TECHNICAL MANUAL

TES-181A

Hoffman Specialty® Steam Heating Systems

DESIGN MANUAL AND ENGINEERING DATA

Bell & Gossett

a xylem brand

STEAM SYSTEM DESIGN

The use of steam as an economical heat transfer medium for both comfort heating and many process applications continues to be just as popular today as it has been for many years. Steam systems when properly controlled, trapped and equipped with modern condensate handling systems, help to conserve valuable energy which is increasingly important in view of today's spiraling fuel costs.

Steam use for comfort heating continues to be a practical heating medium in many parts of the institutional market such as hospitals where steam is required for other purposes and in many industrial plants where it is used for process applications. In fact, it mates well with hot water hydronic heating when this type system is desired by making use of heat exchanger-converter units. In many older buildings, one or two pipe steam heating systems continue to perform their important function with only minimal maintenance.

With the continuing popularity of steam as a heat transfer medium, a working knowledge of steam systems is necessary for those who are associated with the design, sales, purchasing, installation or maintenance of steam systems. To aid in this important work we have prepared this information manual.

Table of Contents

ADVANTAGES OF SPACE HEATING WITH STEAM

The widespread use of steam for space heating today points up a long recognized fact that steam, as a heating medium, has numerous basic characteristics which can be advantageously employed. Some of the most important advantages are as follow:

1. STEAM'S ABILITY TO GIVE OFF HEAT

Properties of saturated steam are shown in Steam Tables and give much information regarding the temperature and the heat (energy) contained in one pound of steam for any pressure. For example, to change one pound of water from 212°F into steam at the same temperature of 212°F at Atmospheric Pressure (14.7 PSIA) requires a heat content of 1150.4 BTU which is made up of 180.1 BTU Sensible Heat (the heat required to raise the one pound of water from 32°F., Freezing, to 212°F) and 970.3 BTU of Latent Heat. The Latent Heat is the heat added to change the one pound of water from 212°F into steam at 212°F. This stored up Latent Heat is required to transform the water into steam and it reappears as heat when the process is reversed to condense the steam into water. Because of this basic fact, the high Latent Heat of vaporization of a pound of steam permits a large quantity of heat to be transmitted efficiently from the boiler to the heating unit or radiator with little change in temperature.

2. STEAM PROMOTES ITS OWN CIRCULATION THROUGH PIPING

For example, steam will flow naturally from a higher pressure (as generated in the boiler) to a lower pressure (existing in the steam mains). Circulation or flow is caused by the lowering of the steam pressure along the steam supply mains and the heating units due to pipe friction and to the condensing process of steam as it gives up heat to the space being heated. **See Figure 1.**

Because of this fact, the natural flow of steam does not require a pump such as needed for hot water heating, or a fan as employed in warm air heating.

3. STEAM HEATS MORE READILY

Steam circulates through a heating system faster than other fluid mediums. This can be important where fast pick-up of the space temperature is desired. It will also cool down more rapidly when circulation is stopped. This is an important consideration in Spring and Fall when comfort conditions can be adversely affected by long heating-up or slow cooling-down periods.

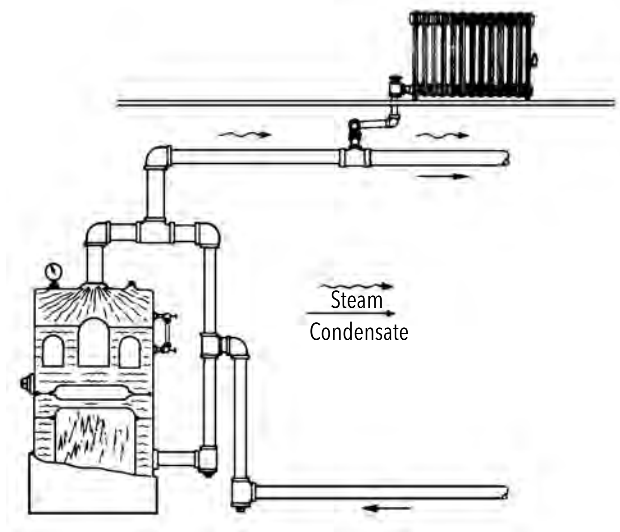

Figure 1

4. FLEXIBILITY OF STEAM HEATING (Figure 2)

Other advantages in using steam as a heating medium can be found in the easy adaptability to meet unusual conditions of heat requirement with a minimum of attention and maintenance. Here are some examples:

(a) Temporary heat during construction is easily provided without undue risks and danger of freeze-ups.

(b) Additional heating units can be added to existing systems without making basic changes to the system design.

(c) Increased heat output from heating units can be easily accomplished by increasing the steam pressure the proper amount.

(d) Steam heating systems are not prone to leak; however, leaks that may occur in the system piping, pipe fittings, or equipment, cause less damage than leaks in systems using hot water. A cubic foot of steam condenses into a relatively small quantity of water. In many cases, a small leak does not cause any accumulation of water at the location of the leak; instead, it evaporates into the air and causes no damage.

Water Content in Boiler

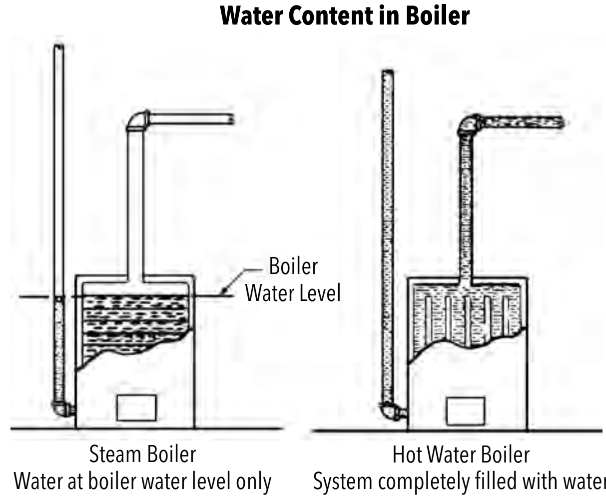

Figure 2

(e) Repair or replacement of system components such as valves, traps, heating units (radiators) and similar equipment, can be made by simply closing off the steam supply. It is not necessary to drain the system and to spend additional time to re-establish circulation. There is less need to worry about freezing since the water in a steam heating system is mainly in the boiler.

Boiler water can easily be protected from freezing during shut-downs, during new construction, repairs or replacement of parts, by installing an aquastat below the boiler water level to control water temperature.

(f) Steam is a flexible medium when used in combination processes and heating applications. These often require different pressures which are easily obtained. In addition, exhaust steam, when available, can be utilized to the fullest advantage.

(g) Steam heating systems are considered to be "lifetime" investments. Many highly efficient systems are in operation today after more than 50 years of service.

5. STEAM IS EASY TO DISTRIBUTE AND CONTROL IN A HEATING SYSTEM

The distribution of steam to heating units or radiators is easy to accomplish with distribution orifices located at the steam inlet to the unit. Metering orifices can also be employed when used with proper controls to maintain steam pressure in accordance with the flow characteristic of the metering orifices. These orifices can be either a fixed type or a variable type. The controls for steam systems are simple and effective. They include those used to control space temperature by the application of "ON-OFF" valves. Modulating controls can also be applied which respond to Indoor-Outdoor temperature conditions to control the quantity of steam flowing to orificed radiators.

6. WHERE STEAM CAN BE USED

Steam can be used as the heating medium for all types of heating units such as convectors, wall in tube radiation, cast iron radiators, unit heaters, unit ventilators, heating and ventilating units, all types of coils in ventilating and air conditioning systems, and steam absorption units used in air conditioning. Steam can be used for all types of systems applicable to a variety of building designs ranging from residences to large industrial, commercial, multi-story apartment groups, offices, churches, or schools.

7. RECOMMENDED STEAM APPLICATIONS

The following applications for the use of steam are recommended to provide trouble-free and efficient heating systems.

(a) Where there is more than one job to do, such as providing comfort heating as well as steam for processes in Industrial Plants, Restaurants Hospitals, Dry Cleaning Plants, Institutions Laundries, etc

(b) Where outdoor air is heated for ventilation (especially in cold climates) ... as in Factories Buildings with Central Air Conditioning, Ventilating Systems, Gymnasiums, Auditoriums, and School Classrooms

(c) Where there is a surplus of steam from processes which can be used for air cooling or water chilling.

(d) Where the heating medium must travel a great distance from the boiler to the heating units, as found in High Multi-Story, Buildings, Long Rambling Buildings, Scattered Buildings Supplied from a Central Station

(e) Where intermittent changes in heat loads are required as in Schools, Churches and Office Buildings

(f) Where central heat control or individual room control of temperature is important as found in Schools, Office Buildings, Hospitals Hotels and Motels

(g) Where there is a chance of freezing, in cold climates or where sub-freezing air is handled.

(h) Where there may be additions or alterations of space or change of occupancy in a building.

(i) Where extra heat is needed as in buildings with large or frequently used doors, Department Stores, Shipping Departments, Warehouses, Airplane Hangars or Garages

ONE-PIPE STEAM HEATING SYSTEMS

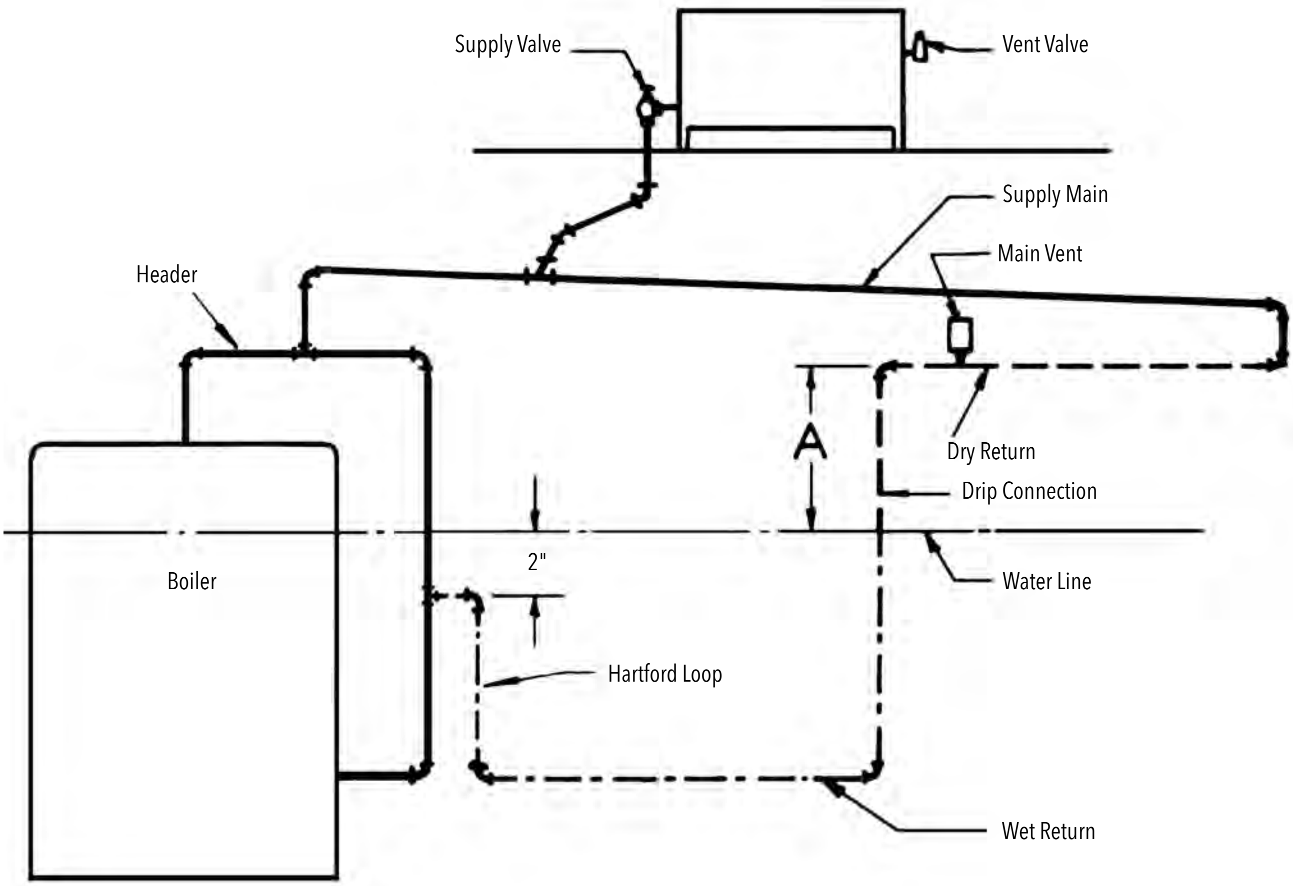

Figure 3
Basic one-pipe up-feed system

Steam heating systems fall into two basic classifications- One-Pipe Systems and Two-Pipe Systems. These names are descriptive of the piping arrangement used to supply steam to the radiator or heating unit and to return condensate from the unit. It is a One-Pipe System when the heating unit has a single pipe connection through which steam flows to it and condensate returns from it at the same time. In a like manner, it is a Two-Pipe System when the heating unit has two separate pipe connections-one used for the steam supply and the other for the condensate return.

Many of the advantages, previously described, of using steam as the heating medium are applicable to the One-Pipe System.

Modern, automatically-fired boilers promote rapid steaming which assures quick pick-up of the space temperature from a cold start. The natural circulation of steam in the system, in combination with the simplicity of piping and air venting, makes this type of system the most economical as well as a desirable method of heating.

A One-Pipe System, properly designed for gravity return of the condensate to the boiler with open type air vents on the end of mains and on each heating unit, requires a minimum of mechanical equipment. The result is a low initial cost for a very dependable system.

The modern One-Pipe System shown in **Figure 3** is a typical illustration of a simple up-feed, gravity one-pipe system. The basic equipment used is described as follows:

ONE-PIPE STEAM HEATING SYSTEMS

BASIC EQUIPMENT OF A ONE-PIPE HEATING SYSTEM

1. A STEAM BOILER (Figure 4)

It is usually automatically fired and equipped with suitable controls to maintain system pressure. It also has the required safety devices or proper burning of fuel, and should be equipped with an automatic water feeder and low water cut-out.

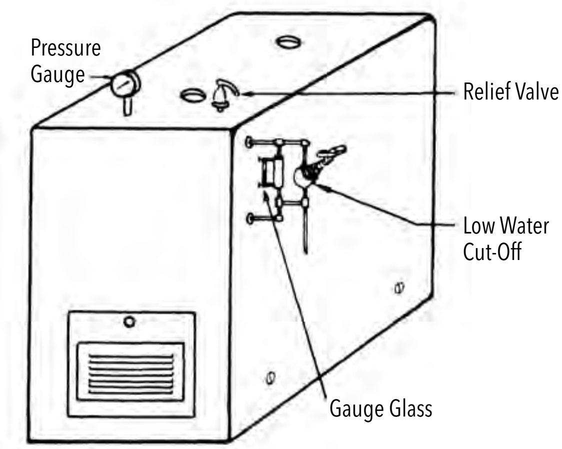

Figure 4

2. HEATING UNITS (Figure 5)

One-Pipe Systems can be designed to use convectors, cast iron radiators, wall fin-tube, and similar heat output units.

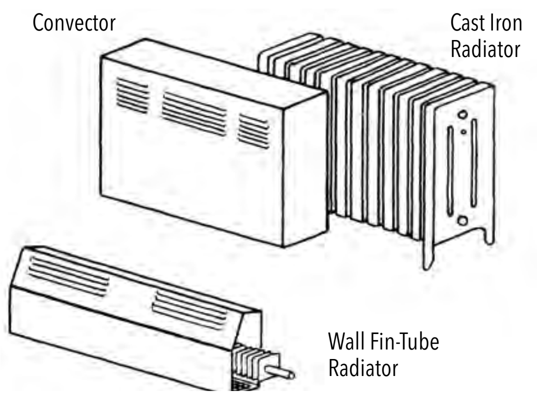

Figure 5

3. AIR VENTS (Figure 6)

Steam cannot circulate or radiators heat until air has been vented from the system. Each heating unit and the end of each steam main must be equipped with an air vent valve.

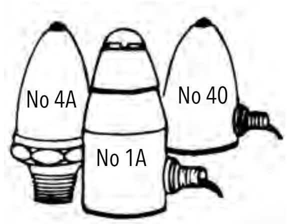

Figure 6

4. RADIATOR VALVES (Figure 7)

Each radiator must be equipped with an angle pattern, radiator supply valve installed at the bottom inlet tapping.

Figure 7

OTHER PARTS OF A ONE-PIPE SYSTEM

The typical system illustrated in **Figure 3** also shows the piping components of a One-Pipe System. The names and functions of these components are as follow:

HEADER: Boilers, depending on their size, have one or more outlet tappings. The vertical steam piping from the tapped outlet joins a horizontal pipe called a "Header". The steam supply mains are connected to this Header.

STEAM SUPPLY MAIN: The supply main carries steam from the boiler to the radiators connected along its length, as shown by **Figure 3**. It also carries condensate accumulation from these units back to the drip connection. When the condensate flow in the supply main is in the same direction as the steam flow, as illustrated, the system is called a parallel flow system.

DRIP CONNECTION: The drip connection is the vertical length of pipe connecting the remote end of the steam supply main to the wet return.

WET RETURN: The return piping which carries the condensate accumulation back to the boiler and is installed below the level of the boiler water line is called a wet return. It is completely filled with water and does not carry air or steam. When the system is first filled with water or is cold, the pressure throughout the system is the same, or balances. Therefore, the water is at the same level in the boiler and drip connection as indicated by the boiler water line.

DRY RETURN: The dry return is that portion of the return main located above the boiler water line. In addition to carrying condensate it also carries steam and air. The end of the Dry Return must be located at the proper height to maintain the minimum required distance for Dimension "A" above the boiler water line.

ONE-PIPE STEAM HEATING SYSTEMS

OPERATING PRINCIPLES OF A ONE-PIPE SYSTEM

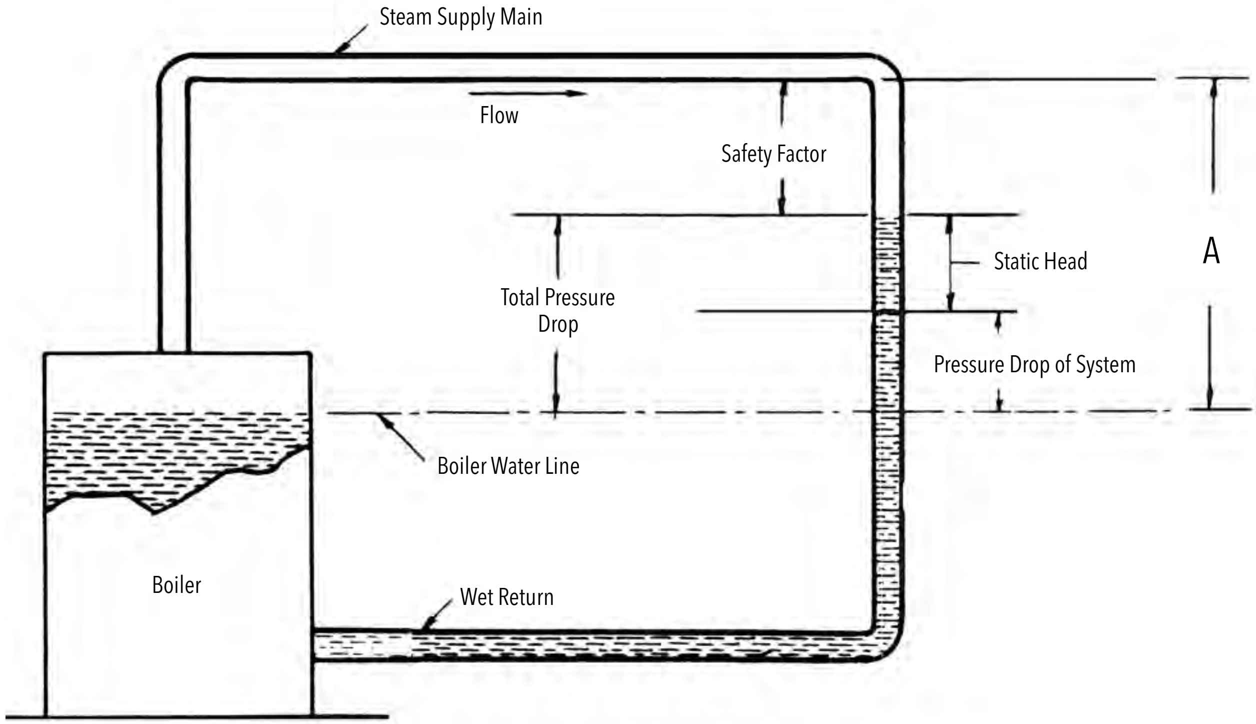

Figure 8
Pressure drop in a one-pipe steam system

The illustration shown in **Figure 8** will be used to describe important operating principles of a one-pipe system. Steam is generated in a boiler when fuel is burned and heat added to the boiler water. This causes an increase in steam pressure at the boiler which results in the flow of steam to the heating units.

As steam flows through the Steam Supply Main, there is loss in steam pressure due to the resistance to flow caused by pipe friction. An additional pressure loss is caused by the condensing process in the steam main, other piping and radiators. The sum of these pressure losses is the Pressure Drop of the system. This pressure drop results in a lower pressure on the surface of the water in the drip connection at the end of the Steam Supply Main as compared to **Figure 8** the higher pressure acting on the surface of the water in the boiler. This difference in pressure causes the water to rise in the drip connection, and the measured difference between these two levels is the Pressure Drop of the system, as shown in **Figure 8**. The end of the steam supply main must be a minimum distance above the boiler water line for any gravity return one-pipe system. Two other factors must be considered and the distance they represent must be added to the system pressure drop to obtain the proper distance. The Static Head represents the height of water required to return the condensate to the boiler. The safety factor represents an additional height to allow for unusual heating up conditions.

HOW TO DETERMINE DIMENSION "A"

The following examples will show how to determine values for arriving at the location of the end of the Steam Supply Main above the boiler water line, denoted by dimension "A". **See Figure 8**.

1. For a small system having a total heat loss not more than 100,000 BTU /Hr., piping is sized on the basis of 1/8 PSI. Therefore, the three distances will be as follows:

Pressure Drop of System (1/8 PSI)	= 3 1/2" of water
Static Head (Friction of wet return)	= 3 1/2" of water
Safety Factor (Twice the Static Head)	= 7" of water
Total distance	= 14" of water

For a small system it is standard practice to make the minimum distance for Dimension "A" not less than 18".

2. For a larger system, assume that the piping was sized for a total pressure drop of 1/2 PSI. The three distances would then be as follows:

Pressure Drop of System (1/2 PSI)	= 14" of water
Static Head (Friction of wet return)	= 4" of water
Factor of Safety (Twice the Static Head)	= 8" of water
Total distance	= 26" of water

It is standard practice for a system based on 1/2 PSI pressure drop to make the minimum distance for Dimension "A" not less than 28".

ONE-PIPE STEAM HEATING SYSTEMS

HARTFORD LOOP

The Hartford Loop is a special arrangement of return piping at the boiler. Its purpose is to reduce likelihood of an insufficient quantity of water creating a low water condition which can cause damage to a steam heating boiler. It came into general use in 1919 and was primarily designed for use with heating systems having gravity wet returns.

Prior to its development it was customary to install a Check Valve in the Wet Return. This Check Valve was a continual source of trouble. In the first place, it added resistance to the return piping which increased the pipe friction in the return piping. It was often noisy due to the chattering of the check valve disc, and was subject to malfunction because of dirt and scale lodging under its seat. Should a system leak occur in the Wet Return and the Check Valve fail to close, the boiler would not be protected against damage caused by a low water condition in the boiler. The Hartford Loop, which corrected this condition, is explained as follows:

The application of the Hartford Loop to our simple piping system is shown in **Figure 9**. In effect, it consists of two loops of pipe forming two U-tubes. The first loop is around the boiler and the second loop is composed of the Drip Connection, the Wet Return, and a short riser called the Loop Riser. In the first loop, an Equalizer line runs from the boiler header down the side of the boiler to the boiler return connection. A pressure balance is maintained in this loop because the steam pressure on the top of the water in the Equalizer pipe is the same as the pressure on the top of the water in the boiler.

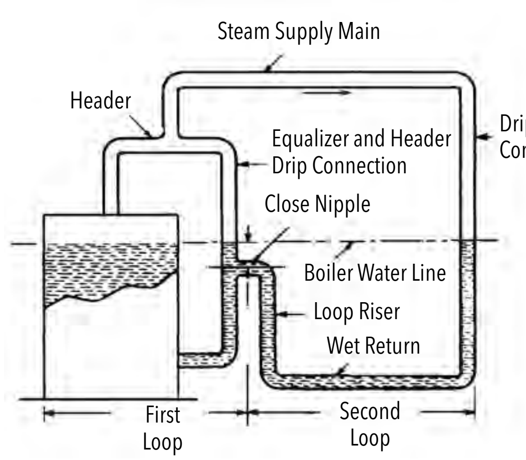

Figure 9
The Hartford Loop

In the second loop, the short riser is connected to the Equalizer pipe with a Close Nipple, the center of which must be not less than 2" below the boiler water line. Here again, there is a balance between the pressure on the top of the water in the Equalizer pipe, in the boiler, and in the Drip.

Should a leak occur in the Wet Return or bad operating conditions be experienced, the boiler could drain down only until the water line fell to the bottom of the nipple connection between the Loop Riser and the Equalizer pipe. Sufficient water will still remain in the boiler to cover the crown sheet or section and prevent damage to the boiler.

It is important to use a Close Nipple to construct the Hartford Loop. If a long nipple is used at this point and the water line of the boiler becomes low, water hammer noise will occur. Some designers prefer using a "Y" fitting which, if used, also must be 2" below the boiler water line.

The use of a Hartford Loop is not recommended in any system where the condensate is returned to the boiler by a condensate pump. It can be a source of noise resulting from the introduction of relatively cold water to the boiler at the hottest boiler water temperature.

ONE-PIPE STEAM HEATING SYSTEMS

TYPES OF ONE-PIPE HEATING SYSTEMS

One-Pipe steam heating systems can be described by the piping arrangement as being one of several different types. When the condensate returns directly to the boiler it is called a Gravity System. This is in contrast to the system that requires mechanical means, such as a Condensate Pump, to return the condensate to the boiler.

To insure proper flow of steam, air and condensate in a One-Pipe system the proper slope or pitch must be given to the steam supply and dry return mains. They must be uniformly pitched not less than one inch in 20 feet in the direction of the gravity flow of condensate. No pitch is required for Wet Returns. Typical examples of different piping arrangements and types are shown and explained as follows:

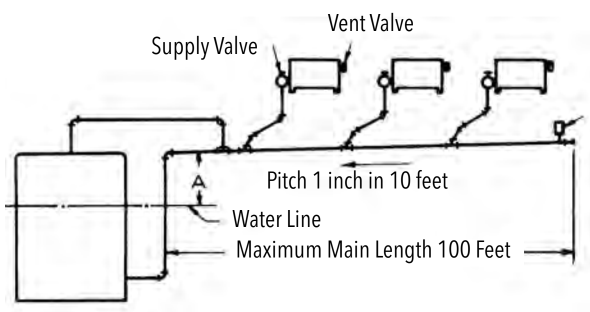

Figure 10

COUNTER-FLOW SYSTEM

In this type of system, steam flows counter to, or in the opposite direction to, the condensate flow. This type of system is limited to small residential installations, especially those with unexcavated basements.

The Steam Main must be pitched upwards from the boiler towards the end of the main one inch in 10 feet to facilitate proper condensate return to the boiler. The size of the Steam Main will depend on the load requirements and must be *one pipe size larger* than that required for other types on One-Pipe Systems. The Dimension "A" at the Boiler must be of sufficient height above the Boiler Water Line to return the condensate to the boiler as previously described and as shown in **Figure 8**.

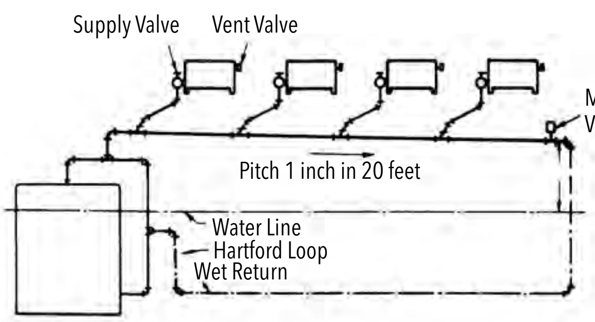

Figure 11

PARALLEL FLOW SYSTEMS

In this type of system, steam and condensate flow in the same direction in the horizontal steam or return mains. **Figure 11** shows the piping arrangement for a Parallel Flow System with Wet Return. The condensate returns to the Boiler through a Hartford Loop**.** **Figure 12** shows a Parallel Flow System with a Dry Return. At the end of the Dry Return Main it drops below the Boiler Water Line to become a Wet Return and the condensate is returned to the boiler through a Hartford Loop. In both Dry Return and Wet Return systems Dimension "A" must be maintained to a proper height above the Boiler Water Line to return the condensate by gravity to the Boiler.

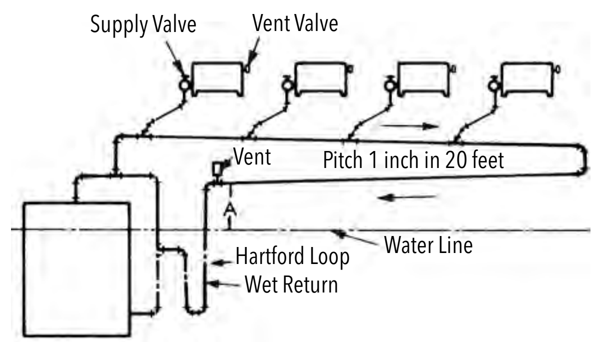

Figure 12

PARALLEL FLOW UP-FEED SYSTEM

This type of system is installed in buildings which have more than one floor. Steam is distributed from a basement main. **Figure 13** shows the piping arrangement for a three-story structure. It uses a Wet Return with the end of the Main and the heel of each Up Feed Riser dripped to the Wet Return. This method of connection is desirable as it keeps the horizontal main free of condensate accumulations and assures unobstructed free flow of steam. The Up-Feed Risers supply steam to the second and third floor radiators. **Figure 14** shows details of the drip connections.

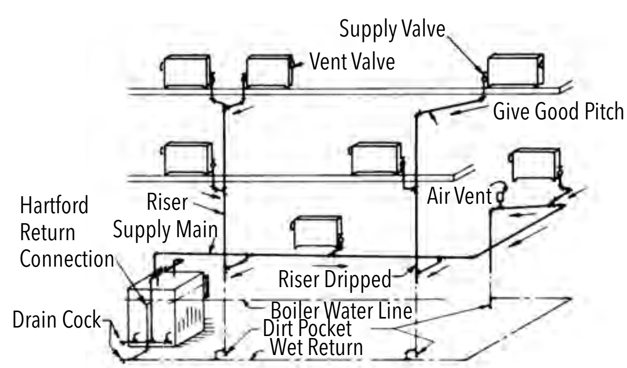

Figure 13

ONE-PIPE STEAM HEATING SYSTEMS

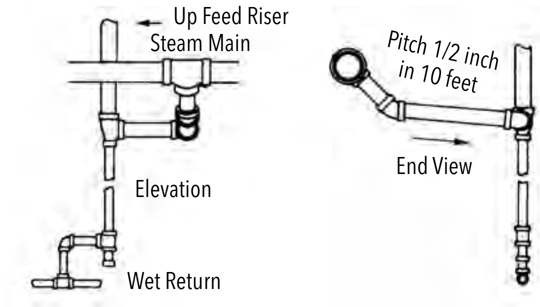

Figure 14

The Up-Feed Branch Connection to the first floor radiators are not dripped. **Figure 15** shows the details of these connections.

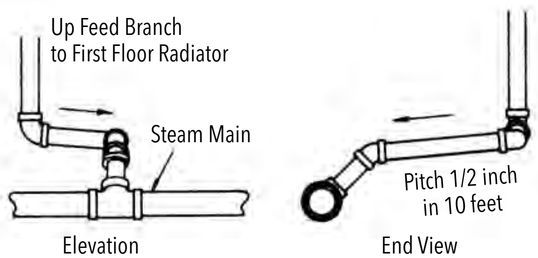

Figure 15

If the up-feed runout to a riser is not to be dripped to the Wet Return, the horizontal pipe must be increased one pipe size and a greater pitch provided as shown in **Figure 16.**

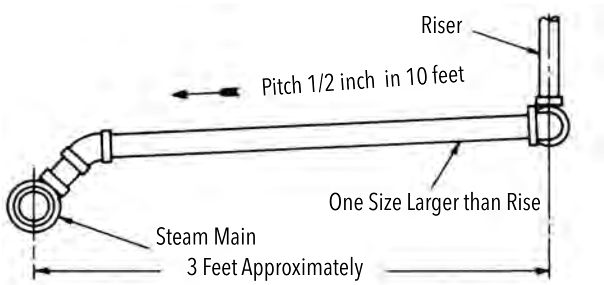

Figure 16

The two methods of connecting a branch or runout to an up-feed riser are shown by **Figure 17**. The connection at 45° provides less obstruction to the free flow of steam to the riser which must also handle the reverse flow of condensate from the riser.

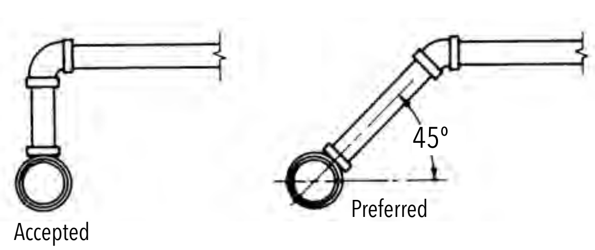

Figure 17
Method of taking branches from main

PARALLEL FLOW DOWN-FEED SYSTEM

When a one-pipe system distribution main is overhead above the radiator, such as in an attic or ceiling space, it is known as a Down-Feed System. It is a parallel flow system but, unlike the Up-Feed System, the Down-Feed Risers have steam and condensate flowing in the same direction, as shown in **Figure 18**.

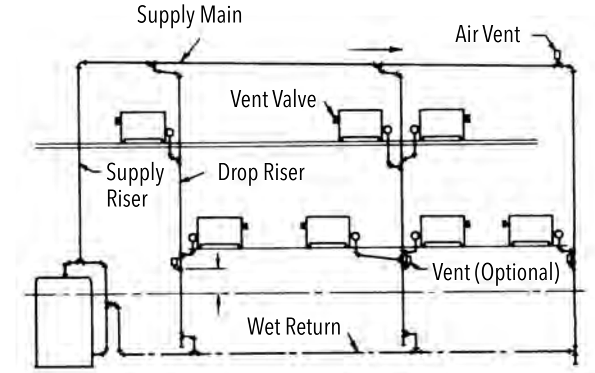

Figure 18

To obtain good performance, the main steam supply riser should be located so it can run directly between the boiler header and the overhead supply main. All down-feed runout connections are taken from the bottom of the horizontal supply main to assure the least accumulation of condensate in the main. The piping details for these runouts from the bottom of the main are shown by **Figure 19.** The Main Vent can be installed at the end of the horizontal main or on the Down-Feeder Risers below the first floor radiator as an optional location. When they are installed on the riser they must be located so that the Dimension "A" is maintained at the proper height above the Boiler Water Line to prevent the Vent Valve from being closed by water rising to a sufficient height to raise the Vent Valve float.

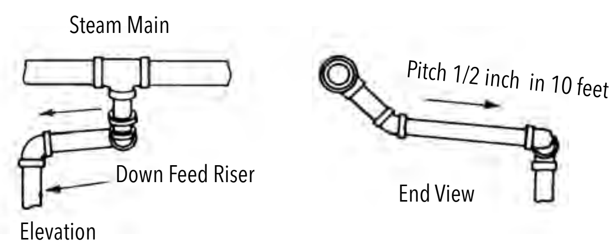

Figure 19

RETURN OF CONDENSATE TO BOILER BY MECHANICAL MEANS

When there is insufficient height to maintain Dimension "A" at its minimum above the Boiler Water Line so condensate cannot be returned directly to the Boiler by gravity, the use of a Condensate Pump becomes necessary, as shown in **Figure 20**.

ONE-PIPE STEAM HEATING SYSTEMS

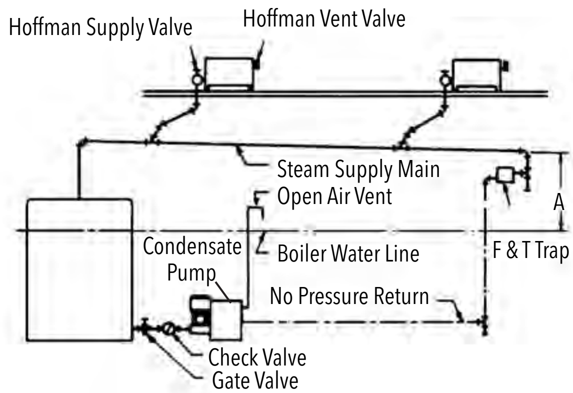

Figure 20
One-pipe system with condensate pump

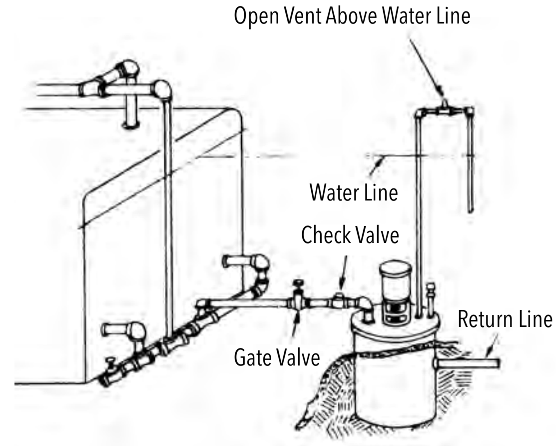

Figure 21
Underground condensate pump

When a Condensate Pump is used, the Boiler Pressure, the end of the Steam Main Pressure, or Boiler Water Line elevation, have no bearing on the height of the end of the Steam Main as long as it is above the maximum water level in the Condensate Pump Receiver. The Return Main begins at the discharge of a Float and Thermostatic Trap located at the end of the Steam Supply Main, as shown. The Trap is sized to handle the entire maximum condensate load of the system if it is a single main, or the connected load of each, individual main. It functions to discharge air and condensate accumulations into the return, and to close against the passage of steam. The Return Main for this type of system is sometimes referred to as a "no pressure return" because the open vent on the Condensate Pump Receiver maintains it at atmospheric pressure.

The piping must be uniformly pitched to the Pump Receiver without pockets which will trap air and prevent the gravity flow of condensate to the Receiver. The pump discharge is connected directly to the boiler return opening without the use of a Hartford Loop, which can cause noise when used with a pumped discharge of condensate. It is not good practice to install the discharge of a Float and Thermostatic Trap directly into, or close to, the Pump Receiver because the flash of steam vapor may be sufficient to affect the pump operation. It is not good practice to cover the return piping when a pump is used to return the condensate to the boiler. For best pump operation, the condensate temperature in the Receiver should not exceed 200°F.

USE OF UNDERGROUND CONDENSATE PUMP

Often there are construction conditions which will not permit the condensate to be returned by gravity to the condensate pump receiver if it is installed above the floor as shown in **Figure 20**. An Underground Condensate Pump installed below the floor level is then used as shown in **Figure 21**. The "no pressure return" main is also installed below the floor level in a trench so that the condensate will flow by gravity to the inlet of the Underground Receiver. The Underground Pump can be installed close to the boiler or at a remote location as required. For a remote installation, the pump discharge pipe can be installed at the ceiling or can run in the trench below the floor back to the boiler return connection

BOILER RETURN TRAP INSTALLATION

The use of a Boiler Return Trap is another method used to return condensate to a low pressure heating boiler by mechanical means. This method is more costly than the use of a condensate pump. **Figure 22** shows the installation of a Boiler Return Trap and is included for information purposes only. When it becomes necessary to service or change such installation it is recommended that a condensate pump be used, as shown in **Figure 20**.

Figure 23 illustrates the piping arrangement and the equipment used for a down-feed one-pipe Paul System. This system can be designed for either an up feed or down-feed system having gravity return of condensate to the boiler, or the condensate can be returned to the boiler using a condensate pump as shown. The important difference of the Paul System from that of an ordinary one-pipe system is the elimination of air from each radiator using vacuum created by a motor-operated air-line pump, or by using either a steam- or water-operated air exhauster using a jet principle for creating a vacuum by removing air.

Each radiator is equipped with an air-line valve installed on the end of the radiator, or other type heating unit, opposite the radiator supply valve. The air from the radiators is discharged through the airline valve into a separate piping system, maintained under vacuum for the fast removal of air. This airline piping does not carry the condensate from the radiator.

When the system is operating, the air line valve remains open for the removal of air from the radiation due to the vacuum in the air-line piping. This results in fast circulation of steam to each radiator. When steam reaches the air-line valve, it closes. It continues to remain closed until air accumulations cause it to cool sufficiently to open again so air can be removed and steam can fill the space it occupied.

