Dave Sellers

PRINCIPLES OF STEAM HEATING

6

6–1. STEAM IN HEATING SYSTEMS

When saturated steam is supplied to a radiator or convector the useful heating is caused primarily by condensation of the steam. The heat of condensation is of course equal to the heat of vaporization (latent heat), listed as h_{fg} in tables 2–2 and 2–3. For example, at atmospheric pressure (14.7 psi) a pound of dry saturated steam, in condensing, gives up 970.3 Btu (h_{fg} from Table 2–3). During this process of condensation, just as in vaporization, the temperature remains constant (at 212 F when the pressure is 14.7 psi). Reference to Table 2–3 shows that when steam is allowed to condense at a lower pressure than atmospheric (partial vacuum) the h_{fg} values are somewhat larger than 970.3, and the saturation or condensation temperatures are lower than 212 F. In utilizing this latent heat in radiators or convectors, provision must be made for removing the condensate (water) from them. The condensate is at the same temperature as the steam until it is chilled by cool piping or radiator surface not in contact with the steam. Additional heat flow to a space, often amounting to 30 to 60 Btu per pound of steam, can occur as the liquid subcools.

Since the magnitude of h_{fg} does not greatly change over the pressure range employed in heating systems, pressure has little effect on the heat delivered by each pound of steam. However, low-pressure steam is also low-temperature steam, and since heat transfer depends primarily on the temperature difference existing between the radiator surface and the ambient air, low-pressure steam with a given radiator surface supplies less heat in a given time than high-pressure steam. The specific volume of low-pressure steam is greater than that of high-pressure steam, and so a low-pressure system, in addition to requiring somewhat greater surface in the radiators, also requires larger pipe sizes to transmit the greater volume of steam.

Radiators capable of heating a building satisfactorily in zero weather would greatly overheat the building in mild weather unless the steam were turned on and off or unless the steam temperature could be varied. With

subatmospheric steam systems this latter arrangement is used, and the steam pressure (and its temperature) is reduced in mild weather to suit the heat demand. Such cooler radiators supply less heat and prevent the necessity of frequent control through operation of steam valves. With many steam-heating systems, particularly single-pipe arrangements, partly closing a valve does not give satisfactory temperature modulation.

Because radiators and convectors are discussed in detail in Chapter 9, only two brief definitions will be given at this point. A *radiator* is a heating device placed in a space so that it can direct its radiation to part or all of the heated space. It delivers heat by radiation, convection, and conduction. A *convector* is a heating device arranged to deliver heat to the air largely by convection currents. Convectors are enclosed, or concealed from the heated space.

It is always desirable to express the capacity of a radiator or convector under stated conditions in terms of Btuh (Btu per hour) or Mbh (1000 Btu per hour). However, there still exists an old unit, called by various names, such as *equivalent direct radiation* (EDR) or *square feet of radiation*, or even *feet of radiation*. Unit EDR (equivalent direct radiation) is defined as heat delivery at a rate of 240 Btuh. Thus one square foot of EDR, if the term is used in this way, does not imply 144 sq in. of heater surface, but means a heat delivery of 240 Btuh for each indicated EDR of a given radiator or convector. For rating of radiators and convectors, it is usually considered that the output is produced when steam is condensing at 215 F in 70 F ambient space. (In the case of hot-water heating an EDR is often taken as 150 Btuh.)

6-2. STEAM FLOW IN SYSTEMS

Steam flows through pipes because of pressure differences. Heat added in a boiler causes the water to change into a vapor (steam), with a resulting increase in volume. For example, at atmospheric pressure (212 F) the volume occupied by one pound of saturated steam is about 1600 times as great as the volume occupied by one pound of water; that is, $v_g/v_f = 26.80/0.01672 = 1604$ (from Table 2-2). When one pound of water is converted into steam in a heating system at a pressure above atmospheric, the steam displaces and drives out the air. As heat is transferred from the pipes and radiators the steam condenses to water and a resultant small volume. The shrinkage in volume causes a lower pressure within the system, and the rate of steam flow from the boiler increases because of the reduced pressure. Thus two forces influence the pressure difference necessary for steam flow: first, application of heat at the boiler, with evaporation and an accompanying increase in volume; second, heat removal in the radiators and pipes, with an accompanying shrinkage in volume and reduction in pressure as condensation occurs. If the heat removal is greater than the

heat supply, the pressure of the system falls. When the heat supply at the boiler is diminished in airtight systems, transmission of heat in decreasing amount will continue while the steam pressure falls far into the vacuum region.

Flow of steam in a heating system is resisted by the skin friction of the various conduits, particularly in pipes, fittings, and valves. Condensation commences whenever saturated steam loses heat, so that in most pipes the fast-flowing steam rushes alongside a low-velocity stream of condensate. If the drainage slope of the pipe is such that the steam flow is counter to the water flow, friction is increased and therefore larger pipes must be used.

6-3. BASIC STEAM CIRCULATION

An enclosed vessel, Fig. 6-1, partly filled with water, may be placed over a fire and connected to a pipe loop, and when the water boils, the

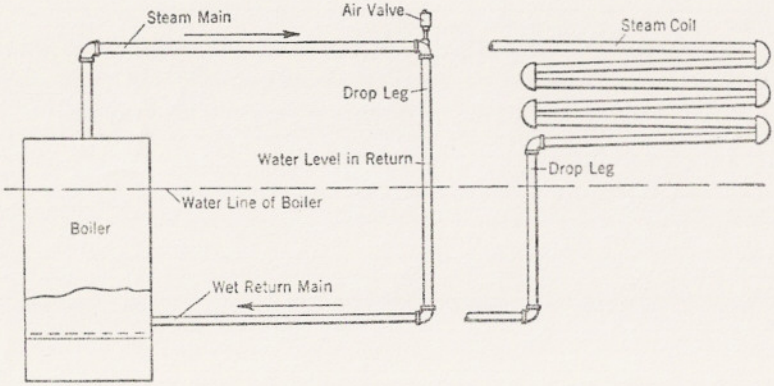

FIG. 6-1. Basic steam-circulation system, with representative pipe coil at right.

steam forming above the water in the vessel will cause an increase in pressure and compress the air in the pipes. Since the pressures throughout the system are about equal, the water line in the drop leg at the end of the steam main will be almost level with the water line in the boiler. These water surfaces will be level with each other as long as the respective pressures are equal. If air is vented from the drop leg above the water line, as steam starts to flow, this water level will rise because the friction drop reduces the pressure below that in the boiler. This pressure difference will also exist when part of the steam in the main condenses, and will be balanced by an increased head of water in the drop leg.

When a steam coil such as is shown at the right in Fig. 6-1 transfers so much heat that the pressure in the drop leg is 1 psi less than boiler pressure, the water line in the drop leg will rise about 28 in. higher than the

water level in the boiler. If the flow of steam is retarded, as by a valve or small-size pipe, the difference in pressure and water levels will increase. Thus in certain steam-heating systems the permissible difference between the water level in the boiler and that in the drop leg is the governing factor in choosing pipe sizes.

6-4. AIR VENTING

Water at normal temperature usually contains air some of which separates when the water is heated. Pipes and radiators contain air when steam is first developed. To heat the surfaces the air must be removed before the steam can enter, and must be vented continually thereafter.

Acceptable air valves allow the air to pass out, but close when steam starts to escape. Figure 6-2 shows a common construction. The float

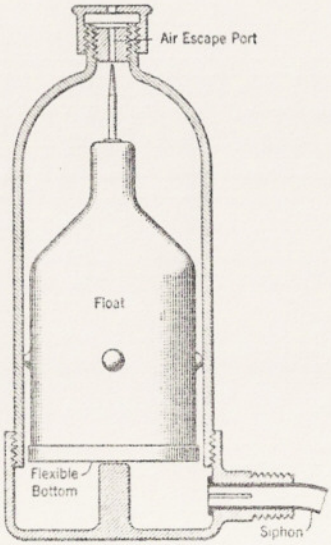

FIG. 6-2. Simple radiator air valve.

contains a small amount of volatile fluid which, expanding or contracting in response to temperature changes, moves the flexible bottom of the float outward, or allows it to spring inward. When the steam pressure increases above the pressure of the atmosphere, any air that is present is forced out through the escape port. After the relatively cool air escapes and the hot steam enters the valve, the confined fluid within the float chamber expands and thereby forces the flexible bottom downward. This action lifts the float and forces the valve pin into the port opening, thereby closing the valve. The float chamber will also rise if for any reason the valve

becomes filled with water. Thus no water can escape through the valve. The loosely fitting siphon is used when the valve is placed on radiators, to return to the radiator any accumulation of water within the valve.

With air valves of the type shown in Fig. 6–2, air will return through the escape port when the steam pressure drops to that of the atmosphere, and the entire steam system will eventually fill with air as the steam condenses. The air must again be forced out of the system whenever heat is required, and its ejection involves a considerable delay in heat distribution.

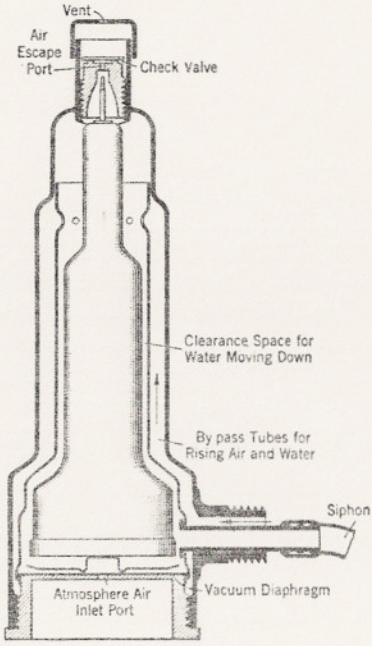

Fig. 6–3. Vacuum type of radiator valve.

It is better to employ air valves which will retard the return of air into the system for several hours, and which thus permit steam circulation at a temperature below that of the boiling point of water at atmospheric pressures. Figure 6–3 shows the construction of one such air valve. It operates to discharge air and to prevent escape of water and steam just as does the ordinary air valve, but when the steam pressure drops and the air tends to return to the system the check valve at the vent port closes to prevent such counterflow. As the steam in the system condenses, a vacuum will be formed if no air re-enters the system. The check valve, having functioned first, is then supplemented by atmospheric air pressure, which, acting through the port at the bottom of the valve, presses on the

vacuum diaphragm, forces the float to rise, and thus positively seals the venting port. In Fig. 6–3 there are air and water bypass passages which separate the air from any carried-over water and permit the water to flow downward around the float to the siphon, while the air rises to the vent.

The advantage of this type of air valve over the ordinary kind is its ability to keep the radiator warm for some time after the fire in the boiler has been allowed to die down. This is possible because the temperature at which water boils becomes lower as the absolute pressure decreases, and steam at 160 F or an even-lower temperature will circulate throughout an extensive piping and radiator system if air can be prevented from entering. The added cost for the vacuum-type valve is justified by the longer intervals between firing in mild weather, and by the better temperature control which can be obtained.

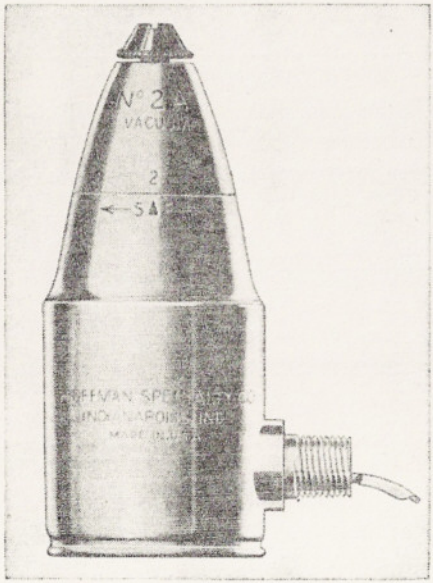

Fig. 6–4. Air-vent valve for radiator, with adjustable port. (Courtesy Hoffman Specialty Manufacturing Corporation.)

