight

16 ounces = 1 pound.
100 pounds = 1 hundredweight.
20 cwt. = 1 ton.

Table 27-00. Comparison of Wire Gauges

Gauge No.	American or Brown & Sharpe's	Birmingham or Stubs	Wash. & Moen	Imperial S. W. G.	London or Old English	United States Standard
0000000			.490	.500		.500
000000	.5800		.460	.464		.46875
00000	.5165		.430	.432		.4375
0000	.4600	.454	.3938	.400	.454	.40625
000	.4096	.425	.3625	.372	.425	.375
00	.3648	.380	.3310	.348	.38	.34375
0	.3249	.340	.3065	.324	.34	.3125
1	.2893	.300	.2830	.300	.3	.28125
2	.2576	.284	.2625	.276	.284	.265625
3	.2294	.259	.2437	.252	.259	.25
4	.2043	.238	.2253	.232	.238	.234375
5	.1819	.220	.2070	.212	.22	.21875
6	.1620	.203	.1920	.192	.203	.203125
7	.1443	.180	.1770	.176	.18	.1875
8	.1285	.165	.1620	.160	.165	.171875
9	.1144	.148	.1483	.144	.148	.15625
10	.1019	.134	.1350	.128	.134	.140625
11	.09074	.120	.1205	.116	.12	.125
12	.08081	.109	.1055	.104	.109	.109375
13	.07196	.095	.0915	.092	.095	.09375
14	.06408	.083	.0800	.080	.083	.078125
15	.05707	.072	.0720	.072	.072	.0703125
16	.05082	.065	.0625	.064	.065	.0625
17	.04526	.058	.0540	.056	.058	.05625
18	.04030	.049	.0475	.048	.049	.05
19	.03589	.042	.0410	.040	.040	.04375
20	.03196	.035	.0348	.036	.035	.0375
21	.02846	.032	.03175	.032	.0315	.034375
22	.02535	.028	.0286	.028	.0295	.03125

Gauge No.	American or Brown & Sharpe's	Birmingham or Stubs	Wash. & Moen	Imperial S. W. G.	London or Old English	United States Standard
23	.02257	.025	.0258	.024	.027	.028125
24	.02010	.022	.0230	.022	.025	.025
25	.01790	.020	.0204	.020	.023	.021875
26	.01594	.018	.0181	.018	.0205	.01875
27	.01420	.016	.0173	.0164	.0187	.0171875
28	.01264	.014	.0162	.0148	.0165	.015625
29	.01126	.013	.0150	.0136	.0155	.0140625
30	.01003	.012	.0140	.0124	.01372	.0125
31	.008928	.010	.0132	.0116	.0122	.0109375
32	.007950	.009	.0128	.0108	.0112	.01015625
33	.007080	.008	.0118	.0100	.0102	.009375
34	.006305	.007	.0104	.0092	.0095	.00859375
35	.005615	.005	.0095	.0084	.009	.0078125
36	.005000	.004	.0090	.0076	.0075	.00703125
37	.004453		.0085	.0068	.0065	.006640625
38	.003965		.008	.0060	.0057	.00625
39	.003531		.0075	.0052	.005	
40	.003145		.007	.0048	.0045	
41	.002800			.0044		
42	.002494			.004		
43	.002221			.0036		
44	.001978			.0032		
45	.001761			.0028		
46	.001568			.0024		
47	.001397			.002		
48	.001244			.0016		
49	.001018			.0012		
50	.0009863			.001		