ONE-PIPE STEAM HEATING SYSTEMS

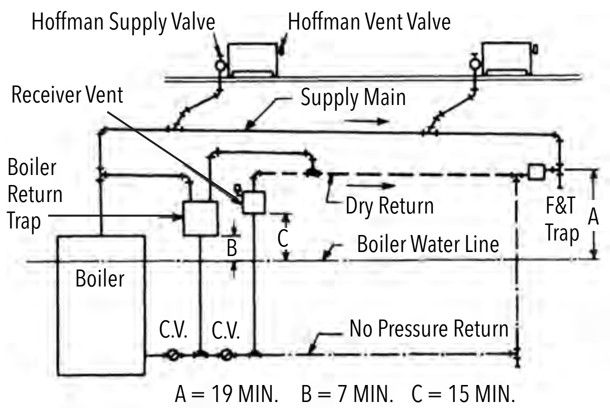

Figure 22
One pipe system with boiler return trap

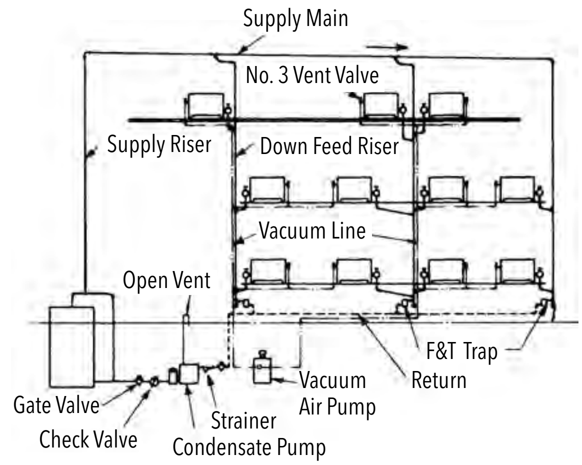

Figure 23
One-pipe air line Paul System

BOILER RETURN TRAP INSTALLATION

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ONE-PIPE STEAM HEATING SYSTEMS

CONTROL AND DISTRIBUTION OF STEAM

The distribution of steam in a One-Pipe system can be accomplished by varying or controlling the venting rate at which air leaves the radiator. It cannot be accomplished by throttling the Radiator Supply Valve. The following discussion will explain the proper use of the Radiator Supply Valve and the Radiator Vent Valve.

RADIATOR SUPPLY VALVE

Each radiator or heating unit is equipped with a Radiator Supply Valve installed at the bottom inlet connection. This Radiator Supply Valve should be an angle type, globe pattern valve because steam must enter the heating unit through this valve, and condensate must leave the unit through the same valve and at the same time. Never use a straight through type, globe pattern valve in the horizontal because it has a natural obstruction to the free flow of steam and condensate. This Radiator Supply Valve can be used only to turn the radiator "off" completely, or to turn it "on" full open. It cannot be used in a throttled position to accomplish control of steam distribution. The reason can be explained by referring to **Figure 24** and **Figure 25**. A throttled valve creates objectionable noise and if there is a sufficient number of valves left in throttled positions, condensate cannot be readily returned to the boiler.

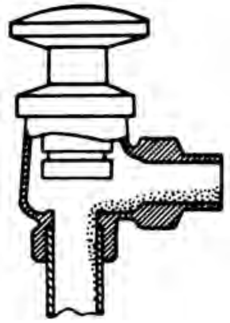


Figure 24

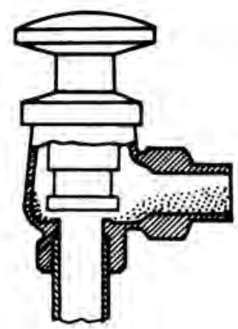


Figure 25

VENT VALVE

The device used to eliminate air properly from a One-Pipe Steam System is called a Vent Valve. Steam cannot reach or enter the radiator as long as air is present in sufficient quantity to cause air binding which will block the flow of steam to the radiator. Each radiator must be equipped with an air vent valve installed in the vent tapping located on the end opposite the Supply Valve. A Main Vent should be installed at the end of the Steam Supply Main or at any other location where piping arrangement will cause air accumulation.

RADIATOR VENT VALVES

Radiator Vent Valves are designed for use in low pressure steam systems. They are available to cover a wide range of air venting capacities related to their operating pressure. The operating pressure is that pressure at which the valve will always function to open and close the vent port during the venting cycle. The operating pressure range is as low as 1 PSI to a maximum of 11 PSI. The higher the operating pressure for any particular valve, the lower will be its venting capacity. They are made in two basic types:

1. Non-vacuum or open vent type. It is made with a single port which is non adjustable. It is also available with an adjustable port which is used for proportional venting.
2. A vacuum type. It has an adjustable port which is used for proportional venting.

The functions of a Vent Valve are:

1. To permit air to be pushed out or vented by the pressure created at the boiler so steam can occupy the space in the piping and heating units.
2. To close when steam reaches it and prevent its escape through the vent port.
3. To close when water reaches it and prevent its escape through the vent port.
4. To re-open when steam temperature has dropped sufficiently, or water has drained away to permit the air venting process to continue.
5. Vacuum Vent ONLY -An additional feature to prevent the return of air to the system.

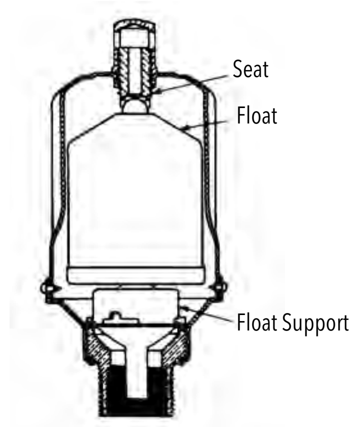


Figure 26

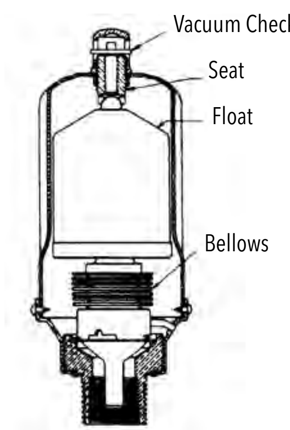


Figure 27

ONE-PIPE STEAM HEATING SYSTEMS

END OF MAIN VENT

The use of End of Main Vents is particularly important for large One-Pipe Systems. The quick venting of air from the horizontal steam supply main assures prompt distribution of steam to the vertical runouts to first floor radiators and to risers which supply steam to radiators on upper. floors. An End of Main Vent is always made with a single port and has a much larger venting capacity than a Radiator Vent. **Figure 26** shows the construction of non-vacuum, or open vent type. It has all the features of smaller radiator vents and functions in the same manner. Main Vents are never constructed for adjustable, variable venting because once all the air has been expelled from the steam main, the vent port is closed and has no further function until the next boiler firing or heating-up cycle occurs.

Figure 27 shows the construction of a vacuum type End of Main Vent. It has all the features of a smaller vacuum type Radiator Vent and functions in the same manner.

INSTALLATION OF END OF MAIN VENTS

End of Main Vents should always be installed near the end of the steam main, as shown by **Figure 28**. It is never good practice to install a Main Vent on the last fitting at the end of the steam supply main. When incorrectly installed as shown by **Figure 29**, the float can be damaged by water surge which can create a very high pressure. Although this high surge pressure lasts only a fraction of a second, it is long enough to cause the float to be collapsed. This damage is referred to as damage by "Water Hammer."

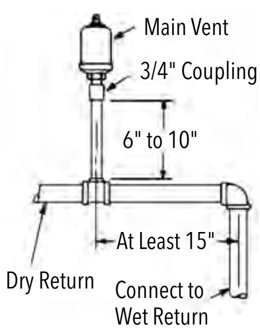


Figure 28

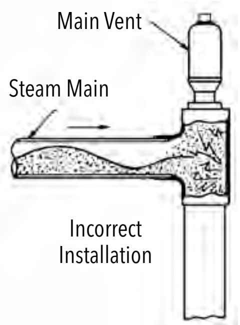


Figure 29

PROPORTIONAL VENTING

It is not possible to get steam to all radiators at the same time using single port air vents. Some radiators are larger than others, requiring a large quantity of air to be vented before they are fully heated, and those closest to the boiler receive steam first, and are often fully heated before the most remote, or farthest a way from the boiler, receive any steam. This condition is not too serious for a small one-pipe system but is undesirable and more serious for a reasonably large one-pipe systems. This condition can be corrected by the use of adjustable vent valves which permit modulation or proportional venting. **Figure 30** shows such construction of a 1A Valve having these variable venting rate features.

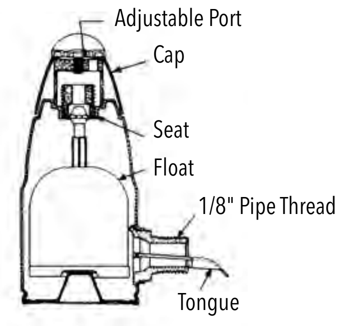


Figure 30

It is apparent that by using this vent valve on all radiators the venting rate of air from radiators near the boiler can be controlled to a minimum by selecting the smallest port. For the remote radiation the venting rate can be controlled by selecting ports which will control the venting rate up to the maximum. It is, therefore, possible to make adjustment so all radiators will receive steam about the same time.

NON-VACUUM AND VACUUM ONE-PIPE SYSTEM

The vent valves described previously have been of the open vent type. Either the single port or the variable modulating port are suitable for use in a non-vacuum type system. These systems operate in a pressure range from zero gauge pressure when the system is cold to a higher pressure, usually in the range of 2 to 3 PSIG. As has been pointed out, steam cannot enter a radiator until the air has been vented. The air is pushed ahead of the steam under a pressure sufficient to reach the farthest radiator, and when all the air is vented, the steam fills the space in the radiators which was filled with air. When the pressure drops to zero PSIG, or atmosphere, air will again enter the radiator from which it will again be vented during the next firing cycle, which is usually in response to action of a room thermostat.

A vacuum system operates over a pressure range from below atmosphere to a higher pressure above atmosphere, which is usually in the range from 2 to 3 PSIG, the same as required for the non-vacuum system. The vacuum, or pressure below atmosphere, in this system is not produced mechanically but is induced by the cooling of the radiators to a temperature lower than the temperature when filled with steam at atmospheric pressure.

CONTROL AND DISTRIBUTION OF STEAM

It is readily apparent that in order to have an induced vacuum in this type of system, the radiator must first be filled with steam under pressure, the same as required for a non-vacuum system. In addition, the radiators must be equipped with vacuum type Air Vents such as shown in section in **Figure 31.** This vacuum vent has all the features required of an ordinary air vent plus the very important feature for preventing the return of air through the vent valve. This feature includes a vacuum check which permits the induced vacuum to form, once the radiator has been filled with steam and begins to cool on the "off" firing cycle. When this occurs, the. vacuum diaphragm in the base of the valve snaps upward to raise the float so the float needle, or valve, will engage the port and close it. This Vacuum Vent Valve is also equipped with six variable ports which can be adjusted so all radiators will receive steam at about the same time. In a vacuum system, all vents on the radiators and the main vents must be of a vacuum type.

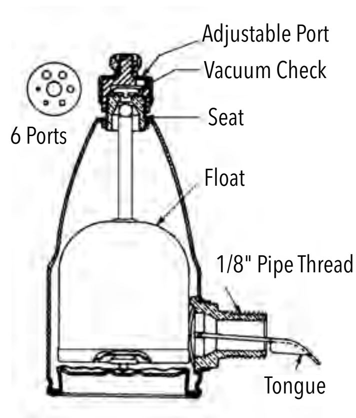


Figure 31

A One-Pipe Vacuum System was an economical and comfortable way to heat when coal-fired steam boilers were very common. However, since automatic oil or gas-fired boilers replaced most of the coal-fired installations, the use of a One-Pipe Vacuum System has not been as ideally suited as it had been during the coal-fired era. An understanding of what takes place under the two methods of firing a steam boiler will help to make clear the difference in performance of a vacuum system.

1. **Coal-Fired System:** In a coal-fired system, either stoker or hand-fired, after the drafts have been closed or the thermostat has stopped the stoker, there is still a large amount of heat stored in the fuel bed. This heat is transferred to the water, which is kept boiling. When the temperature of the boiler water drops, the pressure goes below atmosphere as the induced vacuum is created. There is enough heat stored up in the fuel bed to feed the radiators for a considerable length of time. The speed with which the system pressure drops to a vacuum depends not only on the heat required by the room, but also upon the heat retained in the fuel bed and the heated parts of the boiler. If there is a large amount of heat, as in the coal-fired installation, the vacuum will form rather slowly because cooling will take place gradually. As soon as the firing is again started or the dampers opened, the temperature of the boiler increases and steam will flow to the radiators.

   The room thermostat may be satisfied before the radiators are completely heated through and the firing device stopped. This will still permit the system to go to vacuum, but if this happens on the first cycle, the radiators will all be partly filled with air. As long as air is confined to the radiators no harm is done, but if this entrapped air should escape into the main, distribution would be affected when the next firing cycle starts. The higher the vacuum and the faster it is formed, the greater the chances of this taking place. The flow formation of the vacuum in a coal-fired system, however, is not apt to disturb the relationship between air and steam.

2. **Oil- and Gas-Fired Systems:** In either a gas- or oil-fired installation there is no fuel bed, and as soon as the thermostat has extinguished the gas flame or the oil flame, the heat input from that source ceases immediately. The only heat storage capacity is the metal parts and in some cases the refractory in the fire box of the boiler. This does not amount to a great deal of BTU content. The result is that the system goes to vacuum much quicker than a coal-fired system would, and in all probability reaches a higher vacuum, provided always that the system is tight. If each radiator of the system was completely heated, and all of them were full of steam when the vent valves shut,no harm can come from this rapid descent of pressure and temperature. If, however, in the first cycle, the firing device was shut off by the room thermostat before the radiators were heated through, then the air content may prevent effective steam distribution on the next cycle. The vacuum forms so quickly that the air which is left in the radiators expands rapidly in a gas- or oil-fired system. It might even go to a higher vacuum than in a coal-fired installation, in which case the air would expand more and might occupy the entire space in the radiator. In fact, it might creep down through the risers into the mains. This would seriously affect the distribution during the next cycle.

In addition, the cycling in an oil- or gas-fired installation occurs at shorter intervals than in a coal fired, and therefore, the opportunity of mixing air with the steam and letting it go into the mains occurs more frequently in the oil- or gas-fired system than in the coal-fired. For this reason, vacuum vent valves are not ordinarily recommended for use with gas- or oil-fired installations.

ONE-PIPE STEAM HEATING SYSTEMS

Occasionally, One-Pipe Steam Heating Systems are supplied from a high pressure source, such as a District Heating Main (commercial or street supply service). The high pressure steam from this source is reduced to a suitable, low pressure for use in the heating system by means of a Pressure Reducing Valve. For this type of service, the End of Main Vent should be a high pressure type such as recommended for Unit Heaters. The high pressure vent will open and close to expel air as long as its design pressure is not exceeded. **Figure 32** shows the proper method of installing a low pressure Main Vent and a high pressure vent to cover a wider range of venting demands for high pressure systems supplied with steam from District Heating mains. The regular low pressure vent will take care of the initial, large accumulation of air until it is closed. The high pressure vent will continue to operate to take care of smaller accumulations at the higher system pressure.

There are times when large capacity steam supply mains require more than one End of Main Vent to take care of the initial air accumulation. This same piping arrangement can be used for two main vents of the same size or venting capacity.

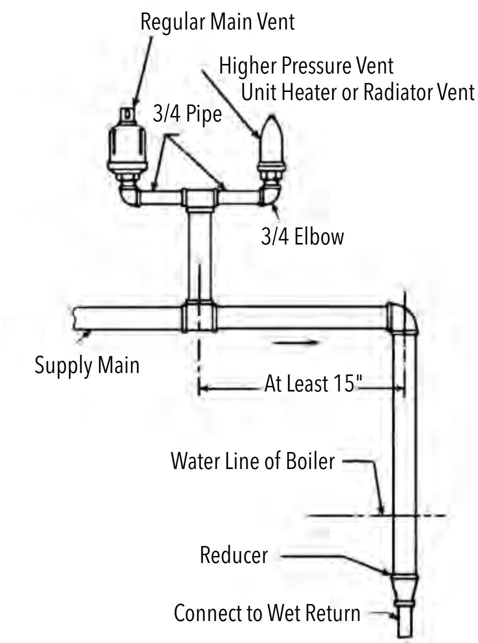


Figure 32

PIPING FUNDAMENTALS

There are essential factors which must be considered in the design and installation of the piping for a One Pipe Steam System. To insure proper flow of steam, air and condensate, the steam supply main and the dry return must be installed with a uniform pitch.

Figure 33 shows the proper method of reducing the size of the pipe when installed horizontally. Use an eccentric reducing coupling instead of a regular reducing coupling. The use of the eccentric coupling permits the continuance of uniform pitch without forming a water pocket which does restrict flow.

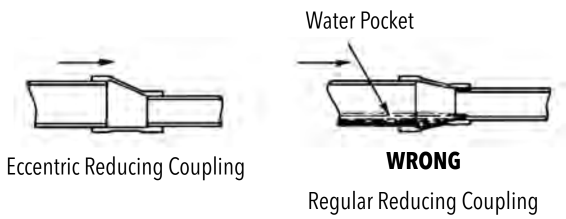


Figure 33

When a horizontal main is installed without uniform pitch, usually due to improper or insufficient support, it can cause a water pocket due to a sag in the piping. **Figure 34** shows such a condition and it is evident that the free flow of steam and air is impossible until the condition is corrected. Such a sag can cause noise when steam reaches the water pocket on a cold start. Also, depending on its location, it can be responsible for water hammer noise and, sometimes, destructive damage to valves and vents.

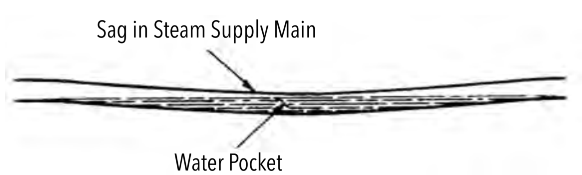


Figure 34

There are times when it is necessary to install a steam main around a construction obstruction such as a steel beam or girder. The details of the proper method of piping around such an obstruction is shown in **Figure 35**. Steam and air will flow above the obstruction and the condensate will flow below.

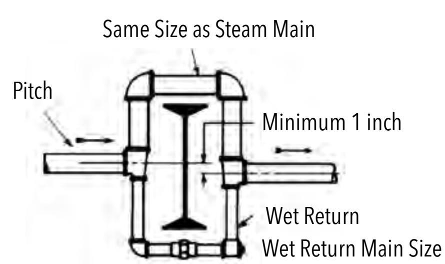


Figure 35

Another type of obstruction is a doorway which interferes with uniform pitch of a dry return which must be located above the floor. **Figure 36** shows how to pipe around the doorway. The condensate will flow in the piping below the doorway and air will flow through the Air Loop over the top.

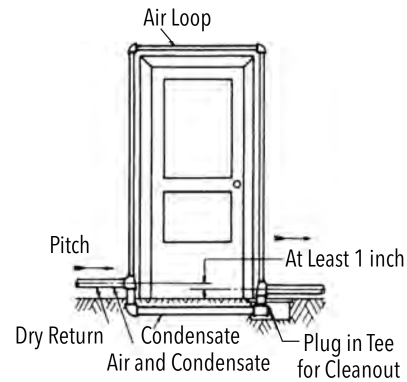


Figure 36

SOURCES OF TROUBLE AT THE RADIATOR

A radiator which does not set level because of a sagging floor, as shown in **Figure 37**, can be the cause of noise and poor heating. Condensate remains trapped in the lower end of the radiator, thus setting up an obstruction to the steam trying to enter the radiator. Water hammer can result from this condition. The radiator should be set as level as possible. On a One-Pipe system, a slight pitch of the radiator towards the supply end is helpful.

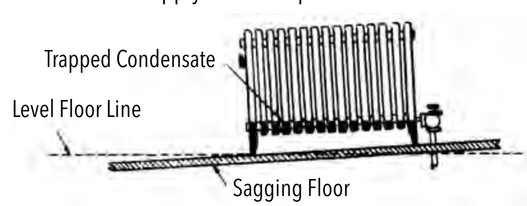


Figure 37

HOW TO ADD A RADIATOR TO AN EXISTING SYSTEM

Figure 38 shows the "wrong" and "right" way to add a new radiator to an existing One-Pipe Steam Heating System. Incorrect installation will cause the supply pipe from the existing radiator to fill with water (condensate) and thus block the passage of steam to the added radiator.

The "right method," allows the steam from the supply line to go directly to the additional radiator. It is very important to pitch this Steam Supply piping so condensate can flow back to the return system.

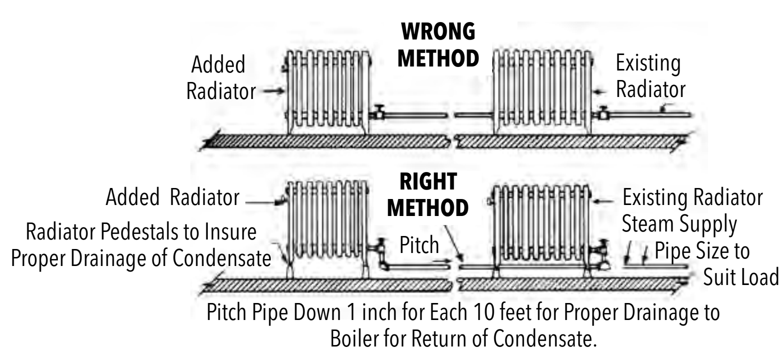


Figure 38

TWO-PIPE STEAM HEATING SYSTEMS

In the first part of the manual, One-Pipe Systems are discussed and the various system types, components, and installation practices are described and illustrated.

Although a Two-Pipe System has fundamental differences from those of a One-Pipe System, many components and piping installation practices are common to both systems. Also, the Two-Pipe System employs many advantages of using steam as a heating medium. They are applicable to a variety of structures from small residences to large commercial buildings, office buildings, apartment, and industrial complexes.

Two-Pipe Systems are designed to operate at pressures ranging from sub-atmospheric (vacuum) to high pressure. Although they use many practical piping arrangements to provide up-flow or down-flow systems, they are conveniently classified by the method of condensate return to the boiler. Condensate can be returned to the boiler by gravity or by use of any one of several mechanical return means.

By definition, it is a Two-Pipe System when the heating unit (radiator) has two separate pipe connections- one used for the steam supply and the other for the condensate return. Basic descriptions of the various systems, their operating principles, and the proper use and function of the required mechanical equipment will be given in this section of the manual. This will provide the necessary background knowledge to install, maintain, or up-grade older existing systems.

BASIC EQUIPMENT FOR TWO-PIPE SYSTEMS

The mechanical components for Two-Pipe Systems are used in various combinations depending on the type system or its design. Some of these same components are used for One-Pipe Systems and were described for these systems. However, all equipment used for Two-Pipe Systems will be described and discussed as follows:

1. A STEAM BOILER

Boilers are manufactured as either a sectional cast iron boiler or as a steel boiler. The modern boiler is automatically fired using coal, oil, or gas, as the fuel. They are provided with suitable pressure controls, safety firing devices, and protection against damage due to low water conditions.

Package boilers are also extensively used and these are furnished complete, with all equipment factory assembled and tested.

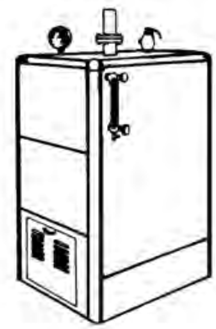


Figure 39

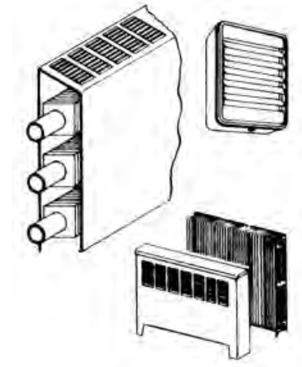


Figure 40

2. HEATING UNITS

Two-Pipe Systems use a variety of heat output units such as cast iron radiators, convectors, wall fin-tube radiation, unit ventilators, and unit heaters.

3. RADIATOR SUPPLY VALVES

The Supply Valve for a Two-Pipe System is installed at the inlet connection. These valves are globe type and made in several different patterns which include angle, straightway, right hand, or left hand. The angle pattern valve is generally used because it is readily adaptable to most heating unit installations. The straightway pattern is easily applied in a vertical connection to a convector when the inlet connection is below the unit. Right and left hand patterns are used when piping connections can be simplified by eliminating extra pipe elbows and pipe nipples. These valves can be obtained with a modulating feature or can be of the non-modulating type.

Depending on the required use and to prevent leakage around the valve stem, they are furnished with several methods of packing: regular packing nut, spring packed, or packless using a metal bellow or a special diaphragm. **Figure 41** shows an ordinary supply valve which uses a packing nut to tighten the stem packing to prevent steam leakage. This valve has a rising stem and requires several turns of the handle to open the valve port fully.

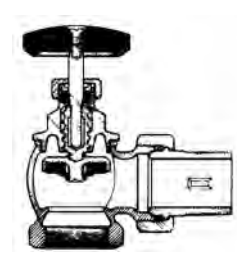


Figure 41

TWO-PIPE STEAM HEATING SYSTEMS

BASIC EQUIPMENT FOR TWO-PIPE SYSTEMS

Figure 42 shows a supply valve which uses a spring packed stem. The stem packing is tightened by the compression of a spring. It has a non-rising **Figure 42** stem, and for non-modulating applications it requires several turns of the handle to open the valve fully. It can also be obtained to open fully with one turn or less and, when equipped with a dial and pointer, it is a modulating type. Modulating valves can also be obtained with a cone or throttling nut at the bottom of the disc face to permit a variable port opening with valve lift or travel.

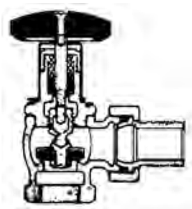


Figure 42

Figure 43 shows a truly packless type supply valve. No packing gland is used and leakage around the stem is prevented by the metal bellows. This valve is always a modulating type valve. Another design of a packless type is shown in **Figure 44** which uses a special diaphragm and lifting mechanism. Both of these packless types are ideal for Vacuum Two-Pipe Systems because their construction prevents air leakage into the system as well as steam leakage from the system.

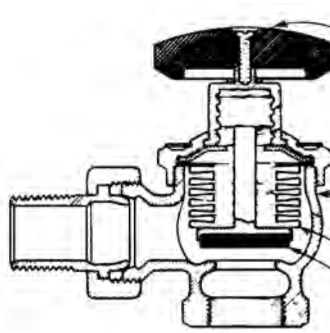


Figure 43

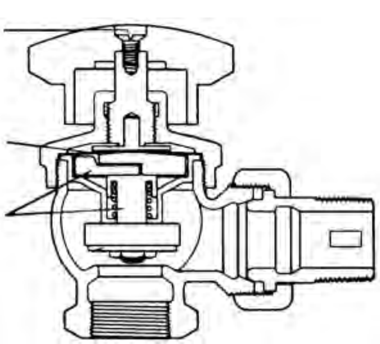


Figure 44

4. THERMOSTATIC TRAPS

Thermostatic Steam Traps are the most common of all types used in Two-Pipe Steam Heating Systems. They are used because they are (a) simple in construction, (b) small in size and weight, and (c) have adequate capacity for usual heating system pressure. They are designed to open in response to pressure and temperature to discharge air and condensate and to close against the passage of steam. The temperature at which a Thermostatic Trap will open is variable but is at the required number of degrees below the saturated temperature for the existing steam pressure. This difference between the saturated steam temperature and the trap opening temperature is called "Temperature Drop."

Balanced Pressure Type Thermostatic Traps are those which have been described. They employ thermal elements made from metal bellows, a series of diaphragms, or special cells made from diaphragms. **Figure 45** shows an example of two types of Radiator Thermostatic Traps. There are some types which employ special shapes of bimetal for the thermal elements.

The radiator return connection is equipped with a Thermostatic Trap. They are also used as drip traps and to handle condensate from fan-coil units such as a unit heater. For these and similar applications, a cooling leg (an adequate length of pipe) must be installed between the equipment or drip and the Thermostatic Trap. The cooling leg permits the condensate accumulation to cool sufficiently to open the trap to discharge the condensate to prevent flooding of the equipment.

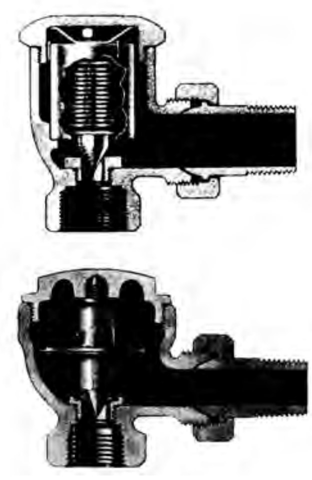


Figure 45

TWO-PIPE STEAM HEATING SYSTEMS

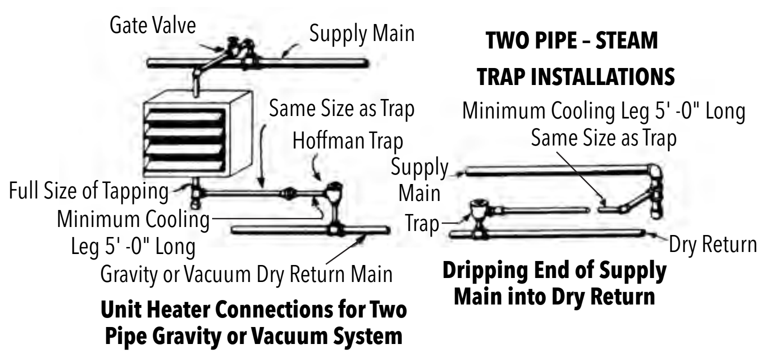


Figure 46

Thermostatic Traps are made in angle, straightway, swivel, and vertical patterns and can be used over a wide range of pressures from subatmospheric (vacuum) to high pressure steam.

5. MECHANICAL STEAM TRAPS

There are several different Mechanical Traps which are used in Two-Pipe Steam Heating Systems. They include Float and Thermostatic Traps, Float Traps, Inverted Bucket Traps, and Open or Upright Bucket Traps. All of these have different operating characteristics which make them applicable to a variety of heat output units, steam main or riser drips.

(a) Float and Thermostatic Traps: A Float and Thermostatic Trap is shown by **Figure 47** and is often called an F&T Trap. This trap opens and closes in response to the raising and lowering of the float due to changes in level of condensate entering the trap body. The discharge of condensate is continuous due to the throttling action of the pin or valve in the seat port. The Thermostatic Air By-pass remains open when air or condensate is present at a temperature below its designed closing temperature. When steam enters the trap body the Thermostatic Air By-pass is closed. An F&T Trap will discharge condensate at any temperature up to a temperature very close to the saturated steam temperature corresponding to the pressure at the trap inlet. For this reason, cooling legs are not required and condensate accumulations are kept free from steam lines or from the equipment being served. An F&T Trap is a first choice selection for dripping the end of a steam main, the heels of up-feed steam risers, and the bottom of down-feed steam risers.

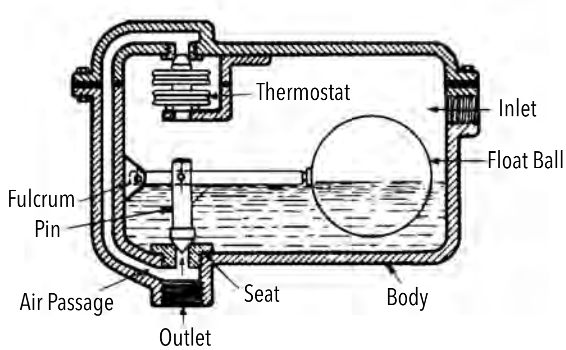


Figure 47

They are also excellent choices for handling the condensate from fan-coil units such as unit heaters, unit ventilators, and ventilating coils. **Figure 48** shows application of an F&T Trap to a unit heater.

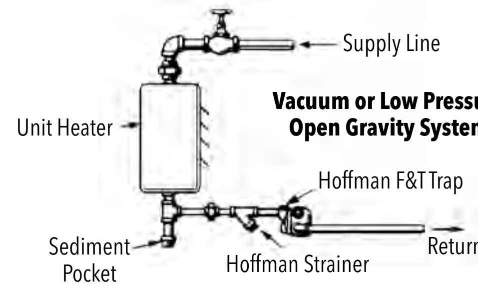


Figure 48

(b) Float Trap: A Float Trap does not have an Internal Thermostatic Air By-pass. There are applications where air is not a problem and they can therefore be used to handle condensate only. There are specifications where a Float Trap is called for, using an external Thermostatic Air By-pass. The external bypass is a Thermostatic Trap piped around the inlet and outlet of the trap body.

(c) Inverted Bucket Trap: Figure 49 shows an Inverted Bucket Trap. It is so named because the "float" is an inverted bucket which operates the leverage mechanism to open and close the trap. This trap will operate to discharge condensate at any temperature up to the saturated temperature corresponding to the steam pressure at the trap inlet. It operates in cycles at a frequency depending on the condensate load being handled. This Inverted Bucket Trap is simple in construction and can be chosen to handle condensate for many industrial types of requirements and will handle condensate from heating fan-coil units which must be lifted to discharge to discharge to return mains located above the equipment as shown by **Figure 50**. Before an Inverted Bucket Trap is put into operation it must be "primed" or filled with water. They operate best at or near full load conditions and where loads do not vary over a wide range.

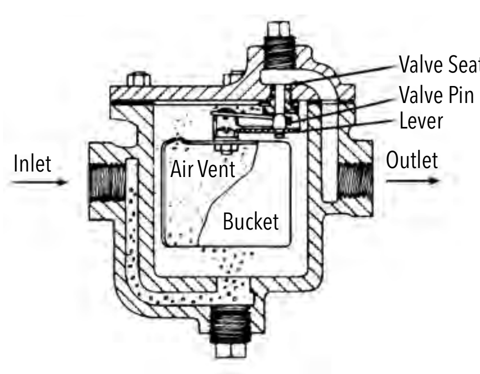


Figure 49

TWO-PIPE STEAM HEATING SYSTEMS

BASIC EQUIPMENT FOR TWO PIPE SYSTEMS

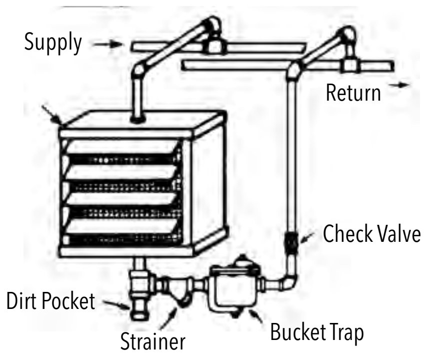


Figure 50

(d) Upright Bucket Trap: Figure 51 shows an Upright Open Bucket Trap. This trap operates on the principle of condensate entering the trap body to "float" the upright bucket. This action closes the discharge port. As condensate continues to enter the trap body it rises to fill the space surrounding the bucket until it overflows into the bucket. When this occurs, the bucket sinks and the discharge port is opened. Steam pressure then discharges the condensate and the operating cycle is repeated**. Figure 51** The construction of this trap is not as simple as that of an Inverted Bucket Trap but the trap will operate to handle applications having wide variations of load or pressure.

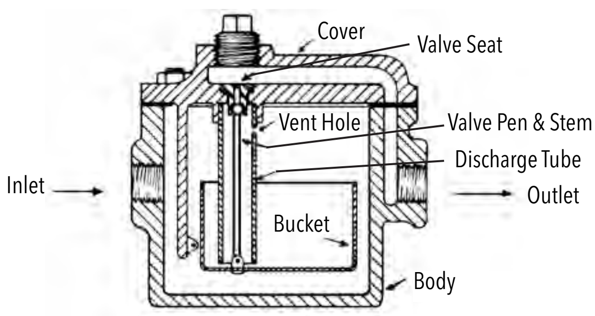


Figure 51

6. MAIN VENT AIR VALVES

For small Two-Pipe installations designed to return condensate directly to the boiler by gravity, air must he vented from the system. A Main Vent Air Valve must be used at the proper location. These vents are the same as those used for One-Pipe Systems and are installed at the end of the return main ahead of the point where it drops below the boiler water line to become a wet return.

Figure 26 on **page 12** shows the construction of a Main Vent Valve. It is designed to permit air accumulations to be discharged (vented) to the atmosphere and to close against the passage of steam or water. The proper installation of a Main Vent is shown by **Figure 28** on **page 13**. The incorrect method of installation is also illustrated.

TYPES OF TWO-PIPE SYSTEMS

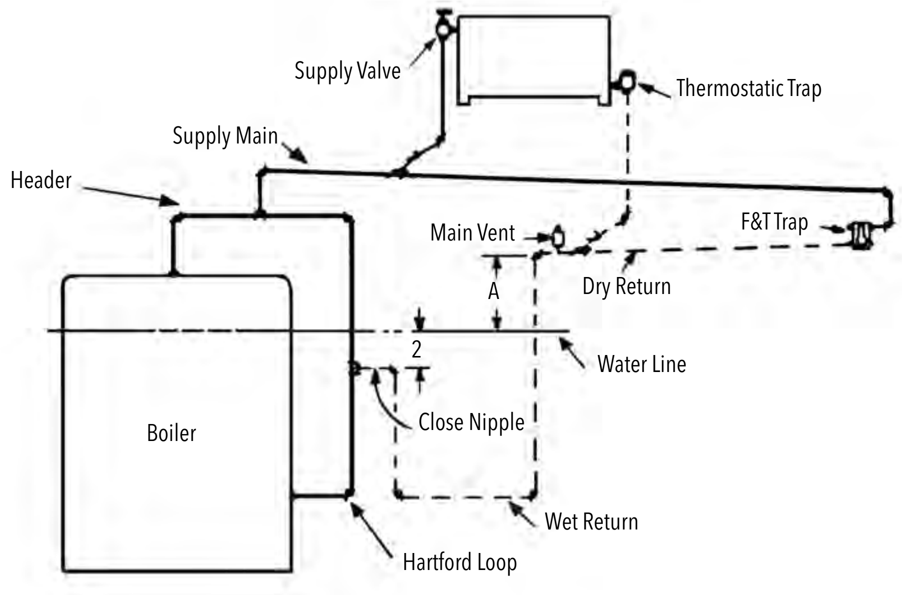


Figure 52

The following discussions and drawings of systems and components will provide basic information of the various types of Two-Pipe Steam Heating Systems.

These types include Gravity Return Systems and Mechanically Returned Systems which operate with atmospheric pressure in the return piping. The basic components used for these systems have been discussed. Other types are called Vacuum Systems and some of the components are different from other systems and will be covered later, under another section.

1. GRAVITY RETURN SYSTEM

A very simple Two-Pipe Gravity Return System is shown diagrammatically in **Figure 52**. Steam from the boiler is carried by the supply main to the radiator through a supply valve. Condensate and air are discharged from the radiator into the dry return through a thermostatic trap. Also, condensate and air accumulation in the supply main is dripped through a float and thermostatic trap into the dry return. The end of the dry return is equipped with a main vent through which all the air in the system is discharged or vented to the atmosphere. This main vent must be located ahead of the pipe connection where the dry return drops below the boiler water line and becomes a wet return. See **Figure 28**, **page 13.** Condensate from this gravity return system is returned to the boiler through a Hartford Loop as shown by **Figure 9** and described in the One-Pipe section of this manual. Dimension "A" must not be less than 28" for each 1.0 PSIG pressure maintained at the boiler. This requirement is a limitation which makes this gravity return type of system suitable for smaller sized installations requiring a boiler pressure between 1/2 PSIG and 1 PSIG. The definition and function of the system components are explained using **Figure 52**.

(a) STEAM SUPPLY HEADER: Boilers, depending on their size, have one or more outlet tappings. The vertical steam pipe from the tapped or flanged outlet joins the horizontal pipe called a "Header." The steam supply mains are connected to this "Header."

(b) STEAM SUPPLY MAIN: The Supply Main carries steam from the boiler to the radiators connected along its horizontal length. An upfeed runout carries steam to the radiator through a supply valve connected to its inlet connection. The radiator outlet connection is equipped with a thermostatic trap. The trap discharge is connected by the return runout piping to the Dry Return Main.

(c) DRY RETURN MAIN: A portion of the Return Main is designated "Dry Return" to distinguish it from the portion shown as "Wet Return." It is defined as a "Dry Return" because it is located above the boiler water line in this gravity return system. It carries air and condensate which has been discharged from the radiator thermostatic traps and from the float and thermostatic trap dripping the end of the steam supply main.

(d) WET RETURN: That portion of the return main which carries condensate back to the boiler and is installed below the boiler water line level is called a "Wet Return." It is completely filled with water and does not carry air or other non-condensible gases. When the system is first filled with water or is cold, the pressure throughout the system is at atmosphere. Because of this equal pressure, the water level in the boiler and in the drip piping connected to the Wet Return is at the same level as indicated by the boiler water line. Dimension "A" must be a minimum of 28" for each 1.0 PSIG pressure carried at the boiler.

TYPES OF TWO-PIPE SYSTEMS

(e) HARTFORD LOOP: The Hartford Loop is a special arrangement of return piping at the boiler. Its purpose is to reduce the likelihood of an insufficient quantity of water creating a low water condition which could cause damage to the boiler. It is used with gravity return systems having Wet Returns. The operating principle of the Hartford Loop is described in the One-Pipe section of this manual and is shown in **Figure 9**.

PIPING FUNDAMENTALS AND FUNCTION OF EQUIPMENT

There are certain essential factors which must be applied to the design and installation of a gravity return system to insure proper flow of steam and condensate and the elimination of air from the system. These are illustrated and discussed in the following.

(a) PITCH OF MAINS: Pitch is the amount of slope given a pipe. For horizontal supply mains and Dry Returns it must not be less than 1/4" in 10 feet in the direction of steam and condensate flow. (**See Figure 52**). Uniform pitch of the steam supply and Dry Return is required to assure free flow of steam without interference from the condensate. **Figure 34** on **page 16** shows a condition of nonuniform pitch caused by a sag in the horizontal steam supply or dry return main. The pocket causes a restriction to the free flow of steam when in the steam supply main and will also interfere with the free flow of air if the pocket is in the dry return main. A pocket in either the steam or return main will interfere with good steam distribution. Such a sag in a pipe is usually due to insufficient or improper pipe supports or adjustment of the supports. These defects are usually easy to correct.