A recent improvement in automatic air valves for room-located radiators and convectors adds a manually selective escape orifice to the features of the valve in Fig. 6–3. With this scheme the radiators nearest the source of steam, which would normally be the first ones to become warm, may have the rate of air escape adjusted so that they will be retarded in heating speed as compared with the more remote radiators. Figure 6–4 illustrates such a valve. On this particular valve, at the top, can be seen the adjustable escape-port section which makes possible better balance of a system.

The air inside the steam mains cannot all escape through the air valves on the radiators. Sometimes the radiator valves are closed, shutting off communication between the radiator air valves and the main. Therefore auxiliary air valves are required on the ends of all steam mains where they enter the drop pipes and become wet. The location most likely to trap the

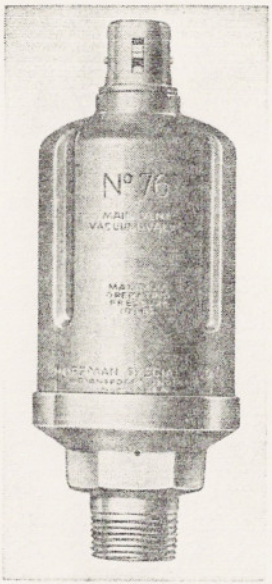

Fig. 6–5. Air-vent valve for steam line. Vacuum type with double air lock. (Courtesy Hoffman Specialty Manufacturing Corporation.)

air accumulation is on the boiler side of the last radiator and close above the water level. However, an air vent placed too close to the water line may be closed by a backsurge of water. The best location for the air vent on a steam main, all things considered, is in the top of the main just above its final drop below the boiler water line (see Fig. 6–8). Figure 6–5 shows a double-air-lock vacuum valve for mounting on a steam main. The mechanism inside this valve resembles that shown in Fig. 6–3. Figure 6–6 shows an eliminator of different design. This particular type is applicable to hot-water systems as well as to steam systems. When water enters this type of eliminator the float rises and thereby closes the outlet, which remains closed until sufficient air collects in the chamber to drop the float, thus venting the air out at the top. If steam instead of cooled condensate enters the eliminator, the thermostatic element, which is filled with a volatile liquid, expands and forces the outlet shut, preventing escape of steam. A check valve prevents air from entering if the system is under vacuum.

6-5. TYPES OF STEAM SYSTEMS. DEFINITIONS

Numerous classifications of steam systems have been made and are in use. Many of these classifications and names, however, are merely subclassifications of the basic types, which are summarized as follows:

One-pipe, air-vent gravity return
Two-pipe, air-vent gravity return
Vapor system, simple gravity return
Vapor system, with return trap
Mechanical return system, nonvacuum
Vacuum return system, with vacuum pump

Most of the commonly named modifications to the system classifications indicated are concerned with indicating the direction of steam and condensate flow, and for these, indicative names—such as "upfeed mains," "downfeed systems," and "overhead mains"—are in common use.

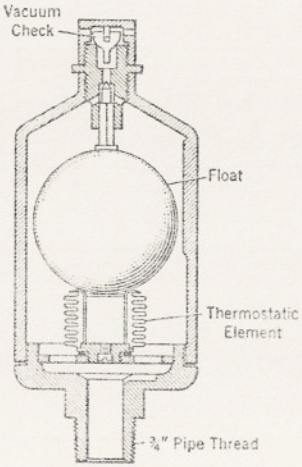

FIG. 6-6. Air eliminator. (Courtesy Sarco Company, Inc.)

Figure 6-7 illustrates a one-pipe air-vent steam system. A study of the illustration will reveal the one-pipe aspect: The supply main which starts at the boiler continues as a single pipe until it finally drops down at the end of its run to constitute the vertical drop leg and, finally, the wet return, which carries the condensate back to the boiler. Steam from the supply main passes through connecting piping to the radiators. Each of the radiators has a single angle-valve screwed into a low position on the radiator. The condensate which forms in the radiator moves counterflow to the direction of the steam entering the radiator, and after re-entering the steam main it flows in the same direction as the steam until it enters the drop leg leading down to the wet return. Systems of this type are

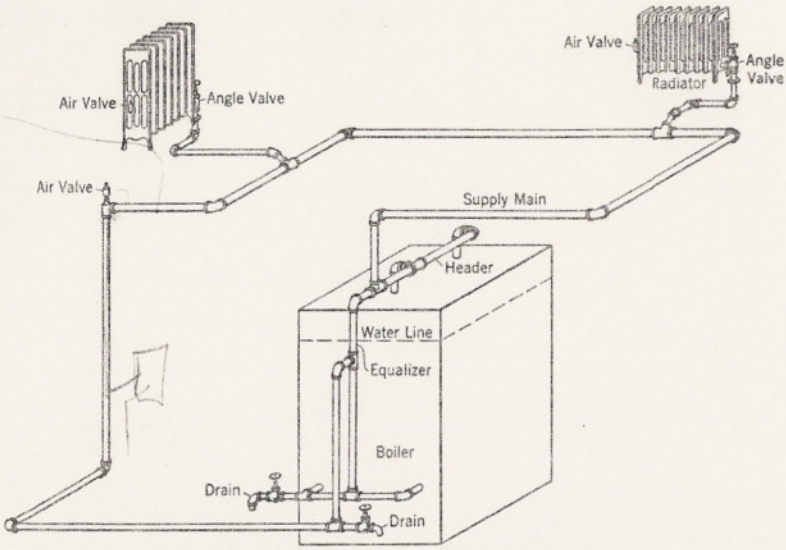

FIG. 6-7. One-pipe air-vent steam system.

noisy, since water and steam can surge back and forth in the radiator. It is therefore necessary to make the connecting pipes from the radiator to the main sufficiently large to serve the double purpose of steam supply and condensate return. Notice that an air vent is required on each radiator. It is also advisable to supply one on the return main in case a closure of valves on some of the radiators might make it impossible to vent the steam main and thereby prevent satisfactory operation of the system. In this and in other systems as well, steam supply mains and return pipes often run along a basement ceiling. Where possible, mains should be pitched in such a direction as to permit condensate to flow along with the steam, and where this is done the high point of the main, exclusive of risers to other floors, is normally at the boiler. A pitch of 1 in. in 20 ft is desirable, and the lowest radiation should be at least 24 in. above the boiler water line.

An *overhead main* runs horizontally, or nearly so, at an elevation higher than the radiators it serves, and is supplied by a vertical main riser. Such overhead mains are frequently placed in the attics of multistoried buildings.

Supply risers are vertical pipes that pass from story to story to convey steam to the radiators or convectors on several floors. Risers are known as either upfeed or downfeed risers, the latter being those in which the steam flows downward to the radiators or convectors from an overhead main.

Return risers are those vertical pipes that take the condensate from the radiators or coils on the several stories of a building and convey it to the return main. Return risers are always of the downfeed type.

A *return main* is a nearly horizontal line of pipe that receives the condensate from the heating system and returns it to the boiler or otherwise disposes of it. The return main

is usually run in the basement, but in any event it must be below all heat-transmitting surfaces, and must drain the supply mains.

A *dry-return main* is one that is run above the water line of the boiler. In some types of steam heating, a dry return conveys both water and steam, while in others which have traps at all points communicating with the steam main, a dry return carries only water and air.

A *wet-return main* is one that is run below the water line of the boiler or equivalent device and is filled with water at all times. As a rule, a wet-return main is preferable to a dry-return main, except where the main is subject to freezing temperatures. Supply mains must not be connected with other supply mains except through water seals or non-return traps. This type of connection is necessary because the unequal frictional resistance to steam flow, and the rate of heat dissipation in each main, produce different pressures at the respective return ends of the mains. If these ends are open through dry returns to each other, turbulence and noise will result. When the ends of supply mains in any one system are sealed separately by dropping into a wet return, the water-column level in each vertical pipe will be sufficiently different from the others to balance the pressure difference.

A *drip pipe*, or *relief*, or *bleeder*, is a pipe used to drain condensate away from the foot of supply risers or from low points, pockets, or traps in the steam main. Drip pipes usually drain into wet returns. Drip pipes in steam mains are employed at points where reduction or increase in the size of the main occurs, and where eccentric reducing fittings cannot be used. This drainage serves to prevent water hammer, by relieving the main of the water which would otherwise accumulate at such locations. The extreme end of the steam main must always be connected so as to drain the condensate into the wet-return main.

6–6. TWO-PIPE AIR-VENT SYSTEM

The two-pipe system differs from the previously described one-pipe system in that separate circuits are provided for the supply and return parts of the system. Figure 6–8 illustrates a representative two-pipe system. It will be noticed that the supply main from the boiler slopes away from the boiler and eventually connects into the return line. This connection can be made by means of a vertical downdrop water leg or, as shown, through a trap. Traps are described in detail later in this chapter, but at this point it should be mentioned that the purpose of a steam trap is to prevent the passage of steam from one part of a system to another while permitting condensate and air to pass. Thus, with the return line at lower pressure than the supply line, the trap prevents steam from passing into the return line but enables the water (condensate) and air to be removed from the supply line.

The supply main can serve risers which supply radiators on different floors, or direct radiator connections can be made into the main. With a two-pipe system, inlet to the radiator does not have to be made at a low point. However, for condensate drainage, a low point on the heater device is connected through a valve into the return piping of the system. Valves are required at both the inlet and the outlet to the radiator, and the return lines—where they are not in the form of risers—should slope toward the

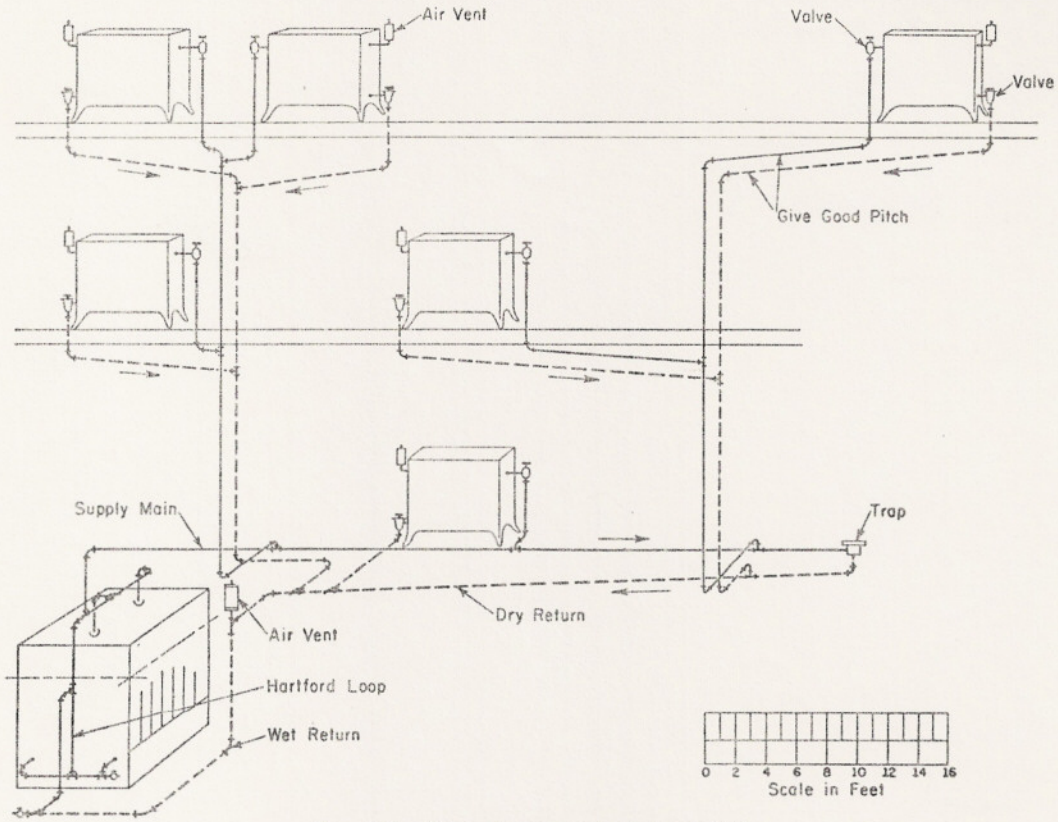

FIG. 6-8. Two-pipe air-vent steam system.

boiler. In the diagram, the return is indicated as a dry return; that is, it is located at a position above the water level in the boiler. It becomes a wet return when it enters the vertical drop leg from which the condensate flows into the boiler. An air vent is supplied on each radiator and, in starting, the steam from the supply main enters the radiator and expels through the air vent the air which was previously trapped in the radiator. The return line is also supplied with an air vent and this serves to vent, from both the supply main and the return main, air which is not otherwise vented at the radiators.