Table 27-00. Conversion Factors—Lineal

Millimeter	1.	10.	25.4	301.8	911.4	1 000.	....	....	....
Centimeter	.1	1.	2.54	30.18	91.44	100.00	....	....	....
Inch	.039 37	.393 70	1.	12.00	36.00	39.37	39 370.	63 360.	72 960.
Foot	.003 28	.032 80	.083 33	1.	3.00	3.280 83	3 280.	5 280.	6 080.
Yard	.001 09	.010 93	.027 78	.333 33	1.	1.093 611	1 093.	1 760.	2 026.66
Meter	.001 00	.010 00	.025 1	.301 801	.914 402	1.	1 000.0	1 609.35	1 853.19
Kilometer	.000 001	.000 01	.000 025	.000 315	.000 914	.001 00	1.	1.609 35	1.853 19
Miles	.000 000 62	.000 006 2	.000 015 8	.000 189	.000 568	.000 621	.621 37	1.	1.151 5
Knots	.000 000 54	.000 005 2	.000 013 7	.000 164 5	.000 493 4	.000 51	.539 61	.868 4	1.

Table 27-00. Conversion Factors—Surface

Square Millimeters	1.	100.	615.2	92 900.	836 100.	1 000 000.
Square Centimeters	.01	1.	6.452	927.	8 361.	10 000.
Square Inches	.001 55	.155	1.	144.	1 296.	1 550.016
Square Feet	....	.001 077	.006 94	1.	9.	10.764
Square Yards	....	.000 12	.000 77	.111 11	1.	1.196
Square Meters	....	.000 1	.000 645	.092 9	.836	1.

Table 27-00. Conversion Factors—Cubical

1 Liter = 33.81 fl. ozs. 1 Gal. = 128 fl. ozs. 1 Qt. = 32 fl. ozs. 1 fl. oz. = 29.57 cu. cms.

Cubic Centimeters	1.	16.387	1 000.	3 785.4	28 315.	764 552.	1 000 000.
Cubic Inches	.061	1.	61.023	231.	1 728.	46 656.	61 023.
Liters or Cubic Decimeters	.001	.016 387	1.	3.785	28.315	764.55	1 000.
Gallons	.000 261	.001 33	.264 17	1.	7.48	201.97	264.17
Cubic Feet	.000 035	.000 578	.035 314	.133 68	1.	27.	35.314
Cubic Yards	.000 001 3	.000 021 4	.001 308	.001 95	.037 037	1.	1.308
Cubic Meters	.000 001 0	.000 016	.001 000	.003 785	.028 32	.765	1.

Table 27-00. Conversion Factors—Weight

Grains	1.	15.432	437.5	7 000.	15 432.36	14 000 000.	15 680 000.
Grams	.061 799	1.	28.349	453.59	1 000.	907 180.	1 016 041.6
Ounces Avoirdupois	.002 280	.035 27	1.	16.	35.274	32 000.	35 840.0
Pounds Avoirdupois	.000 143	.002 205	.062 5	1.	2.201 6	2 000.	2 240.0
Kilograms	.000 064	.001	.028 35	.453 59	1.	907.18	1 116.04

Table 27-00. Conversion Factors—Pressure

Atmospheres	1.	.068 03	.033 42	.029 47	.001 316	.000 472 6	.00 009 677
Pounds per sq. inch	14.7	1.	.491 3	.433 2	.019 34	.006 917	.00 142 3
Inches of Mercury 32° Fahr.	29.922	2.036	1.	.881 8	.039 37	.014 14	.00 289 5
Feet of Water 60° Fahr	33.94	2.309	1.134	1.	.044 64	.016 03	.00 328 3
Millimeters of Mercury 32° Fahr.	760.	51.7	25.398	22.399	1.	.359 2	.07 355
Pounds per sq. foot	2 116.	143.916	70.7	62.35	2.781	1.	.20 48
Kilograms per sq. meter	10 333.	702.925	345.331	304.566	13.596	1.883	1.