(b) REDUCING PIPE SIZE: When it is necessary to reduce the size of a horizontal main, an eccentric coupling should always be used as shown in **Figure 33, page 16**. The use of a regular reducing coupling destroys uniform pitch and creates a water pocket which interferes with the free flow of condensate and steam.

(c) OBSTRUCTIONS: Occasionally it is necessary to install a steam main around a structural obstruction such as an "I" Beam or girder. **Figure 35** on **page 16** shows the proper way to run the steam main above the obstruction while the condensate is carried below. Also, there are obstructions to the installation of a Dry Return Main such as a doorway shown in **Figure 36** on **page 16.** The return main is run in a trench below the doorway with an air loop over the top.

2. MECHANICAL CONDENSATE RETURN SYSTEM

A. CONDENSATE PUMP

The principal difference in this system from that described for the Gravity Return System is the method used to return the condensate to the boiler. When systems increase in size and higher steam pressures are required to obtain steam circulation, the condensate cannot be returned to the boiler by gravity. Some type of mechanical means must be used to perform this return function. The accepted method for a modern steam heating system is to use a Condensate Pump. **Figure 53** shows a typical Two-Pipe Steam Heating System using a Condensate Pump. The Condensate Pump consists of a Receiver on which is mounted single or multiple electric motor-driven Pump Assemblies, Float Switches,

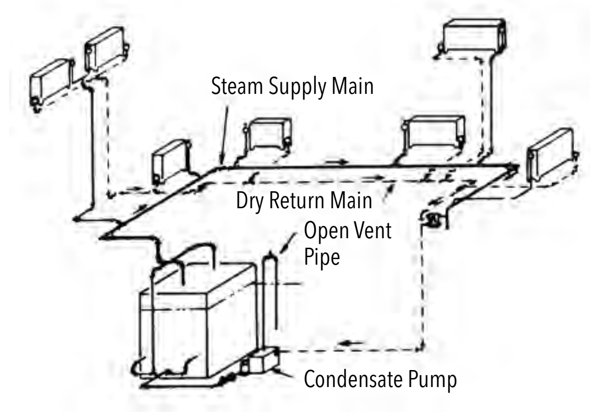


Figure 53

and other electrical controls as required by the heating system. A Condensate Pump must be located at the low point of the return and as close to the Boiler as possible. The Return Main must be uniformly pitched 1/4" in 10 feet to the receiver inlet so condensate can flow by gravity into the Receiver. This is important so that condensate and air can be separated as it flows in the piping and the air can be vented to atmosphere through the receiver vent connection.

When it is necessary to locate the Condensate Pump below the floor level to obtain uniform pitch to the receiver inlet, the Condensate Pump can be installed in a pit or an underground type can be used. **Figure 54** shows the application of an Underground Condensate Pump.

The condensate discharge piping between the pump and the boiler must be equipped with a discharge Check Valve located in the discharge piping adjacent to the pump discharge connection. A Shut-off Valve must be installed adjacent to the Check Valve and between it and the Boiler to permit easy servicing of the pump unit.

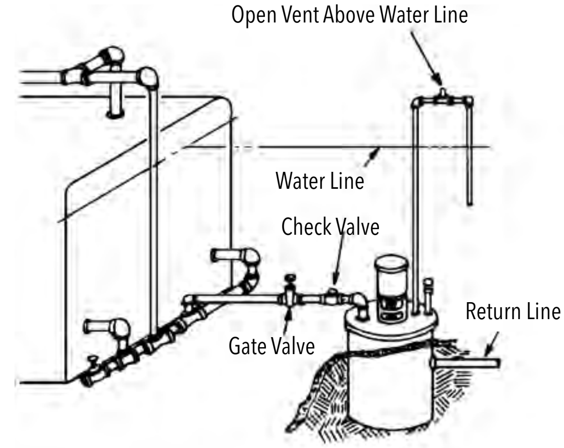


Figure 54
Underground Condensate Pump

TYPES OF TWO-PIPE SYSTEMS

The operating cycle of a Condensate Pump is shown by **Figure 55** and **Figure 56**. As condensate flows by gravity into the Receiver, the water level rises until the Float reaches its top position, at which time the electrical Contacts in the Float Switch are closed to start the pump Motor. As condensate is pumped from the Receiver, the water level is lowered sufficiently to cause the electrical contacts to open and stop the pump Motor when the float has reached its lowest position. This cycle is repeated as of ten as necessary to handle the condensate from the system and maintain the Boiler Water Level at the proper height.

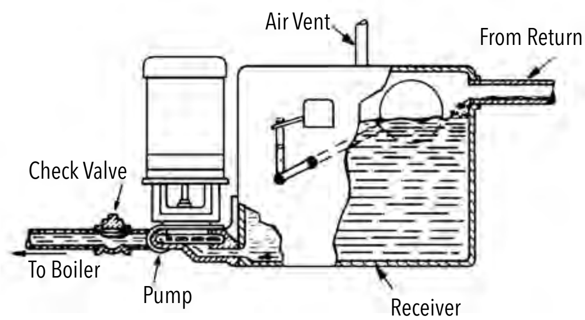


Figure 55

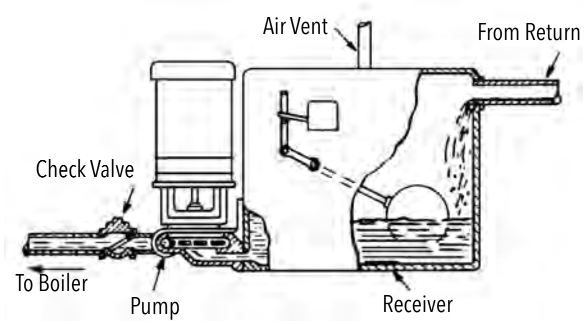


Figure 56

The Float Switch for a Condensate Pump is adjustable so the quantity of water being discharged for each operating cycle can be set to satisfy the water level condition required for the boiler. Duplex Condensate Pumps are often equipped with Mechanical Alternators which will cause the lead pump to operate every other time. The alternator is designed to cause both pumps to operate together to provide double capacity when the system condensate load demands such operation.

B. BOILER RETURN TRAP

Another method used to return condensate mechanically to the boiler is a Boiler Return Trap. A Boiler Return Trap is not a single device but consists of two separate units; (1) a Boiler Return Trap and (2) a Receiver Vent connected in a special way to the Boiler by a system of piping and check valves so condensate can be returned to the boiler and air vented from the system during cycles of operation. **Figure 57** shows the piping arrangement at the boiler when a Boiler Return Trap is used with a Two-Pipe System. A few Boiler Return Trap installations are still being used but they are costly and are usually limited in pressure to 15 PSIG. It is more desirable to install a Condensate Pump for new installations or to replace the Boiler Return Trap when servicing of an old unit is required.

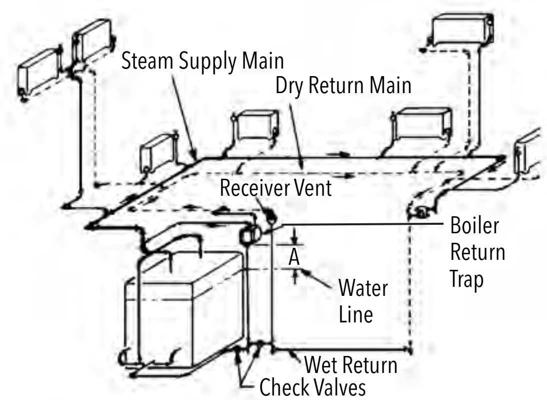


Figure 57

C. BOILER WATER MAKE-UP CONDENSATE UNITS

Steam heating systems are often supplied with steam from the same boiler which supplies steam to an industrial process or for a similar use. In some cases, not all the condensate can be returned to the condensate pump receiver where it can be pumped back to the boiler. It cannot be returned because there is normal waste in the process or the condensate becomes contaminated and is unfit for boiler use. When these conditions exist, make-up water must be supplied to the condensate pump receiver to insure that the boiler water level can be maintained to prevent shutdown due to a low water condition.

There are several methods used to supply makeup water to a Condensate Pump Receiver. **Figure 58** shows one method using a float type Make-up Water Valve. It maintains a sufficient quantity of water in the pump receiver as required by the level of make-up water and condensate in the receiver. When make-up water is required by the boiler, the condensate pump cannot be operated by a float switch in the receiver, but is operated by a boiler water level control or pump controller located on the boiler at the water line. When the boiler water reaches a predetermined "low" condition the pump is started and restores the boiler water line to the proper level. The pump controller can be provided with a low-water cut-off, or a separate low-water cut-off can be used. The boiler can also be equipped with an auxiliary feeder which will maintain a protective boiler water level during an emergency of current failure to the pump motor, or a motor burnout. **Figure 59** shows an arrangement for auxiliary make-up water feeder.

TYPES OF TWO-PIPE SYSTEMS

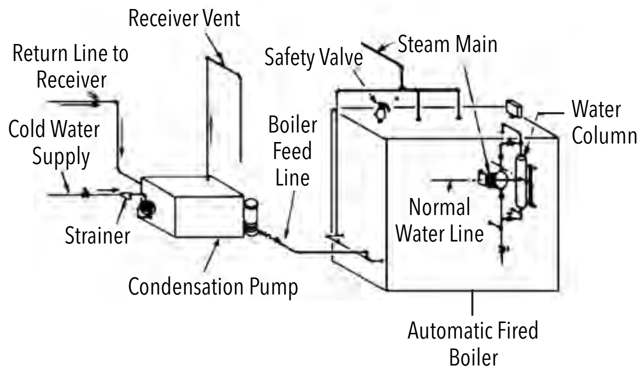


Figure 58

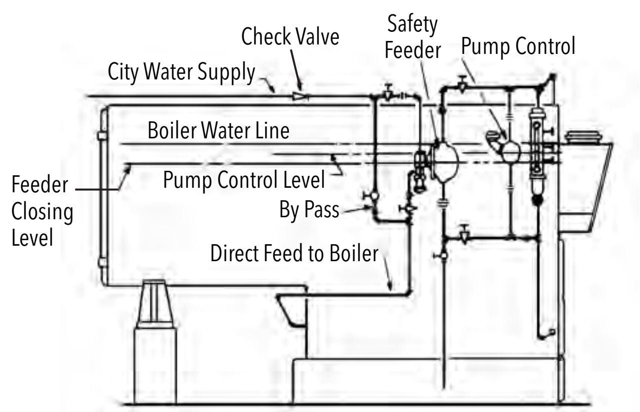


Figure 59

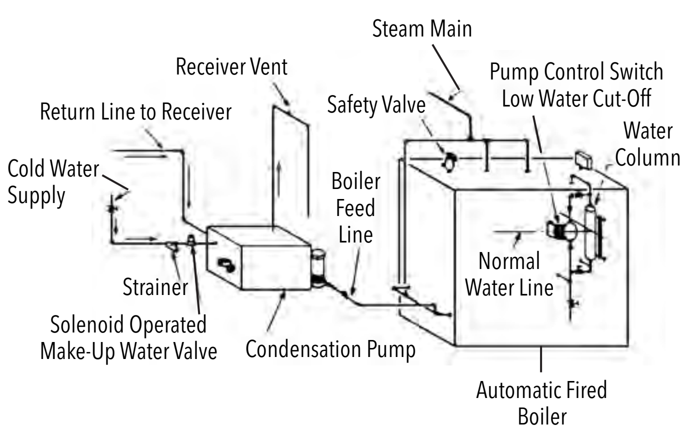


Figure 60

Make-up water can also be supplied to the condensate pump receiver using an electric solenoid or an electric motor-operated valve. This valve is controlled by a float switch in the receiver to maintain the proper level of make-up water and condensate in the receiver. **Figure 60** shows this type of an arrangement for make-up water. Usually, the electric motor-operated valve or solenoid valve arrangement is used where large quantities of make-up water must be provided. The capacity of a float-operated make-up water valve is usually too small for these larger requirements.

3. VACUUM SYSTEMS

When Two-Pipe Steam Heating Systems become large and require long runs of steam and return piping, there is a large volume of air in the system which, if not properly expelled, can be a deterrent to steam distribution to the heating units. The result of a slow elimination of air from the system is a slowing up of the condensate return process to the boiler during the heating-up period. Steam can leave the boiler faster than the condensate can be returned using normal means, causing a low water level in the boiler. The boiler firing device controls will stop the fuel burning equipment due to this low water condition and, in turn, the boiler water make-up equipment will over-supply the boiler during the heating-up period. When the normal rate of condensate returning to the boiler is established, the boiler will be too full and the steam space in the boiler will be flooded. Such conditions cannot be corrected automatically but require manual attention.

One method used to overcome this steam distribution problem and its adverse condensate return problem is to use a Vacuum Pump for the quick elimination of air from the heating units and the steam and return piping. The ordinary Two-Pipe Vacuum System uses a Vacuum Pump designed for steam heating systems. Several means are employed for producing vacuum but all are classified as meeting the requirements of a low vacuum type pump. Low vacuum type pumps, depending on the size, are rated to handle a definite quantity of air to maintain an average vacuum of five and one-half inches of mercury (5 1/2" Hg) at a condensate temperature of 160°F at the pump. The pump is normally controlled to "cut in" at 3" Hg and "cut-out" at 8" Hg.

A. TWO-PIPE VACUUM SYSTEM

The basic equipment used in a Two-Pipe Vacuum System is the same as that used in the Gravity Return or Mechanical Return Systems which have been described. The application of the vacuum pump to the system is the only difference. The vacuum pump not only eliminates air from the system but also returns the condensate to the boiler. There are several methods used to accomplish this and they will be explained in this section of the manual.

Figure 61 shows a Duplex Vacuum Pump of the low vacuum type connected to the boiler of a Two-Pipe System. The vacuum pump shown is a Jet type design. It consists of a two compartment Receiver, motor-driven centrifugal Pumps, Jet Vacuum Producers, and electrical Controls for starting and stopping the pumps.

TYPES OF TWO-PIPE SYSTEMS

The Controls consist of Float Switches, Vacuum Switches, Motor Starters, and any others as may be specified for the system. When the Vacuum Pump is applied to Two-Pipe Steam Heating Systems the condensate must flow by gravity to the lower compartment inlet connection strainer. It is important that the piping be uniformly pitched so that the gravity flow of condensate in the return piping will not obstruct the free flow of air from the system into the vacuum pump. There are times when the vacuum pump cannot be installed to permit the gravity flow of condensate. The proper way to handle this problem will be discussed later.

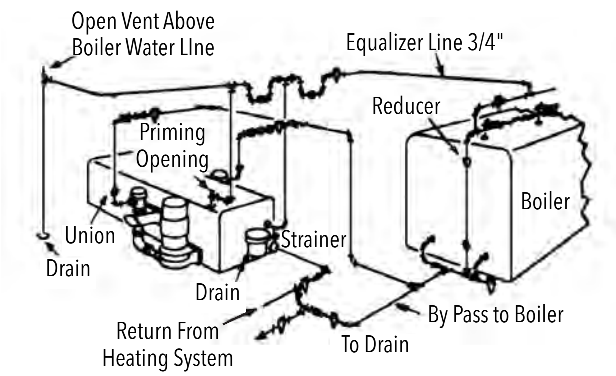


Figure 61

(1) JET VACUUM PRODUCER: One of the simplest and a very common type of vacuum pump used for steam heating systems employs a Jet Vacuum Producer as shown by **Figure 62**. The Jet Vacuum Producer is very effective in producing high vacuum when condensate temperatures are consistent with the vacuum required. It is designed to take full advantage of the fact that water in motion will entrain or pick up air, water, and gases. It can also maintain high operating efficiencies because it does not have close working clearances which can wear. Referring to **Figure 62**, the operation is explained as follows: water is discharged by a centrifugal pump through the nozzle into the ejector throat and tube. As the water at high velocity passes through the ejector gap, air and non-condensible gases from the system, and condensate from the lower compartment of the receiver, are picked up or entrained in the ejector gap by the water jets. This action creates a vacuum which is sufficient to exhaust all the air from the heating system.

The vacuum pump using the Jet Vacuum Producer can also return condensate to the boiler to meet the requirements of the system condensate load and the boiler pressure.

For an illustration of a cycle of operation of a Jet Type Vacuum Pump, refer to **Figure 63, Figure 64,** and **Figure 65**.

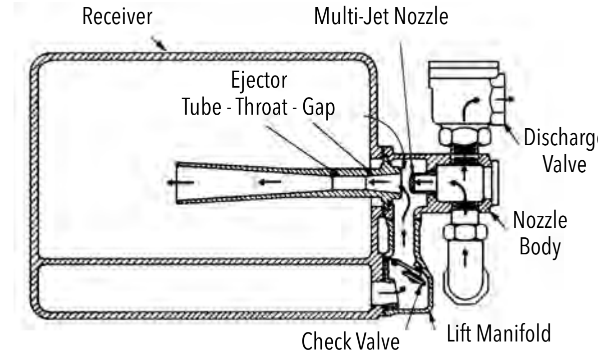
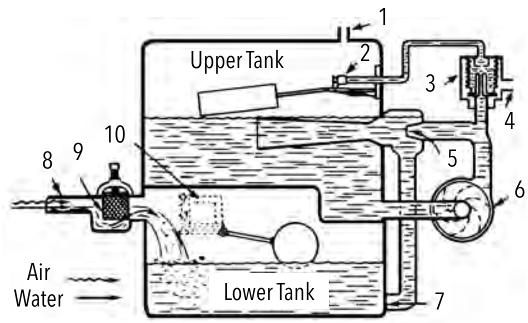


Figure 62



1 Open Vent
2 Pilot Valve
3 Discharge Valve
4 To Boiler
5 Vacuum Jets
6 Pump
7 Check Valve
8 Inlet to Pump From Returns
9 Strainer
10 Float Switch For Starting Pump Motor

Figure 63

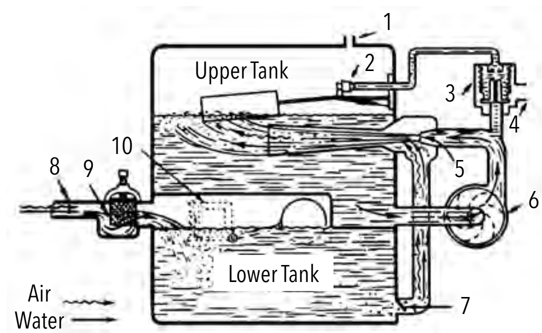


Figure 64

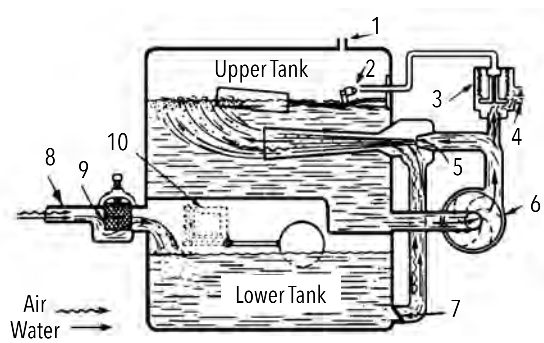


Figure 65

TYPES OF TWO-PIPE SYSTEMS

Figure 63 shows the centrifugal Pump (6) stopped. Condensate from the system flows by gravity into the lower compartment of the receiver. The Float Switch (10) cannot start the pump because the condensate level is too low. "Hurling water" (condensate) is at the proper level in the upper compartment and cannot flow backwards into the lower compartment because Check Valve (7) is closed.

Figure 64 shows the operating condition when sufficient condensate has entered the lower compartment to cause the Float Switch (10) to close the electrical contacts to the centrifugal pump Motor (6). "Hurling water" from the upper compartment flows to the pump suction and is pumped through the Jet Vacuum Producer (5) and back into the upper compartment. During this time, the vacuum created by the water jets discharging across the ejector gap draws air, non-condensible gases, and water vapor, from the system return piping and by entrainment discharges them into the upper compartment. Here the air and non-condensible gases are separated from the condensate and vented to the atmosphere through the open Vent (1). Simultaneously, the condensate from the system is being lifted from the lower compartment through the Check Valve (7) now open to the upper compartment. As shown in **Figure 64**, the Pilot Valve (2) is closed, and the Discharge Valve (3) is also closed due to pump pressure acting internally on the bellows in a manner designed to keep the valve on the seat port.

As this operation continues, the condensate in the upper compartment has reached a level sufficient to operate the float actuated Pilot Valve (2) as shown in **Figure 65**. The Pilot Valve (2) opens, which releases the pump pressure which existed internally on the bellows and the Discharge Valve (3) opens to discharge the accumulated condensate to the boiler through the Discharge Opening (4). The same pump which creates the vacuum in the Jet Vacuum Producer also discharges condensate to the boiler. When the condensate level in the upper compartment has been sufficiently reduced, the Pilot Valve closes which, in turn, causes the Discharge Valve to close.

This cycle is repeated as often as required by the operation of the Float Switch (10). Also, the pump can be started by the Vacuum Control Switch (not shown) to pump "Hurling water" to maintain the required vacuum on the system return piping. The Vacuum Control Switch and the Float Switch (1) are electrically connected in parallel. They are controlled independently by the condensate level in the lower compartment or by the system vacuum being maintained in the return piping.

There are other arrangements of Vacuum Pumps using the operating principle of the Jet Vacuum Producer, but instead of using a two compartment receiver employing a pilot-operated discharge valve to return condensate to the boiler, they use a separate condensate pump operated by a float switch, independent of the pump used to create and maintain the required system vacuum. **Figure 66** shows this type of pump. These combinations are usually applicable to large systems requiring higher vacuum, large air capacity, or where the condensate must be discharged to a boiler operating at high pressure.

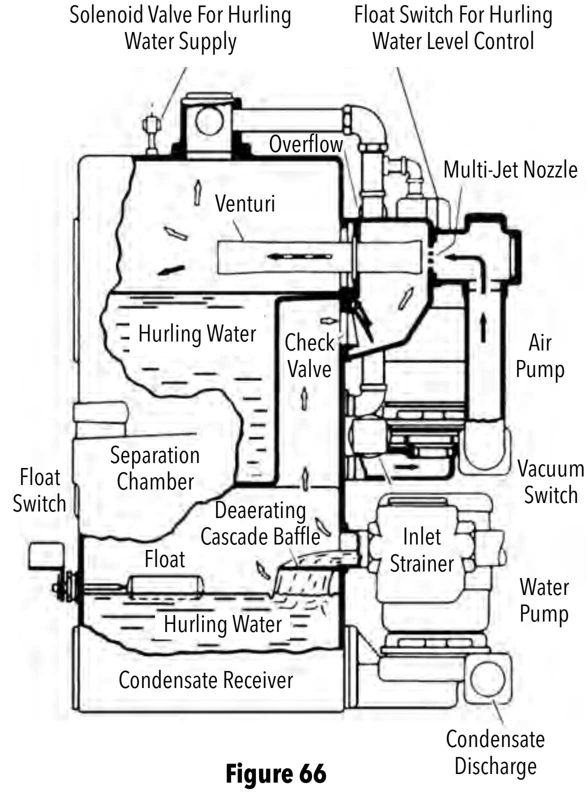


Figure 66

This type of vacuum pump also provides a maximum flexibility to meet individual system needs. It provides a selection of air capacity to meet the requirements of the system air evacuation and condensate pumps best suited for the condensate return needs.

Figure 67 shows a Duplex Vacuum Pump of this type connected to a boiler of an ordinary two pipe vacuum system. The pump shown consists of a condensate receiver equipped with vertical centrifugal pumps which are controlled by float switches to discharge the condensate accumulation to the boiler. The air evacuation unit consists of a separation chamber which contains "Hurling water" which is used to produce the system vacuum by employing the jet vacuum producer in combination with separate motor operated air pumps. The air pumps are controlled by vacuum switches responsive to the vacuum in the condensate receiver and system return piping.

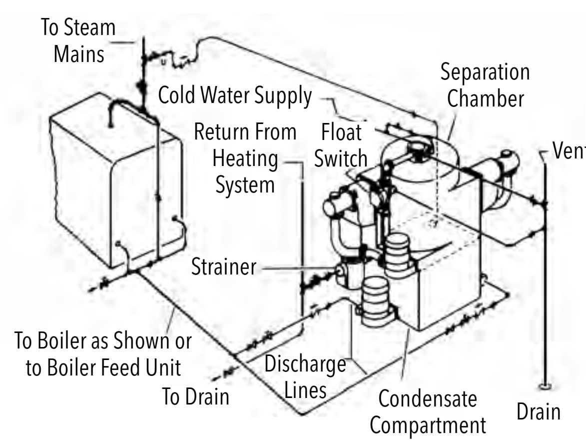


Figure 67

TYPES OF TWO-PIPE SYSTEMS

The cut-away view of this type of vacuum pump shown by **Figure 66** will be used to explain the simplicity of its operation. There are two separate and independent cycles of operation for this type of pump-one for air evacuation and the other for condensate return. The Return Main must be properly pitched to assure gravity flow of condensate to the receiver inlet strainer.

An independent air evacuation cycle begins when the vacuum switch responding to system requirements, starts the centrifugal air pump. This pump circulates "Hurling water" from the separation chamber through the multi-jet nozzle, the venturi, and returns it to the separation chamber. The water, forced at high velocity across the gap between nozzle and venturi, entrains air and gases in multiple jet streams, creating a smooth, steady vacuum in the condensate receiver and system. The mixture is discharged through the venturi into the separation chamber where the air and gases separate from the "Hurling water" and are vented. When the desired vacuum has been produced in the system, the vacuum switch stops the pump, and the check valve at the air suction inlet to separation chamber closes, preventing the return of air to the system.

Replacement of the "Hurling water" evaporated from the separation chamber is controlled by a solenoid valve connected to a water supply and actuated by a float switch.

The condensate return cycle begins when a float switch starts a water pump on condensate rise. The condensate is pumped to the boiler until the preset, low float switch setting has been reached.

This design of a vacuum pump offers advantages other than larger air capacity and higher discharge pressure. Some systems may have conditions requiring the "Hurling water" temperature maintained at some predetermined maximum limit. This is accomplished by installing a temperature limit switch in the separation chamber which will open the solenoid valve to admit cooling water as required to maintain the "Hurling water" temperature. The overflow connection maintains the "Hurling water" at the proper level.

Also, conditions may exist, such as leaking traps or improper connection of traps discharging too close to the vacuum pump, where the temperature of condensate fluctuates intermittently to critically high levels. Operation under such conditions could cause the condensate to vaporize or flash into steam and prevent vacuum control. A temperature limit switch can be installed on the condensate receiver to prevent the air pumps from operating when such a condition exists. Upon temperature drop, the vacuum switches will again control operation of the air pumps.

In selecting pumps of this type for low vacuum heating systems the normal operating range for the air evacuation pump is from 3" Hg to 8" Hg for an average vacuum of 5 1/2" Hg. The air pump capacity for the normal vacuum system should have a capacity of 0.3 to 0.5 CFM of air removal for each 1000 sq. ft. EDR connected load. The larger air capacity is recommended for systems up to 10,000 sq. ft. EDR. These ratings are based on 5 1/2" Hg vacuum at 160°F. When vacuum heating systems have excessive air leakage which cannot be controlled, it is necessary to increase the CFM air removal to compensate for the leakage above normal requirements. When this is done, the system vacuum should continue to be operated from 3" Hg to 8" Hg vacuum.

Using a vacuum pump to meet the basic needs of a two-pipe system also requires means for selecting choices of operation. The three most common requirements are obtained by the setting of a Selector Switch to provide one of the following: (1) *"Automatic" Operation* This permits the pump motor to be started when the system vacuum falls below the vacuum switch lowest setting usually 3" Hg and stopped when the vacuum exceeds the maximum setting-usually 8" Hg. The pump can also be started and stopped by a float switch in the lower compartment to return condensate accumulation to the boiler. (2) *"Continuous" Operation* When the selector switch is turned to this position the pump runs continuously without regard to vacuum or condensate accumulation. (3) *"Float Only"* When desirable, the pump can be selected to start and stop only in response to the condensate accumulation returning from the system.

2. DISPLACEMENT TYPE VACUUM PUMP:

In addition to the Jet Type Vacuum Pumps illustrated and discussed, there are other types used in Vacuum Heating Systems. These vary somewhat in design and operating principle but fall under the general classification of Displacement Type Vacuum Pumps. Performance characteristics are similar to those of a Jet Type Vacuum Pump but are more critical in maintaining top performance because of the close clearances required to create a vacuum. Also, they cannot handle as large a volume of air during a cold system start up as can be handled by a Jet Vacuum Producer. It is very important to eliminate air from the system on start up in order to avoid poor steam distribution and its resulting slow return of condensate to the boiler.

Good performance of a two-pipe vacuum system requires that the return condensate flow by gravity to the receiver of the vacuum pump. This means that the receiver inlet must be the low point of the system return and the return piping must be uniformly pitched to this low point. When this is done, the condensate will flow through the return piping and occupy the lower half of the pipe bore. The space above the condensate is then available for the uninterrupted flow of air, non-condensible gases, and water vapor from the system piping and heating units. These products can be easily evacuated from the system in response to the operation of the vacuum pump.

There are, at times, structural design conditions which must be met which prevent gravity flow of the condensate directly to the low point of the vacuum pump. There are three methods employed to deal with this problem. There are serious disadvantages to two of these methods; namely, lift fitting and accumulator tanks used as lift fittings. These will be described along with the acceptable means of using mechanically pumped accumulators.

TYPES OF TWO-PIPE SYSTEMS

(a) Mechanically Pumped Accumulator: This piece of equipment consists of a condensate pump and receiver combination which is located at the low point so that condensate on the system side of the receiver inlet can flow by gravity to this receiver. The receiver is under the same system vacuum conditions as that of the vacuum pump because an air line is connected between the top of the receiver and the main vacuum return main located at the higher elevation. The condensate is pumped to this same vacuum return main. **Figure 68** and **Figure 69** show two piping arrangements for mechanically pumped accumulators. One uses a receiver located above the floor while the other uses an underground receiver. The pump and motor units for this type of service must use mechanical seals for the rotating shafts to prevent air leakage into the system. The float switches must also be of a type which will prevent air leakage. The pump and motor unit for this equipment can be selected for capacity and discharge pressure to meet the system conditions.

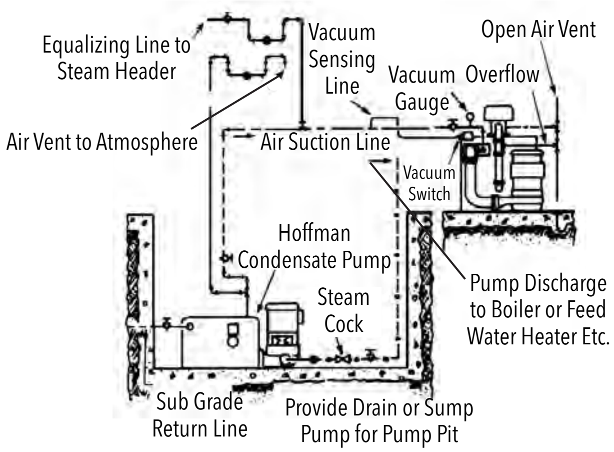


Figure 68

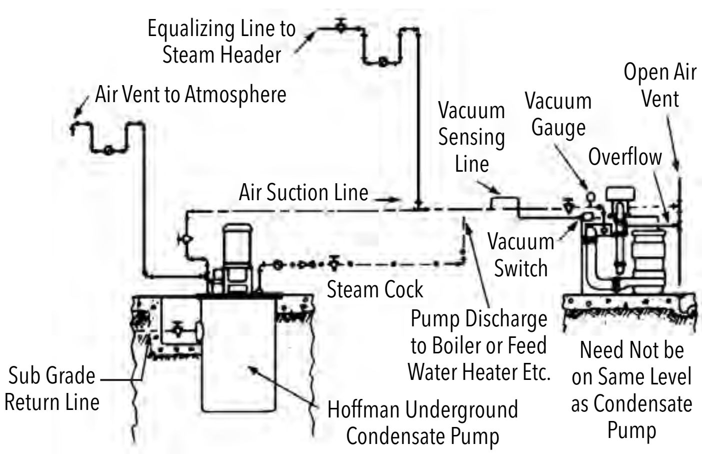


Figure 69

(b) Lift Fitting: When commercial lift fittings are not available they can be constructed using standard pipe fittings as shown by the details in **Figure 70**. It is important to keep the distance between the center line of the vacuum return pipe and the center line of the 90° elbows at the top of the lift to the minimum by using close pipe nipples as shown. The inverted "U" at the top of the lift is required so that the "slug" of water being lifted is continuously carried over the top by siphon action into the vacuum return pipe. This lifting process continues during the time the open end of the lift pipe in the lift pocket is sealed with water. A step lift using commercial lift fitting is shown by **Figure 71.**

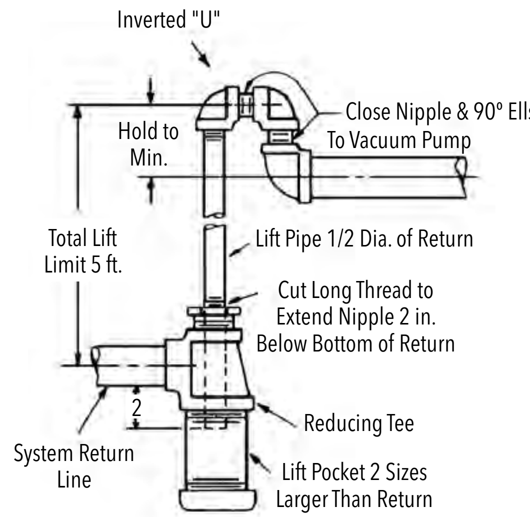
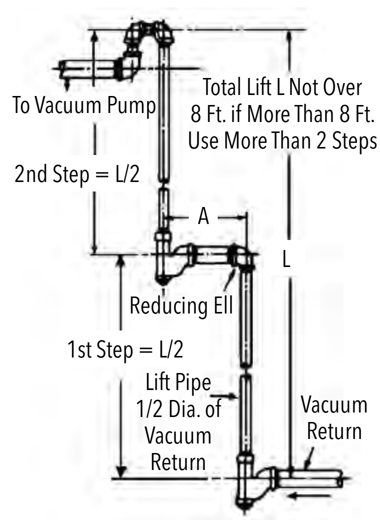


Figure 70



Vacuum Return Pipe Size Inches	A Inches
1	7
1 1/4	8
1 1/2	9
2	10
2 1/2	14
3	15
4	18
5	21
6	24

Maximum Length A

Figure 71

TYPES OF TWO-PIPE SYSTEMS

Lift fittings have been used often in vacuum return mains to lift the condensate and air in slugs from a low point in the system return to a higher elevation of the vacuum pump. Lift fittings as shown here should always be avoided because they are a source of trouble and noise. Condensate and air can be lifted by such an arrangement provided there is sufficient vacuum available (1" Hg for every foot of lift) above that required by the system, and if the temperature of the condensate is always well below that of the vapor pressure of the required vacuum as given by steam tables for saturated steam.

Some of the disadvantages are:

(1) Their use requires constant operation of vacuum pumps to obtain results.
(2) Vacuum pumps must produce higher vacuums, above that of system use, to obtain the required lift.
(3) High temperature of condensate which will flash into steam at the lower pressures can destroy operation of the lift fitting and cause considerable objectionable noise.
(4) Operation of vacuum pumps on "Float Only" controls cannot be used. This "Float Only" is a desirable feature for many installations such as schools, for shutdown at night or for weekends and holidays.

(c) Accumulator Tank: The installation of an Accumulator Tank used as a lift is shown by **Figure 72**. One view is for a tank above the floor line at the low point of the vacuum return main while the other view uses an underground tank below the floor line.

Although the use of an Accumulator Tank is an improvement over a Lift Fitting installed directly in the return line, described under (b), it still has all the undesirable disadvantages given for the lift fitting except that the condensate accumulated does have a chance to cool somewhat before being lifted to the vacuum pump.

The accumulator tank must be installed adjacent to the vacuum pump so the copper tubing connecting the vacuum switch to the accumulator tank can be kept short and properly pitched, Water accumulation in this line will cause false operation of the vacuum switch. The float switch, normally installed in the vacuum pump receiver, must be changed to the accumulator tank and properly wired to the vacuum pump motor.

B. TWO-PIPE SUB-ATMOSPHERIC CONTROL SYSTEM

A . Two-Pipe Sub-Atmospheric System is also called a *Variable* Vacuum System or a High Differential Vacuum System. It differs from the ordinary two-pipe low vacuum system in that a controllable vacuum is maintained in both the supply side and the return side of the system. The heat emission or heat output from the radiators is controlled to meet a building comfort requirement by controlling the partial vacuum in the system. Varying the steam

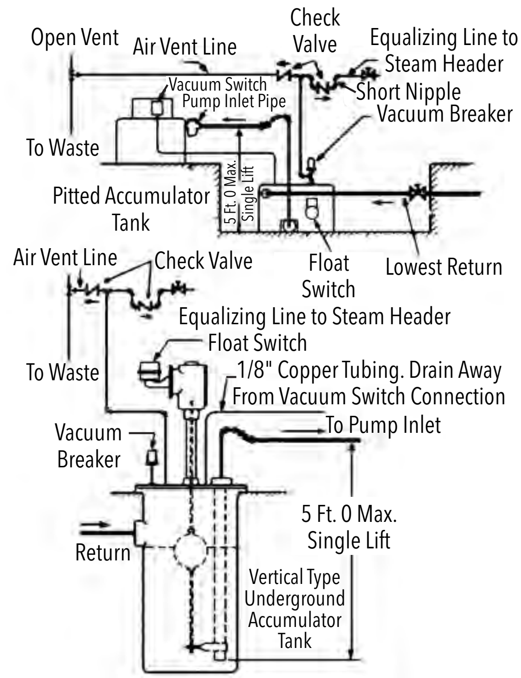


Figure 72
Piping to Vacuum Pump Where Return Main is Below Pump Flow Level

temperature and rate of flow is accomplished by the proper control of the system pressure and the differential pressure across the radiator. In mild weather, this partial vacuum may be maintained as high as 25" Hg while in the coldest weather a steam pressure above atmospheric pressure may be required. These systems employ controls to modulate the steam flow to the various zones of a building in response to changes in outdoor temperature, indoor temperature, or other factors related to the weather. Some systems employ orifices for steam distribution while other types omit the orifice. The controls for these Sub-Atmospheric Systems are usually accomplished by use of proprietary designs and patents.

C. TWO-PIPE ORIFICE SYSTEMS

The piping arrangements, pipe sizing and radiation for a Two-pipe Orifice Steam System is the same as required for a Two-Pipe Vacuum System. Orifices are installed at the inlet to each radiator and are sized to meet the heat output for the radiator. The quantity of steam supplied to each radiator is metered by controlling the differential pressure across the orifice. By varying this differential pressure, it is possible to heat the radiator fully or to heat it partially to provide the required comfort condition. In order to accomplish this variable heat output, the orifices must be accurately drilled with an orifice wall thickness not less than 1/8". When this is done, the steam flow through the orifice will vary with the square root of the differential pressure, provided that the ratio of absolute pressure on the orifice outlet to the absolute pressure on the orifice inlet is greater than 58%.

Some orifice systems use distribution orifices only. These are thin-cup type and cannot be used for metering purposes. They do provide an easy and effective means to get even distribution of steam to all radiators in a system at about the same time on starting up a cold system.

TYPES OF TWO-PIPE SYSTEMS

HEATING SYSTEM SUPPLIED BY REMOTE BOILER

Not all steam heating systems are designed to be supplied with steam from a separate boiler sized for the particular heating load. Often, steam is supplied to the heating system through pressure reducing valves at stations located remotely from a high pressure boiler which generates steam at higher pressure than can be used by the heating system. These remote boiler plants are necessary in large industrial plants to supply steam for a variety of process loads as well as for heating. Other examples where they are required are: college-campuses, apartment and housing projects, campus-type hospital groups, and similar applications.

Another source of steam supply for heating purposes is District Heating or street steam systems. District Heating companies are usually classed as utilities and are subject to certain local regulations and codes. The building heating system design must take these factors into consideration to assure adequate safeguards and compliance.

Figure 73 shows a pressure reducing valve station supplying steam for a heating system. Also shown is a return condensate pump employed to pump condensate back to a hot well or boiler feed accumulator tank or receiver.

Figure 74 shows the pressure reducing valve station employing a pump to return condensate to a vacuum return line.

Pressure reducing valves are designed to meet the requirements of two classes of service. (1) When the application requires a valve which will shut off tightly under "no load" condition to prevent a buildup of pressure on the downstream or low pressure side, a single seated valve must be used and its use is described as "Dead-End" service. (2) Pressure reducing valves are also designed with double seated valve bodies. They are simpler in construction, have increased capacity, and are usually less expensive. They will not shut off absolutely tight but can be used for "Continuous Flow" service. This means that under "no heat requirement" condition there can be a build-up of pressure on the low pressure side due to this leakage. Many heating system applications will condense enough steam to permit a small leakage through a double seated valve. When this condition exists, a double seated valve can be used.