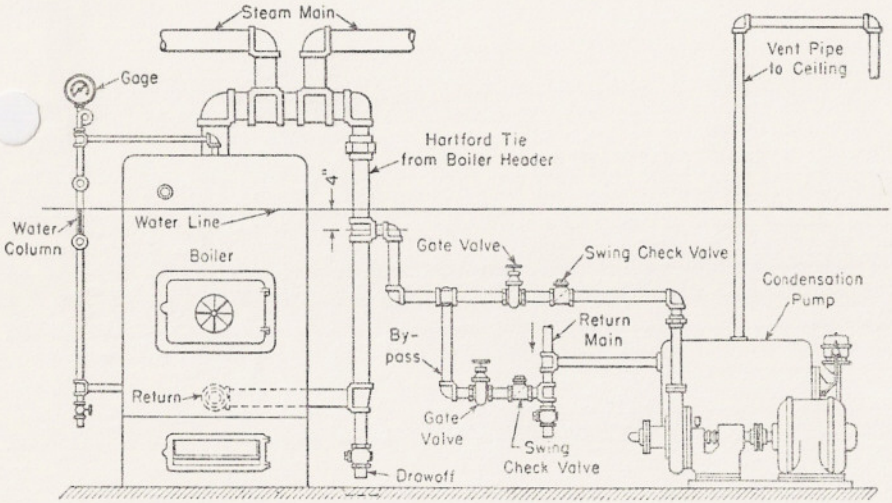

FIG. 6–9. Connections at boiler for a two-pipe air-vent steam system employing a condensate-return pump.

Where a water leg is employed in the operation of a system of this type, it is usually desirable to have the lowest radiation not less than 24 in. above the water line in the boiler. In some cases this is not possible, and where water might flood low-lying radiation a condensate return pump is used to force the water back into the boiler. Return pumps are ordinarily centrifugal pumps and as such cannot pump air. The air is vented to the outside through the air vents on the radiators and on the return line. Large systems frequently make use of the condensate return-pump arrangement (Fig. 6–9). Although a two-pipe air-vent system may drop slightly into the vacuum region, it is not particularly designed to operate in this range and should normally be thought of as a positive pressure system using either gravity or mechanical return.

6–7. VAPOR SYSTEMS

The vapor system differs from the two-pipe air-vent system in that the air elimination is accomplished usually at a central point or points, and

thus no air-vent valves are used on the radiators or convectors. Because it is possible for a system of this type to operate well in the vacuum region, it is customary to use packless valves, which have no opening to outside air around their stems. Traps are employed on each radiator or convector. As stated before, traps permit the passage of condensate and air and prevent the passage of steam. Thus, under ideal operating conditions, a supply of steam enters the radiator and as fast as this condenses it passes out through the trap to the return side. Thus the steam itself is not permitted to short-circuit through the radiator into the return side before it has given up its latent heat in condensing.

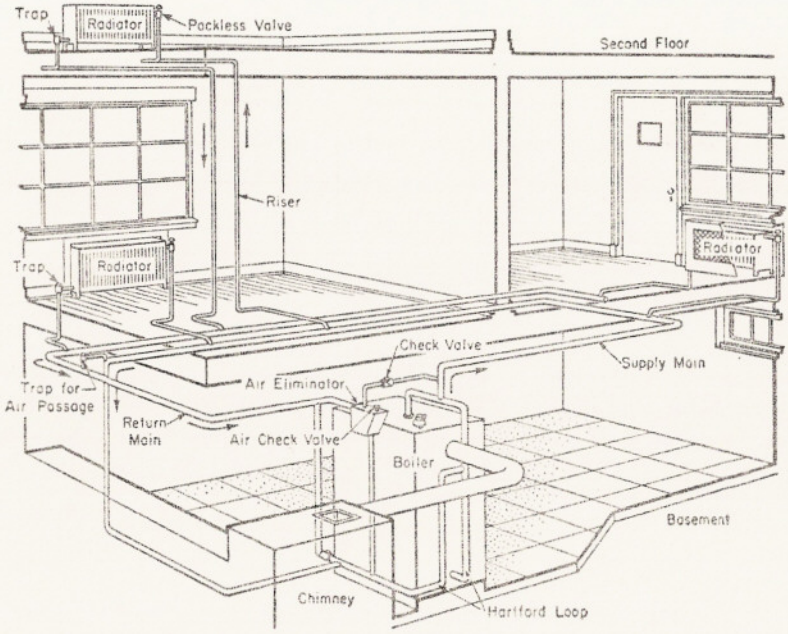

FIG. 6-10. Diagrammatic layout of vapor (steam) system.

Figure 6-10 shows in diagrammatic fashion a layout of such a system. As before, it is desirable for the supply main to drain away from the boiler into the return, and for the return main to connect into the boiler feed line by means of a vertical drop leg. The extreme end of the supply main also connects through a vertical drop leg into the wet return, which feeds the boiler. Water will, of course, stand at different heights in the two vertical drop legs, depending upon the pressure difference in the two parts of the system. It is desirable in a system of this type for the lowest connected radiation to be at least 24 in. above the water level in the boiler, and where this cannot be accomplished other provisions for returning the

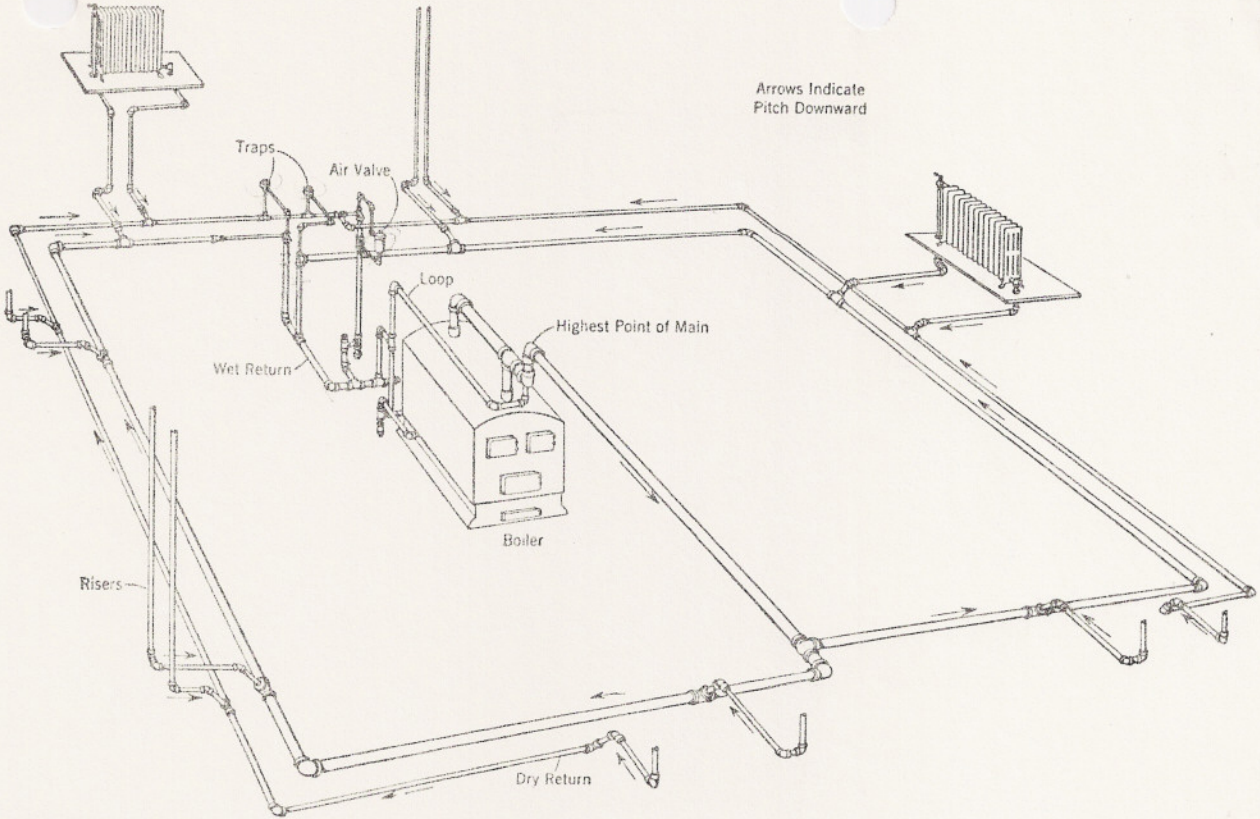

FIG. 6–11. Sketch of vapor heating system.

water to the boiler must be considered. The air in the supply main which does not enter the radiators and leave them through the outlet traps can pass through a connecting trap into the return main, as shown in the illustration. Normal air found in the radiators also enters the return main and moves through the dry-return part of this main to the central air eliminator of the system. Here, after some cooling, the air is eliminated to the outside, and the eliminator is so designed that when the system is under partial vacuum, outside air cannot enter through it. Because of the tightness with which vapor systems can be built, it is possible for them to operate well into the subatmospheric region. With air excluded, as long

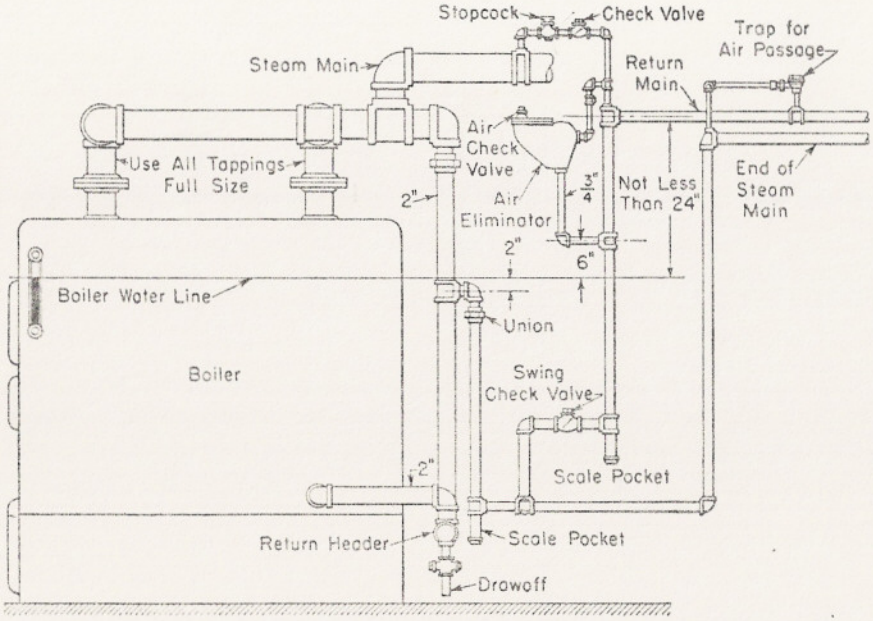

FIG. 6-12. Vapor system showing connections to a boiler with multiple outlets.

as the water-steam in the boiler is hotter than the water-steam medium in the radiators, steam can continue to flow to the radiators and maintain heating at a reduced rate even though no fuel is being burnt. It is easy to control a system of this type, because the boiler pressure can be operated over a range of positive to subatmospheric pressures and because it is possible to use a graduated type of valve at the radiator.

Figure 6-11 shows a more extensive vapor system. It can be seen that there are two complete steam-main circuits connecting to the main from the boiler, and also two return circuits. The air from the supply mains is vented through traps into the return, and the air from all sources is eliminated through the single air valve of the system. By using the

water legs at the end of the steam mains and also at the end of the return, it is possible to offset frictional-loss differences in the various runs of the system by different elevations of the water in the vertical drop legs.