Table 27-00. Conversion Factors—Power

Foot pounds per min.	1.		44.256 7	60.	433.980				33 000.	44 256.7
Kg.—meters per min.		1.	6.118 72	8.295 32	60.				4 562.42	6 118.7
Watts	.022 595 4	0.163 433	1.	1.355 73	9.805	7.581 6	31.646 9	69.769 6	745.650	1 000.
Foot pounds per sec.			0.737 612	1.					550.	737.612
Kg.—meters per sec.			0.102 979		1.				76.040	101.979
B.t.u. per min.			0.056 877 6			1.			42.410 8	56.877 6
Lb. Centigrade per min.			0.031 598 7				1.		23.561 5	31.598 7
Kg.—deg. Cent. large calories			0.014 332 9					1.	10.687 3	14.332 9
Horsepower			0.001 341 1	0.001 818 18	0.013 150 9	0.023 578 9	0.042 442	0.093 568	1.	1.341 1
Kilowatts			0.001	0.001 355 73	0.009 805 9	0.017 581 6	0.031 646 9	0.060 760	0.745 65	1.

Table 27-00. Conversion Factors—Heat Energy

Joule	1.	1.355 73	9.805 96	745.650	1 000.	1 054.90
Foot-pound	0.737 612	1.	7.233 00	550.	737.612	778.104
Kilogram-meter	0.101 979	0.138 255	1.	76.040 4	101.979	107.577
Horsepower-seconds	0.001 341	0.001 818	0.013 150 9	1.		1.414 7
Kilowatt-seconds	0.001	0.001 355 73	0.009 805 96		1.	1.054 9
British thermal units	0.000 917 9	0.001 285 17	0.009 295 67			1.
Pounds Centigrade	0.000 526 6	0.000 713 986	0.005 164 26			.555 5
Watt-Hours	0.000 277 77	0.000 376 591	0.002 723 88	0.207 125	0.277 778	.293 02
Large Calories	0.000 238 882	0.000 323 859	0.002 342 47			.251 99
Horsepower hours	0.	0.000 000 505 05	0.			.000 392 982
Kilowatt-hours	0.	0.000.000 376 591	0.000 002 723 8			.000 293 027

Joule	1 898.81	3 600.	4 186.17	2 684 340.	3 600 000.
Foot-pound	1 400.59	2 655.4	3 087.77	1 980 000.	2 655 403.
Kilogram-meter	193.639	367.123	426.900	273 745.	367 123.
Horsepower-seconds	2.546 5	4.828	5.614 12	3 600.	4 828.
Kilowatt-seconds	1.898 8	3.6	4.186 17		
British thermal units	1.800 0	3.112 66	3.968 32	2 544.65	3 412.66
Pounds Centigrade	1.	1.895 92	2.204 62	1 413.69	1 895.92
Watt-hours	0.527 44	1.	1.162 82	745.650	1 000.
Large Calories	0.453 59	0.859 975	1.	641.240	859.975
Horsepower hours	0.000 707 36	0.001 341 1	0.001 559 4	1.	1.341 11
Kilowatt-hours	0.000 527 44	0.001	0.001 162 8	0.745 65	1.

DEFINITION OF TERMS

Joule = Watt-second = Energy expended in one second by one standard ampere flowing through a resistance of one standard ohm.
Foot-pound = 1 ÷ 778 B.t.u.'s = Energy expended by the force of one pound working through a distance of one foot.
Kilogram-meter = 7.233 Foot-pounds = Energy expended by the force of one kilogram working through a distance of one meter.
Horsepower = 33,000 Foot-pounds per min. = Work done by 33,000 foot pounds of energy exerted in one minute = 550 ft.-lb. per sec.
Kilowatt = 1,000 Watts = Kilowatt-second = Work at the rate of one Joule per second.
British thermal unit = 778 + Foot-pounds = Heat required to raise the temperature of one pound of water one degree fahr. at temperature of max. dens. 39° F.
Pound Centigrade = 1400.59 Foot-pounds = Heat required to raise the temperature of one pound of water one degree cent. at temperature of max. density 4 deg. cent.
Calories—Large—3087.77 Foot-pounds = Heat required to raise the temperature of kilogram of water one deg. cent. at temperature of max. density 4 deg. cent.