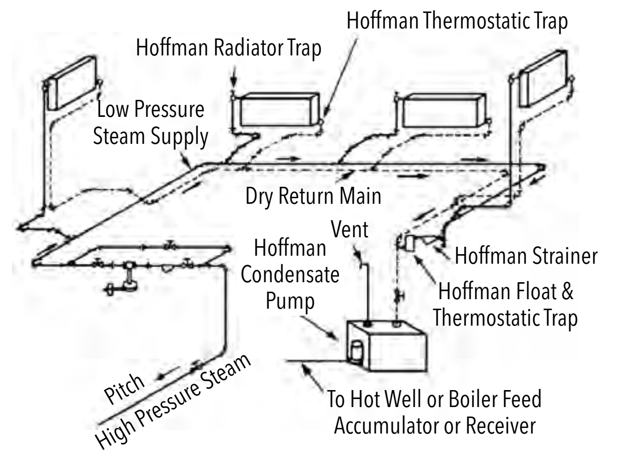


Figure 73

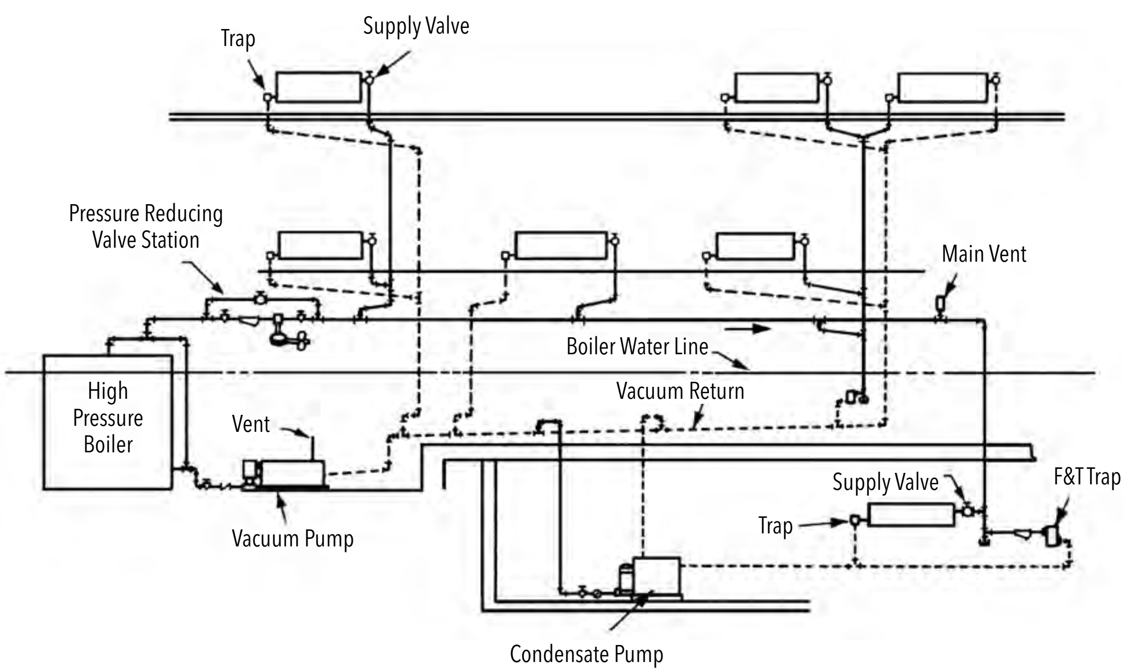


Figure 74

TYPES OF TWO-PIPE SYSTEMS

Single or double seated pressure reducing valves are available in several different types to meet a variety of needs. A few are illustrated by the following diagrams. **Figure 75** shows a spring loaded diaphragm.

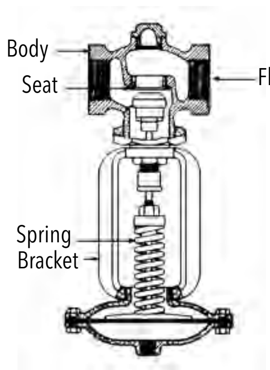


Figure 75

One of the most dependable types used with modern steam heating systems is known as a pressure regulator with external pilot. It has the ability to respond to large changes in load and wide variations in the supply inlet or initial pressure. **Figure 77** shows this type of valve installed for a heating system.

Figure 77

It is not always possible to reduce from a high initial pressure to a required low pressure using a single pressure reducing valve for heating systems. **Figure 78** shows a double-reduction installation with two valves in series. The first valve reduces the initial high pressure to an intermediate pressure. The second valve reduces the intermediate pressure to the desired low pressure. This series type of installation has advantages of reducing velocity noises, reducing wire drawing of valve or seat, and permitting more reasonable diaphragm sizes. This is particularly true for single seated, direct acting, spring loaded valves. The pipe size between the primary and secondary valve should be at least one pipe size larger than the primary valve size.

For heating systems, the pipe size should be of an area that will assure a velocity of between 6,000 and 8,000 feet per minute. The distance between the primary and the secondary valve should be ample to give good steam volume between the two valves to prevent excessive turbulence and "hunting." This distance should never be less than 20 feet. Some type pressure reducing valves require a greater intermediate pressure volume than this 20 foot length provides. If it is impractical to provide the necessary additional length, the required

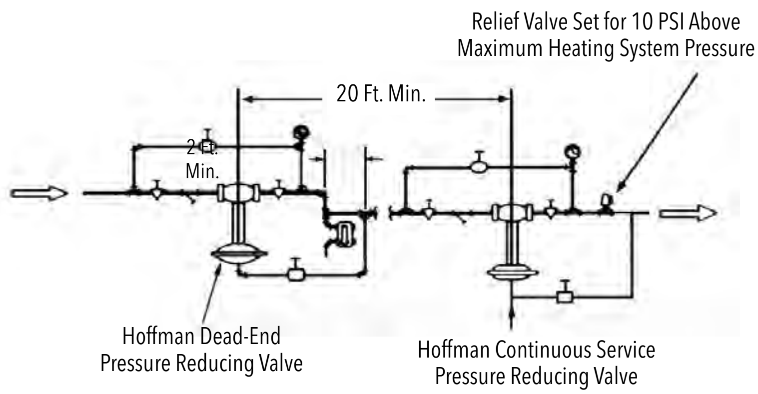


Figure 78

volume can be obtained by the addition of a receiver, or volume tanks, connected to the shorter length of pipe. In many cities, District Heating service is available. Some of these system distribution mains are maintained at low pressure, below 15 PSIG, while other distribution systems are at medium and high pressures.

Figure 79 shows a typical installation of a pressure reducing valve supplying steam to a heating system from street steam.

District Heating Systems do not return condensate to the remote boiler plant but instead, discharge it to the sewer. Two methods are used to measure the steam consumption for a steam system. One method uses a steam flow meter to measure the quantity of steam as it enters the building where it is used. The more common method uses condensate meters at the common point where all the system condensate lines terminate. A condensate meter is simple in design, low in cost, accurate at all loads, and can be applied to either low or high pressure steam distribution.

Condensate meters must not operate under pressure but are made for either gravity or vacuum installations. The detail piping arrangements for four standard installations are shown on the next page.

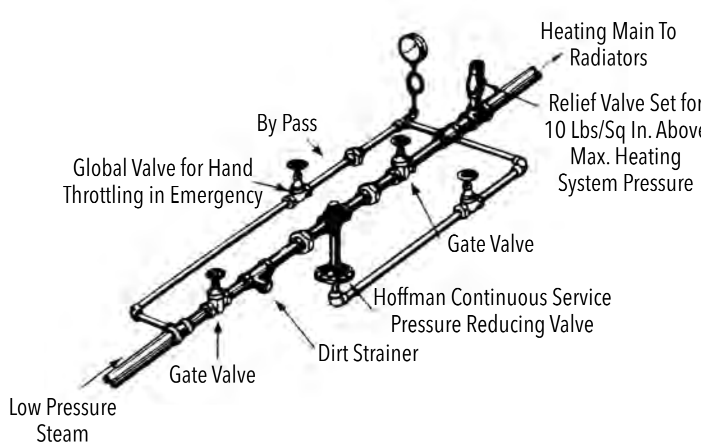


Figure 79

TYPES OF TWO-PIPE SYSTEMS

Figure 80 shows a typical gravity installation. A continuous flow type Float and Thermostatic Trap is shown ahead of the meter with a float-type Air Vent at the trap discharge. This vent discharges all the air from the system because some meter constructions will not handle air.

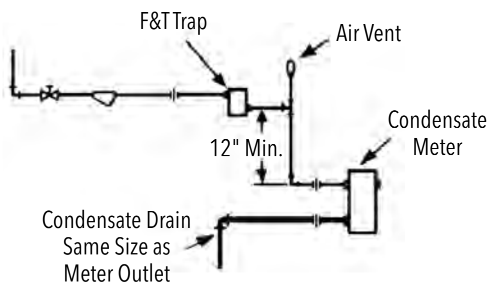


Figure 80

Figure 81 shows the installation where an intermittent type of Bucket Trap is used. A Receiver must always be used with an intermittent operating type trap so as not to overload the meter when discharging. The float-type vent is also used on the receiver.

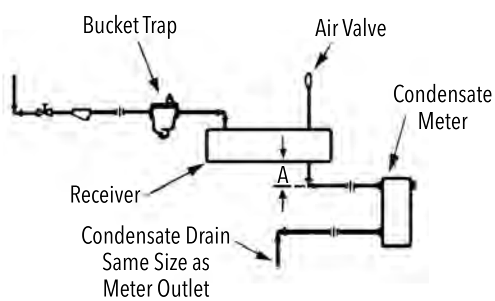


Figure 81

Figure 82 shows the installation of a Float and Thermostatic Trap ahead of the meter for a vacuum installation. The by-pass air vent line around the meter is required for this vacuum installation.

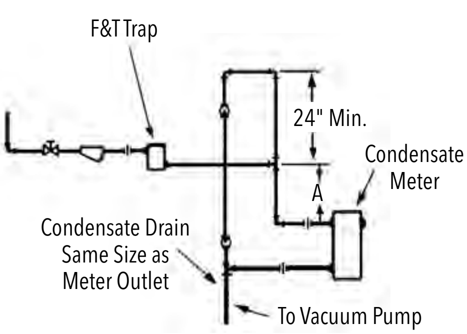


Figure 82

Figure 83 shows a vacuum installation without the use of a master trap. This can be used when other traps in the system are performing without leakage. The air by-pass piping is also required. The following table gives Dimension "A" for a variety of meter sizes for installations as shown by **Figures 81** through **83**.

Meter Capacity Lb. Per Hr.	250	500	750	1500	3000	6500	12000
	Dimension A – Inches						
Figure 81	4	4	8	8	8	12	12
Figure 82	12	12	15	15	15	18	18
Figure 83	12	12	15	15	15	18	18

Local codes in most cities do not permit the discharge of condensate into the sewer at temperatures above a specific amount. This is usually between 140°F and 150°F. In order to assure control of this maximum temperature requirement and to extract all the heat possible from the condensate before discharging it to the sewer, a method of salvaging this heat is required.

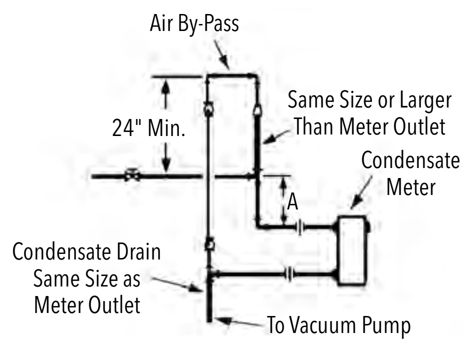


Figure 83

Figure 84 shows a method of discharging the condensate through an Economizer to preheat domestic water before it goes to the primary water heater. If the condensate is pumped, it should first be discharged into a vented receiver from which it can flow by gravity through the economizer, or preheater, and the flow meter.

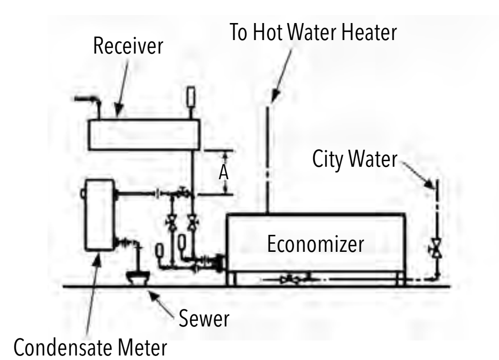


Figure 84

DEFINITIONS OF HEATING TERMS

The definitions given in this section are only those applying to heating and particularly as used in this Book. It is realized that some do not define the terms for all usages, but in the interest of clearance and space this sacrifice was made.

Absolute Humidity: The weight of water vapor in grains actually contained in one cubic foot of the mixture of air and moisture.

Absolute Pressure: The actual pressure above zero. It is the atmospheric pressure added to the gauge pressure. It is expressed as a unit pressure such as lbs. per sq. in. abso.

Absolute Temperature: The temperature of a substance measured above absolute zero. To express a temperature as absolute temperature add 460°F to the reading of a Fahrenheit thermometer or 273°F to the reading of a Centigrade one.

Absolute Zero: The temperature (-460°F. approx.) at which all molecular motion of a substance ceases, and at which the substance contains no heat.

Air: An elastic gas. It is a mechanical mixture of oxygen and nitrogen and slight traces of other gases. It may also contain moisture known as humidity. Dry air weighs 0.075 lbs. per cu. ft.

One BTU will raise the temperature of 55 cu. ft. of air one degree F.

Air expands or contracts approximately 1/490 of its volume for each degree rise or fall in temperature from 32°F.

Air Change: The number of times in an hour the air in a room is changed either by mechanical means or by the infiltration of outside air leaking into the room through cracks around doors and windows, etc.

Air Cleaner: A device designed for the purpose of removing air-borne impurities, such as dust, fumes, and smokes. (Air cleaners include air washers and air filters.)

Air Conditioning: The simultaneous control of the temperature, humidity, air motion, and air distribution within an enclosure. Where human comfort and health are involved, a reasonable air purity with regard to dust, bacteria, and odors is also included.
The primary requirement of a good air conditioning system is a good heating system.

Air Infiltration: The leakage of air into a house through cracks and crevices, and through doors, windows, and other openings, caused by wind pressure and/or temperature difference.

Air Valve: See Vent Valve.

Atmospheric Pressure: The weight of a column of air, one square inch in cross section and extending from the earth to the upper level of the blanket of air surrounding the earth. This air exerts a pressure of 14.7 pounds per square inch at sea level, where water will boil at 212°F. High altitudes have lower atmospheric pressure with correspondingly lower boiling point temperatures.

Boiler: A closed vessel in which steam is generated or in which water is heated by fire.

Boiler Heating Surface: The area of the heat transmitting surfaces in contact with the water (or steam) in the boiler on one side and the fire or hot gases on the other.

Boiler Horse Power: The equivalent evaporation of 34.5 lbs. of water per hour at 212°F. to steam at 212°F. This is equal to a heat output of 33,475 BTU per hour, which is equal to approximately 140 sq. ft. of steam radiation (EDR).

British Thermal Unit (BTU): The quantity of heat required to raise the temperature of 1 lb. of water 1°F. This is somewhat approximate but sufficiently accurate for any work discussed in this Book.

Bucket Trap (Inverted): A float trap with an open float. The float or bucket is open at the bottom. When the air or steam in the bucket has been replaced by condensate the bucket loses its buoyancy and when it sinks it opens a valve to permit condensate to be pushed into the return.

Bucket Trap (Open): The bucket (float) is open at the top. Water surrounding the bucket keeps it floating and the pin is pressed against its seat. Condensate from the system drains into the bucket. When enough has drained into it so that the bucket loses its buoyancy it sinks and pulls the pin off its seat and steam pressure forces the condensate out of the trap.

Calorie (Small): The quantity of heat required to raise 1 gram of water 1°C (approx.).

Calorie (Large): The quantity of heat required to raise 1 kilogram of water 1°C (approx.).

Centigrade: A thermometer scale at which the freezing point of water is 0°F and its boiling is 100°F. In this country it is only used in scientific and laboratory work.

Central Fan System: A mechanical indirect system of heating, ventilating, or air conditioning consisting of a central plant where the air is heated and/or conditioned and then circulated by fans or blowers through a system of distributing ducts.

Chimney Effect: The tendency in a duct or other vertical air passage for air to rise when heated due to its decrease in density.

Circulating Pipe (Hot Water System): The pipe and orifice in a Hoffman Panelmatic Hot Water Control System through which the return water by-passes the boiler until the temperature of the circulating stream is too low at which time part of it is replaced by the correct quantity of hot boiler water to restore its temperature.

DEFINITIONS OF HEATING TERMS

Coefficient of Heat Transmission (Over-all)-U-: The amount of heat (BTU) transmitted *from air to air* in one hour per square foot of the wall, floor, roof, or ceiling for a difference in temperature of one degree Fahrenheit *between the air on the inside and outside of the wall, floor, roof, or ceiling.*

Column Radiator: A type of direct radiator. (This radiator has not been listed by manufacturers since 1926.)

Comfort Line: The effective temperature at which the largest percentage of adults feel comfortable. Comfort Zone (Average): The range of effective temperatures over which the majority of adults feel comfortable.

Concealed Radiator: See Convector.

Condensate: In steam heating, the water formed by cooling steam as in a radiator. The capacity of traps, pumps, etc., is sometimes expressed in lbs. of condensate they will handle per hour. One pound of condensate per hour is equal to approximately 4 sq. ft. of steam heating surface (240 BTU per hour per sq. ft.).

Conductance (Thermal)-C-: The amount of heat (BTU) transmitted *from surface to surface*, in one hour through one square foot of a material or construction *for the thickness or type under consideration* for a difference in temperature of one degree Fahrenheit between the two surfaces.

Conduction (Thermal): The transmission of heat through and by means of matter.

Conductivity (Thermal)-k-: The amount of heat (BTU) transmitted in one hour through one square foot of a homogenous material one inch thick for a difference in temperature of one degree Fahrenheit between the two surfaces of the material.

Conductor (Thermal): A material capable of readily transmitting heat by means of conduction.

Convection: The transmission of heat by the circulation (either natural or forced) of a liquid or a gas such as air. If natural, it is caused by the difference in weight of hotter and colder fluid.

Convector: A concealed *radiator*. An enclosed heating unit located (with enclosure) either within, adjacent to, or exterior to the room or space to be heated, but transferring heat to the room or space mainly by the process of convection. A shielded heating unit is also termed a convector. If the heating unit is located exterior to the room or space to be heated, the heat is transferred through one or more ducts or pipes.

Convertor: A piece of equipment for heating water with steam without mixing the two. It may be used for supplying hot water for domestic purposes or for a hot water heating system.

Cooling Leg: A length of uninsulated pipe through which the condensate flows to a trap and which has sufficient cooling surface to permit the condensate to dissipate enough heat to prevent flashing when the trap opens. In the case of a thermostatic trap a cooling leg may be necessary to permit the condensate to drop a sufficient amount in temperature to permit the trap to open.

Degree-Day: (Standard) A unit which is the difference between 65°F. and the daily average temperature when the latter is below 65°F. The "degree days" in any one day is equal to the number of degrees F. that the average temperature for that day is below 65°F.

Dew-Point Temperature: The air temperature corresponding to saturation (100 per cent relative humidity) for a given moisture content. It is the lowest temperature at which air can retain the water vapor it contains.

Direct-Indirect Heating Unit: A heating unit located in the room or space to be heated and partially enclosed, the enclosed portion being used to heat air which enters from outside the room.

Direct Radiator: Same as *radiator:*

Direct-Return System (Hot Water): A two-pipe hot water system in which the water after it has passed through a heating unit, is returned to the boiler along a direct path so that the total distance traveled by the water from each radiator is the shortest feasible. There is, therefore, a considerable difference in the lengths of the several circuits composing the system.

Domestic Hot Water: Hot water used for purposes other than for house heating such as for laundering, dish washing, bathing, etc.

Down-Feed One-Pipe Riser (Steam): A pipe which carries steam downward to the heating units and into which the condensation from the heating units drains.

Down-Feed System (Steam): A steam heating system in which the supply mains are above the level of the heating units which they serve.

Dry-Bulb Temperature: The temperature of the air as determined by an ordinary thermometer.

Dry Return (Steam): A return pipe in a steam heating system which carries both water of condensation and air.

Dry Saturated Steam: Saturated steam containing no water in suspension.

Equivalent Direct Radiation (E.D.R.): See Square Foot of Heating Surface.

Extended Heating Surface: Heating surface consisting of ribs, fins, or ribs which receive heat by conduction from the prime surface.

DEFINITIONS OF HEATING TERMS

Extended Surface Heating Unit: A heating unit having a relatively large amount of extended surface which may be integral with the core containing the heating medium or assembled over such a core, making good thermal contact by pressure, or by being soldered to the core or by both pressure and soldering. (An extended surface heating unit is usually placed within an enclosure and therefore functions as a convector.)

Fahrenheit: A thermometer scale at which the freezing point of water is 32° and its boiling point is 212° above zero. It is generally used in this country for expressing temperature.

Flash (Steam): The rapid passing into steam of water at a high temperature when the pressure it is under is reduced so that its temperature is above that of its boiling point for the reduced pressure. For example: If hot condensate is discharged by a trap into a low pressure return or into the atmosphere, a certain percentage of the water will be immediately transformed into steam. It is also called re-evaporation.

Float & Thermostatic Trap: A float trap with a thermostatic element for permitting the escape of air into the return line.

Float Trap: A steam trap which is operated by a float. When enough condensate has drained (by gravity) into the trap body the float is lifted which in turn lifts the pin off its seat and permits the condensate to flow into the return until the float has been sufficiently lowered to close the port. Temperature does not effect the operation of a float trap.

Furnace: That part of a boiler or warm air heating plant in which combustion takes place. Sometimes also the complete heating unit of a warm air heating system.

Gauge Pressure: The pressure above that of the atmosphere. It is the pressure indicated on an ordinary pressure gauge. It is expressed as a unit pressure such as lbs. per sq. in. gauge

Grille: A perforated covering for an air inlet or outlet usually made of wire screen, pressed steel, cast-iron or other material.

Head: Unit pressure usually expressed in ft. of water.

Heat: That form of energy into which all other forms may be changed. Heat always flows from a body of higher temperature to a body of lower temperature. See also: Latent Heat, Sensible Heat, Specific Heat, Total Heat, Heat of the Liquid.

Heat of the Liquid: The heat (BTU) contained in a liquid due to its temperature. The heat of the liquid for water is zero at 32°F. and increases 1 BTU approximately for every degree rise in temperature.

Heat Unit: In the foot-pound-second system, the British Thermal Unit (BTU) in the centimeter-gram-second system, the calorie (cal.).

Heating Medium: A substance such as water, steam, or air used to convey heat from the boiler, furnace, or other source of heat to the heating units from which the heat is dissipated.

Heating Surface: The exterior surface of a heating unit. See also Extended Heating Surface.

Heating Unit: Radiators, convectors, base boards, finned tubing, coils embedded in floor, wall, or ceiling, or any device which transmits the heat from the heating system to the room and its occupants.

Horsepower: A unit to indicate the time rate of doing work equal to 550 ft.-lb. per second, or 33,000 ft.-lb. per minute. One horsepower equals 2545 BTU per hour or 746 watts.

Hot Water Heating System: A heating system in which water is used as the medium by which heat is carried through pipes from the boiler to the heating units.

Humidstat: An instrument which controls the relative humidity of the air in a room.

Humidity: The water vapor mixed with air.

Insulation (Thermal): A material having a high resistance to heat flow.

Latent Heat of Evaporation: The heat (BTU per pound) necessary to change 1 pound of liquid into vapor without raising its temperature. In round numbers this is equal to 960 BTU per pound of water.

Latent Heat of Fusion: The heat necessary to melt one pound of a solid without raising the temperature of the resulting liquid. The latent heat of fusion of water (melting 1 pound of ice) is 144 BTU.

Mechanical Equivalent of Heat: The mechanical energy equivalent to 1 BTU which is equal to 778 ft. – lb.

Mil-Inch: One one-thousandth of an inch (0.001").

One-Pipe Supply Riser (Steam): A pipe which carries steam to a heating unit and which also carries the condensation from the heating unit. In an up feed riser steam travels upwards and the condensate downward while in a down feed both steam and condensate travel down.

One-Pipe System (Hot Water): A hot water heating system in which one-pipe serves both as a supply main and also as a return main. The heating units have separate supply and return pipes but both are connected to the same main.

One-Pipe System (Steam): A steam heating system consisting of a main circuit in which the steam and condensate flow in the same pipe. There is but one connection to each heating unit which must serve as both the supply and the return.

Overhead system: Any steam or hot water system in which the supply main is above the heating units. With a steam system the return must be below the heating units; with a water system, the return may be above the heating units.

DEFINITIONS OF HEATING TERMS

Panel Heating: A method of heating involving the installation of the heating units (pipe coils) within the wall, floor or ceiling of the room.

Panel Radiator: A heating unit placed on, or flush with, a flat wall surface and intended to function essentially as a radiator. Do not confuse with panel heating system.

Plenum Chamber: An air compartment maintained under pressure and connected to one or more distributing ducts.

Pressure: Force per unit area such as lb. per sq. inch. See Static, Velocity, and Total Gauge and Absolute Pressures. Unless otherwise qualified, it refers to unit static gauge pressure.

Pressure Reducing Valve: A piece of equipment for changing the pressure of a gas or liquid from a higher to a lower one.

Prime Surface: A heating surface having the heating medium on one side and air (or extended surface) on the other.

Radiant Heating: A heating system in which the heating is by radiation only. Sometimes applied to Panel Heating System.

Radiation: The transmission of heat in a straight line through space.

Radiator: A heating unit located within the room to be heated and exposed to view. A radiator transfers heat by radiation to objects "it can see" and by conduction to the surrounding air which in turn is circulated by natural convection.

Recessed Radiator: A heating unit set back into a wall recess but not enclosed.

Reducing Valve: See Pressure Reducing Valve.

Re-Evaporation: See Flash.

Refrigeration, Ton of: See Ton of Refrigeration.

Register: A grille with a built-in damper or shutter.

Relative Humidity: The amount of moisture in a given quantity of air compared with the maximum amount of moisture the same quantity of air could hold at the same temperature. It is expressed as a percentage.

Return Mains: The pipes which return the heating medium from the heating units to the source of heat supply.

Reverse-Return System (Hot Water): A two-pipe hot water heating system in which the water from the several heating units is returned along paths arranged so that all radiator circuits of the system are practically of equal length.

Roof Ventilator: A device placed on the roof of a building to permit egress of air.

Sensible Heat: Heat which only increases the temperature of objects as opposed to latent heat.

Specific Heat: In the foot-pound-second system, the amount of heat (BTU) required to raise one pound of a substance one degree Fahrenheit. In the centimeter- gram-second system, the amount of heat (cal.) required to raise one gram of a substance one degree centigrade. The specific heat of water is 1.

Split System: A system in which the heating is accomplished by radiators or convectors and ventilation by separate apparatus.

Square Foot of Heating Surface: Equivalent direct radiation (EDR). By definition, that amount of heating surface which will give off 240 BTU per hour when filled with a heating medium at 215°F. and surrounded by air at 70°F. The equivalent square foot of heating surface may have no direct relation to the actual surface area.

Static Pressure: The pressure which tends to burst a pipe. It is used to overcome the frictional resistance to flow through the pipe. It is expressed as a unit pressure and may be either in absolute or gauge pressure. It is frequently expressed in feet of water column or (in the case of pipe friction) in mil-inches of water column per ft. of pipe.

Steam: Water in the vapor phase. The vapor formed when water has been heated to its boiling point, corresponding to the pressure it is under. See also Dry Saturated Steam, Wet Saturated Steam, Super Heated Steam.

Steam Heating System: A heating system in which the heating units give up their heat to the room by condensing the steam furnished to them by a boiler or other source.

Steam Trap: A device for allowing the passage of condensate and air but preventing the passage of steam. See Thermostatic, Float, Bucket Trap.

Superheated Steam: Steam heated above the temperature corresponding to its pressure.

Supply Mains: The pipes through which the heating medium flows from the boiler or source of supply to the run-outs and risers leading to the heating units.

Tank Regulator: See Temperature Regulator.

Temperature Regulator: A piece of equipment for controlling the admission of steam to a hot water (or other liquid) heating device in the correct quantities so that the temperature of the liquid will remain constant.

Thermostat: An instrument which responds to changes in temperature and which directly or indirectly controls the room temperature.

DEFINITIONS OF HEATING TERMS

Thermostatic Trap: A steam trap which opens by a drop in temperature such as when cold condensate or air reaches it and closes it when steam reaches it. The temperature sensitive element is usually a sealed bellows or series of diaphragm chambers containing a small quantity of volatile liquid.

Ton of Refrigeration: The heat which must be extracted from one ton (2,000 lbs.) of water at 32°F. to change it into ice at 32°F. in 24 hours. It is equal to 288,000 BTU/24 hours, 12,000 BTU/hour, or 200 BTU/ minute.

Total Heat: The latent heat of vaporization added to the heat of the liquid with which it is in contact.

Total Pressure: The sum of the static and velocity pressures. It is also used as the total static pressure over an entire area, that is, the unit pressure multiplied by the area on which it acts.

Trap: See Steam Trap, Thermostatic Trap, Float Trap, and Bucket Trap.

Two-Pipe System (Steam or Water): A heating system in which one pipe is used for the supply main and another for the return main. The essential feature of a two-pipe hot water system is that each heating unit receives a direct supply of the heating medium which cannot have served a preceding heating unit.

Unit Heater: A heating unit consisting of a heat transfer element, a housing, a fan with driving motor, and outlet deflectors or diffusers. It is usually suspended from the ceiling and its heat output is controlled by starting and stopping the fan by a room thermostat. The circulation of the heating medium (steam or hot water) is usually continuous. It is used mostly for industrial heating.

Unit Pressure: Pressure per unit area as lbs. per sq. in.

Up-Feed System (Hot Water or Steam): A heating system in which the supply mains are below the level of the heating units which they serve.

Vacuum Heating System (Steam): A one- or two-pipe heating system equipped with the necessary accessory apparatus to permit the pressure in the system to go below atmospheric.

Vapor: Any substance in the gaseous state.

Vapor Heating System (Steam): A two-pipe heating system which operates under pressure at or near atmospheric and which returns the condensation to the boiler or receiver by gravity.

Velocity Pressure: The pressure used to create the velocity of flow in a pipe. It is expressed as a unit pressure.

Ventilation: Air circulated through a room for ventilating purposes. It may be mechanically circulated with a blower system or it may be natural circulation through an open window, etc.

Vent Valve (Steam): A device for permitting air to be forced out of a heating unit or pipe and which closes against water and steam.

Vent Valve (Water): A device permitting air to be pushed out of a pipe or heating unit but which closes against water.

Warm Air Heating System: A warm air heating plant consists of a heating unit (fuel-burning furnace) enclosed in a casing, from which the heated air is distributed to the various rooms of the building through ducts. If the motive head producing flow depends on the difference in weight between the heated air leaving the casing and the cooler air entering the bottom of the casing, it is termed a gravity system. A booster fan may, however, be used in conjunction with a gravity-designed system. If a fan is used to produce circulation and the system is designed especially for fan circulation, it is termed a fan furnace system or a central fan furnace system. A fan furnace system may include air washer, filters, etc.

Wet Bulb Temperature: The lowest temperature which a water-wetted body will attain when exposed to an air current.

Wet Return (Steam): That part of a return main of a steam heating system which is completely filled with water of condensation.

Wet Saturated Steam: Saturated steam containing some water particles in suspension.

ENGINEERING DATA AND TECHNICAL INFORMATION

This section contains necessary Engineering technical information used in the design, installation, maintenance or revamping of low pressure steam heating systems. The Tables, Charts, and Examples, including properties of saturated steam, steam capacities of ASTM Schedule 40(S) Pipe and return pipe capacities used in One-Pipe and Two-Pipe Steam Heating Systems. Also included is data for determining pipe friction allowance, steam velocity, system pressure drops and similar information.

PROPERTIES OF SATURATED STEAM (APPROX.)

Absolute Pressure	Gage Reading at Sea Level	Temp. °F.	Heat in Water B.T.U. per Lb.	Latent Heat in Steam (Vaporization) B.T.U. per Lb.	Volume of 1 Lb. Steam Cu. Ft.	Wgt. of Water Lbs. per Cu. Ft.
INCHES OF MERCURY	VACUUM—INCHES OF MERCURY					
0.18	29.7	32	0.0	1076	3306	62.4
0.50	29.4	59	27.0	1061	1248	62.3
1.0	28.9	79	47.0	1049	653	62.2
2.0	28	101	69	1037	341	62.0
4.0	26	125	93	1023	179	61.7
6.0	24	141	109	1014	120	61.4
8.0	22	152	120	1007	93	61.1
10.0	20	161	129	1002	75	60.9
12.0	18	169	137	997	63	60.8
14.0	16	176	144	993	55	60.6
16.0	14	182	150	989	48	60.5
18.0	12	187	155	986	43	60.4
20.0	10	192	160	983	39	60.3
22.0	8	197	165	980	36	60.2
24.0	6	201	169	977	33	60.1
26.0	4	205	173	975	31	60.0
28.0	2	209	177	972	29	59.9
29.0	1	210	178	971	28	59.9
30.0	0	212	180	970	27	59.8
POUNDS PER SQ. INCH	PRESSURE—POUNDS PER SQ. INCH					
14.7	0	212	180	970	27	59.8
15.7	1	216	184	968	25	59.8
16.7	2	219	187	966	24	59.7
17.7	3	222	190	964	22	59.6
18.7	4	225	193	962	21	59.5
19.7	5	227	195	960	20	59.4
20.7	6	230	198	958	19	59.4
21.7	7	232	200	957	19	59.3
22.7	8	235	203	955	18	59.2
23.7	9	237	205	954	17	59.2
25	10	240	208	952	16	59.2
30	15	250	219	945	14	58.8
35	20	259	228	939	12	58.5
40	25	267	236	934	10	58.3
45	30	274	243	929	9	58.1
50	35	281	250	924	8	57.9
55	40	287	256	920	8	57.7
60	45	293	262	915	7	57.5
65	50	298	268	912	7	57.4
70	55	303	273	908	6	57.2
75	60	308	277	905	6	57.0
85	70	316	286	898	5	56.8
95	80	324	294	892	5	56.5
105	90	332	302	886	4	56.3
115	100	338	309	881	4	56.0
140	125	353	325	868	3	55.5

Table 1

RELATIONS OF ALTITUDE, PRESSURE & BOILING POINT

Altitude Feet	ATMOSPHERIC PRESSURE ABSOLUTE		BOILING POINT OF WATER °F. (GAGE PRESSURE PSI)				
	Inches of Mercury (Barometer)	Lbs. per Sq. In.	0	1	5	10	15
—500	30.46	14.96	212.8	216.1	227.7	239.9	250.2
—100	30.01	14.74	212.3	215.5	227.2	239.4	249.9
Sea Level	29.90	14.69	212.0	215.3	227.0	239.3	249.7
500	29.35	14.42	211.0	214.4	226.3	238.7	249.2
1000	28.82	14.16	210.1	213.5	225.5	238.1	248.6
1500	28.30	13.90	209.4	212.7	225.0	237.6	248.2
2000	27.78	13.65	208.2	211.7	224.1	236.8	247.7
2500	27.27	13.40	207.3	210.9	223.4	236.3	247.2
3000	26.77	13.15	206.4	210.1	222.7	235.7	246.7
3500	26.29	12.91	205.5	209.2	222.1	235.1	246.2
4000	25.81	12.68	204.7	208.4	221.4	234.6	245.7
4500	25.34	12.45	203.7	207.5	220.7	234.0	245.2
5000	24.88	12.22	202.7	206.8	220.1	233.4	244.7
6000	23.98	11.78	200.9	205.0	218.7	232.4	243.8
7000	23.11	11.35	199.1	203.3	217.3	231.3	242.9
8000	22.28	10.94	197.4	201.6	216.1	230.3	242.0
9000	21.47	10.55	195.7	200.0	214.8	229.3	241.3
10000	20.70	10.17	194.0	198.4	213.5	228.3	240.4
11000	19.95	9.80	192.2	196.8	212.3	227.3	239.6
12000	19.23	9.45	190.6	195.2	211.1	226.3	238.7
13000	18.53	9.10	188.7	193.6	209.9	225.4	237.9
14000	17.86	8.77	187.2	192.3	208.8	224.5	237.2
15000	17.22	8.46	185.4	190.6	207.6	223.6	236.4

Table 2

ENGINEERING DATA AND TECHNICAL INFORMATION

VELOCITY OF STEAM

To find the approximate velocity of low pressure steam (ft. per second) through a pipe, multiply the condensation in pounds per hour by the volume of steam in cubic feet per pound corresponding to the steam pressure. Divide this result by 25 times the internal area of the pipe.

The pipe area is found in the Table ASTM Schedule 40(S) Pipe Dimensions (page 75) and the volume per cu. ft. is found in the Table "Properties of Saturated Steam."

Example. What is the velocity of steam at 5 lbs. per sq. inch flowing through a 2" pipe at a rate to produce 175 lbs. of condensate per hour?

1 pound of steam at 5 lbs. pressure = 20 cu. ft.
Internal area of 2" pipe = 3.36

The velocity of steam is:

$$\frac{175 \times 20}{25 \times 3.36} = \frac{3500}{84} = 41.7 \text{ ft. per sec.}$$

Table 3 gives the capacity of ASTM Schedule 40(S) pipe expressed in square feet EDR. The values are obtained from charts published by the American Society of Heating, Refrigerating and Air Conditioning Engineers' 1967 GUIDE.

STEAM CAPACITY[a] OF ASTM[b] SCHEDULE 40(S) PIPE AT INTERNAL PRESSURES OF 3.5 and 12 PSIG[c]

FLOW EXPRESSED IN SQ. FT. EDR

NOMINAL PIPE SIZE INCHES	PRESSURE DROP—PSI PER 100 FT. IN LENGTH													
	1/16 PSI (1 oz.)		1/8 PSI (2 oz.)		1/4 PSI (4 oz.)		1/2 PSI (8 oz.)		3/4 PSI (12 oz.)		1 PSI		2 PSI	
	SAT. PR. 3.5	PSIG 12	SAT. PR. 3.5	PSIG 12	SAT. PR. 3.5	PSIG 12	SAT. PR. 3.5	PSIG 12	SAT. PR. 3.5	PSIG 12	SAT. PR. 3.5	PSIG 12	SAT. PR. 3.5	PSIG 12
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
¾"	36	44	56	64	80	96	116	140	144	172	168	200	240	292
1"	68	84	104	124	148	184	216	264	272	328	324	380	456	548
1¼"	144	180	212	264	312	384	444	552	560	680	648	800	928	1120
1½"	224	280	326	400	480	588	696	840	872	1040	984	1216	1440	1720
2"	432	536	648	776	936	1140	1344	1640	1680	2040	1920	2360	2840	3400
2½"	696	860	1032	1240	1512	1840	2160	2640	2720	3280	3120	3800	4600	5480
3"	1272	1520	1860	2200	2640	3240	3840	4640	4760	5720	5520	6680	7800	9600
3½"	1848	2200	2680	3200	3960	4872	5640	6800	6960	8400	8000	9680	11800	13800
4"	2560	3200	3800	4640	5640	6760	7920	9600	9800	12000	11520	13840	16800	19600
5"	4800	5720	6720	8400	9760	12000	14280	17000	17520	21000	20400	24400	30000	34400
6"	7680	9200	11280	13200	15840	19400	22800	28000	28800	34400	33600	40000	47600	56800
8"	15600	19200	22280	28000	32400	40000	45600	57200	58000	70800	66000	82000	96000	118000
10"	28800	35200	40800	50400	60000	72800	84000	104000	104800	128000	120000	148000	170800	208000
12"	45600	54800	66000	78000	93600	113600	132000	160000	164000	198000	192000	230000	271200	324000

a—Based on Moody Friction Factor where flow of condensate does not inhibit the flow of steam.
b—American Society for Testing Materials Schedule. The number 40 refers to the ASTM Schedule. The letter (S) refers to the former designation of standard weight pipe.
c—The flow rates at 3.5 psig can be used to cover saturated pressures from 1 to 6 psig, and the rates at 12 psig can be used to cover saturated pressures from 8 to 16 psig with an error not exceeding 8%.

Table 3

ENGINEERING DATA AND TECHNICAL INFORMATION

Where condensate must flow counter to steam flow, the governing factor is the velocity which will not interfere with the condensate flow. The capacity limit for horizontal pipe at various pitches is given in **Table 4**.

COMPARATIVE CAPACITY OF STEAM LINES AT VARIOUS PITCHES FOR STEAM AND CONDENSATE FLOWING IN OPPOSITE DIRECTIONS[a]

PITCH of PIPE	(PITCH OF PIPE IN INCHES PER 10 FT. / VELOCITY IN FT. PER SEC. / CAPACITY IN SQ. FT. EDR)															
	¼ inch		½ inch		1 inch		1½ inch		2 inch		3 inch		4 inch		5 inch	
PIPE SIZE INCHES	Capac-ity	Max. Vel.	Capac-ity	Max. Vel.	Capac-ity	Max. Vel.	Capac-ity	Max. Vel.	Capac-ity	Max. Vel.	Capac-ity	Max. Vel.	Capac-ity	Max. Vel.	Capac-ity	Max. Vel.
¾	12.8	8	16.4	11	22.8	13	25.6	14	28.4	16	33.2	17	38.6	22	42.0	22
1	27.2	9	36.0	12	46.8	15	51.2	17	59.2	19	69.2	22	76.8	24	82.0	25
1¼	47.2	11	63.6	14	79.6	17	98.4	20	108.0	22	125.2	25	133.6	26	152.4	31
1½	79.2	12	103.6	16	132.0	19	149.6	22	168.0	24	187.2	26	203.2	28	236.8	33
2	171.6	15	216.0	18	275.2	24	333.2	27	371.6	30	398.4	32	409.6	32	460.0	33

a—From research sponsored by ASHRAE

Table 4

Maximum Velocity in Steam Piping: The capacity of a steam pipe depends on these factors:

(1) The quantity of condensate in the pipe

(2) The direction of condensate flow

(3) The pressure drop in the pipe. The total pressure drop of a system should not exceed one half of the supply pressure when steam and condensate are flowing in the same direction.

When steam and condensate flow in the same direction, only the pressure drop need be considered. When steam flows counter to condensate the velocities in **Table 4** must not be exceeded.

Table 5 gives steam pipe capacities in square feet EDR when steam flow is counter to condensate flow in either One-Pipe or Two-Pipe Systems.