Figure 6–12 shows, in more detail, representative connections such as might be employed at the boiler for a system of the type shown in Fig. 6–11. On larger-size boilers it frequently happens that more than one outlet tapping is made available, and where this is the case all of them should be used.

Vapor systems are usually satisfactory in operation because quiet circulation of steam without water hammer or air binding can be realized, and because control through variation in steam temperature is also possible. However, systems of this type usually are designed for effective operation only at low steam pressures, and consequently relatively large pipe sizes are employed. Since the condensate returns to the boiler by gravity many systems have headroom limitations. It is always desirable to have 24 in. or more elevation between the boiler water line and the end of the return main. Where this is not possible, it is necessary to employ the return-trap arrangement described in the following section.

6–8. VAPOR SYSTEM WITH RETURN TRAP

By making use of a return-trap system, it is possible to operate the vapor system at somewhat higher steam pressures, and smaller pipe sizes are possible. However, even with a return-trap system, there are elevation limitations which must be observed in locating the return trap with respect to the return main and boiler water level. In general, slightly better control can be obtained with a return trap system than with conventional water legs.

Figure 6–13 shows the essential connections at the boiler when a return trap is used. Certain critical dimensions are indicated on the drawing and should be observed when laying out systems of this type. Return traps, which are also known as alternating receivers, have, in general, a float-actuated mechanism which in one position closes a communicating port to the boiler above the water line and opens a corresponding port communicating with the dry return. In its other position, the float mechanism reverses the port openings. With the receiver open to the return system, water drains by gravity into the receiver, and as this fills the float rises. When sufficient condensate accumulates in the receiver, the float, at its high position, trips a mechanism which closes off the connection to the dry-return main and at the same time opens the return trap to the boiler. With the water in the receiver above the boiler water level, and with pressure equalized in the receiver and boiler, the water drains by gravity into the boiler. The float follows the dropping water level in the receiver until it retrips the valve mechanism to connect with the return side, and a refilling

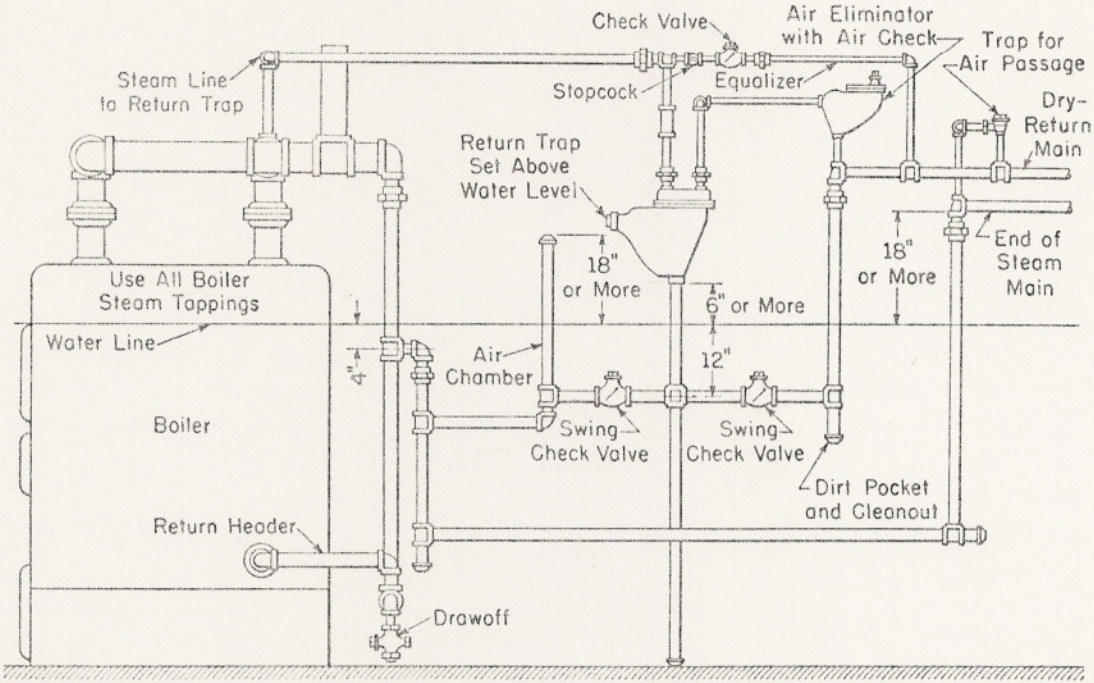

FIG. 6-13. Representative return-trap-system connections at boiler.

of the receiver occurs. The alternating receiver (return trap) merely makes it possible to use the vapor system more advantageously under adverse or limiting conditions; however, most new designs avoid the use of return traps.

Figure 6–14 shows one design of an alternating receiver. As water enters through the bottom inlet the float rises and on reaching its top posi-

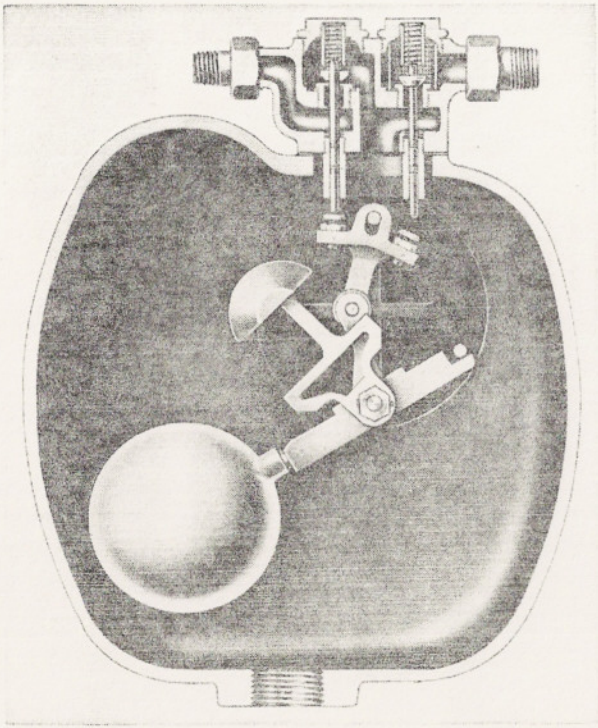

FIG. 6–14. Alternating receiver or return trap for a boiler. (Courtesy Sarco Company, Inc.)

tion it moves the weight past its center equilibrium point. When this occurs the weight quickly swings far to the right and in so doing opens the equalizing-line valve while at the same time the vent-line valve is closed. With the weight in this position, water flows from the bottom outlet of the receiver into the boiler until the float reaches its low position, at which point the weight shifts to the left and thereby reverses the vent connections to start the filling cycle.

Alternating receivers are available in a variety of forms and are designed for low-pressure operation to 15 psig, medium pressure to 45 psig, and high pressure to 100 psig. The higher-pressure units permit use of high lifts, as

for example when a return, high above the heating units, is required to lead the water to a conveniently placed receiver feeding into the return trap. The receiver may need to be vented to the atmosphere to provide a low-pressure region for the lift.

6-9. VACUUM SYSTEM EMPLOYING VACUUM-PUMP RETURN

A vacuum steam-heating system differs from a vapor system in that a vacuum pump is used on the return side to maintain continuously a reduced pressure (usually subatmospheric) on the return side of the system. The vacuum-return pump removes the condensate and air from the return side, delivering the condensate to the boiler and the air to waste. With effective removal of the air, rapid circulation of steam is possible, smaller pipe sizes can be employed, and low-pressure steam or steam at partial vacuum can be fed to the radiators.

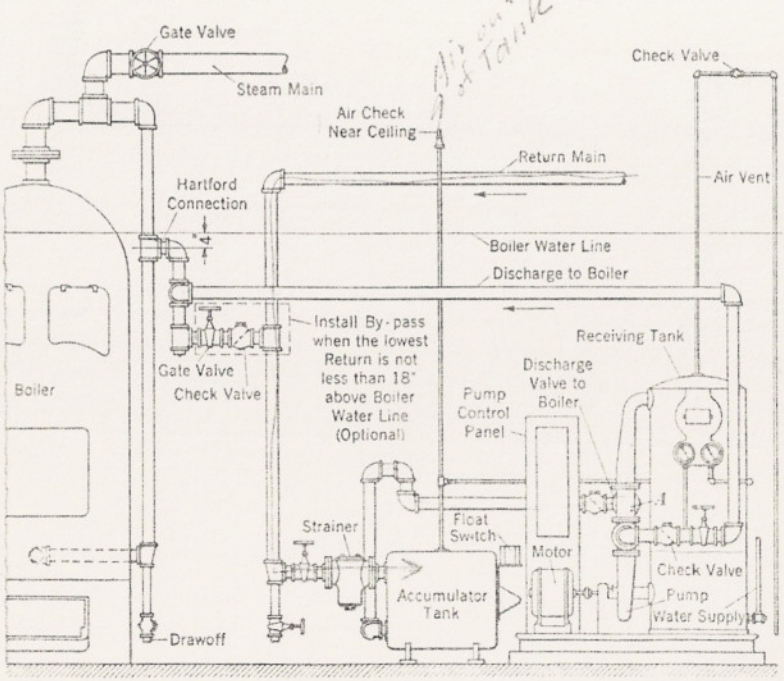

FIG. 6-15. Vacuum-system return connections and pump. (Courtesy C. A. Dunham Company.)

From a control viewpoint, the vacuum system is most versatile; however, the initial investment in the vacuum pump and the subsequent maintenance costs limit the use of this system to buildings larger than conventional residences. Vacuum systems are used in the majority of large steam-heated buildings, where the small pipe sizes are particularly desir-

able and where rapid well-balanced steam circulation is essential. Vacuum-system radiators and convectors have design and installation features similar to those found in vapor systems. However, the connections at the boiler return are so decidedly different that these are separately shown in Fig. 6–15, which illustrates the return arrangements of one type of vacuum system. Condensate and air enter the accumulator tank, which is placed at a low point to which the system can drain. If this location is below the pump suction level, lift fittings make it possible for the pump to lift water from the accumulator tank through as much as 4 ft per lift fitting. A float-

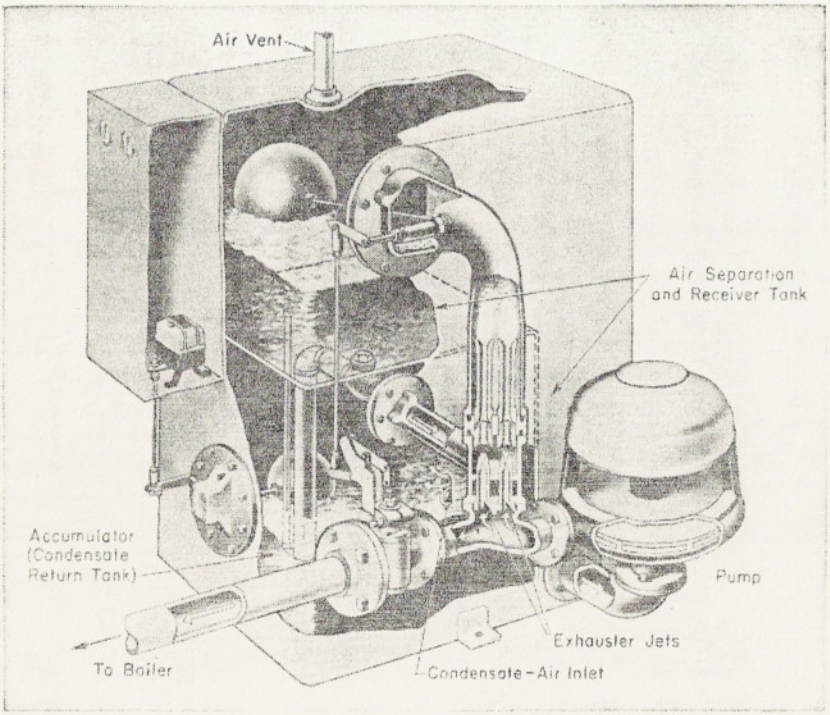

FIG. 6–16. Vacuum pump and tanks for a vacuum steam system. (Courtesy C. A. Dunham Company.)

operated switch on the accumulator tank starts and stops the pump motor. Since centrifugal pumps cannot effectively pump air, the pump itself is not connected to the accumulator but receives suction water from the receiving tank. The pump delivers this water at high velocity through the jets at A, where the kinetic effect of the water jets in a combining tube aspirates (pulls) water and air from the accumulator. In a diffuser tube the kinetic energy of the jets is sufficient to compress the air-water mixture to the receiving-tank pressure, which is slightly above atmospheric. The

mixture in the receiving tank is separated, the air passing to the outside and the water remaining to recirculate through the pump. Only a portion of the water discharge from the pump is required by the aspirating jets, and the remainder is sent directly into the boiler as feed water. An automatic control maintains the proper level of water in the receiving tank by closing down on the discharge valve to the boiler feed line until the level is re-established by additional water entering from the jet circuit.