STEAM PIPE CAPACITIES FOR LOW PRESSURE SYSTEMS

(For Use on One-Pipe Systems or Two-Pipe Systems in which Condensate Flows Against the Steam Flow)

NOMINAL	CAPACITY IN SQUARE FEET EDR				
	TWO-PIPE SYSTEMS		ONE-PIPE SYSTEMS		
	Condensate Flowing Against Steam		Supply Risers Up-feed	Radiator Valves & Vertical Connections	Radiator and Riser Horizontal Runouts
	VERTICAL	HORIZON-TAL			
A	B	C[c]	D[b]	E	F[c]
¾	32	28	24	....	28
1	56	56	44	28	28
1¼	124	108	80	64	64
1½	192	168	152	92	64
2	388	362	288	168	92
2½	636	528	464	....	168
3	1128	800	800	....	260
3½	1548	1152	1144	....	476
4	2044	1700	1520	....	744
5	4200	3152	....	....	1112
6	7200	5600	....	....	2180
8	15000	12000	....	....	....
10	28000	22800	....	....	....
12	46000	38000	....	....	....

NOTES:

a—Do not use Col. B for pressure drops of less than 1 oz./100 ft. of equivalent length of run. Use Table 3 instead.

b—Do not use Col. D for pressure drops less than ⅔ oz./100 ft. of equivalent length of run except for pipe size 3" and over. Use Table 3 instead.

c—Pitch of horizontal runouts to risers and radiators should not be less than ½ inch per ft. Where this pitch cannot be obtained, for runouts 8 ft. in length or over, increase one pipe size larger than shown in this table.

Table 5

ENGINEERING DATA AND TECHNICAL INFORMATION

In all the pipe sizing tables given in this section, the *Length of Run* is defined as the *Equivalent Length of Run*. The Equivalent Length consists of the measured length plus the allowance for fittings and valves which add resistance to the regular friction of the straight run of pipe. **Table 6** gives the value in feet to be added to the measured length of pipe.

It is not possible at the beginning of the pipe sizing process to know the Equivalent Length of Pipe so it is necessary to make an assumption. The accepted rule is to measure the actual length and double it to get the total Equivalent Length. After the pipe sizing is done for the proper pressure drop per 100 ft. of length based on the assumption, then a reexamination can be made to determine if any changes are required which will change the pipe sizing.

FRICTIONAL ALLOWANCE FOR FITTINGS IN FEET OF PIPE (TO BE ADDED TO ACTUAL LENGTH OF RUN) STEAM

SIZE OF PIPE INCHES	LENGTH IN FEET TO BE ADDED IN RUN				
	STANDARD ELBOW	SIDE OUTLET TEE	GATE VALVE	GLOBE VALVE	ANGLE VALVE
½″	1.3	3	0.3	14	7
¾″	1.8	4	0.4	18	10
1″	2.2	5	0.5	23	12
1¼″	3.0	6	0.6	29	15
1½″	3.5	7	0.8	34	18
2″	4.3	8	1.0	46	22
2½″	5.0	11	1.1	54	27
3″	6.5	13	1.4	66	34
3½″	8	15	1.6	80	40
4″	9	18	1.9	92	45
5″	11	22	2.2	112	56
6″	13	27	2.8	136	67
8″	17	35	3.7	180	92
10″	21	45	4.6	230	112

EXAMPLE:
MEASURED LENGTH = 132.0 FT.
4″ GATE VALVE = 1.9 FT.
4'4″ ELBOWS = 36.0 FT.
EQUIV. LENGTH 169.9 FT.

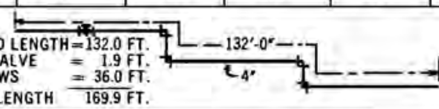


Table 6

RETURN MAIN AND RISER CAPACITIES FOR LOW PRESSURE SYSTEMS—SQ. FT. EDR.

	PIPE SIZE INCHES	1/32 psi or ½ oz. Drop per 100 ft.			1/24 psi or ⅔ oz. Drop per 100 ft.			1/16 psi or 1 oz. Drop per 100 ft.			⅛ psi or 2 oz. Drop per 100 ft.			¼ psi or 4 oz. Drop per 100 ft.			½ psi or 8 oz. Drop per 100 ft.		
		WET	DRY	VAC.	WET	DRY	VAC.	WET	DRY	VAC.	WET	DRY	VAC.	WET	DRY	VAC	WET	DRY	VAC.
	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V	W	X	Y
MAINS	¾	.....	.....	.....	.....	.....	168	.....	.....	400	.....	.....	568	.....	.....	800	.....	.....	1132
	1	500	248	.....	580	284	572	700	320	700	1000	412	996	1400	460	1400	.....	.....	1976
	1¼	852	520	.....	992	596	976	1200	672	1200	1700	868	1707	2400	964	2400	.....	.....	3392
	1½	1352	824	.....	1572	944	1552	1900	1060	1900	2700	1360	2696	3800	1512	3800	.....	.....	5360
	2	2800	1880	.....	3240	2140	3260	4000	2300	4000	5600	2960	5680	8000	3300	8000	.....	.....	11320
	2½	4720	3040	.....	6320	3472	5440	6720	3800	6720	9400	4920	9520	13400	5440	13400	.....	.....	18920
	3	7520	5840	.....	8520	6240	8720	10720	7000	10720	15000	9000	15200	21400	10000	21400	.....	.....	30240
	3½	11000	7880	.....	13200	8800	13000	16000	10000	16000	22000	12920	22720	32000	14320	32000	.....	.....	45200
	4	15520	11720	.....	18320	13400	18000	22000	15000	22000	31000	19320	31240	44000	21520	44000	.....	.....	62000
	5	.....	.....	.....	.....	.....	31520	.....	.....	38720	.....	.....	54800	.....	.....	77600	.....	.....	109200
	6	.....	.....	.....	.....	.....	50400	.....	.....	62000	.....	.....	88000	.....	.....	124000	.....	.....	175200
RISERS	¾	.....	192	.....	.....	192	572	.....	192	700	.....	192	996	.....	192	1400	.....	.....	1976
	1	.....	452	.....	.....	452	976	.....	452	1200	.....	452	1704	.....	452	2400	.....	.....	3392
	1¼	.....	992	.....	.....	992	1552	.....	992	1900	.....	992	2696	.....	992	3800	.....	.....	5360
	1½	.....	1500	.....	.....	1500	3260	.....	1500	4000	.....	1500	5680	.....	1500	8000	.....	.....	11320
	2	.....	3000	.....	.....	3000	5440	.....	3000	6720	.....	3000	9520	.....	3000	13400	.....	.....	18920
	2½	.....	.....	.....	.....	.....	8720	.....	.....	10720	.....	.....	15200	.....	.....	21400	.....	.....	30240
	3	.....	.....	.....	.....	.....	13000	.....	.....	16000	.....	.....	22720	.....	.....	32000	.....	.....	45200
	3½	.....	.....	.....	.....	.....	17920	.....	.....	22000	.....	.....	31240	.....	.....	44000	.....	.....	62000
	4	.....	.....	.....	.....	.....	31520	.....	.....	38720	.....	.....	54800	.....	.....	77600	.....	.....	109200
	5	.....	.....	.....	.....	.....	50400	.....	.....	62000	.....	.....	88000	.....	.....	124000	.....	.....	175200

Table 7

ENGINEERING DATA AND TECHNICAL INFORMATION

HORIZONTAL STEAM SUPPLY RUN-OUTS TO RADIATOR FROM MAIN OR RISER AND RADIATOR SUPPLY VALVE SIZES					
	SIZE IN INCHES				
	CAPACITY EDR				
	0-25	26-75	76-150	151- 200	201-400
HORIZONTAL RUN-OUT	¾	1	1¼	1½	2
VERTICAL CONNECTION	½	¾	1	1¼	1½
SUPPLY VALVE	½	¾	1	1¼	1½

Table 8

HORIZONTAL RETURN RUN-OUTS TO RADIATOR FROM MAIN OR RISER AND RADIATOR TRAP SIZE			
	SIZE IN INCHES		
	CAPACITY EDR		
	0-200	201-400	401-700
HORIZONTAL RUN-OUT	¾	¾	1
VERTICAL CONNECTION	½	¾	1
TRAP	½	¾	1

Table 9

EQUIVALENT CAPACITIES OF PIPES OF SAME LENGTH—STEAM												
Size	NO. OF SMALL PIPES EQUIVALENT TO ONE LARGE PIPE											
	½"	¾"	1"	1¼"	1½"	2"	2½"	3"	3½"	4"	5"	6"
½"	1.00	2.27	4.88	10.0	15.8	31.7	52.9	96.9	140	205	377	620
¾"		1.00	2.05	4.30	6.97	14.0	23.3	42.5	65	90	166	273
1"			1.00	2.25	3.45	6.82	11.4	20.9	30	44	81	133
1¼"				1.00	1.50	3.10	5.25	9.10	12	19	37	68
1½"					1.00	2.00	3.34	6.13	9	13	23	39
2"						1.00	1.67	3.06	4.5	6.5	11.9	19.6
2½"							1.00	1.82	2.70	3.87	7.12	11.7
3"								1.00	1.50	2.12	3.89	6.39
3½"									1.00	1.25	2.50	4.25
4"										1.00	1.84	3.02
5"											1.00	1.65
6"												1.00

Table 10

This table may be used to find the number of smaller pipes equivalent in steam carrying capacity to one larger pipe. It may also be used to find the size of a larger pipe equivalent to several smaller ones. The pipes in either case must be of the same lengths.

EXAMPLE 1 – Find the number of 1" pipes each 50 ft. long equivalent to one 4" pipe 50 ft. long.

SOLUTION 1 – Follow down column headed 4" to the point opposite 1" in vertical column, and it is found that it will take 44 of the 1" pipes in parallel to equal one 4" pipe in steam carrying capacity.

EXAMPLE 2 – Find the size of one pipe equivalent to four 2" pipes in steam carrying capacity.

SOLUTION 2 – Find 2" in vertical column headed "Size" and follow across horizontally until closest number to 4 is found. The nearest to 4 is 4.5. Following this column up it is found that the size is 3.5". One 3.5' pipe is, therefore, equivalent in steam carrying capacity to approximately four 2" pipes.

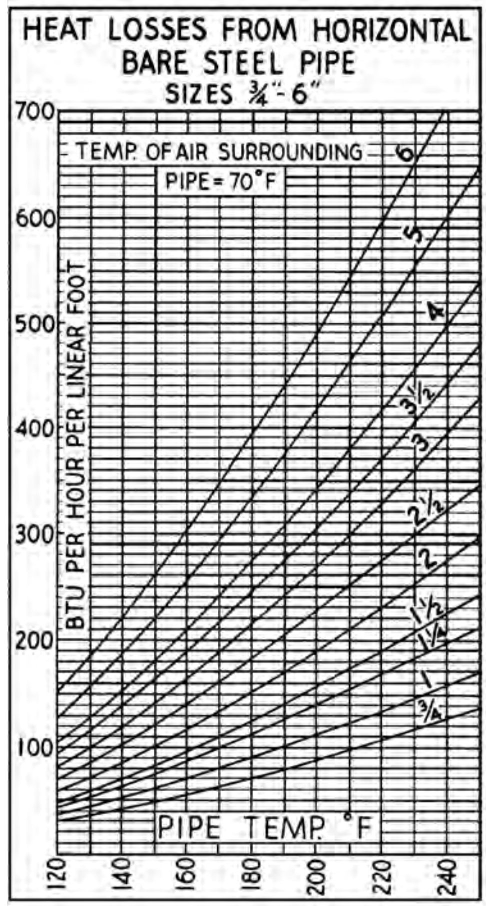


Chart 1

ENGINEERING DATA AND TECHNICAL INFORMATION

HEAT LOSSES FROM COVERED PIPE
85 PERCENT MAGNESIA TYPE
BTU PER LINEAR FOOT PER HOUR PER °F TEMPERATURE DIFFERENCE (SURROUNDING AIR ASSUMED 75°F)

Pipe Size	Insulation Thickness, Inches	MAX. TEMP. OF PIPE SURFACE °F.				
		125	175	225	275	325
½	1	.145	.150	.157	.160	.162
¾	1	.165	.172	.177	.180	.182
1	1	.190	.195	.200	.203	.207
	1½	.160	.165	.167	.170	.175
1¼	1	.220	.225	.232	.237	.245
	1½	.182	.187	.193	.197	.200
1½	1	.240	.247	.255	.260	.265
	1½	.200	.205	.210	.215	.219
2	1	.282	.290	.297	.303	.307
	1½	.230	.235	.240	.243	.247
	2	.197	.200	.205	.210	.217
2½	1	.322	.330	.340	.345	.355
	1½	.260	.265	.270	.275	.280
	2	.220	.225	.230	.237	.242
3	1	.375	.385	.395	.405	.415
	1½	.300	.305	.312	.320	.325
	2	.253	.257	.263	.270	.277
3½	1	.419	.430	.440	.450	.460
	1½	.332	.339	.345	.352	.360
	2	.280	.285	.290	.295	.303
4	1	.460	.470	.480	.492	.503
	1½	.362	.370	.379	.385	.392
	2	.303	.308	.315	.320	.327
5	1	.545	.560	.572	.585	.600
	1½	.423	.435	.442	.450	.460
	2	.355	.360	.367	.375	.382
6	1	.630	.645	.662	.680	.693
	1½	.487	.500	.510	.520	.530
	2	.405	.415	.420	.430	.437
8	1	.790	.812	.835	.850	.870
	1½	.603	.620	.635	.645	.660
	2	.495	.507	.517	.527	.540

Table 11

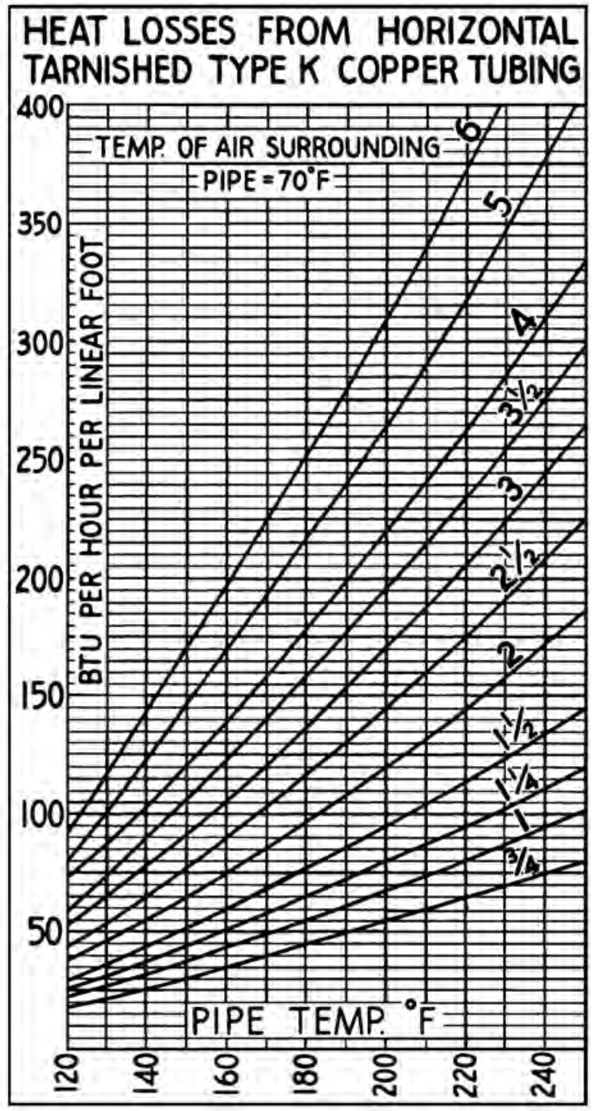


Chart 2

ENGINEERING DATA AND TECHNICAL INFORMATION

EXAMPLE FOR USING RE-EVAPORATION CHART

PROBLEM

Condensate at 5 PSI gage is delivered into a 20" Hg vacuum line at a rate of 500 lbs. per hour. What is the total re-evaporation?

Solution No. 1

Assume that the operation is at sea level and that condensate temperature is the same as that of steam at 5 PSI gage. A vacuum of 20" Hg is equal to a pressure of about 10 PSI (gage) below atmosphere. Then the pressure drop is:

$$5 - (-10) = 15 \text{ PSI}$$

Erect a perpendicular at 5 PSI (gage) and draw a horizontal through its intersection with the 15 PSI pressure drop curve. This intersects the vertical scale at 6.8 lbs. re-evaporation per 100 lbs. of condensate. (See solid line on Chart.) Total re-evaporation is:

$$5 \text{ X } 6.8 = 34.0 \text{ lbs. per hour}$$

Solution No. 2

Assume operation is at Denver. The Table "Highest and Lowest Elevations of Principal Cities" of this section gives the altitude of Denver as 5500 ft. On the Table "Altitude, Pressure, etc." the atmospheric pressure for this altitude is given as 12.00 PSI. The absolute pressure of 5 PSI (gage) at Denver is therefore:

$$12.0 + 5 = 17.0 \text{ PSI absolute}$$

Dash lines trace out this solution and it is found that the re-evaporation is at the rate of 9.2 lbs. per 100 lbs. of condensate. Total re-evaporation is:

$$5 \text{ X } 9.2 = 46.0 \text{ lbs. per hour}$$

It can be seen that it is very important to correct for altitude at these pressures. When the pressures are higher, say in the neighborhood of 100 PSI gage, the correction for altitude becomes of minor importance as can be seen by working out a similar problem for a higher pressure.

This chart also shows the necessity of cooling the condensate before it is delivered to the receiver of a vacuum pump. In order to reduce the re-evaporation to zero, which is desirable so as not to interfere with the operation of the pump, the condensate temperature must be reduced to that of saturated steam under return line pressure. This temperature can be found in the Table "Properties of Saturated Steam." Return pipes on vacuum systems should, therefore, never be insulated.

If the re-evaporation of a relatively high pressure line is to be used, say to augment a heating system, then it is advantageous to insulate all return lines of the high pressure system so that as much heat as possible may be retained.

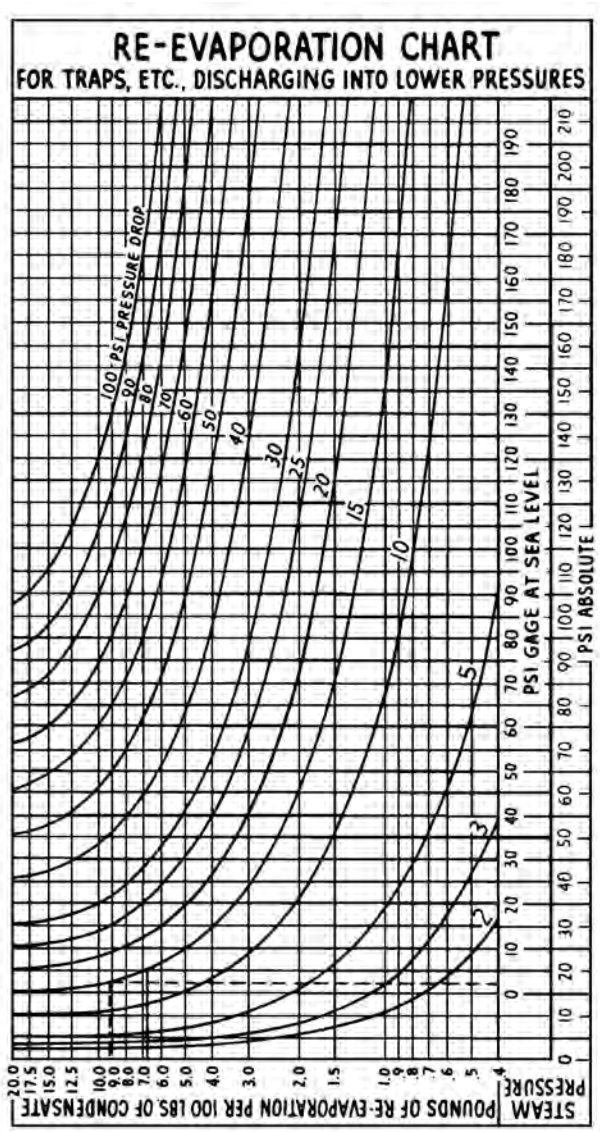


Chart 3

TYPICAL PIPING CONNECTIONS FOR ONE-PIPE AND TWO-PIPE STEAM HEATING SYSTEMS

The piping and radiator connections shown in this section are diagrammatic and illustrate the proper method of making piping connections. They are not dimensional and cannot be scaled for pipe size or product size.

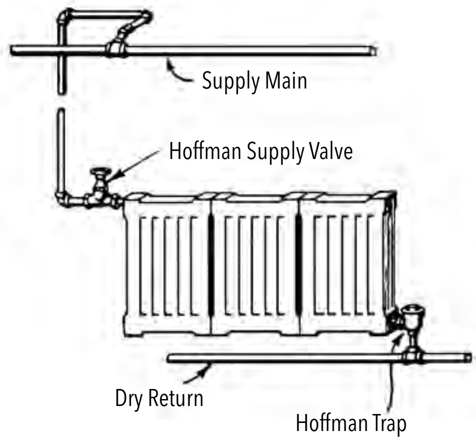
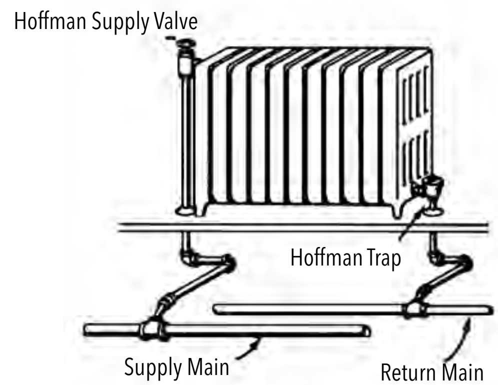
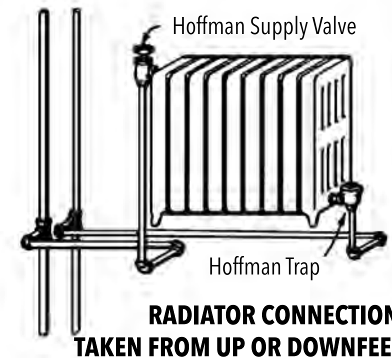


Figure 85
TWO PIPE STEAM SYSTEMS - RADIATOR CONNECTIONS

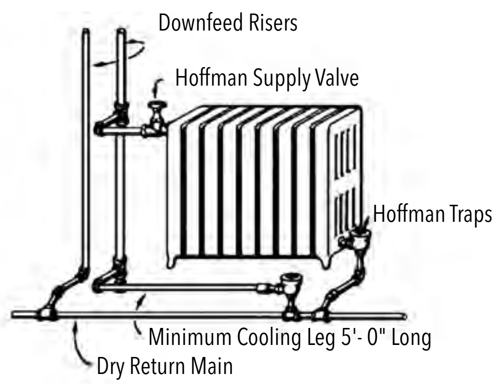


UPFEED CONNECTIONS TO RADIATOR



RADIATOR CONNECTIONS TAKEN FROM UP OR DOWNFEED RISERS

Figure 87
TWO PIPE STEAM SYSTEMS - RADIATOR CONNECTIONS



DRIPPING HEEL OF DOWN FEED RISER INTO DRY RETURN

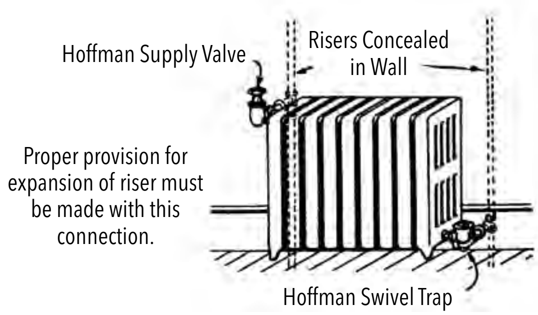
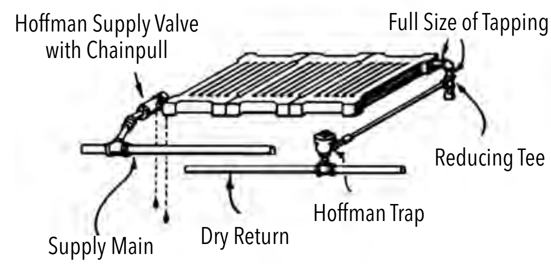
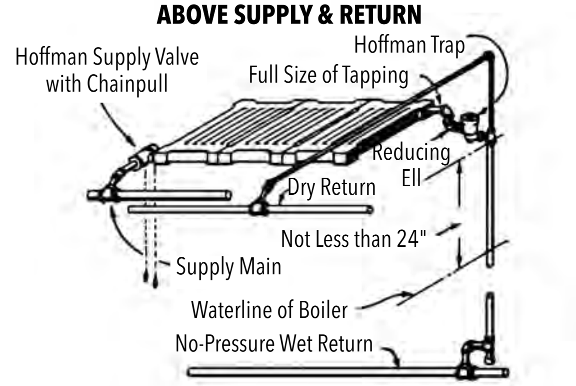


Figure 86
TWO PIPE STEAM SYSTEMS - RADIATOR CONNECTIONS



CONNECTIONS TO CEILING RADIATOR LOCATED ABOVE SUPPLY & RETURN



CONNECTIONS TO CEILING RADIATORS WITH RETURN BLED INTO WET RETURN

Figure 88
TWO PIPE STEAM SYSTEMS - RADIATOR CONNECTIONS

TYPICAL PIPING CONNECTIONS FOR ONE-PIPE AND TWO-PIPE STEAM HEATING SYSTEMS

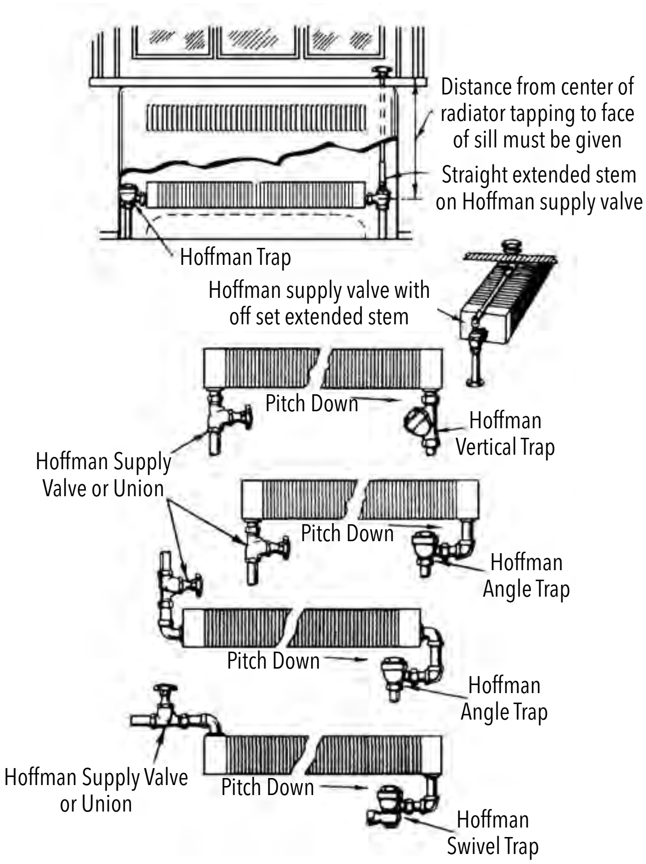


Figure 89
TWO PIPE STEAM SYSTEMS - CONVECTOR CONNECTIONS

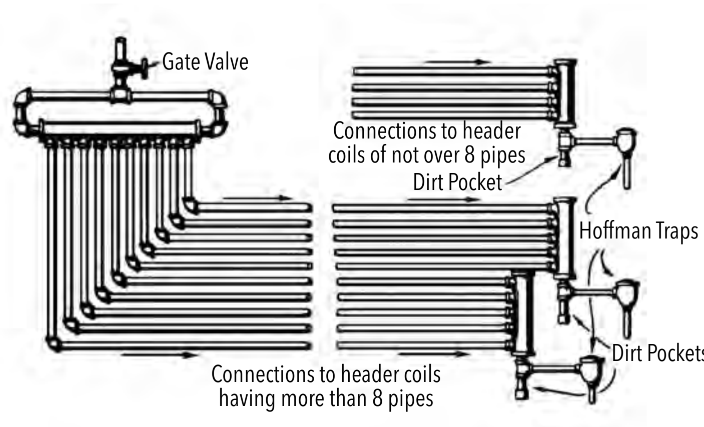


Figure 90
EXPOSED PIPE COILS - TWO PIPE STEAM

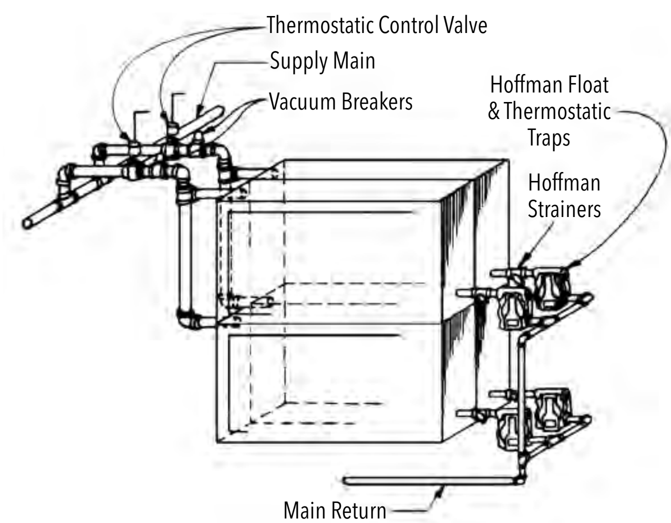


Figure 91
BLAST COILS CONTROLLED BY THERMOSTATIC SUPPLY VALVES

TYPICAL PIPING CONNECTIONS FOR ONE-PIPE AND TWO-PIPE STEAM HEATING SYSTEMS

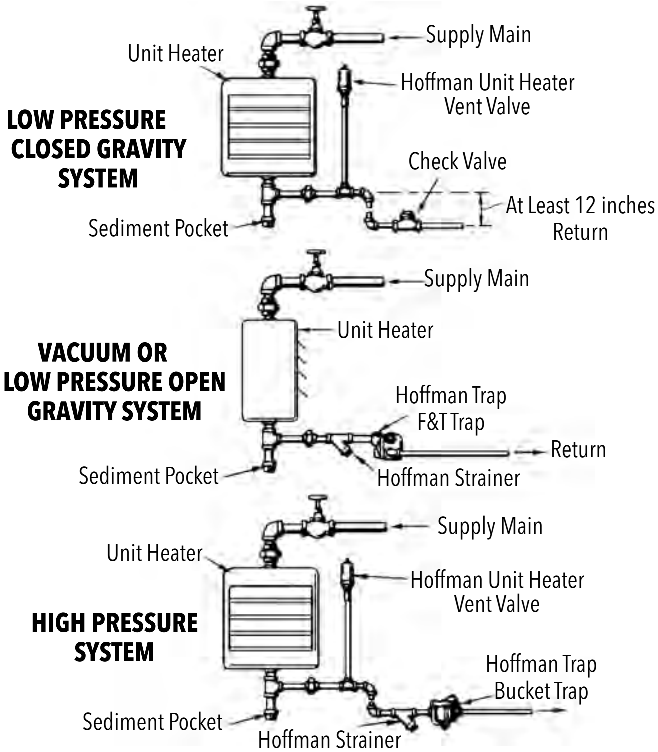
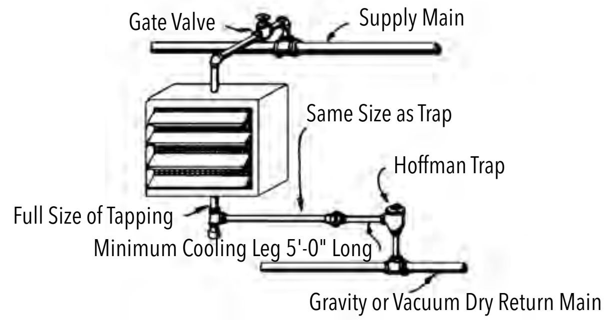
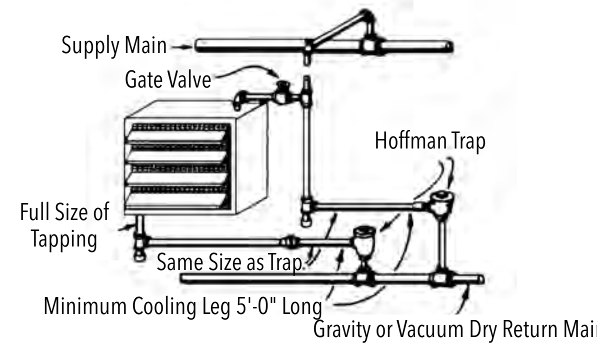


Figure 92
PIPING CONNECTIONS FOR UNIT HEATERS (STEAM)

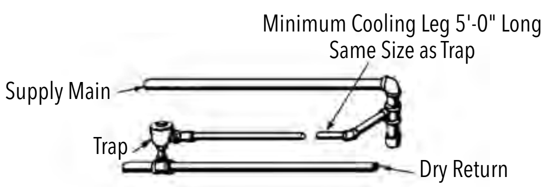


UNIT HEATER CONNECTIONS FOR TWO PIPE GRAVITY OR VACUUM SYSTEM

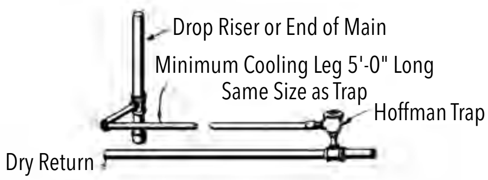


UNIT HEATER CONNECTIONS FOR TWO PIPE GRAVITY OR VACUUM SYSTEM WITH SUPPLY BRANCH DRIPPED THRU TRAP

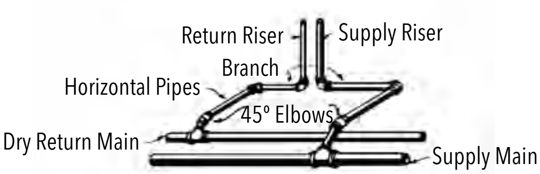
Figure 93



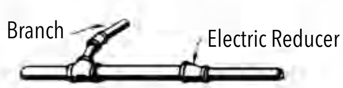
DRIPPING END OF SUPPLY MAIN INTO DRY RETURN



DRIPPING DROP RISER OR END OF MAIN INTO DRY RETURN

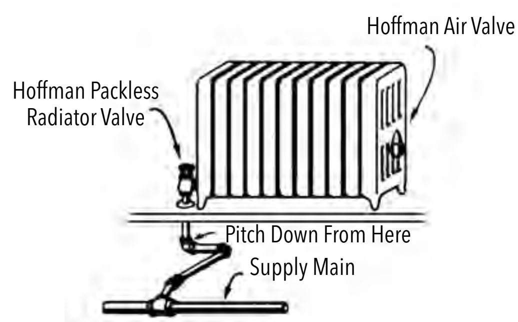


CONNECTIONS FOR UPFEED RISERS

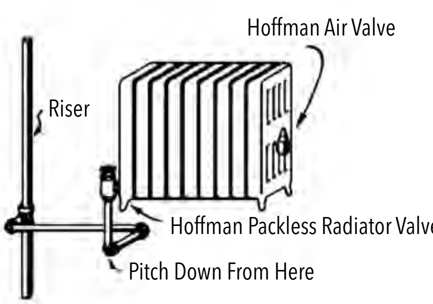


METHOD OF REDUCING SIZE OF MAIN

Figure 94
TWO PIPE – STEAM - TRAP INSTALLATIONS



UPFEED CONNECTION TO RADIATOR



RADIATOR CONNECTION TAKEN FROM UP OR DOWN FEED RISER

Figure 95
ONE PIPE STEAM SYSTEM - RADIATOR CONNECTIONS

TYPICAL PIPING CONNECTIONS FOR ONE-PIPE AND TWO-PIPE STEAM HEATING SYSTEMS

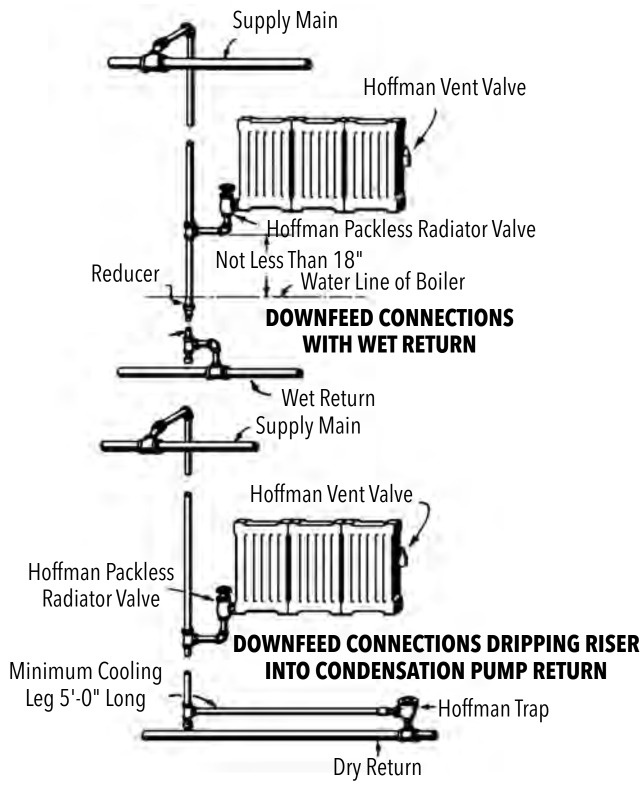
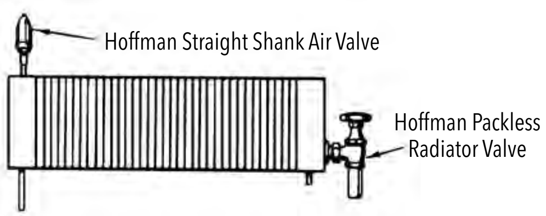


Figure 96
ONE PIPE STEAM SYSTEMS - RADIATOR CONNECTIONS



VENTING CONCEALED RADIATOR OF HEADER TYPE – AIR VENT TAPPING IN TOP

ONE PIPE STEAM SYSTEMS - CONVECTOR CONNECTIONS

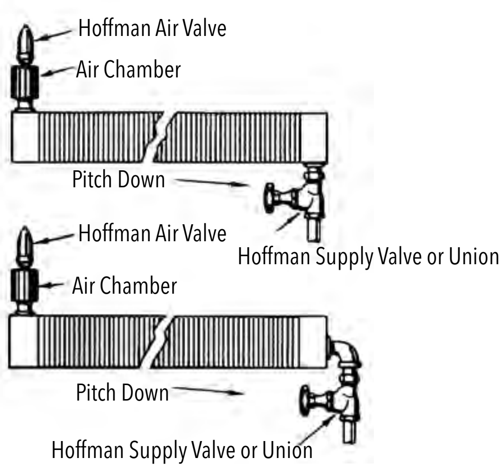


Figure 97
ONE PIPE STEAM SYSTEMS - RADIATOR CONNECTIONS

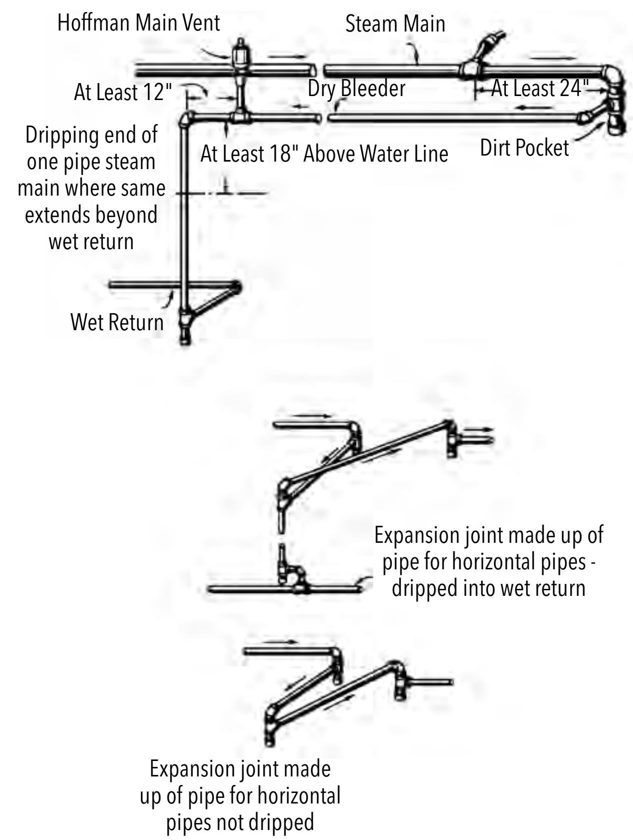


Figure 98
ONE PIPE STEAM SYSTEMS - RADIATOR CONNECTIONS

TYPICAL PIPING CONNECTIONS FOR ONE-PIPE AND TWO-PIPE STEAM HEATING SYSTEMS

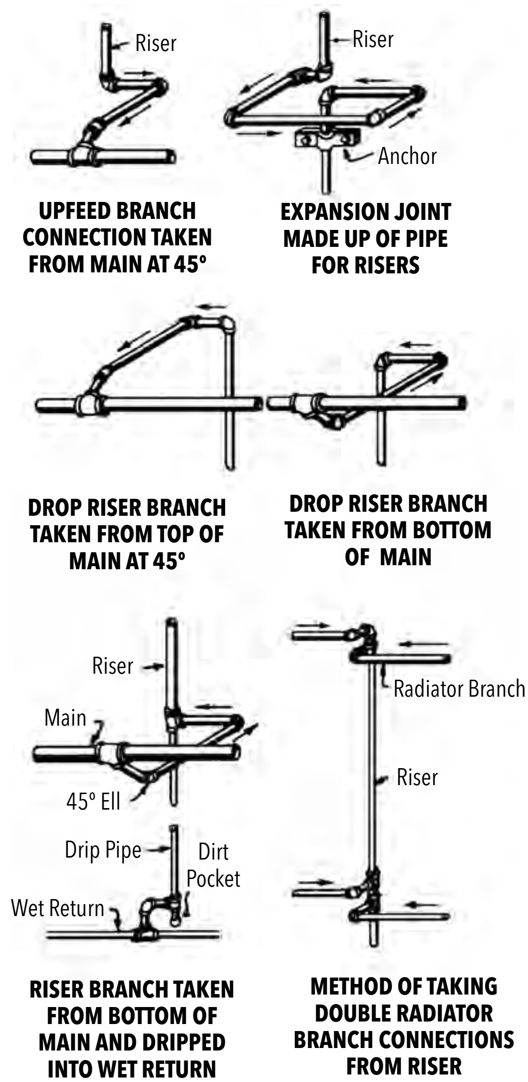
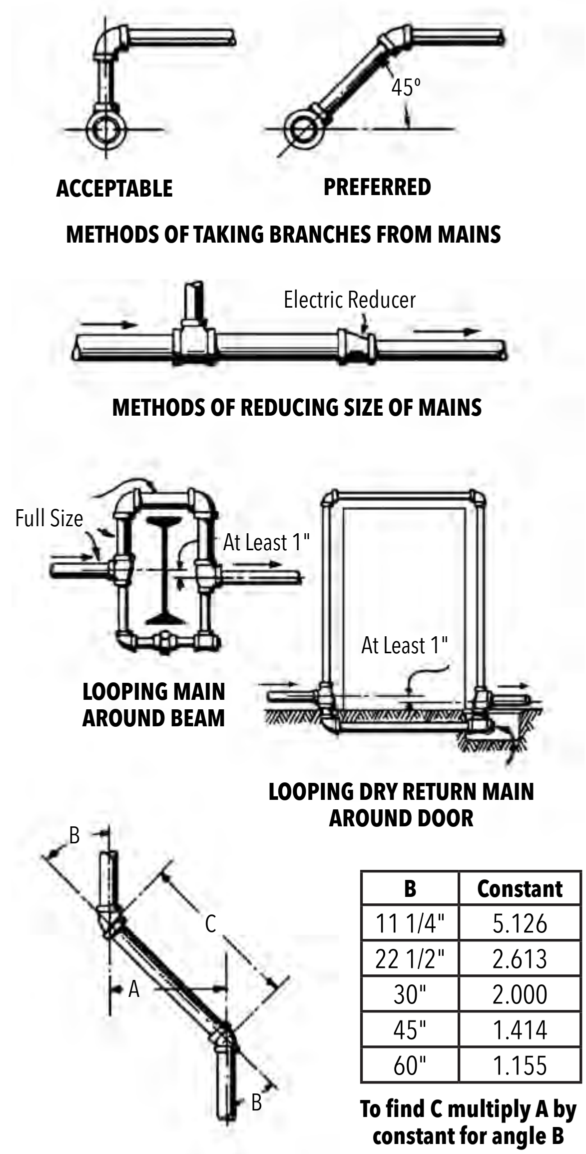


Figure 99



B	Constant
11 1/4"	5.126
22 1/2"	2.613
30"	2.000
45"	1.414
60"	1.155

To find C multiply A by constant for angle B

Figure 100

STEAM HEATING SYSTEMS DESIGNS

Steam heating systems are identified by one or more of the following combinations which relate to the features of the system: (1) Piping arrangement, (2) Pressure range and (3) Methods employed for the return of condensate.

(1) When identified by piping arrangement, they can be either one-pipe systems or two-pipe systems. They can be either up-feed or downfeed depending on the direction of steam flow in the risers. They can be described as a wet return system or a dry return system depending on the location of the condensate return main above or below the water line of the boiler or of the condensate receiver.

(2) Systems are called vapor systems when they operate at pressures ranging from low pressure to vacuum without the use of a vacuum pump. It is a vacuum system when it operates over this same pressure range using a vacuum pump.

A system is a low pressure system when it employs pressure ranging from 0 to 15 PSIG. It is a high pressure system when it operates at pressure above 15 PSIG.

(3) Systems are described as gravity systems when condensate returns directly to the boiler by gravity against the pressure causing the steam flow. When the condensate cannot be returned by gravity, mechanical means must be used to return the condensate accumulation to the boiler. The return mains can be at atmospheric pressure for non-vacuum systems or under vacuum when a vacuum pump is used. In either system, using mechanical means for condensate return, the condensate must flow by gravity to the receiver.

DESIGN PRESSURE DROPS

Steam distribution in a heating system depends on the pressure drop selected for the system. The steam supply piping should be sized so that the pressure drop for the developed length of branches from the same supply are as nearly uniform as possible for the same supply pressure. The pressure drop is expressed in a unit length of 100 feet of equivalent length of run.

The pressure drop in common use for steam piping is as follows:

Return piping is always sized for the same pressure drop as selected for the supply piping.

System Pressure PSIG	Pressure Drop per 100 ft.	Total System Pressure Drop in Steam Supply Piping
Vacuum Systems	2 - 4 oz.	1 - 2 PSI
1	2 oz.	1 - 4 oz.
2	2 oz.	8 oz.
5	4 oz.	1 1/2 PSI
10	8 oz.	3 PSI
15	1 PSI	4 PSI

Arranging the system piping so that the total distance of the supply from the boiler to the radiator is the same as the distance of the return from the radiator back to the boiler, gives the desirable resistance to and from the radiator. It is important that the piping be sized to handle the full design load conditions.

A heating system operates at less than half of its design load conditions during an average winter. However, the pick-up load required to raise the temperature of the metal in the piping up to the steam temperature and to raise the indoor temperature up to design conditions requires a large amount of heat even during moderate winter outdoor temperature. This increased pick-up value is usually considered to be 143 percent of the design load.

SIZING BOILER CONNECTIONS AND HEADER PIPING

Steam boilers for heating systems have one or more outlets. When there is more than one, all the outlets should be used. The vertical pipe to the supply main or the header should be the same size as the boiler outlet tapping and never reduced except at the supply main or header. The header should be sized on the basis of the maximum load that must be carried by any part of it. The sizing can be done from the same capacity table as used for sizing the steam mains. Increasing the size of the header to a larger pipe size than required can be undesirable when the pick-up load during the heating-up period is considered.

The return header is the same size as the required return main at the boiler. If the boiler has more than one return tapping, both should be used and sized the same as the return main.

DESIGNING A ONE-PIPE STEAM SYSTEM

The piping for a one-pipe gravity air vent system in which the total equivalent length of run does not exceed 200 ft. can be sized from **Tables 3, 5, and 7**. Instructions for their use is as follows:

1. Where steam and condensate flow in the same direction in the steam main, dripped run-outs to up-feed risers, and down-feed risers, use **Table 3,** Col. 2 (1 oz. per 100 ft. pressure drop).
2. Where run-outs to risers are not dripped and steam and condensate flow in the opposite directions, use **Table 5**, Col. F. Use the same for run-outs to radiators.
3. For up-feed risers carrying condensate back from the radiators, use **Table 5**, Col. D.
4. For down-feed systems, the main riser between the boiler and the overhead attic main, which does not carry condensate from the radiators, use **Table 5**, Col. B.
5. Size radiator supply valves and vertical connections from **Table 5**, Col. E.
6. Use **Table 7**, Col. 0 for sizing dry return mains.
7. Use **Table 7**, Col. N for sizing wet return mains.

STEAM HEATING SYSTEMS DESIGNS

Additional notes on the installation of the piping for a one-pipe system is as follows:

1. Supply and dry-return mains should not be pitched less than 1/2" in 10 ft.
2. Horizontal run-outs to risers and radiators should not be less than 1/2" per foot. Where this pitch cannot be obtained for run-outs over 8 ft. in length, increase the pipe one size larger.
3. It is not desirable to have a supply main smaller than 2".
4. All supply mains, run-outs to risers, or risers should be dripped where necessary.
5. Where supply mains are decreased in size they should be dripped or eccentric reducer coupling should be used as shown by **Figure 33** or **Figure 100**.

The following six (6) steps are required for a complete design of a one-pipe steam system.

STEP 1 – Calculate the heat loss for each room or space. Use ASHRAE GUIDE,1=B,=R or other available publications. Heat loss calculations are not performed for this example.

STEP 2 – Decide on type and size of heating units for each room. Consult ASHRAE GUIDE and manufacturers' catalogs for specific information.

A. Convert the BTU per hour heat loss for each room to Equivalent Direct Radiation n-1 sq. ft. EDR = 240 BTU per hour.

B. Locate heating unit to suit space and wall exposure. This location is usually on exposed outside walls and under or near windows so as to replace heat losses where they occur.

NOTE: Example shows size only of required heat output for a unit and not a specific make of unit.

STEP 3 – Make a piping layout to obtain the most direct piping arrangement for supplying steam from the boiler to the most remote radiator, and for the return of condensate.

Usually a double circuit main has the advantage of quick and uniform delivery of steam to the heating units. **Figure 101** shows the piping layout for the two circuit system.

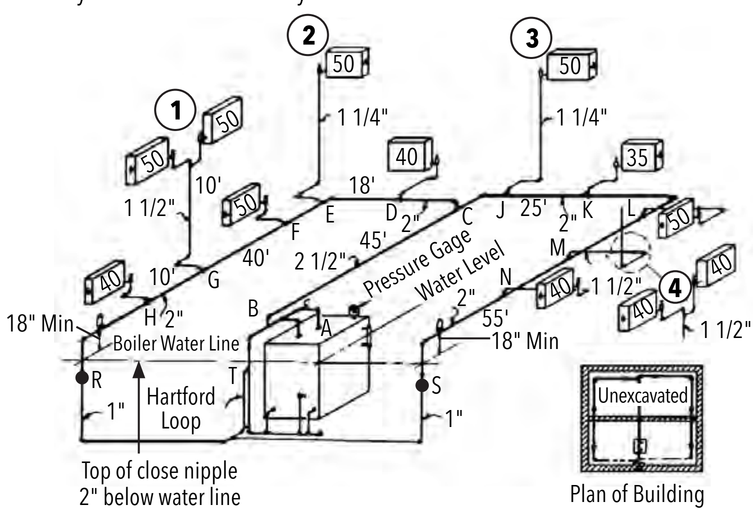


Figure 101

A. Measure actual length of main and risers from the boiler to most remote radiator to establish maximum pipe length.

Section ABCG	= 103 ft.
Riser (1)	= 20 ft.
Total	123 ft.

B. Determine Total Equivalent Length of piping by making proper allowance for fitting. See T**able 6.**

Main	
2-2Y2" elbows	= 10 ft.
3-2Y2" side outlet tees	= 33 ft.
2-2" elbows	= 8.6 ft.
Riser	
1-2" side outlet tee	= 8 ft.
1-1/2" side outlet tee	= 7 ft.
3-1/2" elbows	= 10.5 ft.
	Total 77.1 ft.
Total Equivalent Length	
	= 123 ft. + 77 ft.
	= 200 ft.

STEP 4 – Determine System Pressure Drop and Boiler Pressure:

A. For Gravity system where equivalent length is less than 200 ft. the total pressure drop should not exceed 2 oz. For equivalent length greater than 200 ft. the total pressure drop should not exceed 4 oz.

Because the Total Equivalent Length of this example is not more than 200 ft. the piping will be sized on the basis of a total pressure drop of 2 oz.

B. Boiler Pressure should not be less than twice the total pressure drop. The Boiler Pressure will be maintained at a minimum practical control limit of 1/4 PSI.

STEP 5 –Size supply mains, risers, return piping.

A. Divide total system pressure drop by equivalent length of run to obtain pressure drop for 100 ft. length.

B. Consult **Table 3**, Col. 2 for supply main sizes. The minimum horizontal supply main should not be less than 2".

C. Consult **Table 7**, Col. N for wet return main sizes. Size returns on basis of same pressure drop for 100 ft. length as that selected for supply main.

D. Consult **Table 5** for sizing the following pipe connections:

(1) Use Col. F for horizontal run-out to up-feed riser or to radiator.

(2) Use Col. E for vertical connection to radiator and radiator supply valve size.

(3) Use Col. D for up-feed supply riser.

STEP 6 – Select a boiler, from manufacturers' catalog, having a net I= B = R rating equal to or greater than the total connected load; in this example 535 sq. ft. EDR. If domestic hot water is to be heated from this steam boiler no allowance need be made unless there are more than two bathrooms. Consult I = B = R recommendation for additional allowance when required.

STEAM HEATING SYSTEMS DESIGNS

PIPE SIZES FOR ONE-PIPE SYSTEM

Description	Selection	Load Sq. Ft. EDR	Pipe Size
Horizontal Run-out to Radiator		35 40 50	1 1/4" 1 1/4" 1 1/4"
Vertical Connection Radiator Supply Valve		35 40 50	1 1/4" 1 1/4" 1 1/4"
Up-feed Riser	(1) (4) (2) & (3)	100 80 50	1 1/2" 1 1/4" 1 1/4"
Supply Main	C to R C to S B to C	280 255 535	2" 2" 2 1/2"
Boiler Header	A to B	535	2 1/2"
Wet Return Main	R to T S to T	280 235	1" 1"

DESIGNING A TWO-PIPE SYSTEM

Two-pipe steam heating systems can be designed with piping arranged for an up-feed system or for a downfeed system from a steam supply main located in an attic or pipe space. All two-pipe systems are similar in all basic respects. Attention must be given to the details of dripping the ends of the steam main, the bottom or heel of up-feed risers through steam drip traps.

The use of a condensate pump for gravity open vented return system or the use of vacuum pumps for a vacuum system is the preferred method of returning condensate to the boiler. Using these mechanical means eliminates the problem of establishing the proper operating pressure for the system.

Capacity tables for sizing steam and return piping are contained in the section of Engineering Data and Technical Information.

Additional information for the piping arrangement for two-pipe systems is as follows:

1. Steam and return mains should be pitched not less than 1/4" in 10 ft.
2. Horizontal run-outs to riser and radiators should be pitched not less than 1/2" per foot. Where this pitch cannot be obtained for runouts over 8 feet in length increase the pipe one size larger than called for in the capacity table.
3. It's not desirable to have a steam supply main smaller than 2".
4. When required, the ends of steam mains, bottom of up-feed riser or run-outs to up-feed risers should be dripped through steam traps into the dry return main. The ends of all downfeed steam risers should also be dripped through steam traps to the dry return.
5. Return mains should always be pitched to provide gravity flow of condensate to the pump receiver inlet. Vacuum returns should also be installed without the use of lift fitting. Refer to **Figure 68** or **Figure 69** for proper method of using a mechanically pumped accumulator instead of lift fitting.

The following six (6) steps are required for a complete design of a two-pipe steam heating system. The example shown in **Figure 102** will be used to illustrate the step by step procedure.

STEP 1 – Calculate the heat loss for each room or space. Many publications are available which can be used to determine the heat loss, such as ASHRAE GUIDE & DATA BOOK, and I= B = R Heat Loss Calculation Guide. Heat loss calculations are not performed for this example.

STEP 2 – Decide on the type of heating unit to be used and the size required for each room. Specific information must be obtained from manufacturers' literature and their catalog ratings. With this information available it can be used as follows.

A. Convert the BTU per hour heat loss from each room to square feet Equivalent Direct Radiation. 1 sq. ft. EDR = 240 BTU per hr.

B. Locate the heating units on the floor plan to suit available space and wall exposure. The preferable location is on an outside wall and under or near windows. This is done so as to replace the heat losses where they occur and to offset them where they have the highest rate of loss. The example shows the size of the heating units in sq. ft. EDR but does not specify the type of unit.

STEP 3 – Determine Equivalent Length of Piping. A. Make a piping layout to obtain the most direct piping arrangement for supplying steam from the boiler to the most remote radiator and for the return of condensate to the boiler. For this example, the application of a heating system to a long, narrow, two-story building was chosen, as shown by **Figure 102**, with the boiler located in one end of the building. The supply and return piping for the radiators are connected to obtain the shortest length of pipe for steam flow to the radiator and condensate flow from the radiators. The steam supply piping begins at the boiler and ends at the most remote radiator, while the return piping begins at the most remote radiator and ends at the condensate pump adjacent to the boiler. When the piping arrangement has been determined, the procedure is as follows:

(1) Measure the actual length of steam supply main, run-outs and risers from the boiler to the most remote radiator to establish the maximum pipe length, as shown by **Figure 102.**

Section	ABCDE	= 235 ft.
Riser	(6)	= 15 ft.
Total		250 ft.

(2) Determine the Total Equivalent length of piping by making proper allowance for fittings. Table 6 shows the Friction Allowance for fittings which must be added to the measured length to determine the equivalent length. Because the selection of pipe sizes for the system is based on the Total Equivalent Length of Piping, it is necessary to begin the design process based on an assumed equivalent length. This assumed value is usually considered to be double the actual measured length. For this example the assumed equivalent length will be 250 ft. x 2 = 500 ft. The actual equivalent length of pipe can be checked after the piping has been sized, based on the assumed length.

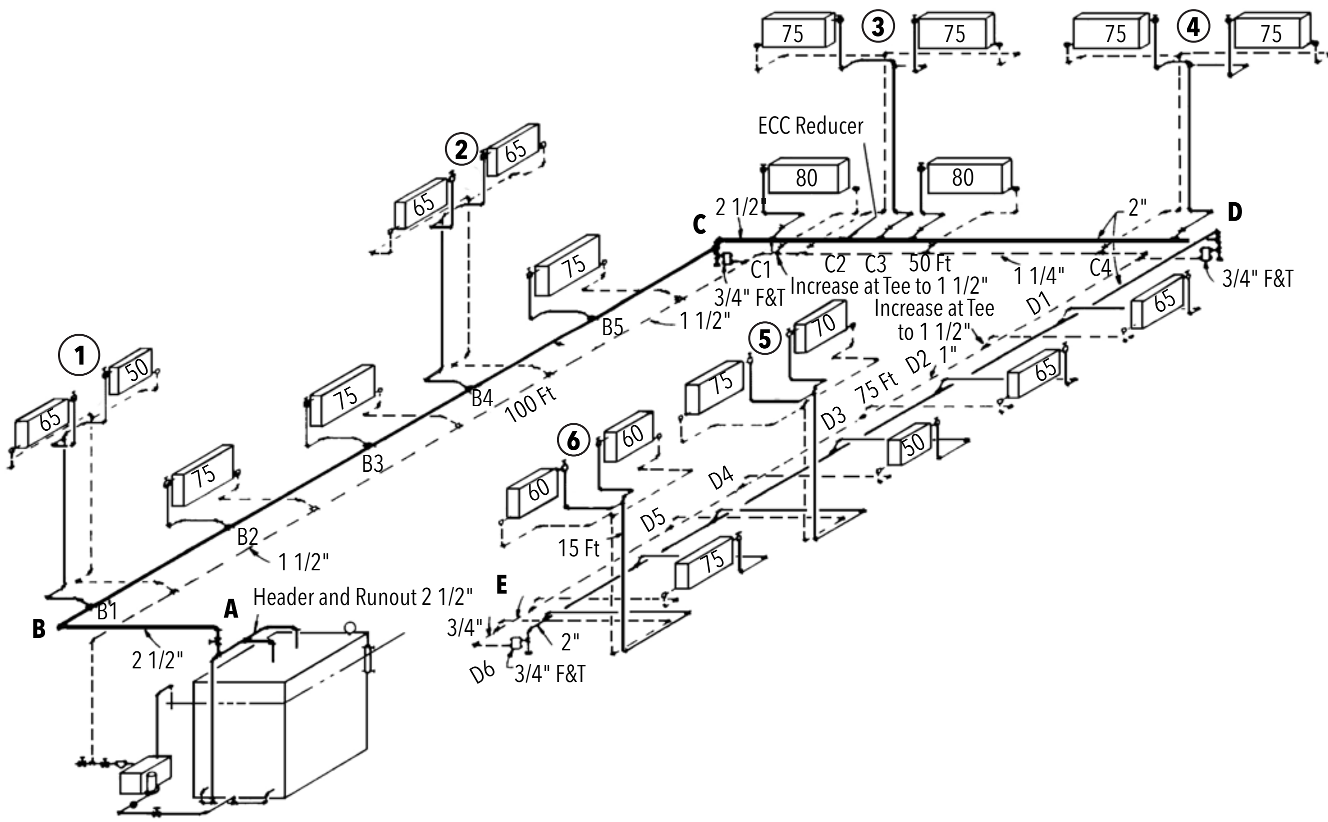


Figure 102
(See detailed drawings – Figures 102A, 102B and 102C)

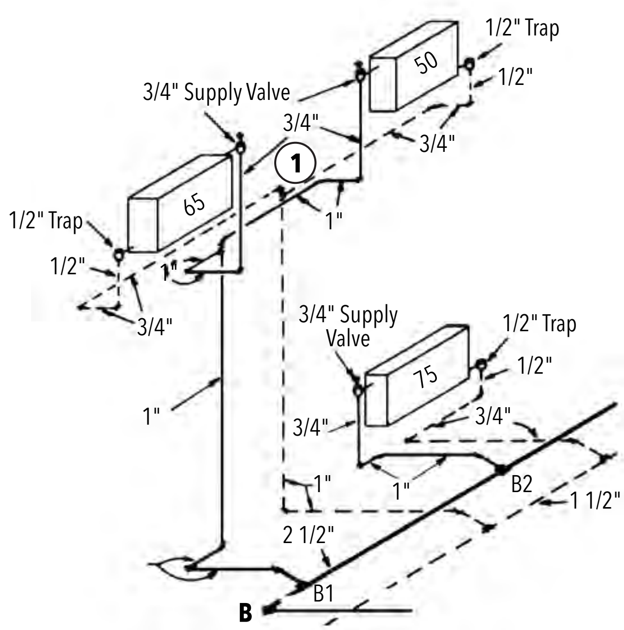


Figure 102A

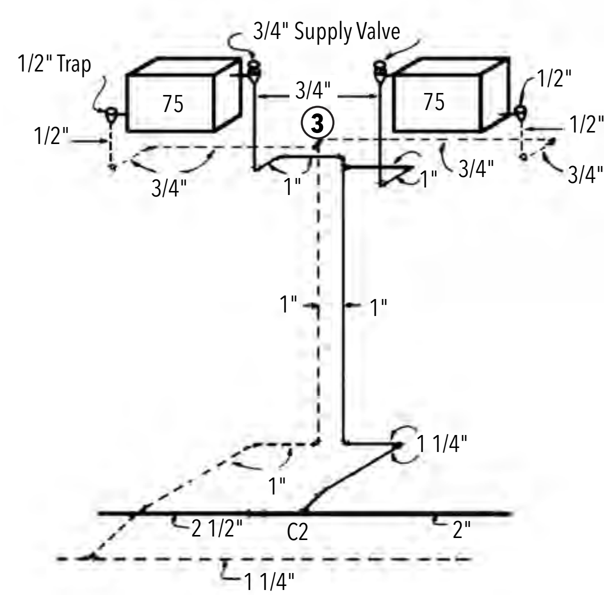


Figure 102B

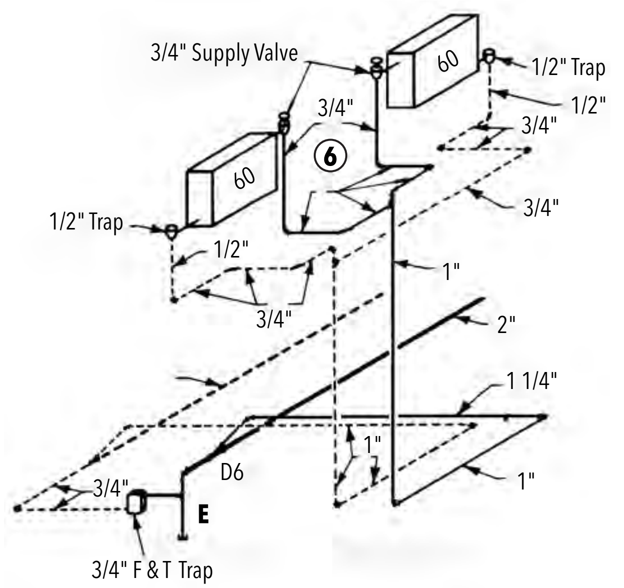


Figure 102C

STEP 4 – Determine System Pressure. Before pipe sizes can be selected for the system, several factors must be considered which pertain to the steam pressure at the boiler and system pressure drop under full load conditions. These factors include the following:

A. The total system pressure drop must not exceed one-half the boiler pressure when steam and condensate flow in the same direction.

B. The pressure drop must be kept at a value which will not create excessive velocities. For systems operating below 15 PSI the velocities should not be more than 5000 ft. per minute for quiet operation for concurrent flow. For the example shown in **Figure 102** the boiler pressure of 3 PSI was selected for full load conditions. The system pressure drop, therefore, must not exceed 1.5 PSI (24 oz.)

STEP 5 – Size the Supply and Return Piping.

A. From **Step 3** the Equivalent Length of Piping was assumed to be 500 ft. From Step 4 the maximum system pressure drop has been established as 1.5 PSI (24 oz.) Therefore, for 500 ft. Equivalent Length, the Pressure Drop per 100 ft. of length will be approximately 5 oz. However, **Table 3**, Col. 6 shows the nearest pressure to be 4 oz. per 100 ft. and the capacities shown in this column will be used to size the steam main. When sizing supply mains for low pressure steam heating systems it not desirable to have a horizontal supply main smaller than 2".

B. The horizontal steam run-out to risers and risers will be sized from **Table 3** Col. 6, and **Table 4**, 5" pitch column.

C. The return mains, horizontal run-outs to risers, and the return risers will be sized from **Table 7**, Col. U for a pressure drop of 1/4 PSI (4 oz.) which is the same as that used for sizing the steam piping.

D. The horizontal steam run-out to radiators from steam main or riser and the supply valve vertical connection and valve size are selected from **Table 8**. The horizontal return run-outs to radiators from return mains or risers and radiator trap sizes are selected from **Table 9**.

E. Referring to **Figure 102** and using the pipe capacity tables, the following tabulation shows the pipe sizes selected for the design example. The sizes are also shown in the piping layout of **Figure 102**.

TABULATION OF STEAM PIPE SIZES FOR FIG. 102

PIPE SECTION LOCATION	SQUARE FEET EDR		PIPE SIZE (Inches)	REMARKS
	ACTUAL LOAD	PERMITTED LOAD		
D6 to D1	520	936	2	Table 3—Col. 6
C4 to C2	900	936	2	Table 3—Col. 6
C1 to B1	1450	1512	2½	Table 3—Col. 6
Boiler Header	1450	1512	2½	Table 3—Col. 6
Runout to Riser[1]	115	152.4	1¼	Table 4—5" Pitch
Riser[1]	115	148	1	Table 3—Col. 6
Runout to Riser[3]	150	152.4	1¼	Table 4—5" Pitch
Riser[3]	150	312	1¼	Table 3—Col. 6

TABULATION OF RETURN PIPE SIZES FOR FIG. 102

PIPE SECTION LOCATION	SQUARE FEET EDR		PIPE SIZE (Inches)	REMARKS
	ACTUAL LOAD	PERMITTED LOAD		
D6 to D2	455	460	1	Table 7—Col. U
D1 to C2	900	964	1¼	
C1 to B1	1450	1512	1½	
B1 to Pump Receiver	1450	1512	1½	
Runout to Riser[1]	115	460	1 (Min.)	
Riser[1]	115	460	1	
Runout to Riser[3]	150	460	1	
Riser[3]	150	460	1	

TABULATION OF RADIATOR CONNECTION FOR FIG. 102

SQUARE FEET EDR LOAD	STEAM SUPPLY*			RETURN PIPE**		
	RUNOUT FROM RISER OR MAIN (Inches)	VERTICAL CONNECTION (Inches)	VALVE SIZE (Inches)	RUNOUT FROM RISER OR MAIN (Inches)	VERTICAL CONNECTION (Inches)	TRAP SIZE (Inches)
50	1	¾	¾	¾	½	½
65	1	¾	¾	¾	½	½
80	1	¾	¾	¾	½	½

*From Table 8. **From Table 9.
[1] See Fig. 102A. [3] See Fig. 102B. Note: Fig. 102C shows detailed pipe sizing at End of Main and Riser

F. Pipe sizing up to this point has been made on the basis of an assumed equivalent length. The actual equivalent length can now be determined and the pressure drop per 100 ft. of pipe length can also be determined. The following tabulation shows the result of this check for system pressure drop.

Steam main measured length	250 ft.
Allowance for fittings	
Steam Main	
1 – 2 1/2" Ell	5.0
2 – 2 1/2" S.O. Tee	22.0
1 – 2 1/2" G.V.	1.1
3 – 2 1/2" Ell	15.0
1– 2 1/2" S.O. Tee	11.0
1– 2" Ell	4.3
2-2" S.O. Tee	16.0
Runout from Main	
2 – 1" Ell	4.4
Runout from Riser	
1 – 1" S.O. Tee	5.0
2 – 1" Ell	4.4
1 – 3/4" Angle Valve	10.0
	348.2 ft.

As can be seen, with an actual Total Equivalent length of 348.2 Ft. and a total system pressure drop of 24 oz. (1.5 PSI), the pressure drop per 100 feet is 6.8 oz. Therefore, a study of Table 3 shows that there will be no changes in pipe size if the sizing is made on the basis of an 8 oz. drop (Col. 8) instead of a 4 oz. drop (Col. 6).

G. Determine steam velocity in the system with 3 PSI boiler pressure, 2 1/2" main pipe size, 362 Lb./Hr. (Lb./Hr. = $\frac{EDR}{4}$) load in system. 5000 ft. per minute maximum. By calculation using steam velocity formula in Engineering Data Section we get a velocity of 3996 ft. per minute.

STEP 6 – Select a boiler from manufacturer's catalog having a net rating (I=B=R or SBI) equal to or greater than total connected loa.d to 1450 sq. ft. EDR.

Often, domestic water is heated by the same boiler used to supply steam for heating. The size of the heater and storage tank depends on the water requirements. Proper sizing information is available from many sources such as ASHRAE GUIDE, I=B= R, SBI, and several manufacturers of heaters and coils.

STEP 7 – Select Float and Thermostatic Traps to drip the main.
MAIN A-C = 470 sq. ft.
MAIN C-D = 460 sq. ft.
MAIN D-E = 520 sq. ft.

A Hoffman 8C Trap will handle 560 sq. ft. EDR (140 lb. of condensate per hour) with a pressure differential of 1/2 pound per square inch.

STEP 8 – Select a condensate pump from Hoffman Condensate catalog to meet system requrements.

Heating System Load	1450 sq. ft.
Boiler pressure	3 PSIG

GENERAL AND USEFUL INFORMATION

ABBREVIATIONS USED IN HEATING

Term	Abbreviation
Absolute	abs
Alternating-Current	a-c
Ampere	amp
Atmosphere	atm
Average	avg
Avoirdupois	avdp
Barometer	bar.
Boiling Point	bp
Brake Horsepower	bhp
Brake Horsepower-Hour	bhp-hr
British Thermal Unit	Btu
British Thermal Units per Hour	Btuh
Calorie	cal
Centigram	cg
Centimeter	cm
Cubic	cu
Cubic Centimeter	cc
Cubic Foot	cu ft
Cubic Feet per Minute	cfm
Cubic Feet per Second	cfs
Degree	deg or °
Degree, Centigrade	C
Degree, Fahrenheit	F
Diameter	diam
Direct-Current	d-c
Feet per Minute	fpm
Feet per Second	fps
Foot	ft
Foot-Pound	ft-lb
Freezing Point	fp
Gallon	gal
Gallons per Minute	gpm
Gallons per Second	gps
Gram	g
Horsepower	hp
Horsepower-Hour	hp-hr
Hour	hr
Inch	in.
Inch-Pound	in.-lb
Kilogram	kg
Kilowatt	kw
Melting Point	mp
Meter	m
Miles per Hour	mph
Millimeter	mm
Minute	min
Ounce	oz
Pound	lb
Pounds per Square Inch	psi
Pounds per Square Inch, Gage	psig
Pounds per Square Inch, Absolute	psia
Revolutions per Minute	rpm
Revolutions per Second	rps
Second	sec
Specific Gravity	sp gr
Specific Heat	sp ht
Square Foot	sq ft
Square Inch	sq in.
Volt	v
Watt	w
Watt Hour	whr

Figure 12

INSIDE TEMPERATURES USUALLY SPECIFIED

BUILDING	ROOM	DEG. F.
SCHOOLS	Class Rooms	70-72
	Assembly Rooms	61-72
	Gymnasiums	55-65
	Toilets and Baths	70
	Wardrobe and Locker Rooms	65-68
	Kitchens	66
	Dining and Lunch Rooms	65-70
	Play Rooms	60-65
	Natatoriums	75
THEATRES	Seating Space	68-72
	Lounge Rooms	68-70
	Toilets	68
HOSPITALS	Private Rooms	70-72
	Private Rooms (Surgical)	70-80
	Operating Rooms	70-95
	Wards	68
	Kitchens and Laundries	66
	Toilets	68
	Bath Rooms	70-80
HOTELS	Bed Rooms and Baths	70
	Dining Rooms	70
	Kitchens and Laundries	66
	Ball Rooms	65-68
	Toilets and Service Rooms	68
HOMES		70-72
Stores		65-68
Public Buildings		68-72
Warm Air Baths		120
Steam Baths		110
Factories and Machine Shops		60-65
Foundries and Boiler Shops		50-60
Paint Shops		80

Figure 13

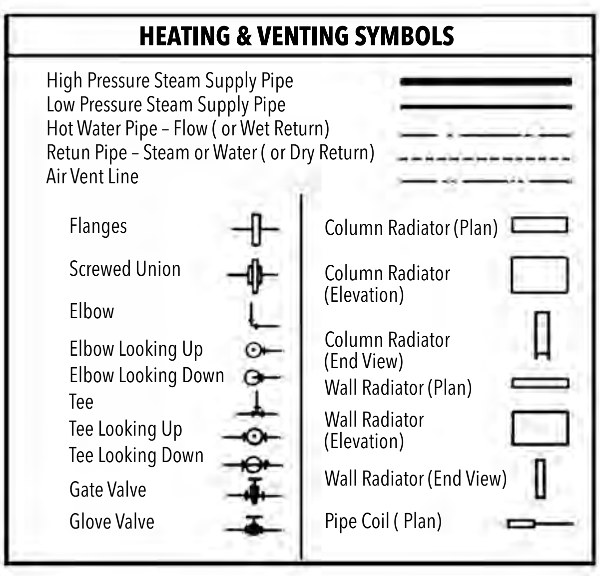
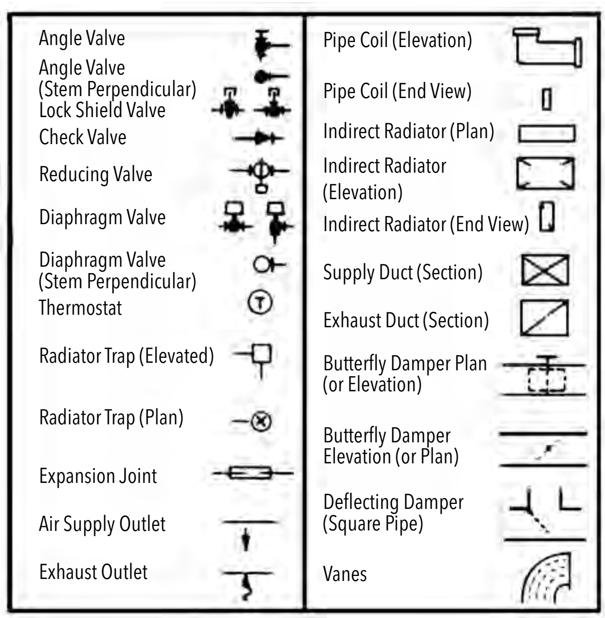


Figure 14



GENERAL AND USEFUL INFORMATION

AIR CHANGES TO BE PROVIDED FOR UNDER AVERAGE CONDITIONS EXCLUSIVE OF AIR REQUIRED FOR VENTILATION	
KIND OF ROOM OR BUILDING	**Number of Changes Taking Place Per Hour**
Rooms, 1—Side Exposed	1
Rooms, 2—Sides Exposed	1½
Rooms, 3—Sides Exposed	2
Rooms, 4—Sides Exposed	2
Rooms with No Windows or Outside Doors	½ to ¾
Entrance Halls	2 to 3
Reception Halls	2
Living Rooms	1 to 2
Dining Rooms	1 to 2
Bath Rooms	2
Drug Stores	2 to 3
Clothing Stores	1
Churches, Factories, Lofts, Etc.	½ to 3

Table 15

Number of Sections	RATING, IN SQUARE FEET							
	3 TUBE	4 TUBE		5 TUBE		6 TUBE		
	25 in.	25 in.	22 in.	25 in.	22 in.	32 in.	25 in.	19 in.
6	9.6	12.0	10.8	14.4	12.6	22.2	18.0	—
10	16.0	20.0	18.0	24.0	21.0	37.0	30.0	23.0
14	22.4	28.0	25.2	33.6	29.4	51.8	42.0	32.2
18	28.8	36.0	32.4	43.2	37.8	66.6	54.0	41.4
22	35.2	44.0	39.6	52.8	46.2	81.4	66.0	50.6
26	41.6	52.0	46.8	62.4	54.6	96.2	78.0	59.8
30	48.0	60.0	54.0	72.0	63.0	—	90.0	69.0
38	60.8	76.0	68.4	91.2	79.8	—	—	87.4

NIPPLES – Sections are assembled with cast- or malleable-iron push nipples.

TAPPINGS – Tappings are standard iron-pipe size. Flow and return tappings are located horizontally opposite the top and bottom nipple ports. Air-vent tappings for water and steam radiators are provided on the end section opposite the supply section.

PIPE FITTINGS – Iron-pipe size plugs or bushings or both, may be furnished with each radiator.

PAINTING – Each section or radiator assembled by the manufacturer is given one priming coat.

NUMBER OF SECTIONS – The stock assemblies are shown in the table. When assemblies of more sections than those listed are required, the maximum number should not exceed 56, to avoid damage in shipping and handling. Consult manufacturers' catalogs.

The square foot of equivalent direct steam radiation is defined as the ability to emit 240 BTU per hour, with steam at 215°F. in air at 70°F. These ratings apply only to radiators installed exposed in a normal manner; not to radiators installed behind enclosures, grilles, etc.

AIR CHANGES TO BE PROVIDED FOR UNDER AVERAGE CONDITIONS EXCLUSIVE OF AIR REQUIRED FOR VENTILATION

Add 1– air change to above for rooms with a fireplace. While the most accurate way of estimating air infiltration considers leakage through walls and clearance between window frames and sash, the table at left is sufficiently accurate for practical purposes. Where windows are equipped with metal weatherstripping, use 1/2 air change less per hour than shown above.

For example: A room with 2 sides exposed, windows weatherstripped, would be figured at 1- airchange instead of 1 1/2.

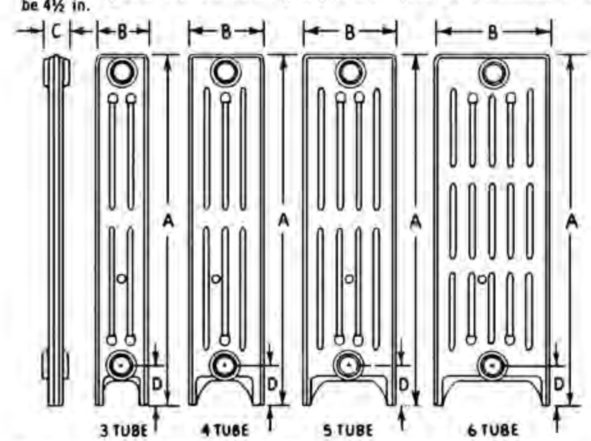
EXTRA SOURCES OF HEAT

In auditoriums, like a church, a theater or in factories where a large number of people are gathered a considerable amount of heat is liberated which should be subtracted from the BTU required to heat the room. A person at rest gives off about 400 BTU per hour, while one at work gives off somewhere between 500 and 800.

Electricity used in the room may be quite an item. The total wattage of electricity actually used multiplied by 3.4 should be subtracted from the required heating load (BTU per hour). In factories where gas or oil is used as in process work, the BTU per hour consumed should be subtracted from the required heating load. The heating values of various kinds of gas and oil may be found in the engineering data section.

Number of Tubes per Section	Catalog Rating per Section Sq. Ft.	SECTION DIMENSIONS IN INCHES				
		A HEIGHT ①	B WIDTH		C SPACING	D LEG HEIGHT ①
			Min.	Max.		
3	1.6	25	3¼	3½	1¾	2½
4	1.6	19	4 7/16	4 13/16	1¾	2½
4	1.8	22	4 7/16	4 13/16	1¾	2½
4	2.0	25	4 7/16	4 13/16	1¾	2½
5	2.1	22	5⅝	6 5/16	1¾	2½
5	2.4	25	5⅝	6 5/16	1¾	2½
6	2.3	19	6 13/16	8	1¾	2½
6	3.0	25	6 13/16	8	1¾	2½
6	3.7	32	6 13/16	8	1¾	2½

①Over-all height and leg height of radiator as made by some manufacturers is 1 inch greater than shown in Columns A and D. Radiators may be furnished without legs. Where greater than standard leg heights are required, this dimension is to be 4½ in.



*From Simplified Practice Recommendation R174-47, U.S. Dept. of Commerce.