Figure 6–16 shows details of a vacuum-pump-return system resembling the one just described—built as an integral unit, however, with the accumulator tank constituting the bottom left compartment and the air-separating tank the top compartment. From the top separating tank, water passes down, through a passage at the right rear of the unit, to the pump inlet. By this arrangement, water flows into the horizontal impeller of the pump, always under a positive pressure. The water delivered by the pump passes both to the boiler-feed system and to the aspirating (exhauster) jets. These jets are at sufficiently reduced pressure to pull water and air from the accumulator tank and then send it to the air-separating tank. The pump is started in response to a float-actuated switch in the condensate accumulator tank. The water flow to the boiler is controlled by a float switch in the upper tank, which restricts boiler flow until an adequate supply of water is available in the recirculation system to assure satisfactory operation.

Figure 6–17 shows the vacuum-pump separation unit of another manufacturer. This unit also employs the jet-aspiration principle for handling the air.

In addition to the jet-aspirated types of vacuum pumps just described, mention should be made of the reciprocating vacuum pump, which resembles an air compressor in its general features, and of the rotor-type pump, which uses an elliptic housing. The rotor-type pump employs part of the condensate, which whirls rapidly inside an elliptic housing to compress and deliver the air. This pump may be driven by motor or by a low-differential-pressure steam turbine. Other designs employ separate air and water pumps on a common motor shaft. Most vacuum pumps can return the condensate directly to the boiler, provided the boiler pressure does not exceed the reasonable limits of a single-stage centrifugal pump. Usually if the boiler pressure is higher than 50 psig a separate boiler-feed pump is necessary, and in any event the pump must be designed especially for the maximum boiler pressure to be encountered.

The suction pressure for a vacuum steam-heating system may be as much as 24 in. of mercury—equivalent to about a 27-ft column of water—although a vacuum as high as this presents maintenance difficulties. High vacuums produce low steam temperatures and provide greater flexibility of the system for meeting the fluctuations of outside temperatures. Some

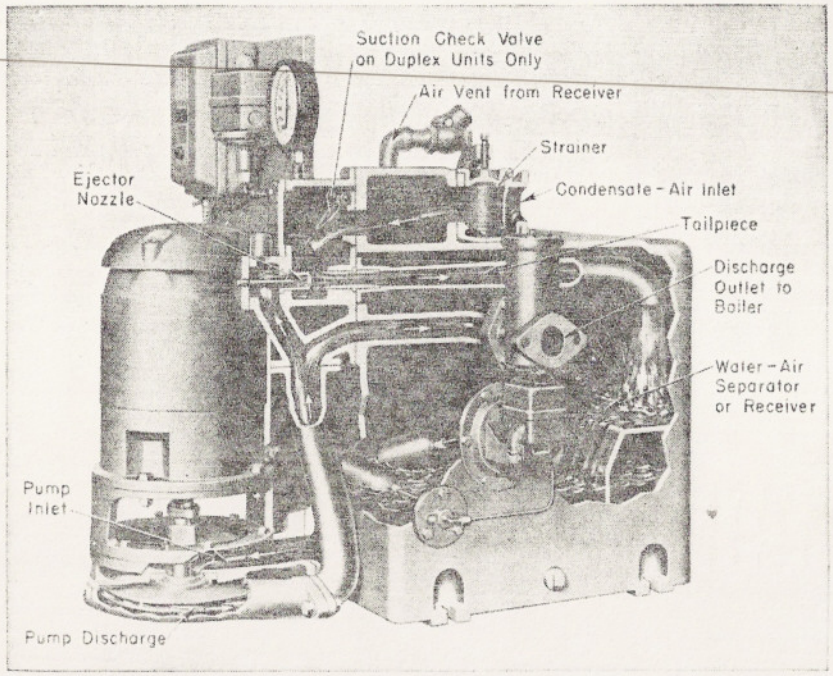

FIG. 6–17. Vacuum pump for vacuum steam system, with vertically mounted motor. (Courtesy Hoffman Specialty Manufacturing Corporation.)

heating systems operate at subatmospheric pressure even on the supply side, and such systems have often been found to be exceptionally economical. This economy is possible because little overheating results, since in mild weather it is possible to set the temperatures of the radiators and convectors in relation to the variations in temperature out of doors.

It should be mentioned that vacuum pumps are often made as double units, in a so-called duplex pattern. A duplex unit is able to handle the overloads that exist when a system is started, because then both units can run; or in the event that one pump fails, the other can continue in use and keep the system in operation. The capacity of the usual vacuum pump varies from some 2500 EDR to 100,000 EDR (600,000 to 24,000,000 Btuh). Representative data on the units of one manufacturer show that a 2500-EDR unit would handle 3.8 gpm and 1.3 cfm of air, while using a $\frac{3}{4}$-hp motor with the discharge pressure 20 psig at pump outlet. It is customary to rate pump units to maintain $5\frac{1}{2}$ in. vacuum with 160 F water. However, the usual units can operate satisfactorily at vacuums as high as 10 in. At reduced capacities or with special pumps, vacuums as high as 25 in. Hg are possible. Because pressure can be accurately controlled in the return system, and because a significant differential between inlet and outlet

pressure can be maintained, it is possible to control the steam and heat input to radiators by means of variable-opening inlet valves. These restrict the flow of steam into the unit, and since steam cannot condense faster than it is supplied, the capacity of the unit is thereby limited. Some systems provide thermostatic control valves on individual radiators. In response to room temperature the control elements limit the opening of the inlet valve and thereby control the amount of steam entering a given radiator.

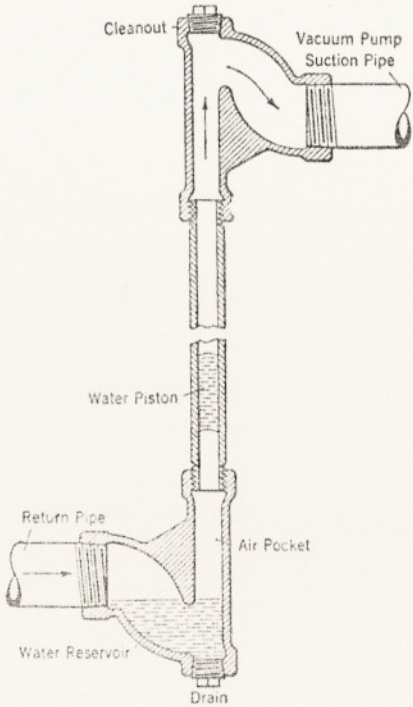

FIG. 6-18. Lift fittings for a vacuum system.

Although it is possible to install heat-transmitting elements below the level of the vacuum pump, this should be avoided whenever possible since such an arrangement involves lifting water which, if sufficiently hot, will partially flash to vapor when the pressure is reduced. Under such conditions pumps do not operate well, and if they fail, partial flooding of low-lying parts of the system can occur. This may stop steam circulation and can contribute to noise production. Difficulties in this connection can be reduced by employing suction lift fittings, one design of which is illustrated in Fig. 6-18. The particular device shown is built from standardized pipe fittings. As can be seen, alternating slugs of liquid or vapor are formed,

and this is equivalent to reducing the effective vertical lift through which the pump operates.

The operating pressure differential in a vacuum system is the difference between the supply pressure and the vacuum on the return side. For example, if the vacuum pump produced 6 in. of mercury vacuum and if steam were supplied at 2 psig, a simple calculation would show that the

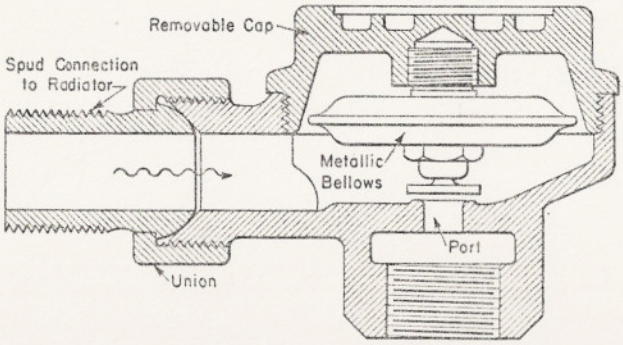

FIG. 6-19. Thermostatic trap of radiator-type design for passing water and air.

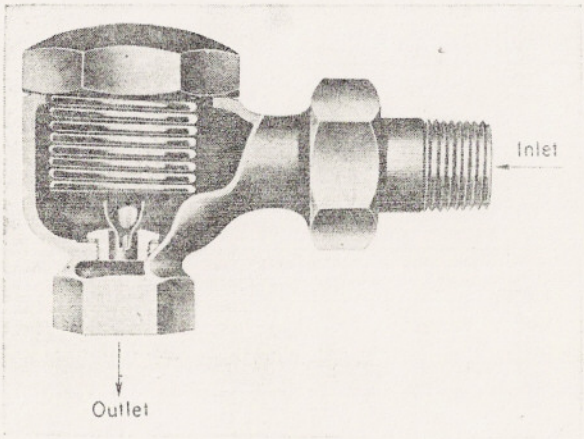

FIG. 6-20. Thermostatic trap of basic radiator design. (Courtesy Sarco Company, Inc.)

operating differential would be approximately 137 in. of water. Because of the significant differential pressure employed in vacuum systems, every connection between the supply main and the return main must be trapped by a thermostatic or mechanical device. One untrapped or leaking connection can defeat the operation of the entire system by passing enough steam to prevent the pump from maintaining a pressure below atmospheric. Vacuum traps are almost exclusively of the thermostatic type, with ports

which open to pass water or air but which close tightly as steam, at its higher temperatures, starts to flow. Thus the return pipes of a vacuum system, when in proper condition, do not transport anything warmer than the condensate, which under vacuum is cooler than atmospheric-pressure steam. Typical thermostatic traps are shown in Figs. 6–19 and 6–20. In these traps the metallic bellows is partly filled with a volatile liquid which expands so as to close the port when vapor warmer than the design temperature enters through the radiator or pipe drain connection. Water at steam temperature can pass through the trap, since it flows along the bottom and does not come into contact with the bellows device to an extent sufficient to make it close the port.

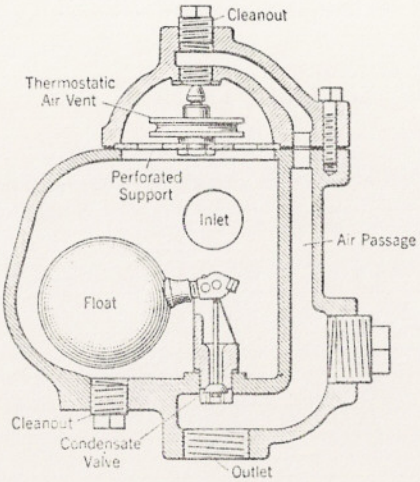

Fig. 6–21. Combination float and thermostatic trap.