Table 16

OUTPUT OF DIRECT CAST IRON RADIATORS†

STEAM PRESSURE (APPROXIMATE)			STEAM OR MEAN WATER TEMP. °F	BTU PER SQ. FT. E.D.R. PER HOUR ROOM TEMPERATURE °F.						
	GAGE*	ABSOLUTE** Lbs. per Sq. In.		80	75	70	65	60	55	50
VACUUM Inches of MERCURY	22.4	3.7	150	93	102	111	120	129	139	148
	20.3	4.7	160	111	120	129	139	148	158	167
	17.7	6.0	170	129	139	148	158	167	178	188
	14.6	7.5	180	148	158	167	178	188	198	209
	10.9	9.3	190	167	178	188	198	209	218	229
	6.5	11.5	200	188	198	209	218	229	240	250
	3.9	12.8	205	198	209	218	229	240	250	261
Lbs. per Sq. Inch	0.0	14.7	212	211	222	233	242	253	264	276
	1.	15.6	215	218	229	240	250	261	273	282
	2.	17	220	229	240	250	261	273	282	296
	6.	21.	230	250	261	273	282	296	308	316
	10.	25.	240	273	282	296	308	316	329	343
	15.	30.	250	296	308	316	329	343	353	364
	27.	42.	270	343	353	364	375	387	400	414
	52.	67.	300	414	421	436	453	462	471	490

*At sea level only.

**At locations other than sea level use temperature only or convert gage reading to absolute pressure. Add gage reading to atmospheric pressure in lbs. per sq. in. for given altitude. To convert vacuum (inches of mercury) to absolute, multiply inches vacuum by 0.49 and deduct from atmospheric pressure (lbs. per sq. in.) for given altitude.

†These outputs also apply quite closely to the output of the "R" type cast iron radiant baseboards. For exact outputs, the catalogs of the manufacturers of the baseboards involved should be consulted.

Table 17

STANDARD COLUMN RADIATION*

SINGLE-COLUMN RADIATORS							TWO-COLUMN RADIATORS							
		HEATING SURFACE—SQUARE FEET							HEATING SURFACE—SQUARE FEET					
No. of Sections	Length 2½"-Sec.	38" 3 Sq. Ft. Sec.	32" 2½ Sq. Ft. Sec.	26" 2 Sq. Ft. Sec.	23" 1⅔ Sq. Ft. Sec.	20" 1½ Sq. Ft. Sec.	No. of Sections	Length 2½"-Sec.	45" 5 Sq. Ft. Sec.	38" 4 Sq. Ft. Sec.	32" 3⅓ Sq. Ft. Sec.	26" 2⅔ Sq. Ft. Sec.	23" 2⅓ Sq. Ft. Sec.	20" 2 Sq. Ft. Sec.
3	7½	9	7½	6	5	4½	3	7½	15	12	10	8	7	6
4	10	12	10	8	6⅔	6	4	10	20	16	13⅓	10⅔	9⅓	8
5	12½	15	12½	10	8⅓	7½	5	12½	25	20	16⅔	13⅓	11⅔	10
6	15	18	15	12	10	9	6	15	30	24	20	16	14	12
7	17½	21	17½	14	11⅔	10½	7	17½	35	28	23⅓	18⅔	16⅓	14
8	20	24	20	16	13⅓	12	8	20	40	32	26⅔	21⅓	18⅔	16
9	22½	27	22½	18	15	13½	9	22½	45	36	30	24	21	18
10	25	30	25	20	16⅔	15	10	25	50	40	33⅓	26⅔	23⅓	20
11	27½	33	27½	22	18⅓	16½	11	27½	55	44	36⅔	29⅓	25⅔	22
12	30	36	30	24	20	18	12	30	60	48	40	33	28	24
13	32½	39	32½	26	21⅔	19½	13	32½	65	52	43⅓	34⅔	30⅓	26
14	35	42	35	28	23⅓	21	14	35	70	56	46⅔	37⅓	32⅔	28
15	37½	45	37½	30	25	22½	15	37½	75	60	50	40	35	30
16	40	48	40	32	26⅔	24	16	40	80	64	53⅓	42⅔	37⅓	32
17	42½	51	42½	34	28⅓	25½	17	42½	85	68	56⅔	45⅓	39⅔	34
18	45	54	45	36	30	27	18	45	90	72	60	48	42	36
19	47½	57	47½	38	31⅔	28½	19	47½	95	76	63⅓	50⅔	44⅓	38
20	50	60	50	40	33⅓	30	20	50	100	80	66⅔	53⅓	46⅔	40
21	52½	63	52½	42	35	31½	21	52½	105	84	70	56	49	42
22	55	66	55	44	36⅔	33	22	55	110	88	73⅓	58⅔	51⅓	44
23	57½	69	57½	46	38⅓	34½	23	57½	115	92	76⅔	61⅓	53⅔	46
24	60	72	60	48	40	36	24	60	120	96	80	64	56	48
25	62½	75	62½	50	41⅔	37½	25	62½	125	100	83⅓	66⅔	58⅓	50
26	65	78	65	52	43⅓	39								
27	67½	81	67½	54	45	40½								

*This table cover column radiation manufactured prior to 1926.

Table 18

STANDARD COLUMN RADIATION*

THREE-COLUMN RADIATORS

No. of Sections	Length 2½"-Sec.	HEATING SURFACE SQUARE FEET 45" 6 Sq. Ft. Sec.	38" 5 Sq. Ft. Sec.	32" 4½ Sq. Ft. Sec.	26" 3¾ Sq. Ft. Sec.	22" 3 Sq. Ft. Sec.	18" 2¼ Sq. Ft. Sec.
3	7½	18	15	13½	11¼	9	6¾
4	10	24	20	18	15	12	9
5	12½	30	25	22½	18¾	15	11¼
6	15	36	30	27	22½	18	13½
7	17½	42	35	31½	26¼	21	15¾
8	20	48	40	36	30	24	18
9	22½	54	45	40½	33¾	27	20¼
10	25	60	50	45	37½	30	22½
11	27½	66	55	49½	41¼	33	24¾
12	30	72	60	54	45	36	27
13	32½	78	65	58½	48¾	39	29¼
14	35	84	70	63	52½	42	31½
15	37½	90	75	67½	56¼	45	33¾
16	40	96	80	72	60	48	36
17	42½	102	85	76½	63¾	51	38¼
18	45	108	90	81	67½	54	40½
19	47½	114	95	85½	71¼	57	42¾
20	50	120	100	90	75	60	45
21	52½	126	105	94½	78¾	63	47¼
22	55	132	110	99	82½	66	49½
23	57½	138	115	103½	86¼	69	51¾
24	60	144	120	108	90	72	54
25	62½	150	125	112½	93¾	75	56¼
26	65	156	130	117	97½	78	58½
27	67½	162	135	121½	101¼	81	60¾

FOUR-COLUMN RADIATORS

No. of Sections	Length 3"-Sec.	HEATING SURFACE SQUARE FEET 45" 10 Sq. Ft. Sec.	38" 8 Sq. Ft. Sec.	32" 6½ Sq. Ft. Sec.	26" 5 Sq. Ft. Sec.	22" 4 Sq. Ft. Sec.	18" 3 Sq. Ft. Sec.
3	9	30	24	19½	15	12	9
4	12	40	32	26	20	16	12
5	15	50	40	32½	25	20	15
6	18	60	48	39	30	24	18
7	21	70	56	45½	35	28	21
8	24	80	64	52	40	32	24
9	27	90	72	58½	45	36	27
10	30	100	80	65	50	40	30
11	33	110	88	71½	55	44	33
12	36	120	96	78	60	48	36
13	39	130	104	84½	65	52	39
14	42	140	112	91	70	56	42
15	45	150	120	97½	75	60	45
16	48	160	128	104	80	64	48
17	51	170	136	110½	85	68	51
18	54	180	144	117	90	72	54
19	57	190	152	123½	95	76	57
20	60	200	160	130	100	80	60
21	63	210	168	136½	105	84	63
22	66	220	176	143	110	88	66
23	69	230	184	149½	115	92	69
24	72	240	192	156	120	96	72
25	75	250	200	162½	125	100	75

*This table cover column radiation manufactured prior to 1926

Table 19

HEATING SURFACE—THREE TUBE RADIATORS
FOR STEAM OR HOT WATER

No. of Sections	Length Inches	HEATING SURFACE—SQUARE FEET 240 B.T.U. PER SQ. FT. PER HOUR				
		36″ HEIGHT 3½ Sq. Ft. Per Sec.	30″ HEIGHT 3 Sq. Ft. Per Sec.	26″ HEIGHT 2⅓ Sq. Ft. Per Sec.	23″ HEIGHT 2 Sq. Ft. Per Sec.	20″ HEIGHT 1¾ Sq. Ft. Per Sec.
2	5	7	6	4⅔	4	3½
3	7½	10½	9	7	6	5¼
4	10	14	12	9⅓	8	7
5	12½	17½	15	11⅔	10	8¾
6	15	21	18	14	12	10½
7	17½	24½	21	16⅓	14	12¼
8	20	28	24	18⅔	16	14
9	22½	31½	27	21	18	15¾
10	25	35	30	23⅓	20	17½
11	27½	38½	33	25⅔	22	19¼
12	30	42	36	28	24	21
13	32½	45½	39	30⅓	26	22¾
14	35	49	42	32⅔	28	24½
15	37½	52½	45	35	30	26¼
16	40	56	48	37⅓	32	28
17	42½	59½	51	39⅔	34	29¾
18	45	63	54	42	36	31½
19	47½	66½	57	44⅓	38	33¼
20	50	70	60	46⅔	40	35
21	52½	73½	63	49	42	36¾
22	55	77	66	51⅓	44	38½
23	57½	80½	69	53⅔	46	40¼
24	60	84	72	56	48	42
25	62½	87½	75	58⅓	50	43¾

NOTE—These data apply to obsolete radiators manufactured since 1926 but not manufactured today.

Table 20

HEATING SURFACE FOUR TUBE RADIATORS
FOR STEAM OR HOT WATER

No. of Sections	Length Inches	HEATING SURFACE—SQUARE FEET 240 B.T.U. PER SQ. FT. PER HOUR				
		37″ HEIGHT 4¼ Sq. Ft. Per Sec.	32″ HEIGHT 3½ Sq. Ft. Per Sec.	26″ HEIGHT 2¾ Sq. Ft. Per Sec.	23″ HEIGHT 2½ Sq. Ft. Per Sec.	20″ HEIGHT 2¼ Sq. Ft. Per Sec.
2	5	8½	7	5½	5	4½
3	7½	12¾	10½	8¼	7½	6¾
4	10	17	14	11	10	9
5	12½	21¼	17½	13¾	12½	11¼
6	15	25½	21	16½	15	13½
7	17½	29¾	24½	19¼	17½	15¾
8	20	34	28	22	20	18
9	22½	38¼	31½	24¾	22½	20¼
10	25	42½	35	27½	25	22½
11	27½	46¾	38½	30¼	27½	24¾
12	30	51	42	33	30	27
13	32½	55¼	45½	35¾	32½	29¼
14	35	59½	49	38½	35	31½
15	37½	63¾	52½	41¼	37½	33¾
16	40	68	56	44	40	36
17	42½	72¼	59½	46¾	42½	38¼
18	45	76½	63	49½	45	40½
19	47½	80¾	66½	52¼	47½	42¾
20	50	85	70	55	50	45
21	52½	89¼	73½	57¾	52½	47¼
22	55	93½	77	60½	55	49½
23	57½	97¾	80½	63¼	57½	51¾
24	60	102	84	66	60	54
25	62½	106¼	87½	68¾	62½	56¼

NOTE—These data apply to obsolete radiators manufactured since 1926 but not manufactured today.

Table 21

HEATING SURFACE FIVE TUBE RADIATORS
FOR STEAM OR HOT WATER

No. of Sections	Length Inches	HEATING SURFACE—SQUARE FEET 240 B.T.U. PER SQ. FT. PER HOUR				
		37″ HEIGHT 5 Sq. Ft. Per Sec.	32″ HEIGHT 4⅓ Sq. Ft. Per Sec.	26″ HEIGHT 3½ Sq. Ft. Per Sec.	23″ HEIGHT 3 Sq. Ft. Per Sec.	20″ HEIGHT 2⅔ Sq. Ft. Per Sec.
2	5	10	8⅔	7	6	5⅓
3	7½	15	13	10½	9	8
4	10	20	17⅓	14	12	10⅔
5	12½	25	21⅔	17½	15	13⅓
6	15	30	26	21	18	16
7	17½	35	30⅓	24½	21	18⅔
8	20	40	34⅔	28	24	21⅓
9	22½	45	39	31½	27	24
10	25	50	43⅓	35	30	26⅔
11	27½	55	47⅔	38½	33	29⅓
12	30	60	52	42	36	32
13	32½	65	56⅓	45½	39	34⅔
14	35	70	60⅔	49	42	37⅓
15	37½	75	65	52½	45	40
16	40	80	69⅓	56	48	42⅔
17	42½	85	73⅔	59½	51	45⅓
18	45	90	78	63	54	48
19	47½	95	82⅓	66½	57	50⅔
20	50	100	86⅔	70	60	53⅓
21	52½	105	91	73½	63	56
22	55	110	95⅓	77	66	58⅔
23	57½	115	99⅔	80½	69	61⅓
24	60	120	104	84	72	64
25	62½	125	108⅓	87½	75	66⅔

NOTE—These data apply to obsolete radiators manufactured since 1926 but not manufactured today.

Table 22

HEATING SURFACE—SIX TUBE RADIATORS
FOR STEAM OR HOT WATER

NO. OF SEC-TIONS	LENGTH INCHES	HEATING SURFACE—SQUARE FEET 240 B.T.U. PER SQ. FT. PER HOUR				
		37″ HEIGHT 6 SQ. FT. PER SEC.	32″ HEIGHT 5 SQ. FT. PER SEC.	26″ HEIGHT 4 SQ. FT. PER SEC.	23″ HEIGHT 3½ SQ. FT. PER SEC.	20″ HEIGHT 3 SQ. FT. PER SEC.
2	5	12	10	8	7	6
3	7½	18	15	12	10½	9
4	10	24	20	16	14	12
5	12½	30	25	20	17½	15
6	15	36	30	24	21	18
7	17½	42	35	28	24½	21
8	20	48	40	32	28	24
9	22½	54	45	36	31½	27
10	25	60	50	40	35	30
11	27½	66	55	44	38½	33
12	30	72	60	48	42	36
13	32½	78	65	52	45½	39
14	35	84	70	56	49	42
15	37½	90	75	60	52½	45
16	40	96	80	64	56	48
17	42½	102	85	68	59½	51
18	45	108	90	72	63	54
19	47½	114	95	76	66½	57
20	50	120	100	80	70	60
21	52½	126	105	84	73½	63
22	55	132	110	88	77	66
23	57½	138	115	92	80½	69
24	60	144	120	96	84	72
25	62½	150	125	100	87½	75

NOTE—THESE DATA APPLY TO OBSOLETE RADIATORS MANUFACTURED SINCE 1926 BUT NOT MANUFACTURED TODAY.

Table 23

HEATING SURFACE—SEVEN TUBE RADIATORS
FOR STEAM OR HOT WATER

NO. OF SEC-TIONS	LENGTH INCHES	HEATING SURFACE—SQUARE FEET 240 B.T.U. PER SQ. FT. PER HOUR		
		20″ HEIGHT 4¼ SQ. FT. PER SEC.	16½″ HEIGHT 3½ SQ. FT. PER SEC.	13″ HEIGHT 2¾ SQ. FT. PER SEC.
2	5	8½	7	5½
3	7½	12¾	10½	8¼
4	10	17	14	11
5	12½	21¼	17½	13¾
6	15	25½	21	16½
7	17½	29¾	24½	19¼
8	20	34	28	22
9	22½	38¼	31½	24¾
10	25	42½	35	27½
11	27½	46¾	38½	30¼
12	30	51	42	33
13	32½	55¼	45½	35¾
14	35	59½	49	38½
15	37½	63¾	52½	41¼
16	40	68	56	44
17	42½	72¼	59½	46¾
18	45	76½	63	49½
19	47½	80¾	66½	52¼
20	50	85	70	55
21	52½	89¼	73½	57¾
22	55	93½	77	60½
23	57½	97¾	80½	63¼
24	60	102	84	66
25	62½	106¼	87½	68¾

NOTE—THESE DATA APPLY TO OBSOLETE RADIATORS MANUFACTURED SINCE 1926 BUT NOT MANUFACTURED TODAY.

Table 24

HEAT EMISSION ST'D. COLUMN RADIATION*

Number of B.T.U. transmitted per hour per sq. ft. of radiation with low pressure steam when heating room to given temperature.

Temperature of Room	TYPE OF RADIATION				
	3 Col. 26″	3 Col. 32″	3 Col. 38″	Wall.	Coil.
Col. A	Col. B	Col. C	Col. D	Col. E	Col. F
40°F.	309	305	293	362	381
45	301	292	281	347	365
50	290	282	271	335	354
55	279	270	261	322	338
60	269	261	250	310	326
65	258	250	240	297	313
70	247	240	231	285	300
75	236	230	220	275	288
80	226	220	211	261	277
85	216	210	200	251	265
90	206	200	190	242	253
95	196	190	180	228	239
100	186	180	170	215	226
105	176	171	162	203	214
110	167	162	155	192	202
115	158	153	147	181	191
120	149	144	139	171	180
125	140	135	130	160	169
130	130	126	121	150	158
135	121	118	113	140	147
140	113	110	106	130	137

If 500 ϕ of wall radiation is heating a room to 50°F. what will its equivalent be in 32″ column radiation setting in 70°F—Refer to Col. E and find one ϕ wall radiation setting in 50°F. = 335 B.T.U. Then 500 × 335 = 167,500 B.T.U. Refer to Col. C at 70°F. for average Col. Rad. and find 240 B.T.U. per sq. ft. Therefore 167,500 ÷ 240 = 698 ϕ direct equivalent.

*This table covers heat emission from radiation manufactured prior to 1926.

Table 25

HEAT EMISSION OF DIRECT PIPE COILS

USING STEAM AT 215° F. AND ROOM TEMP. 70°

Wall Coils—Coils Placed Vertical—Pipes Horizontal
B.T.U. per linear foot of coil per hour (not lineal feet of pipe)

Size of Pipe Coil Conductor	1″	1¼″	1½″
Single Row	132	162	185
Two Row	252	312	348
Four Row	440	545	616
Six Row	567	702	793
Eight Row	651	796	907
Ten Row	732	907	1020
Twelve Row	812	1005	1135

Ceiling Coils—Coils placed horizontally—pipes horizontal, emission is equal to that of a single row coil.

Allowance must be made, however, if the coil is at the ceiling in a higher temperature. In this case use:

126 B.T.U. per linear foot of pipe for 1″ Coils
156 B.T.U. per linear foot of pipe for 1¼″ Coils
175 B.T.U. per linear foot of pipe for 1½″ Coils

HEAT EMISSION OF HOT WATER WALL COILS

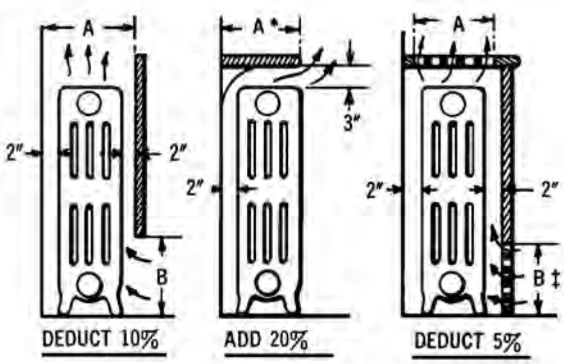
INSTALLATION LIKE STEAM COILS ABOVE

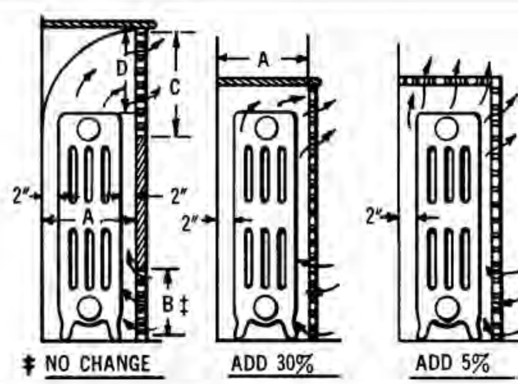
PIPE SIZE	MEAN WATER °F.	ROWS IN COIL						
		1	2	4	6	8	10	12
1″	170	80	155	270	350	400	450	500
	180	90	175	305	395	450	510	565
	190	105	195	345	445	510	570	635
	200	115	220	385	495	565	635	705
1¼″	170	100	190	335	435	490	560	620
	180	110	215	380	485	555	630	700
	190	125	245	425	550	620	710	785
	200	140	270	475	610	690	790	875
1½″	170	115	215	380	490	560	630	700
	180	130	240	430	550	630	710	790
	190	145	270	480	620	710	795	885
	200	160	305	535	690	790	885	985

Table 26

RADIATOR ENCLOSURES

To enclose or partly enclose a radiator reduces its heat output and changes the distribution of heated air in the room. The additional surface usually added to column or tube radiation for various enclosures is indicated below.





*If A is 50% of width of radiator, add 10%; if 150%, add 35%.

‡B = 80% of A C = 150% of A D = A

Example: A room requires 50 sq. ft. radiation radiator recessed flush with wall, — 50 ϕ + 20% = 60 ϕ radiator required. If radiator for same room is to have grille over entire face only, — 50 ϕ + 30% = 65 ϕ required.

Table 27

KITCHEN EQUIPMENT

POUNDS OF STEAM REQUIRED PER HOUR

	Pounds Per Hour
Bain Marie—Per Foot	12.5
Coffee Urn (3)	50
Egg Boiler—3 Compartment	10
Stock Kettle—40 Gallon	37.5
Steam Table—Each 6 Feet	25
Clam and Lobster Steamer	10
Jet for Pot Sink	10
Vegetable Boiler—Per Section	15
Silver Burnisher	50
Plate Warmer—15 feet long	30
Steamer—3 Compartment	75
Usual Pressure Carried 30-35 lbs.	

Table 28

OUTDOOR STORAGE TANKS

(TO KEEP FROM FREEZING)

T^1 = Outdoor Temperature.

T^2 = Temperature of Water.

E.S. = Sq. ft. of Exposed Surface of Tank

Coef. = 2″ Wood Tank—.5 B.T.U. per sq. ft. per hour
Steel Tank—1.5 B.T.U. per sq. ft. per hour

B.T.U. loss from Water per hour = E.S. $\times$ ($T^1 - T^2$) $\times$ Coef.

$$\text{Pounds Steam Required} = \frac{\text{Total B.T.U. Loss from Water per Hour}}{\text{Latent Heat Steam at Pressure Carried}}$$

Sq. ft. Pipe Coil Required = Lbs. of Steam required ÷ Lbs. Steam Condensed per Hour per sq. ft. of Pipe (see Table 30)

To obtain sq. ft. of exposed surface for round tanks multiply diameter by 3.1416 $\times$ Height + Area of Bottom and Top.

NOTE—The above is only to be used for water storage tanks, usually located on roof.

Table 29

HEATING POWER OF LOW PRESSURE STEAM PIPES IN WATER FOR AVERAGE WORKING CONDITIONS

BRASS PIPE					
COL. A	COL. B	COL. C	COL. A	COL. B	COL. C
Temp. Dif.	B.T.U.	Pounds	Temp. Dif.	B.T.U.	Pounds
6°F.	192	.20	70	13,000	13.55
7	240	.25	75	15,000	15.62
8	300	.31	80	17,000	17.70
9	400	.42	85	19,000	20.00
10	480	.50	90	21,000	21.77
15	800	.83	95	23,000	24.00
16	960	1.00	100	25,000	26.05
20	1,440	1.50	110	30,000	31.25
25	2,300	2.40	120	35,000	36.45
30	3,100	3.23	130	40,000	41.60
35	4,000	4.16	140	45,000	46.90
40	5,000	5.23	150	50,000	52.10
45	6,000	6.25	160	55,000	57.30
50	7,200	7.50	170	61,000	63.54
55	8,500	8.85	180	67,000	70.00
60	10,000	10.50	190	73,500	76.60
65	11,500	12.00	200	80,000	83.23

Col. A = temperature difference between steam in pipe and average temperature of the water in the tank in degrees.

Col. B = B.T.U. transmitted per sq. ft. per hour.

Col. C = lbs. of steam condensed per sq. ft. per hour. Iron pipe will condense ½ as much steam as given in table for brass pipe.

Example: Heat 200 gals. water per hour from 50° to 164°F. with brass pipe and steam at 5 lbs. gauge pressure.

Temperature of steam 5 lbs. pressure	= 227°F.
Average temperature of water (50 + 164) ÷ 2	= 107°F.
Temperature difference	= 120°F.
200 gals. water by weight (200 $\times$ 8⅓)	= 1667 lbs.
Temperature rise = 164° − 50°	= 114°F.

B.T.U. required per hr. = 1667 $\times$ 114 = 190,038. From Col. A—120°F.—find in Col. B. that 1 sq. ft. brass pipe will give up 35,000 B.T.U. and in Col. C. this equals 36.45 lbs. of steam per hour. Then: 190,038 ÷ 35,000 = 5.43 sq. ft. brass pipe and as each sq. ft. will condense 36.45 lbs.—36.45 $\times$ 5.43 = 198 lbs. steam required per hour.

Table 30

WATER FLOW IN G.P.M. THRU PIPES FOR VARIOUS PRESSURE DROPS

NOT FOR HEATING SYSTEMS

Pressure Drop Lbs. per Sq. In. per 100 Ft. Run	IRON PIPE SIZE IN INCHES								
	¾"	1"	1¼"	1½"	2"	2½"	3"	3½"	4"
5	5.4	11	19	30	62	109	171	252	353
7	6.4	13	23	36	74	129	203	298	418
10	7.6	15	27	43	88	154	242	357	499
20	10.8	22	38	61	125	218	343	504	706
30	13.2	27	47	76	153	267	420	618	864
40	15.0	31	54	86	176	308	485	714	998
50	17.0	35	60	96	197	345	542	800	1115
75	21.0	43	74	117	242	423	665	978	1365
100	24.0	49	85	136	278	485	769	1130	1578
125	27.0	55	96	152	311	544	858	1260	1765
150	30.0	60	105	166	341	598	939	1380	1930

WATER FLOW IN G.P.M. THRU TYPE L COPPER TUBING

FOR VARIOUS PRESSURE DROPS

Pressure Drop Lbs. per Sq. In. per 100 Ft. Run	TUBING SIZE IN INCHES						
	⅜"	½"	¾"	1"	1¼"	1½"	2"
5	1.2	2.4	6.2	14.	23.	37.	78.
7	1.5	2.9	7.6	16.	28.	45.	93.
10	1.8	3.5	9.4	19.	34.	53.	115.
20	2.8	5.2	14.0	29.	50.	80.	170.
30	3.5	6.5	18.0	37.	63.	100.	215.
40	4.1	7.8	21.0	43.	75.	120.	250.
50	4.7	8.9	24.0	49.	85.	135.	290.
75	5.9	11.0	30.0	61.	105.	170.	360.
100	7.0	13.0	36.0	72.	125.	200.	425.

Table 31

BAROMETRIC PRESSURES IN POUNDS PER SQ. IN.

Barometer Inches	Pressure in Lbs. per Sq. In.
28.00	13.75
28.25	13.87
28.50	13.99
28.75	14.12
29.00	14.24
29.25	14.36
29.50	14.48
29.75	14.61
30.00	14.73
30.25	14.85
30.50	14.98
30.75	15.10
31.00	15.22
31.25	15.34

Table 32

SUCTION LIFT OF PUMPS WITH BAROMETRIC PRESSURE AT DIFFERENT ALTITUDES & EQUIVALENT HEAD OF WATER IN FEET

Altitude	Barometric Pressure	Equivalent Head of Water in Feet	Practical Suction Lift
Sea Level	14.70 lbs. sq. in.	33.95	22 ft.
¼ Mile Above	14.02 lbs. sq. in.	32.38	21 ft.
½ Mile Above	13.33 lbs. sq. in.	30.79	20 ft.
¾ Mile Above	12.66 lbs. sq. in.	29.24	18 ft.
1 Mile Above	12.02 lbs. sq. in.	27.76	17 ft.
1¼ Miles Above	11.42 lbs. sq. in.	26.38	16 ft.
1½ Miles Above	10.88 lbs. sq. in.	25.13	15 ft.
2 Miles Above	9.88 lbs. sq. in.	22.82	14 ft.

HIGHEST TEMP. °F. OF CONDENSATE PERMISSIBLE WITH VACUUM PUMPS

Vacuum Inches of Mercury in Pump Receiver	Highest Permissible Temperature of Condensate °F
15	179
12	187
10	192
8	196
6	201
4	205
2	209
1	210

Table 33

WATER REQUIRED FOR HUMIDIFICATION

The approximate rule for calculating the amount of water required per hour to maintain any desired humidity in a room is:

Multiply the difference between the number of grains of moisture per cubic foot of air at the required room temperature and humidity and the number of grains per cubic foot of outside air at the given temperature and humidity by the cubic contents of the room by the number of air changes per hour and divide the result by 7000 (this method disregards the expansion of air when heated.)

For most localities, it is customary to assume the average humidity of outside air as 30-40%.

Example (see table, page 38)

Grains of moisture at 70° & 40% humidity =3.19
Grains of moisture at 0° & 30% humidity = .17
Grains of moisture to be added per cu. ft. =3.02

Assuming two air changes per hour in a room containing 8000 cu. ft. we have

$$\frac{3.02 \times 8000 \times 2}{7000} = 6.9 \text{ lbs. of water per hour required.}$$

Table 35

LENGTH OF EXPANSION OFFSETS AND BENDS FOR PROPER EXPANSION OF PIPE

Total Expansion in Inches*	FEET OF PIPE IN OFFSET OR U-BEND FOR DIFFERENT DIAMETERS OF PIPE									
	2″	3″	4″	5″	6″	8″	10″	12″	14″	16″
1	11	13	15	17	19	21	23	25	27	30
2	15	18	21	23	26	29	32	35	38	42
3	18	22	26	29	32	36	40	43	48	52
4	21	26	30	34	37	42	47	50	56	58
5	24	30	34	38	41	47	53	57	63	65
6	27	33	37	41	45	52	58	63	69	71
7	30	36	40	44	48	56	62	68	74	
8	32	39	43	47	52	60	66	72		

*—This column shows the total expansion the offset will take care of without a cold strain. In general these amounts can be increased 40 per cent which increase can be taken up in cold strain of the pipe on being made up.

The length of pipe in the expansion piece should be the same whether in the form of a single right-angle offset or double offset or U-bend.

The lengths of arms figured for 12,000 lb. per square inch tension for wrought iron pipe. If steel pipe is used this is good for 16,000 lb. per inch so that the arm will take care of ⅓ more expansion.

EXPANSION OF PIPES IN INCHES PER 100 FT

Temperature Degrees F	Cast Iron	Wrought Iron	Steel	Brass or Copper
0	0.00	0.00	0.00	0.00
50	0.36	0.40	0.38	0.57
100	0.72	0.79	0.76	1.14
125	0.88	0.97	0.92	1.40
150	1.10	1.21	1.15	1.75
175	1.28	1.41	1.34	2.04
200	1.50	1.65	1.57	2.38
225	1.70	1.87	1.78	2.70
250	1.90	2.09	1.99	3.02
275	2.15	2.36	2.26	3.42
300	2.35	2.58	2.47	3.74
325	2.60	2.86	2.73	4.13
350	2.80	3.08	2.94	4.45

Table 34

PROPERTIES OF AIR

DRY AIR					SATURATED AIR				
Temperature Degrees F.	Weight per Cu. Ft. of Dry Air in Lbs.	Ratio to Volume at 70°F.	B.T.U. Absorbed per Cu. Ft. of Air per Deg. F.	Cu. Ft. of Air Raised 1°F. by 1 B.T.U.	Temperature Degrees F.	Vapor Press. Inches of Mercury	Weight of Vapor per Cu. Ft. in Lbs.	B.T.U. Absorbed per Cu. Ft. of Air per Deg. F.	Cu. Ft. of Air Raised 1°F. by 1 B.T.U.
0	.08636	.8680	.02080	48.08	0	0.0383	.000069	.02082	48.04
10	.08453	.8867	.02039	49.05	10	0.0631	.000111	.02039	49.50
20	.08276	.9057	.01998	50.05	20	0.1030	.000177	.01998	50.05
30	.08107	.9246	.01957	51.10	30	0.1640	.000276	.01955	51.15
40	.07945	.9434	.01919	52.11	40	0.2477	.000409	.01921	52.06
50	.07788	.9624	.01881	53.17	50	0.3625	.000587	.01883	53.11
60	.07640	.9811	.01846	54.18	60	0.5220	.000829	.01852	54.00
70	.07495	1.0000	.01812	55.19	70	0.7390	.001152	.01811	55.22
80	.07356	1.0190	.01779	56.21	80	1.0290	.001576	.01788	55.93
90	.07222	1.0380	.01747	57.25	90	1.4170	.002132	.01763	56.72
100	.07093	1.0570	.01716	58.28	100	1.9260	.002848	.01737	57.57
110	.06968	1.0756	.01687	59.28	110	2.5890	.003763	.01716	58.27
120	.06848	1.0945	.01659	60.28	120	3.4380	.004914	.01696	58.96
130	.06732	1.1133	.01631	61.32	130	4.5200	.006357	.01681	59.50
140	.06620	1.1320	.01605	62.31	140	5.8800	.008140	.01669	59.92
150	.06510	1.1512	.01578	63.37	150	7.5700	.010310	.01663	60.14
160	.06406	1.1700	.01554	64.35	160	9.6500	.012956	.01664	60.10
170	.06304	1.1890	.01530	65.36	170	12.2000	.016140	.01671	59.85
180	.06205	1.2080	.01506	66.40	180	15.2900	.019940	.01682	59.45

Table 36

RELATIVE HUMIDITY—PERCENT

Dry Bulb Thermometer Deg. F.	DIFFERENCE BETWEEN DRY AND WET BULB THERMOMETER																										
	1.	2.	3.	4.	5.	6.	7.	8.	9.	10.	11.	12.	13.	14.	15.	16.	17.	18.	19.	20.	21.	22.	23.	24.	26.	28.	30.
32	89	79	69	59	49	39	30	20	11	2	0																
40	92	83	75	68	60	52	45	37	29	23	15	7	0														
50	93	87	80	74	67	61	55	49	43	38	32	27	21	16	11	5	0										
60	94	89	83	78	73	68	63	58	53	48	43	39	34	30	26	21	17	13	9	5	1	0					
70	95	90	86	81	77	72	68	64	59	55	51	48	44	40	36	33	29	25	22	19	15	12	9	6	0		
80	96	91	87	83	79	75	72	68	64	61	57	54	50	47	44	41	38	35	32	29	26	23	20	18	12	7	
90	96	92	89	85	81	78	74	71	68	65	61	58	55	52	49	47	44	41	39	36	34	31	29	26	22	17	13
100	96	93	89	86	83	80	77	73	70	68	65	62	59	56	54	51	49	46	44	41	39	37	35	33	28	24	21
110	97	93	90	87	84	81	78	75	73	70	67	65	62	60	57	55	52	50	48	46	44	42	40	38	34	30	26
120	97	94	91	88	85	82	80	77	74	72	69	67	65	62	60	58	55	53	51	49	47	45	43	41	38	34	31
140	97	95	92	89	87	84	82	79	77	75	73	70	68	66	64	62	60	58	56	54	53	51	49	47	44	41	38

EXAMPLE:
DRY BULB THERMOMETER = 70°
WET BULB THERMOMETER = 59°
DIFFERENCE = 11°
SATURATION = 100% — BAROMETER 30″

FOLLOW THE HORIZONTAL LINE OPPOSITE 70° DRY BULB TEMP. TO THE VERTICAL LINE UNDER 11 = RELATIVE HUMIDITY—51%

Table 37

GRAINS OF MOISTURE PER CUBIC FOOT OF AIR AT VARIOUS TEMPERATURES AND HUMIDITIES

TEMPERATURE DEGREES-F.	RELATIVE HUMIDITY, PERCENT									
	100%	90%	80%	70%	60%	50%	40%	30%	20%	10%
75	9.35	8.42	7.49	6.55	5.61	4.68	3.74	2.81	1.87	0.94
72	8.51	7.66	6.81	5.96	5.11	4.25	3.40	2.55	1.70	0.85
70	7.98	7.18	6.38	5.59	4.79	3.99	3.19	2.39	1.60	0.80
67	7.24	6.52	5.79	5.07	4.35	3.62	2.90	2.17	1.45	0.72
65	6.78	6.10	5.43	4.75	4.07	3.39	2.71	2.04	1.36	0.68
60	5.74	5.17	4.60	4.02	3.45	2.87	2.30	1.72	1.15	0.57
50	4.08	3.67	3.26	2.85	2.45	2.04	1.63	1.22	0.82	0.41
40	2.85	2.56	2.28	1.99	1.71	1.42	1.14	0.86	0.57	0.29
30	1.94	1.74	1.55	1.35	1.16	0.97	0.78	0.58	0.39	0.19
20	1.23	1.11	0.99	0.86	0.74	0.62	0.49	0.37	0.25	0.12
10	0.78	0.70	0.62	0.54	0.47	0.39	0.31	0.23	0.16	0.08
0	0.48	0.43	0.39	0.34	0.29	0.24	0.19	0.14	0.10	0.05

7000 Grains of moisture = 1 pound of water.

Table 38

DRYING TIME & CONDITIONS FOR REPRESENTATIVE MATERIAL

KIND AND THICKNESS OF MATERIAL	Temp. Degrees F.	Drying Time
Bedding	150-190	4-6 Hours
Cereals	110-150	24 Hours
Cocoanut	145-155	30 Min.
Coffee	160-180	2½ Hours
Cores—Oil Sand, Molding ½"-1" Thick	300	4½ Hours
Cores—Oil Sand, Molding	480	10 Hours
Black Sand with Goulac Binder about 8"	480	
6/10 of Time for Oil Sand Cores 16"	700	
Feathers	150-180	
Films—Photographic	90	
Fruits and Vegetables	140	2-6 Hours
Furs	110	
Glue	70-90	2-4 Hours
Glue Size on Furniture	130	4 Hours
Gut	150	
Gypsum Wall Board—Start Wet	350	1 Hour
Gypsum Wall Board—Finish	190	
Gypsum Blocks	350-190	8-16 Hours
Hair Goods	150-190	1 Hour
Hats—Felt	140-180	
Hops	120-180	
Hides—Thin Leather	90	2-4 Hours

KIND AND THICKNESS OF MATERIAL	Temp. Degrees F.	Drying Time
Ink—Printing	70-300	
Knitted Fabrics	140-180	
Leather—Thick Sole	90	4-6 Hours
Lumber—Green hardwood	100-180	3-180 Days
Lumber—Green softwood	160-220	2-14 Days
Macaroni	90-110	
Matches	140-180	
Milk and Other Liquid Foods (Spray Dried)	250-300	Instantaneous
Molds, Green Sand, C.I. Flasks (1 sur. only ex.) 8" T.	600	6 Hours
Molds, Green Sand, C.I. Flasks (1 sur. only ex.) 13" T.	700	13 Hours
Nuts	75-140	24 Hours
Paper—Glued	130-300	
Paper—Treated	140-200	
Rubber	80-90	6-12 Hours
Sand, Loose 1" deep	300	10-15 Min.
Shade Cloth	240	1-2 Hours
Soap	125	12 Hours
Starch	180-200	1-4 Hours
Stock Feed—Mixed	180-220	20-30 Min.
Sugar	150-200	20-30 Min.
Tannin and Other Chemicals (Spray Dried)	250-300	Instantaneous
Terracotta (Air Drying in Conditioning Room)	150-220	12-96 Hours
Wall Board	200-250	12-24 Hours

Table 39

APPROXIMATE FUEL EQUIVALENTS FOR HOUSE HEATING

Tons of Coal At 55% Efficiency 12500 BTU per Lb.	Gallons of Oil At 60% Efficiency 140000 BTU per Gal.	Cubic Feet of Gas At 75% Efficiency 550 BTU per Cu. Ft.
1.0	164.	33300.
6.1	1000.	203600.
.030	4.9	1000.

APPROXIMATE NUMBER OF CUBIC FEET PER TON OF COAL

TYPE	SIZE	CU. FT./TON
ANTHRACITE	EGG	38
	STOVE	39
	CHESTNUT	39
	PEA & SMALLER	40
BITUMINOUS	EGG	48
	NUT	44
	LUMP	45
	SCREENINGS	44
	MINE RUN	42

APPROXIMATE NUMBER OF CUBIC FEET PER TON OF COAL

TYPE	SIZE	CU. FT./TON
COKE	EGG	74
	NUT	71
	PEA	66
	RANGE	74
PETROLEUM COKE	LUMP	74
	SCREENINGS	44
POCAHONTAS	EGG	41
	NUT	44
	LUMP	38
	STOVE	46
	MINE RUN	37

The above table was compiled from information furnished by the following:
Commercial Testing & Engineering Co., Chicago
Anthracite Industries, Inc., New York

Table 40

WATER EVAPORATED AND HEAT REQUIRED FOR DRYING

M = PERCENTAGE OF MOISTURE IN MATERIAL TO BE DRIED
Q = LBS. WATER EVAPORATED PER TON (2000 LBS.) OF DRY MATERIAL
H = BRITISH THERMAL UNITS REQUIRED FOR DRYING PER TON OF DRY MATERIAL

M.	Q.	H.	M.	Q.	H.	M.	Q.	H.
1	20.2	85,624	14	325.6	424,884	35	1,077	1,269,240
2	40.8	108,696	15	352.9	458,248	40	1,333	1,555,960
3	61.9	130,424	16	381.0	489,720	45	1,636	1,895,320
4	83.3	156,296	17	409.6	521,752	50	2,000	2,303,000
5	105.3	180,936	18	439.0	554,680	55	2,444	2,800,280
6	127.7	206,024	19	469.1	588,392	60	3,000	3,423,000
7	150.5	231,560	20	500.0	623,000	65	3,714	4,222,680
8	173.9	257,768	21	531.6	658,392	70	4,667	5,290,040
9	197.8	284,536	22	564.1	694,792	75	6,000	6,783,000
10	222.2	311,864	23	597.4	732,088	80	8,000	9,023,000
11	247.2	339,864	24	631.6	770,392	85	11,333	12,755,960
12	272.7	368,424	25	666.7	809,704	90	18,000	20,223,000
13	298.9	397,768	30	857.0	1,022,840	95	38,000	42,623,000

FORMULA:
$Q = \frac{2000M}{100-M}$, $H = 1120\,Q + 63000$.

The value of H is found on the assumption that the moisture is heated from 62° to 212° and evaporated at that temperature, and that the specific heat of the material is 0.21 ($2000 \times (212 - 62) \times 0.21) = 63000$.

Table 41

HEATING VALUES OF FUELS

All of the values given here are very approximate but sufficiently close for ordinary calculations. If greater accuracy is required the exact value should be obtained from the producer of the fuel in question. The heating value is also referred to as the "calorific value".

COAL

TYPE	BTU PER POUND
ANTHRACITE	13000
SEMI ANTHRACITE	13700
BITUMINOUS	12500
LIGNITE	7000

OIL

GRADE	BTU PER GALLON
No. 1	135000
No. 2	140000
No. 4	155000
No. 5	150000
No. 6	153000

GAS

KIND	BTU PER CUBIC FOOT
NATURAL	1000
MANUFACTURED	550
PROPANE	2250
BUTANE	3000

Table 42

SPECIFIC GRAVITY OF METALS

METAL	SPECIFIC GRAVITY
Water (Basis for Comparison)	1.00
Aluminum	2.55-2.80
Tin (Cast)	7.2-7.5
Steel	7.84
Cast Iron	7.03-7.13
Wrought Iron	7.6-7.9
Brass	8.4-8.7
Copper	8.8-8.95
Lead (Cast)	11.35
Mercury	13.60
Platinum	21.50

Table 44

SPECIFIC HEAT

AVERAGE VALUES 32° TO 212°F.

Air*	.237	Ice	.505
Air†	.169	Iron, Cast	.113
Alcohol	.615	Kerosene	.500
Aluminum	.212	Lead	.030
Ammonia	1.098	Limestone	.217
Ammonia*	.520	Marble	.206
Ammonia†	.391	Mercury	.033
Antimony	.052	Mica	.208
Asbestos	.195	Nickel	.109
Benzole	.340	Nitrogen*	.244
Bismuth	.030	Nitrogen†	.173
Brass	.092	Oak	.570
Brick	.220	Olive Oil	.471
Bronze	.104	Oxygen*	.224
Carbon, Graphite	.126	Oxygen†	.155
Carbon, Dioxide*	.215	Osmium	.031
Carbon, Dioxide†	.168	Paraffin	.589
Carbon, Monoxide*	.243	Petroleum	.504
Carbon, Monoxide†	.173	Platinum	.032
Cement, Portland	.271	Rubber, Hard	.339
Chalk	.220	Sand	.195
Chloroform (Liquid)	.235	Selenium	.068
Chloroform (Gas)	.147	Silicon	.175
Coal	.201	Silver	.056
Cobalt	.103	Steam*	.480
Coke	.0203	Steam†	.350
Concrete	.156	Stones	.200
Copper	.092	Steel	.118
Cork	.485	Sulphuric Acid	.336
Cotton	.362	Tantalum	.033
Ether	.540	Tin	.054
Fuel Oil	.500	Turpentine	.420
Gasoline	.500	Tungsten	.034
Glass	.180	Water	1.000
Gold	.032	Wool	.393
Gypsum	.259	Wood	.327
Hydrogen*	3.41	Zinc	.093
Hydrogen†	2.81		

* = Constant Pressure. † = Constant Volume.

Table 43

LINEAL EXPANSION OF SOLIDS AT ORDINARY TEMPERATURES

MATERIAL	For 1°F. Length = 1"
Aluminum (Cast)	0.00001234
Brass Cast	0.00000957
Brass Plate	0.00001052
Brick (Fire)	0.00000300
Bronze (Copper, 96½ ; Tin, 2½ ; Zinc, 1)	0.00000986
Copper	0.00000887
Glass, Hard	0.00000397
Gold, Pure	0.00000786
Iron, Wrought	0.00000648
Iron, Cast	0.00000556
Lead	0.00001571
Mercury (Cubic Expansion)	0.00009984
Nickel	0.00000695
Porcelain	0.00000200
Silver, Pure	0.00001079
Slate	0.00000577
Steel, Cast	0.00000636
Steel, Tempered	0.00000689
Stone (Sandstone), Dry	0.00000652
Tin	0.00001163
Wood, Pine	0.00000276
Zinc	0.00001407

Table 45

WEIGHTS OF MATERIALS

MATERIAL	Pounds per Cubic Ft.	MATERIAL	Pounds per Cubic Ft.
Aluminum	166.5	Lead	709.7
Ashes	45-50	Lignite	31-47
Barley	37-40	Lime	50-80
Brass—Copper, Zinc		Limestone	156-162
80 20	536.3	Manganese	450
70 30	523.8	Mercury 32°	849.3
60 40	521.3	Mercury 60°	846.8
50 50	511.4	Mercury 212°	834.4
Bronze—Cop. 95 to 80	552.	Nickel	548.7
Tin 5 to 20		Oats	25-30
Cement	90-118	Ore (Iron)	105-215
Charcoal	17-27	Platinum	1333
Clay	95-169	Rye	44-50
Coal (Lump)	50-54	Sand	75-120
Nut Coal & Screenings	53-60	Silver	655.1
Coke	26-30	Slag (Blast Furnace)	37-63
Earth	75-115	Steel	489.6
Gold, Pure	1200.9	Stone	90-120
Copper	552	Tin	458.3
Gravel	90-135	Wheat	44-50
Iron, Cast	450	Zinc	448
Iron, Wrought	480		

Table 46

MELTING POINTS OF METALS

	Deg. F.		Deg. F.
Aluminum	1220	Iron (Cast) Gray	2460-2550
Antimony	1167	Iron (Cast) White	1920-2010
Bismuth	520	Iron, Wrought	2460-2640
Brass (Red)	1870	Lead	622
Bronze	1900	Silver (Pure)	1761
Copper	1981	Steel	2370-2550
Glass	2377	Tin	449
Gold (Pure)	1945	Zinc	787
Solder (Lead-Tin)	350-570		

Table 47

BOILING POINTS OF VARIOUS FLUIDS

	Deg. F.		Deg. F.
Water (Atmospheric Pressure)	212	Turpentine	320
Alcohol	173	Sulphur	832
Sulphuric Acid	620	Linseed Oil	549

Table 48

CONVERSION TABLE
METRIC TO ENGLISH MEASURE

METRIC	ENGLISH
MEASURES OF LENGTH	
1 Kilometer 1000 Meters	0.621 Mile 3281. Feet
1 Meter 100 Centimeters 1000 Millimeters	1.094 Yards 3.28 Feet 39.37 Inches
1 Centimeter 10 Millimeters	0.0328 Feet 0.394 Inches
1 Millimeter	0.0394 Inches
MEASURES OF SURFACE	
1 Sq. Kilometer 1,000,000 Sq. Meters	0.386 Sq. Mile 247.1 Acres 1,195,985 Sq. Yards
1 Sq. Meter 10,000 Sq. Centimeters	1.196 Sq. Yards 10.76 Sq. Feet 1550. Sq. Inches
1 Sq. Centimeter 100 Sq. Millimeters 1 Sq. Millimeter	0.155 Sq. Inch 0.0011 Sq. Feet 0.00155 Sq. Inch

CONVERSION TABLE
METRIC TO ENGLISH MEASURE

METRIC	ENGLISH
MEASURES OF VOLUME AND CAPACITY	
1 Cu. Meter 1000 Liters 1.000.000 Cu. Centimeters	1.308 Cu. Yards 35.31 Cu. Feet 61023.4 Cu. Inches
1 Liter 1,000 Cu. Centimeters	0.264 Gallons (U.S.) 0.220 Gallons (Imperial) 1.057 Quarts (U.S.) 0.880 Quarts (Imperial)
1 Cu. Centimeter 1000 Cu. Millimeters	0.061 Cu. Inches
MEASURES OF WEIGHT	
1 Kilogram 1000 Gram	0.0011 Ton (2000 Lbs.) 2.205 Pounds (Av.)
1 Gram	0.0022 Pounds (Av.) 0.035 Ounces (Av.) 15.43 Grains

NOTE—For conversion of pressures from metric to English units see "Conversion Table—Pressures."

Table 49

CONVERSION TABLE PRESSURES

INCHES		FEET	MILLIMETER	METER	POUNDS		GRAMS	KILOGRAMS		Atmos-
Water	Mercury	Water	Mercury (Hg.)	Water	Per Sq. Inch	Per Sq. Foot	Sq. Centimeter	Per Sq. Meter	Per Sq. Cm.	phere
1.0	0.074	0.083	1.88	0.0254	0.036	5.20	2.53	25.37	0.003	0.0024
13.6	**1.0**	1.13	25.4	0.344	0.490	70.5	34.4	344.4	0.0344	0.0333
12.0	0.884	**1.0**	22.4	0.305	0.433	62.4	30.4	304.5	0.0304	0.0295
0.54	0.039	0.045	**1.0**	0.014	0.019	2.78	1.36	13.6	0.00136	0.00131
39.4	2.89	3.28	73.5	**1.0**	1.422	204.6	100.0	1000.0	0.10	0.0967
27.7	2.04	2.390	51.8	0.703	**1.0**	144.0	70.3	703.1	0.070	0.0680
0.19	0.014	0.016	0.36	0.005	0.0069	**1.0**	0.49	4.88	0.00049	0.00047
0.40	0.03	0.033	0.74	0.01	0.014	2.05	**1.0**	10.0	0.001	0.00097
0.04	0.003	0.0033	0.074	0.001	0.0014	0.205	0.10	**1.0**	0.0001	0.0001
393.8	28.96	32.8	735.5	10.0	14.2	2048.	1000.0	10000.	**1.0**	0.9678
407.	29.92	33.9	760.0	10.3	14.7	2116.	1033.2	10332.	1.03	**1.0**

To use table go to column headed by unit to be converted. Follow this column down to the "1.0" in heavy print and read horizontally across. Example: Convert five kilograms per sq. meter to lbs. per sq. inch. Select column headed kilograms per sq. meter and follow down to "1.0", then to left to column headed lbs. per sq. inch. The number 0.0014 found in this space is the conversion factor by which the number of kilograms per sq. meter must be multiplied to change this quantity to lbs. per sq. inch. Therefore, five kilograms per meter is equal to five times 0.0014, or 0.007 lbs. per sq. inch.

Table 50

INCHES (in.) to CENTIMETERS (cm) to MILLIMETERS (mm)

in.	cm.	mm.
1.00	2.54	25.40
2.00	5.08	50.80
3.00	7.62	76.20
4.00	10.16	101.60
5.00	12.70	127.00
6.00	15.24	152.40
7.00	17.78	177.80
8.00	20.32	203.20
9.00	22.86	228.60
10.00	25.40	254.00
20.00	50.80	508.00
30.00	76.20	762.00
36.00	91.40	914.00
40.00	101.60	1016.00
50.00	127.00	1270.00
60.00	152.40	1524.00
70.00	177.80	1778.00
80.00	203.20	2032.00
90.00	228.60	2286.00
100.00	254.00	2540.00

Table 51

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 1-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.

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 1000 watts, or 3415 B.T.U. when used for heating or the equivalent output of 14.2 sq. ft. of steam radiation.

ONE KILOWATT HOUR—(KW. HR.) equals the consumption of 1000 watts in one hour.

To find the watts consumed in a given electrical circuit, 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. With A.C. current multiply the wattage by the power factor, then divide by 74.6.

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 ohms 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 53

FAHRENHEIT—CENTIGRADE CONVERSION TABLE

F.	C.	F.	C.	F.	C.	F.	C.
−20	−28.9	62	16.7	144	62.2	226	107.8
−18	−27.8	64	17.8	146	63.3	228	108.9
−16	−26.7	66	18.9	148	64.4	230	110.
−14	−25.6	68	20.	150	65.6	232	111.1
−12	−24.4	70	21.1	152	66.7	234	112.2
−10	−23.3	72	22.2	154	67.8	236	113.3
− 8	−22.2	74	23.3	156	68.9	238	114.4
− 6	−21.1	76	24.4	158	70.	240	115.6
− 4	−20.	78	25.6	160	71.1	242	116.7
− 2	−18.9	80	26.7	162	72.2	244	117.8
0	−17.8	82	27.8	164	73.3	246	118.9
2	−16.7	84	28.9	166	74.4	248	120.
4	−15.6	86	30.	168	75.6	250	121.1
6	−14.4	88	31.1	170	76.7	252	122.2
8	−13.3	90	32.2	172	77.8	254	123.3
10	−12.2	92	33.3	174	78.9	256	124.4
12	−11.1	94	34.4	176	80.	258	125.6
14	−10.	96	35.6	178	81.1	260	126.7
16	− 8.9	98	36.7	180	82.2	262	127.8
18	− 7.8	100	37.8	182	83.3	264	128.9

F.	C.	F.	C.	F.	C.	F.	C.
20	− 6.7	102	38.9	184	84.4	266	130.
22	− 5.6	104	40.	186	85.6	268	131.1
24	− 4.4	106	41.1	188	86.7	270	132.2
26	− 3.3	108	42.2	190	87.8	272	133.3
28	− 2.2	110	43.3	192	88.9	274	134.4
30	− 1.1	112	44.4	194	90.	276	135.6
32	0.	114	45.6	196	91.1	278	136.7
34	1.1	116	46.7	198	92.2	280	137.8
36	2.2	118	47.8	200	93.3	282	138.9
38	3.3	120	48.9	202	94.4	284	140.
40	4.4	122	50.	204	95.6	286	141.1
42	5.6	124	51.1	206	96.7	288	142.2
44	6.7	126	52.2	208	97.8	290	143.3
46	7.8	128	53.3	210	98.9	292	144.4
48	8.9	130	54.4	212	100.	294	145.6
50	10.	132	55.6	214	101.1	296	146.7
52	11.1	134	56.7	216	102.2	298	147.8
54	12.2	136	57.8	218	103.3	300	148.9
56	13.3	138	58.9	220	104.4		
58	14.4	140	60.	222	105.6		
60	15.6	142	61.1	224	106.7		

Table 52

CAPACITY OF ROUND STORAGE TANKS

Depth or Length	NUMBER OF GALLONS — INSIDE DIAMETER IN INCHES									
	18	24	30	36	42	48	54	60	66	72
1 In.	1.10	1.96	3.06	4.41	5.99	7.83	9.91	12.24	14.81	17.62
2 Ft.	26	47	73	105	144	188	238	294	356	423
2½	33	59	91	131	180	235	298	367	445	530
3	40	71	100	158	216	282	357	440	534	635
3½	46	83	129	184	252	329	416	513	623	740
4	53	95	147	210	288	376	475	586	712	846
4½	59	107	165	238	324	423	534	660	800	952
5	66	119	181	264	360	470	596	734	899	1057
5½	73	130	201	290	396	517	655	808	978	1163
6	79	141	219	315	432	564	714	880	1066	1268
6½	88	155	236	340	468	611	770	954	1156	1374
7	92	165	255	368	504	658	832	1028	1244	1480
7½	99	179	278	396	540	705	889	1101	1335	1586
8	106	190	291	423	576	752	949	1175	1424	1691
9	119	212	330	476	648	846	1071	1322	1599	1903
10	132	236	366	529	720	940	1189	1463	1780	2114
12	157	282	440	634	864	1128	1428	1762	2133	2537
14	185	329	514	740	1008	1316	1666	2056	2490	2960
16	211	376	587	846	1152	1504	1904	2350	2844	3383
18	238	423	660	952	1296	1692	2140	2640	3200	3806
20	264	470	734	1057	1440	1880	2380	2932	3556	4230

CAPACITY OF RECTANGULAR TANKS

To find the capacity in U.S. gallons of rectangular tanks, reduce all dimensions to inches, then multiply the length by the width by the height and divide the product by 231.

Example:

Tank 56″ long × 32″ wide × 20″ deep
Then 56″ × 32″ × 20″ = 35840 cu. in.
35840 ÷ 231 = 155 gallons capacity

Table 54

USEFUL DATA

Diameter	×	3.1416	= Circumference
Circumference	×	.3183	= Diameter
Diameter²	×	.7854	= Area of Circle
Area of Circle	×	1.2732	= Area of Circumscribed Square
Area of Circle	×	.63662	= Area of Inscribed Square
Diameter of Circle	×	.88623	= Side of Equal Square
Diameter of Circle	×	.7071	= Side of Inscribed Square
Circumference of Circle	×	1.1284	= Perimeter of Equal Square
Side of Square	×	1.4142	= Diameter of Circumscribed Circle
Side of Square	×	1.1284	= Diameter of Equal Circle
Perimeter of Square	×	.88623	= Circumference of Equal Circle
Diameter²	×	3.1416	= Surface of Sphere
Diameter³	×	.5236	= Volume of Sphere
Diameter of Sphere	×	.806	= Dimensions of Equal Cube
Diameter of Sphere	×	.667	= Length of Equal Cylinder
Area of Base	×	⅓ Height	= Volume of Pyramid or Cone
Base	×	½ Height	= Area of Triangle
Radius	×	1.1547	= Side of Inscribed Cube
Square Inches	×	1.2732	= Circular Inches
Square Inches	×	.00695	= Square Feet
Square Feet	×	.111	= Square Yard
Square Yards	×	.0002066	= Acres
Cubic Inches	×	.00058	= Cubic Feet
Cubic Feet	×	.03704	= Cubic Yards
Cubic Inches	×	.004329	= U.S. Gallons
Cubic Feet	×	7.4805	= U.S. Gallons
Cubic Inches	×	.000466	= U.S. Bushels
Cubic Feet	×	.8036	= U.S. Bushels
U.S. Bushels	×	2150.42	= Cubic Inches
U.S. Bushels	×	1.242	= Cubic Feet
U.S. Bushels	×	.046	= Cubic Yards
U.S. Gallons	×	231.0	= Cubic Inches
U.S. Gallons	×	.13368	= Cubic Feet
Cubic Inches Water	×	.036127	= Pounds (Avoirdupois)
Cubic Feet Water	×	62.4283	= Pounds (Avoirdupois)
U.S. Gallons Water	÷	268.8	= Tons
Column of Water 1″ Diameter x 12″ High			= .34 Pounds (Avoirdupois)
Cubic Inch	×	.263	= Pound Average Cast Iron
Cubic Inch	×	.281	= Pound Average Wrought Iron
Cubic Inch	×	.283	= Pound Average Cast Steel
Cubic Inch	×	.3225	= Pound Average Copper
Cubic Inch	×	.3037	= Pound Average Brass
Cubic Inch	×	.26	= Pound Average Zinc
Cubic Inch	×	.4103	= Pound Average Lead
Cubic Inch	×	.2636	= Pound Average Tin
Cubic Inch	×	.4908	= Pound Average Mercury
12 × Weight of Pine Pattern			= Iron Casting
13 × Weight of Pine Pattern			= Brass Casting
14 × Weight of Pine Pattern			= Lead Casting
1—Calorie			= 3.968 B.T.U.
1—B.T.U.			= 0.252 Calorie
1—Pound per Square Inch			= 703.08 Kilogrammes per M²
1—Kilogramme per M²			= 0.00142 Pounds per Square Inch
1—Calorie per M²			= 0.3687 B.T.U. per Square Foot
1—B.T.U. per Square Foot			= 2.712 Calories per M²
1—Calorie per M² per Degree Difference Centigrade			= 0.2048 B.T.U. per Square Foot per Degree Difference, Fahrenheit
1—B.T.U. per Square Foot per Degree Difference Fahrenheit			= 4.882 Calories per M² per Degree Difference Fahrenheit
1—B.T.U. per Pound			= 0.556 Calories per Kilogramme
1—Calorie per Kilogram			= 1.8 B.T.U. per Pound
1—Litre of Coke at 26.3 Lb. per Cu. Ft.			= 0.93 Pounds
1—Lb. of Coke at 26.3 Lb. per Cu. Ft.			= 1.076 Liter
Water Expands in Bulk from 40° to 212°			= One Twenty-Third

Table 55

WEIGHTS OF STEEL & IRON PLATES & SHEETS
U.S. STANDARD GAGE

No. of Gage	Thickness Inches (Approx.)	WEIGHT PER SQUARE FOOT	
		Iron	Steel
0000000	.500	20.00	20.40
000000	.469	18.75	19.12
00000	.437	17.50	17.85
0000	.406	16.25	16.57
000	.375	15.00	15.30
00	.344	13.75	14.02
0	.312	12.50	12.75
1	.281	11.25	11.47
2	.266	10.62	10.84
3	.250	10.00	10.20
4	.234	9.37	9.56
5	.219	8.75	8.92
6	.203	8.12	8.29
7	.187	7.50	7.65
8	.172	6.87	7.01
9	.156	6.25	6.37
10	.141	5.62	5.74
11	.125	5.00	5.10
12	.109	4.37	4.46
13	.094	3.75	3.82
14	.078	3.12	3.19
15	.070	2.81	2.87
16	.062	2.50	2.55
17	.056	2.25	2.29
18	.050	2.00	2.04
19	.044	1.75	1.78
20	.037	1.50	1.53
21	.034	1.37	1.40
22	.031	1.25	1.27
23	.028	1.13	1.15
24	.025	1.00	1.02
25	.022	0.87	0.89
26	.019	0.75	0.76
27	.017	0.69	0.70
28	.016	0.63	0.64
29	.014	0.56	0.57
30	.012	0.50	0.51
31	.011	0.44	0.45
32	.010	0.40	0.41
33	.009	0.37	0.38
34	.009	0.34	0.35
35	.008	0.31	0.32
36	.007	0.28	0.29
37	.007	0.27	0.27
38	.006	0.25	0.25

Table 56

AREAS OF CIRCLES

Size	Area	Size	Area	Size	Area	Size	Area
1/64	.00019	5½	23.758	25	490.87	64	3216.9
1/32	.00077	6	28.274	26	530.93	65	3318.3
1/16	.00307	6½	33.183	27	572.55	66	3421.2
3/32	.00690	7	38.484	28	615.75	67	3525.6
1/8	.01227	7½	44.178	29	660.52	68	3631.6
5/32	.01917	8	50.265	30	706.86	69	3739.2
3/16	.02761	8½	56.745	31	754.76	70	3848.4
7/32	.03758	9	63.617	32	804.24	71	3959.2
1/4	.04909	9½	70.882	33	855.30	72	4071.5
5/16	.07670	10	78.54	34	907.92	73	4185.3
3/8	.11045	10½	86.59	35	962.11	74	4300.8
7/16	.15033	11	95.03	36	1017.8	75	4417.8
1/2	.19635	11½	103.86	37	1075.2	76	4536.4
9/16	.24850	12	113.09	38	1134.1	77	4656.0
5/8	.30680	12½	122.71	39	1194.5	78	4778.3
11/16	.37122	13	132.73	40	1256.6	79	4901.6
3/4	.44179	13½	143.13	41	1320.2	80	5026.5
13/16	.51849	14	153.93	42	1385.4	81	5153.0
7/8	.60132	14½	165.13	43	1452.2	82	5281.0
15/16	.69029	15	176.71	44	1520.5	83	5410.6
1	.7854	15½	188.69	45	1590.4	84	5541.7
1 1/8	.9940	16	201.06	46	1661.9	85	5674.5
1 1/4	1.227	16½	216.82	47	1734.9	86	5808.8
1 3/8	1.484	17	226.98	48	1809.5	87	5944.6
1 1/2	1.767	17½	240.52	49	1885.7	88	6082.1
1 5/8	2.073	18	254.46	50	1963.5	89	6221.1
1 3/4	2.405	18½	268.80	51	2042.8	90	6361.7
1 7/8	2.761	19	283.52	52	2123.7	91	6503.8
2	3.141	19½	298.64	53	2206.1	92	6647.6
2 1/4	3.976	20	314.16	54	2290.2	93	6792.9
2 1/2	4.908	20½	330.06	55	2375.8	94	6939.7
2 3/4	5.939	21	346.36	56	2463.0	95	7088.2
3	7.068	21½	363.05	57	2551.7	96	7238.2
3 1/4	8.295	22	380.13	58	2642.0	97	7389.8
3 1/2	9.621	22½	397.60	59	2733.9	98	7542.9
3 3/4	11.044	23	415.47	60	2827.4	99	7697.7
4	12.566	23½	433.73	61	2922.4	100	7854.0
4 1/2	15.904	24	452.39	62	3019.0		
5	19.635	24½	471.43	63	3117.2		

To find the circumference of a circle when diameter is given, multiply the given diameter by 3.1416. To find the diameter of a circle when circumference is given, multiply the given circumference by .31831.

Table 57

ASTM SCHEDULE 40(S) PIPE DIMENSIONS

DIAMETER			Nominal Thickness In.	CIRCUMFERENCE		TRANSVERSE AREAS		Length of Pipe Per Sq. Ft. of External Surface Ft.	Length of Pipe Containing 1 Cu. Ft.	Nominal Weight Per Ft. in Lbs.	Number of Threads Per Inch of Screw
Nominal Internal In.	Actual External In.	Approx. Internal In.		External In.	Internal In.	External Sq. In.	Internal Sq. In.				
⅛	0.405	0.27	0.068	1.27	0.85	0.13	0.06	9.44	2513.00	0.24	27
¼	0.540	0.36	0.088	1.70	1.14	0.23	0.10	7.08	1383.30	0.42	18
⅜	0.675	0.49	0.091	2.12	1.55	0.36	0.19	5.66	751.20	0.57	18
½	0.840	0.62	0.109	2.63	1.95	0.55	0.30	4.55	472.40	0.85	14
¾	1.050	0.82	0.113	3.30	2.59	0.87	0.53	3.64	270.00	1.13	14
1	1.315	1.05	0.134	4.13	3.29	1.36	0.86	2.90	166.90	1.68	11½
1¼	1.660	1.38	0.140	5.22	4.34	2.16	1.50	2.30	96.25	2.27	11½
1½	1.900	1.61	0.145	5.97	5.06	2.84	2.04	2.01	70.66	2.72	11½
2	2.375	2.07	0.154	7.46	6.49	4.43	3.36	1.61	42.91	3.65	11½
2½	2.875	2.47	0.204	9.03	7.75	6.49	4.78	1.33	30.10	5.79	8
3	3.500	3.07	0.217	11.00	9.63	9.62	7.39	1.09	19.50	7.57	8
3½	4.000	3.55	0.226	12.57	11.15	12.57	9.89	0.96	14.57	9.11	8
4	4.500	4.03	0.237	14.14	12.65	15.90	12.73	0.85	11.31	10.79	8
5	5.563	5.05	0.259	17.48	15.85	24.31	19.99	0.69	7.20	14.62	8
6	6.625	6.07	0.280	20.81	19.05	34.47	28.89	0.58	4.98	18.97	8
8	8.625	8.07	0.276	27.10	25.35	58.43	51.15	0.44	2.82	24.69	8
8	8.625	7.98	0.322	27.10	25.07	58.43	50.02	0.44	2.88	28.55	8
9	9.625	8.94	0.344	30.24	28.08	72.76	62.72	0.40	2.29	33.91	8
10	10.750	10.19	0.278	33.77	32.01	90.76	81.55	0.36	1.76	31.20	8
10	10.750	10.14	0.306	33.77	31.86	90.76	80.75	0.36	1.78	34.24	8
10	10.750	10.02	0.366	33.77	31.47	90.76	78.82	0.36	1.82	40.48	8
12	12.750	12.09	0.328	40.06	37.98	127.68	114.80	0.30	1.25	43.77	8
12	12.750	12.00	0.375	40.06	37.70	127.68	113.10	0.30	1.27	49.56	8

Table 58

SURFACE AREAS AND VOLUMES OF SPHERES

Diameter	Surface Area	Volume	Diameter	Surface Area	Volume
⅛	.04908	.00102	4¼	56.745	40.195
¼	.19636	.00818	4½	63.617	47.712
⅜	.44180	.02761	4¾	70.882	56.115
½	.78540	.06545	5	78.540	65.450
⅝	1.2272	.12783	5¼	86.590	75.766
¾	1.7672	.22090	5½	95.033	87.113
⅞	2.4053	.35077	5¾	103.87	99.542
1	3.1416	.52360	6	113.10	113.10
1⅛	3.9760	.74550	6¼	122.72	127.83
1¼	4.9088	1.0227	6½	132.73	143.79
1⅜	5.9396	1.3611	6¾	143.14	161.03
1½	7.0684	1.7671	7	153.94	179.60
1⅝	8.2956	2.2467	7¼	165.13	199.53
1¾	9.6212	2.8062	7½	176.71	220.88
1⅞	11.045	3.4516	7¾	188.69	243.72
2	12.566	4.1887	8	201.06	268.08
2¼	15.904	5.9640	8¼	213.82	294.00
2½	19.635	8.1812	8½	226.98	321.55

SURFACE AREAS AND VOLUMES OF SPHERES

Diameter	Surface Area	Volume	Diameter	Surface Area	Volume
2¾	23.758	10.889	8¾	240.53	350.77
3	28.274	14.137	9	254.47	381.70
3¼	33.183	17.974	9¼	268.80	414.40
3½	38.484	22.449	9½	283.53	448.92
3¾	44.179	27.612	9¾	298.65	485.30
4	50.266	35.511	10	314.16	523.60

SPHERE FORMULAE

This table can be used for feet, inches or any metric unit. For example, the volume of a 2" diameter sphere is 4.1887 cu. in., and for a 2 ft. diameter, 4.1887 cu. ft. The figures apply to either the exterior or to the interior of a hollow sphere, provided the diamter is measured at the proper place. For example: the capacity of a spherical tank measuring 10 ft. on the inside is 523.60 cu. ft. A float ball having an outside diameter of 6 in. has a volume of 113.10 cu. in.

The area or the volume of a sphere of a diameter not given in the table may be figured from the following simple formulae:

$$S = 4A$$
$$V = 0.524D^3$$

in which:
D = Diameter of the Sphere
A = Area of a Circle of Diameter D
S = Surface Area of Sphere
V = Volume of Sphere

Table 59

DECIMAL EQUIVALENTS OF PARTS OF AN INCH

1/64	.01563	33/64	.51563
1/32	.03125	17/32	.53125
3/64	.04688	35/64	.54688
1/16	.0625	9/16	.5625
5/64	.07813	37/64	.57813
3/32	.09375	19/32	.59375
7/64	.10938	39/64	.60938
1/8	.125	5/8	.625
9/64	.14063	41/64	.64063
5/32	.15625	21/32	.65625
11/64	.17188	43/64	.67188
3/16	.1875	11/16	.6875
13/64	.20313	45/64	.70313
7/32	.21875	23/32	.71875
15/64	.23438	47/64	.73438
1/4	.25	3/4	.75
17/64	.26563	49/64	.76563
9/32	.28125	25/32	.78125
19/64	.29688	51/64	.79688
5/16	.3125	13/16	.8125
21/64	.32813	53/64	.82813
11/32	.34375	27/32	.84375
23/64	.35938	55/64	.85938
3/8	.375	7/8	.875
25/64	.39063	57/64	.89063
13/32	.40625	29/32	.90625
27/64	.42188	59/64	.92188
7/16	.4375	15/16	.9375
29/64	.45313	61/64	.95313
15/32	.46875	31/32	.96875
31/64	.48438	63/64	.98438
1/2	.5	1	1.00000

Table 60

FLUE SIZES AND CHIMNEY HEIGHTS

BOILER CAPACITY SQ. FT. RADIATION		RECTANGULAR FLUE		ROUND FLUE		Height in Feet from Grate
Steam 240 BTU per Sq. Ft.	Hot Water 150 BTU Per Sq. Ft.	Nominal Outside Dimensions of Fireclay Lining, In.	Effective Area Square Inches	Inside Diameter of Lining, Inches	Actual and Effective Area, Sq. In.	
590	973	8½ x 13	70			35
690	1140			10	79	
900	1490	13 x 13	99			
900	1490	8½ x 18	100			
1100	1820			12	113	40
1700	2800	13 x 18	156			
1940	3200			15	177	
2130	3520	18 x 18	195			
2480	4090	20 x 20	234			45
3150	5200			18	254	50
4300	7100			20	314	
5000	8250	24 x 24	346			55

Table 61

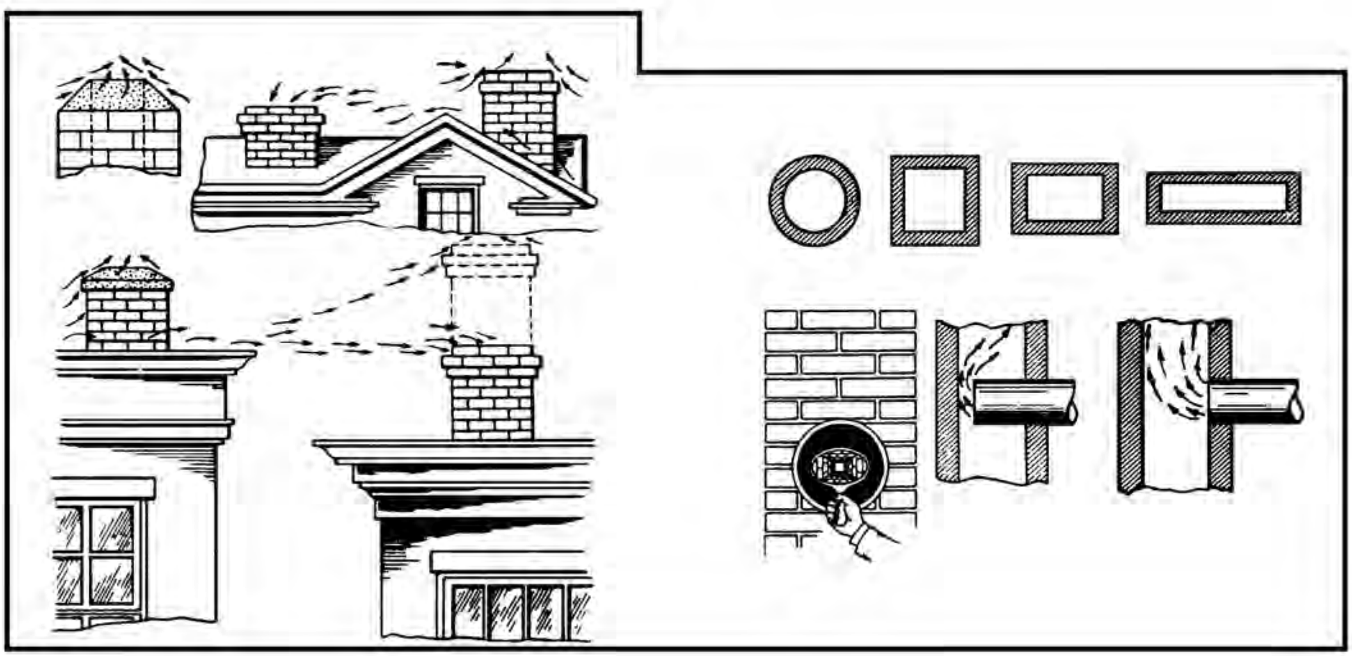


Table 62

COMMON HEATING TROUBLES IN STEAM HEATING SYSTEMS

The following are a few of the most common difficulties experienced with steam or hot water heating systems, and the probable causes.

1. BOILER TROUBLES

A. The Boiler Fails to Deliver Enough Heat

The cause of this condition may be: (a) poor draft: (b) poor fuel: (c) improper attention or firing: (d) boiler too small: (e) improper piping: (f) improper arrangement of sections: (g) heating surfaces covered with soot and (h) improper firing rate of oil or gas burner.

B. The Water Line Is Unsteady

The cause of this condition may be: (a) grease and dirt in boiler: (b) water column connected to a very active section and, therefore, may not be showing actual water level in boiler: (c) boiler operating at excessive output.

C. Water Disappears from Gauge Glass

This may be caused by: (a) priming due to grease and dirt in boiler: (b) too great a pressure difference between supply and return piping causing water to back into return: (c) valve closed in return line: (d) connection of bottom of water column into a very active section or thin waterway: (e) improper connections between boilers in battery permitting boiler with excess pressure to push water into boiler with lower pressure: (f) too high a firing rate.

D. Water Is Carried over into Steam Main

This may be caused by: (a) grease and dirt in boiler: (b) type of boiler not adapted to job: (c) outlet connections too small: (d) using boiler beyond rated capacity: (e) water level carried higher than required: (f) firing rate too high.

E. Boiler Is Slow to Respond

This may be due to: (a) poor draft: (b) inferior fuel: (c) improper attention: (d) accumulation of clinkers on grate: (e) boiler too small for the load: (f) improper firing rate.

F. Boiler Flues Collect Soot Quickly

This may be due to: (a) poor draft: (b) smoky combustion: (c) too low a rate of combustion: (d) excess of air in firebox causing chilling of gases: (f) improper firing rate.

G. Boiler Smokes through Fire Door

This may be due to: (a) defective draft in chimney or incorrect setting of dampers: (b) air leaks into boiler or breeching: (c) gas outlet from firebox plugged with fuel: (d) dirty or clogged flues: (e) improper reduction in breeching size.

H. Pressure Builds up Very Quickly (as indicated by gage) But Steam Does Not Circulate

This is due to grease and dirt in boiler.

2. PIPING TROUBLES

Water hammer is one of the chief causes of noise in steam heating systems and also is the cause of much damage to traps, vents and the like. It is a wave transmitted through a pipe filled, or partially filled, with water. It may have its origin in waves set up by steam passing at a high velocity over condensate collected in the piping. Drawing illustrates how such a wave may be formed. The rapid passage of steam over the surface of the water causes a wave to form as at "B." The rapid condensation of the steam in pockets "A" brings the two slugs of water together with a considerable force, which may be telegraphed through the piping. If this wave is intercepted by say the float of a vent valve, not only will noise result but damage to the float in the valve is almost certain.

A. Water Hammer in Hartford Connection

In the Hartford connection a close nipple should be used between the end of the return and the header drip or equalizing pipe. If the nipple used at this point is too long and the water line of boilers becomes low, hammer will occur. The remedy is to offset the return piping considerably below the water line of the boiler, so that a close nipple can be used in entering header drip and maintain proper boiler water line. Top of close nipple should be 2" below water line.

B. Water Hammer in Mains

(a) Water pocket formed by sagging of the main. An easy and reliable method to check this is to stretch a chalk line between fittings and note relation of pipe to line.

(b) Improper pitch of mains. Check with level.

(c) Water hammer is also caused by too great a pressure drop, due to insufficient pipe sizes, unreamed piping or other restrictions in the line.

(d) Insufficient water line difference between the low point of the horizontal main and boiler water line. In one-pipe gravity systems, this distance should normally be 18" or more and in vapor systems, the ends of dry return mains should be 24" or more above water line, depending on size of installation and pressure drop.

(e) Improper location of Air Valves for venting steam main. (One-pipe Gravity or Vacuum Systems).

(f) Excessive quantities of water in main due to priming boiler or improper header construction. All boiler tappings should be used and connected full size to boiler header.

C. Steam Does Not Circulate to Ends of Mains

(a) dirty boilers: (b) insufficient water line difference: (c) improper venting of mains.

3. RADIATOR TROUBLES

A. Pounding (One-pipe System)

(a) Radiator Supply Valve too small or partially closed.
(b) Radiator pitched away from supply valve.
(c) Vent port of air valve too large, allowing steam to enter radiator too rapidly.

COMMON HEATING TROUBLES IN STEAM HEATING SYSTEMS

B. Radiator Does Not Heat

(a) Improper venting of air: (b) branch supply to radiator too small.
(c) Vent port of Air Valve clogged with dirt (one pipe systems).
(d) Drainage tongue of Air Valve damaged or removed.
(e) Branch supply improperly pitched causing water pocket.

(f) Steam pressure higher than maximum working pressure of Vent Valve. This is especially likely to happen where steam is supplied through a reducing valve from high pressure supply.

(g) Return branch improperly pitched causing water pocket to form, trapping air (Vapor System).

(h) In one pipe Vacuum System, if gas or oil-fired, some radiators may not heat if on previous firing they were not completely heated. Changing to an open (non-vacuum) system by using open vents on radiators and mains would remedy this.

C. Radiator Cools Quickly (One-pipe Vacuum System)

(a) Air leakage into system either through leaky joints or through stuffing box of radiator supply valve if ordinary valve is used.

(b) Dirt in Vacuum Valve preventing formation of vacuum.

(c) On gas or oil-fired system, radiator cools quickly due to rapid formation of vacuum, and it is better to change to open (non-vacuum) system by changing radiator and main vents to non-vacuum (open) type.

4. REAMING AND PITCHING OF PIPING

Many carefully laid out systems are found to have radiators which cannot be completely heated under test. Improper reaming of pipes can be given as the cause of many such conditions. In mains using unreamed pipe, water pockets are formed which frequently cause trouble.

In- many cases troubles caused by burrs can be overcome by repitching the pipe line. If a line having 1/4" pitch in 10 ft. is increased to 1/2" in ft., the capacity is increased 20% in 3/4 and 1" lines and 10-12% in 1 1/4" to 2". "Comparative Capacity of Steam Lines at Various Pitches" in Section 7. The selection of pipe with smooth interior surface is likewise important in its effect on carrying capacity.

5. BOILER CLEANING

After a new installation has been in service for a week or so, grease, oil, scale, core sand, etc., will accumulate in the boiler. It .should then be thoroughly cleaned with some boiler cleaning compound. There are various types of cleaning compounds on the market and it is important to follow the recommendations of the boiler manufacturers in the selection of the proper one and to follow closely the prescribed method of using it.

Air valves should not be installed in a new system prior to the cleaning of the boiler. If temporary heat is required, the air may be vented with pet cocks or with old vent valves.

In systems using vacuum or condensation pumps, the return to pump should be closed off and all condensation passed to drain until the boiler has been cleaned.

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