Most manufacturers have standardized on the dimensions of the various valves and traps so that the same roughing-in dimensions apply to all. Float traps are used for passing condensate from the supply side to the return side of vacuum heating systems, and some of these can also pass air. Figure 6–21 shows a typical combination float and thermostatic trap. Cool air can pass out through the top, but steam flow is blocked by the expanding thermostatic air vent. When sufficient condensate enters, the float rises and opens the condensate valve. The condensate valve closes after the condensate flows out and as the float drops.

Figure 6–22 shows an inverted-bucket trap. This type of trap is used on steam-heating systems but more particularly for a variety of industrial applications with process steam, such as industrial cookers, water heaters, and chemical-process vats. After a trap discharges, water still remains in the trap and, with steam or air in the inverted bucket, the bucket floats in

the high (closed) position like an empty overturned can in a pond. As water, steam, and air enter the trap, the water level slowly rises inside the inverted bucket. Air and uncondensed steam pass to the top of the trap body through the vent-hole in the bucket. When insufficient air and steam remain in the bucket to hold it up, the bucket falls and the escape port opens. The air in the top of the trap, and the water outside and inside the bucket, are blown out by the pressure difference between the supply and return systems until the air and vapor in the bucket again exert their buoyancy to close off the escape port. This design of trap is satisfactory for the operation of high-pressure as well as low-pressure systems, with pressure differentials of 5 to 200 psia. The units with $\frac{1}{2}$-in. and $\frac{3}{4}$-in. pipe sizes have capacities ranging from 900 to 1950 lb of water per hour, and the units with $1\frac{1}{4}$-in. and $1\frac{1}{2}$-in. pipe sizes have capacities ranging from 4350 to 11,000 lb per hour.

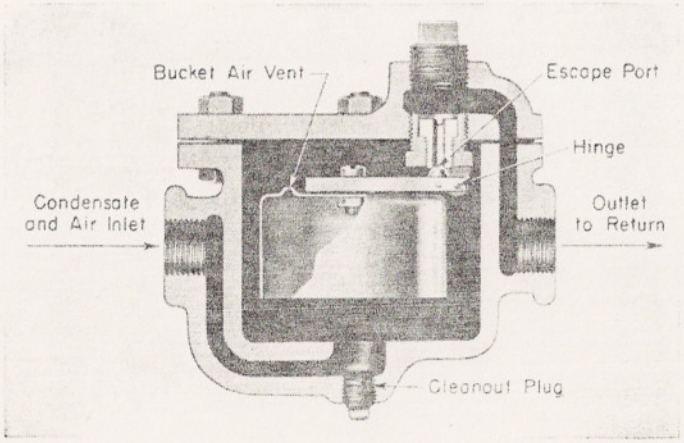

FIG. 6–22. Inverted-bucket trap. (Courtesy Hoffman Specialty Manufacturing Corporation.)

It is desirable to cross-connect the steam and return mains at a convenient point, normally near the boiler. The connection must, however, be made with a check valve employed, and for positive closure it is also desirable to install a stopcock. The equalizer is required to prevent a reverse circulation, which could take place if the pressure in the steam-main side becomes lower than in the return side. This condition may happen when boiler firing is not taking place and an induced vacuum develops on the positive side of the system. Unless pressure is partly equalized from return main to supply main, water may remain in the radiators and return line and thus prevent satisfactory operation. The check valve should be so placed that flow can occur only from the return main to the steam main, and usually check valves of the swing-disk type are desirable.

6-10. AIR-LINE-PUMP SYSTEMS

By use of an air-line pump, it is possible to convert a one-pipe steam heating system, such as that shown in Fig. 6-7, into a modified vacuum system. The most common use of this system is for bettering existing one-pipe, air-vent, gravity-return systems. The change is accomplished by replacing the air valve on each radiator by a thermostatic-type, vacuum-style air valve and connecting the outlets of these valves into a common air-line header which leads to the air pump. The air valve on the steam main is also connected into the air-line header through a thermostatic trap. By removing the air from the system under controlled conditions at a central location, the nuisance of noisy and often leaky air valves on each radiator is eliminated. Moreover, if the system piping and radiator valves are tight, it is possible for the system to drop into the vacuum region, with the corresponding benefit of better temperature control in mild weather. In construction and operation, air-line vacuum pumps are often similar to the vacuum-return pumps described in section 6-9. Any surplus water from the air-separating tank of the pump is not returned to the boiler but is wasted to the sewer.

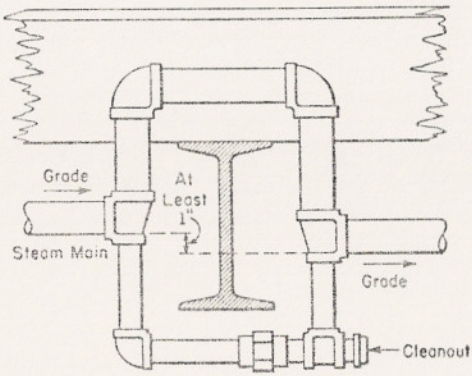

FIG. 6-23. Piping arrangement for steam (or return) main at an obstruction. Lower loop is full size for return main; higher loop is full size for steam main, as shown here.

6-11. STEAM-PIPE ARRANGEMENTS

When a steam main reaches an obstruction through which it cannot pass, it is necessary to install an upper and a lower loop to provide for passage of both vapor and liquid. A method of doing this is shown in Fig. 6-23. Where a double loop is not feasible, it is possible to install a drip connection and to drain water from the main to a return at lower pressure. This will remove the water from the pocket and prevent unsatisfactory flow from water surging back and forth, which otherwise causes noisy and unsatisfactory operation. In the case of a return main reaching

an obstruction, a double loop is also required. However, here the upper air loop can be made of appreciably smaller pipe than the underloop. As an example of comparative sizes at an obstruction, a 2-in. main for steam would need a $1\frac{1}{4}$-in. underloop while a 2-in. return main would need a 1-in. upper air loop.

Whenever the supply main of a two-pipe vapor system requires a fresh start in order to maintain the necessary drainage pitch, or when some obstruction such as a beam is encountered, the main may be raised, provided a drip pipe with a proper trap connecting into the return main is installed. Such an arrangement, called a jump-up, is illustrated in Fig. 6–24. The supply main pitches downward in the direction shown by the

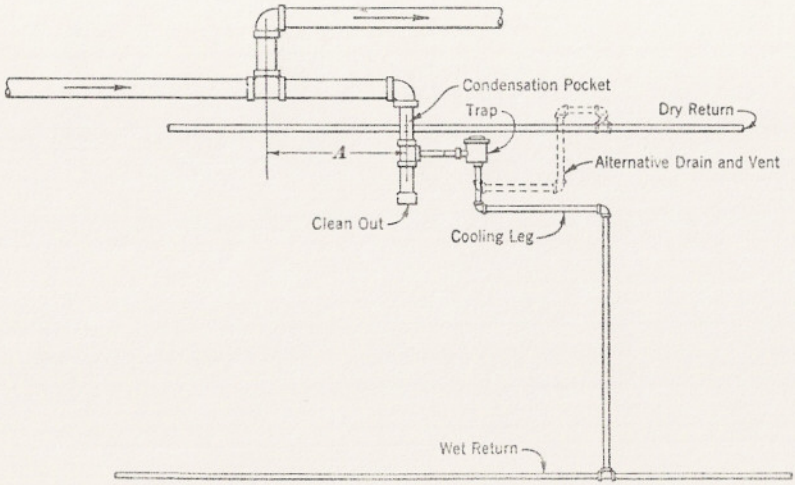

FIG. 6–24. Jump-up in steam main.

arrows. At the bottom of the condensation pocket there is a sediment chamber with a clean-out cap. The trap of Fig. 6–24 has a cooling leg so that the hot condensation may lose some heat before coming in contact with the comparatively cool water in the wet return. When the return main is relatively high above the water line of the boiler and but little lower than the supply main, the trap outlet connection should be made as shown by the dotted line, with an elbow directed down into the top of the return main.

When water whose temperature is higher than the temperature corresponding to the pressure is released into a pipe or other container at a reduced pressure, part of the water will flash into steam. This action occurs at the suction inlet of a pump which is lifting hot water, and at the outlet of a high-pressure steam trap. If the steam for a vacuum system comes from a high-pressure source it is necessary to provide for conservation and disposal of the high-temperature condensate from high-pressure traps.

The trap outlet may be connected to a flash tank, with an interior capacity at least four times that of the trap body. From the top of the flash tank there should be a valved steam connection to the low-pressure steam main, and from the bottom of the tank a thermostatic trap connection into the vacuum return. A typical piping arrangement of this type is shown in Fig. 6–25.

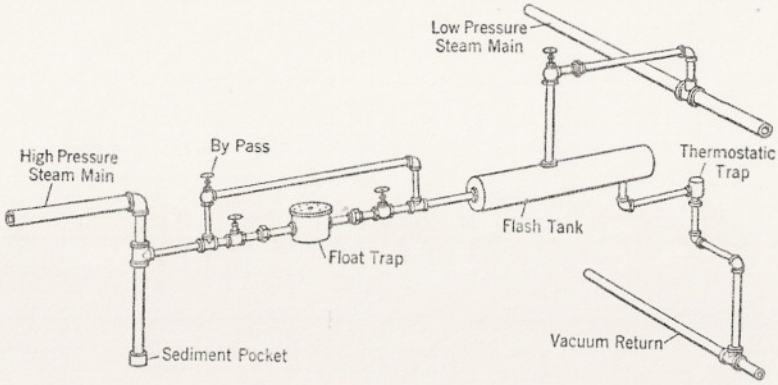

FIG. 6–25. Flash tank for disposal of high-pressure water and steam.

All vapor or temperature-modulating systems of steam heating operate on substantially the same general principle. Graduated radiator supply valves are employed, in which the area of the steam passage at the port responds to a slight movement of the lever handle (Figs. 6–26 and 6–27).

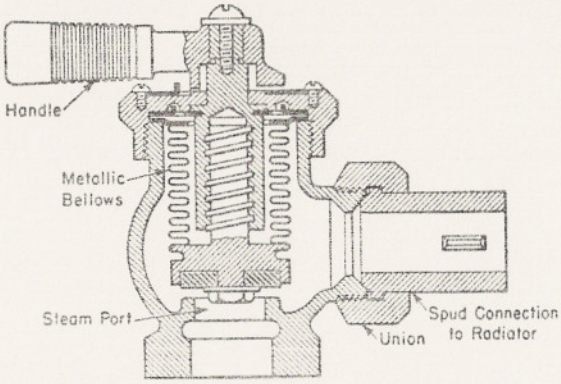

FIG. 6–26. Packless graduated radiator valve.

In such valves a metallic bellows expands and contracts (but does not revolve) as the valve stem rises or descends in response to turning the valve handle. The bellows forms an airtight seal. This tightness is important in any vapor or vacuum system which operates at subatmospheric pressure.

Some valves of this general type have arrangements whereby the area of the valve orifice or of the valve lift may be adjusted to suit the size of each individual radiator. With any given pressure differential the radiator will heat only to the extent shown on the indicator of the valve: $\frac{1}{4}$, $\frac{1}{2}$, $\frac{3}{4}$, etc. This adjustment tends to prevent some radiators from heating more quickly than others, and a vapor heating system, once equalized, remains equalized indefinitely.

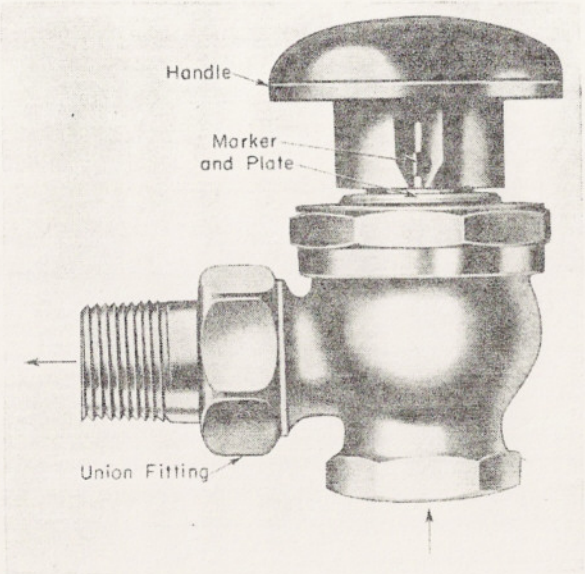

FIG. 6–27. Nonrising-stem packless radiator valve.
(Courtesy C. A. Dunham Company.)

6–12. THE HARTFORD LOOP

Most of the steam-heating-system layouts shown in this chapter use the Hartford loop, which is designed to reduce the possibility of water leaving the boiler by any method except evaporation. In Fig. 6–8 the loop is indicated by the arrow and legend. An equalizing pipe from the steam section of the boiler leads to that point at which the feed-water return to the boiler is connected. This point is located 2 to 4 in. below the normal boiler-water level, and the downflow pipe, which is normally flooded, carries the feed water into the boiler. Under certain conditions of abnormal operation, it is possible to have a pressure much lower than boiler pressure exist in the return system and at the extreme end of the steam supply main. With such reduced pressure it would be possible for water to leave the boiler and back up in the piping, and in some instances even reach the heat-transfer equipment. Such a loss of water from the boiler might endanger the combustion boiler-heating surface. The Hartford

loop can prevent such water loss, because when the water level drops below the return connection point, then steam from the downcomer equalizing line comes in direct contact with the return water, and the boiler pressure acting on both the return water and the boiler water stops the syphon action so that a further drop in the boiler water level is avoided. A similar water-loss condition might arise if, for example, a sudden breakage occurred in the return line. Again, water would not run out of the boiler by syphon action past the point where steam acts at the return inlet. Thus it is customary to provide the Hartford loop as a safety feature for steam boilers. It has no similar use in connection with hot-water systems.

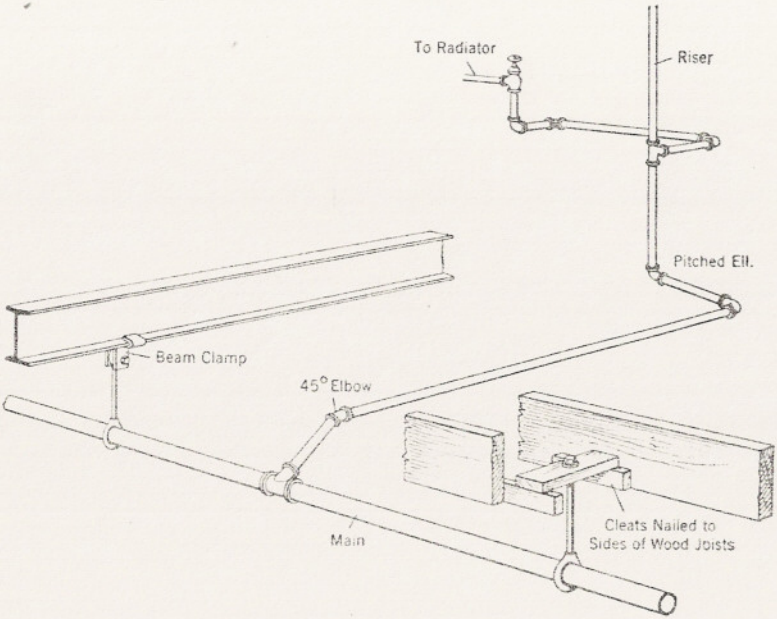

FIG. 6–28. Desirable connections from main to riser and from riser to radiator.

6–13. EXPANSION AND CONTRACTION

In the design and erection of steam-heating plants the problem of expansion cannot be neglected. Care must be taken to have every long steam pipe free for limited movement, and so arranged that small movements of connections or branches are not blocked by building material or by structural beams or columns. A steam main should be supported from overhead with strong hangers, but the branches to the risers must be free to move to take care of expansion. At the bottom of each vertical pipe there should be two elbows. One of these elbows directly on the riser should have more than a true 90-deg angle so as to facilitate drainage. This elbow and a nipple connect with an ordinary elbow placed on its side.

Thus there will always be a nipple or a piece of pipe upon which some swing or spring movement is possible for expansion and contraction of the riser. (See Fig. 6–28.)

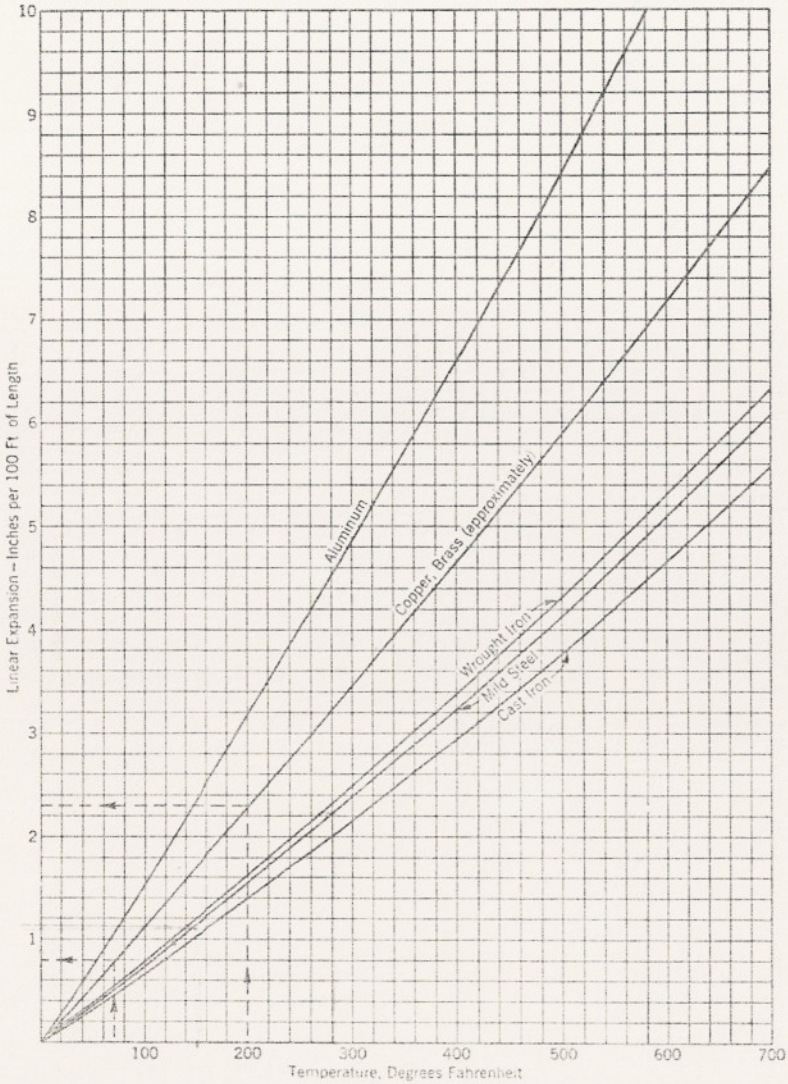

FIG. 6–29. Expansion of pipe, in inches per 100 feet, at various increases in temperature.

Risers, especially, require careful design and installation to provide for expansion, since the necessary drainage pitch of runouts to radiators may easily be lost by the expansion of a tall riser.

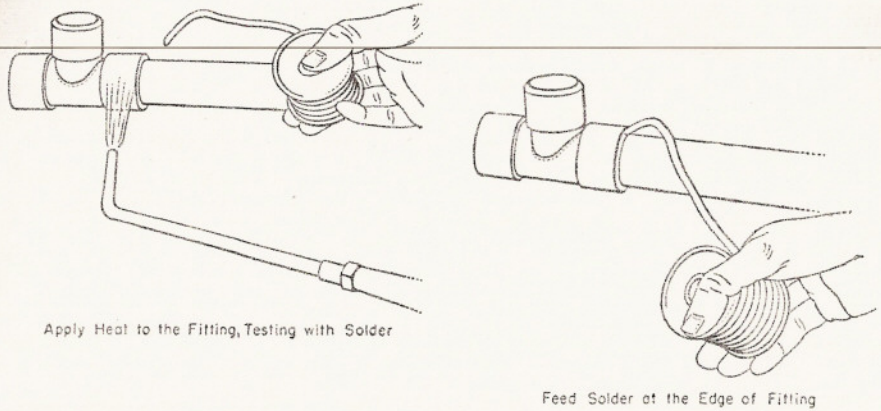

FIG. 7–4. Heating a sweated fitting and adding solder to the hot joint.

7–5), particularly when it will be desirable to disassemble the system or remove valves without the application of heat.

Effective and tight joints can be made with compression-type fittings.

7–3. EQUIVALENT PIPE LENGTH

The resistance to flow of steam or other fluid through a pipe is increased by the presence of valves and fittings, and the carrying capacity is thus

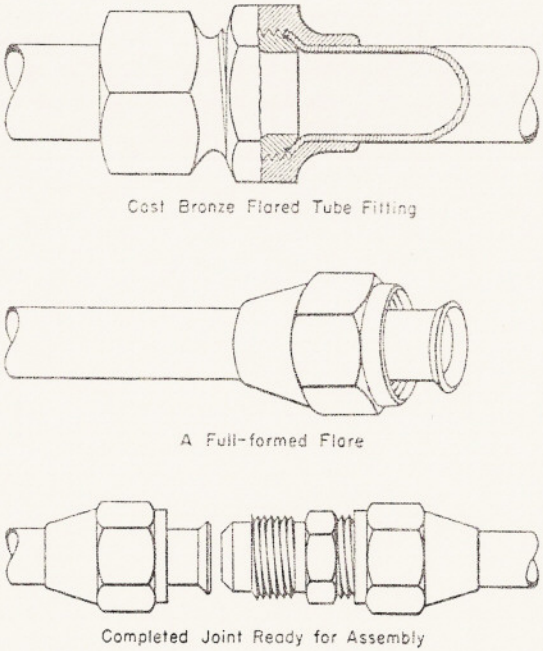

FIG. 7–5. Compression-type (flared) copper fittings.

reduced. Investigators have determined the resistance of valves and fittings in terms of lengths of straight pipe. The resistances thus expressed are added to the measured length of the pipe, and the sum is called the equivalent pipe length. Equivalent pipe length is thus the length of straight pipe of a given size that would have the same resistance as the actual pipe with its bends, fittings, valves, and other devices. In this connection it may be desirable to make use of elbow equivalents, where an

TABLE 7–4

RELATIVE RESISTANCES OF PIPE FITTINGS, BOILER, AND RADIATOR IN TERMS OF ELBOW EQUIVALENTS

(One Elbow Equivalent—Resistance of 90-Deg Elbow)

NAME OF PART	ELBOW EQUIVALENTS	
	Iron Pipe	Copper Tubing
Boiler	3.0	4.0
Radiator	3.0	4.0
Angle radiator valve	2.0	3.0
Open globe valve	12.0	17.0
Open gate valve	0.5	0.7
Tee with side diversion of		
100%	1.8	1.2
50%	4.0	4.0
33%	9.0	11.0
25%	16.0	20.0
90-deg elbow	1.0	1.0
90-deg long-turn elbow	0.5	0.5
45-deg elbow	0.7	0.7
Open return bend	1.0	1.0
Reducing coupling	0.4	0.4

elbow equivalent is equal to a straight run of pipe approximately 25 times the nominal pipe diameter. Thus, for 3-in. pipe, one elbow equivalent is $25 \times 3 = 75$ in., or 6.5 ft of linear run. Elbow equivalents of piping components are given in Table 7–4.

Table 8–2, which applies for smaller pipe sizes, gives the equivalent feet of run of various pipe fittings and can be used where applicable or more convenient.

Example 7–1. A certain 2-in. steam circuit has an actual length of 124 ft and contains three elbows and two globe valves, and one tee through which half of the main flow passes. What is its equivalent length?

Solution: When elbow equivalents are used, select values from Table 7–4 as follows: Elbows, 3; globe valves, two at 12.0; tee, 4.0. Total: 31.0. The total equivalent length is thus

$$31\left(\frac{25 \times 2}{12}\right) + 124 = 253.2 \text{ ft} \qquad \textit{Ans.}$$

Or, by use of Table 8–2, one elbow has the resistance of 4.2 ft of 2-in. pipe; three elbows are equivalent to 12.6 ft of pipe; a globe valve has the resistance of 50 ft of pipe; two globe valves are equivalent to 100 ft of pipe; and a 50 per cent diversion tee is equivalent to 16.7 ft of pipe. Since the actual length of 2-in. pipe is 124 ft, the equivalent length is

$$124 + 12.6 + 100 + 16.7 = 253.3 \text{ ft}$$

As a rough generalization, the equivalent length of a conventional circuit, with the usual take-off points and turns, ranges from 1.5 to 2 times its linear run.

7–4. STEAM PIPING SIZES

Steam flowing through pipe and fittings is retarded by friction loss between the surface of the pipe and the steam, and by turbulence losses in bends and through the restricted passages which are found in valves and other devices. The design of the system should permit uniformly controlled distribution of the steam to various outlets, with minimum noise and with provision for release of air when air is in the system. Elimination and return of condensate presents an additional design problem. Furthermore, it is necessary to balance distribution in a system so that heaters remote from the steam source are not starved in relation to heaters adjacent to the steam source. This can sometimes be done by adjustment of pipe sizes, but it may also be necessary to provide orifice plates for insertion in the inlet of a radiator, to restrict or control the inflow of steam. The system and its radiators must operate satisfactorily not only under conditions of full heating load but also under conditions of partial load. Moreover, in the warming-up period the system is called upon to distribute an excessive amount of steam and to return an equivalent excess of condensate. Thus during this period the system is operating under an overload. It is obvious, therefore, that flexibility must be provided in the system, since the pressure differences causing flow may be radically different between full-load, normal-load, and partial-load conditions.

The laws governing the flow of dry or superheated steam are similar to those which apply to the flow of gases. However, additional complications exist with steam flow, as usually the steam is not 100 per cent dry, and moisture in suspension may be moving with the steam and also on the wetted walls or bottom of a steam pipe. It is also possible for the steam to be moving in one direction while condensate (water) is flowing in the opposite direction—an awkward situation which usually makes it necessary to provide larger pipe sizes to insure satisfactory flow and to reduce the possibility of impact (water-hammer) losses with accompanying noise.

Because of the complexity of the problem, pressure-loss designs based purely on theory and analytical considerations are frequently unsatisfactory. It has been found better to use semiempirical formulations, or compilations of experimental test data, for laying out designs. One of the many equations used for the flow of steam in pipes is the semiempirical Babcock formula, which is

$$W = 87\sqrt{\frac{PdD^5}{\left(1 + \frac{3.6}{D}\right)L}} \tag{7-1}$$

where W = pounds of steam flowing per minute;
P = loss of pressure, in pounds per square inch;
d = weight of one cubic foot of steam (density);
D = inside diameter of pipe, in inches;
L = equivalent length of pipe, in feet.

Because of the complexity of equation 7–1, Table 7–5 has been prepared to simplify its use. The application of the formula with the help of Table 7–5 is shown in several examples that follow.

In connection with steam-heating computations, mention should again be made of the EDR (equivalent direct radiation), which represents a heat delivery of 240 Btuh. A pound of steam in condensing and subcooling delivers approximately 1000 Btu, and consequently 4 EDR are roughly equal to the condensation of 1 lb of steam per hour. Thus

1 lb steam per hour condensing ≡ 4 EDR
1 lb steam per minute = 60 lb steam per hour ≡ 240 EDR
1 EDR ≡ 0.25 lb steam per hour condensing

Example 7–2. What is the capacity of a 3-in. steam supply main 100 ft long with 1.0 psig boiler pressure at inlet, for an allowable pressure drop of $\frac{1}{2}$ oz per sq in.?

Solution:

0.5 oz pressure drop (col. 1) = 1.538
3 in. pipe (col. 2) = 11.183
1 psig pressure (col. 3) = 0.200
100 ft long (col. 4) = 1.00

Therefore the number of pounds of steam per minute is

$$W = 1.538 \times 11.183 \times 0.200 \times 1.00 = 3.44$$

Thus the 3-in. main, under the conditions stated, is capable of supplying $240 \times 3.44 = 825.6$ EDR. *Ans.*

Example 7–3. What will be the capacity of the 3-in. main of example 7–2 if it is 400 ft in equivalent length and if the same pressures apply?

Solution: The values for pressure drop, pipe size, and initial pressure remain the same as in example 7–2, but the factor from Table 7–5, column 4, changes to 0.500, the value for a 400-ft length. Thus

$$W = 1.538 \times 11.183 \times 0.200 \times 0.5 = 1.72 \text{ lb per min}$$

TABLE 7–5

FLOW OF STEAM IN PIPES BY THE BABCOCK FORMULA

Pressure Loss (oz per sq in.)	Col. 1 $87\sqrt{\frac{P}{100}}$	Nominal Pipe Size (in.)	Col. 2 $\sqrt{\frac{D^5}{1+\frac{3.6}{D}}}$	Steam Pressure by Gage (psig)	Col. 3 $\sqrt{d}$	Length of Pipe (ft)	Col. 4 $\sqrt{\frac{100}{L}}$
0.25	1.088	1	0.536	−1	0.187	20	2.240
0.50	1.538	1¼	1.178	−0.5	0.190	40	1.580
1.00	2.175	1½	1.828	0.0	0.193	60	1.290
2	3.076	2	3.710	1	0.200	80	1.120
3	3.767	2½	6.109	2	0.205	100	1.000
4	4.350	3	11.183	3	0.210	120	0.912
5	4.863	3½	16.705	5	0.221	140	0.841
6	5.328	4	23.631	10	0.246	160	0.793
7	5.755	4½	32.134	15	0.269	180	0.741
8	6.152	5	43.719	20	0.289	200	0.710
10	6.878	6	71.762	30	0.325	250	0.632
12	7.534	7	106.278	40	0.357	300	0.578
14	8.138	8	149.382	50	0.387	350	0.538
16	8.700	9	201.833	60	0.414	400	0.500
20	9.727	10	272.592	75	0.451	450	0.477
24	10.655	12	437.503	100	0.506	500	0.447
28	11.509	14	566.693	125	0.556	600	0.407
32	12.304	16	816.872	150	0.602	700	0.378
40	13.756	..	.....	175	0.644	800	0.354
48	15.069	..	.....	200	0.685	900	0.333
80	19.454	..	.....	....	....	1000	0.316
160	27.512	..	.....	....	....	1200	0.289
320	38.908	..	.....	....	....	1500	0.258
480	47.652	..	.....	....	....	2000	0.224

NOTES. The table does not allow for entrained water in steam, condensation in pipe, and roughness in commercial pipe, as found in practice.

Pounds of steam per minute that will flow through a straight pipe for a given condition: W = column 1 × column 2 × column 3 × column 4.

or $$1.72 \times 240 = 412.8 \text{ EDR}$$ *Ans.*

This result illustrates the reduction in capacity caused by the additional length of main, all other conditions remaining the same.

Example 7–4. If the steam pressure of example 7–2 is increased to 10 psig, what will be the capacity of the 3-in. 100-ft pipe for the same pressure drop?

Solution: In Table 7–5 the change from example 7–2 will be found in column 3: 46. Thus

$$W = 1.538 \times 11.183 \times 0.246 \times 1.0 = 4.23 \text{ lb per min}$$

or $$4.23 \times 240 = 1015 \text{ EDR}$$ *Ans.*

Example 7-5. What size of steam main must be used for S22 EDR if the water in the return main is allowed to stand 3.56 in. (2 oz per sq in.) higher than in the boiler, with an initial steam pressure of 1 psig, when the main is 400 ft long?

Solution: S22 EDR is equivalent to S22/240 = 3.425 lb of steam per minute, since 1 lb of steam per minute serves approximately 240 EDR. Thus $W = 3.425$.

From Table 7-5, 2 oz per sq in. = 3.076 (col. 1), 1 psig = 0.2 (col. 2), and 400 ft = 0.5 (col. 3).

From the Babcock equation (eq 7-1) it can be seen that

$$\text{Col. 2} = \frac{W}{\text{col. 1} \times \text{col. 3} \times \text{col. 4}}$$

$$= \frac{3.425}{3.076 \times 0.2 \times 0.5}$$

$$= 11.13$$

Referring to column 2, the factor 11.13 is equivalent to a 3-in. pipe. *Ans.*

Example 7-6. What size of steam main would be required to serve 900 EDR if the main were 272 ft long and had 4 elbows, 1 tee with 50 per cent diversion, 1 angle valve, an initial pressure of 0.5 psig, and a pressure drop of 1 oz per sq in. in 100 ft?

Solution: Assume a pipe size, to obtain the total equivalent length. After computing the result, using this pipe size, the problem should be recalculated if the calculated size is in error more than one-half size. Let a 3-in. pipe be assumed for estimating the total equivalent length. From Table 7-4, the equivalent length of the main is $272 + (4 \times 1 + 4 + 2)[(25 \times 3)/12] = 335$ lin ft, giving a value of 0.550 for column 4. The total drop is 3.35×1 oz per sq in., or 3.35, giving a value of 3.97 for column 1. The initial pressure is 0.50 psig, giving a value of 0.197 for column 3. Thus

$$\text{Col. 2} = \frac{W}{\text{col. 1} \times \text{col. 3} \times \text{col. 4}} = \frac{900/240}{3.97 \times 0.197 \times 0.550} = 8.73 \qquad \textit{Ans.}$$

This factor, from column 2, gives a 3-in. pipe.

Example 7-7. A 5-in. supply main with a measured length of 357 ft has 10 elbows, 1 tee, and 1 globe valve. The load on the main is 8200 EDR. If the initial pressure is 5 psig, what pressure is available at the end of the line?

Solution: Using Table 7-4, the total equivalent length equals $357 + (10 + 1.8 + 12)[(25 \times 5)/12] = 595$ ft, and consequently

$$\text{Col. 1} = \frac{W}{\text{col. 2} \times \text{col. 3} \times \text{col. 4}} = \frac{8200/240}{43.719 \times 0.221 \times 0.407} = 8.67$$

Referring to column 1, the pressure loss through the pipe is very nearly 16 oz per sq in., or 1 psi, and the final pressure would therefore be $5 - 1 = 4$ psig. *Ans.*

Table 7-5 is very readily used in any case where the entire load is considered at the end of the main. When the load is distributed along the main, however, the process of sizing the main is more complicated. For ordinary heating installations, tables have been developed to provide for selection of pipe sizes.

Table 7-6[1] is satisfactory for selecting pipe sizes to use with all low-

[1] Tables 7-6, 7-7, and 7-8 are reprinted, by permission, from *Heating Ventilating Air Conditioning Guide 1956*, Chapter 21.

pressure steam-heating systems. In laying out a design, it is necessary to know the total pressure drop available for distributing the steam between the source and the end of the return, following which the allowable or most suitable pressure drop per 100 equivalent feet of run can be selected. The equivalent length of run of the main and return [and also the equivalent length from the steam source (boiler) to the most distant heating element] must be known, and the direction of flow of condensate—that is, whether it is with or against the direction of steam flow—must also be determined.

Table 7–6 is applicable both to one-pipe and two-pipe systems and can also be used for vapor and vacuum systems. The headings at the top of each column specifically state the conditions of use. When special design usages are required, they will be explained later in the text. For return piping, tables 7–7 and 7–8 are applicable in a similar way. It is desirable to use on the return side the same pressure drop per 100 ft that is used on the supply side of a system.

7–5. PIPE SIZES FOR AIR-VENT SYSTEMS

Air-vent systems of one-pipe design were formerly very common because of low initial cost. In the one-pipe system illustrated in Fig. 6–7, the single pipe from the boiler carries both the supply steam and all the condensate of the system. The combined steam main and return continues until it finally drops and becomes the wet return. A water column automatically collects in the wet return at a sufficiently high level to force the water back into the boiler and counterbalance the pressure difference between the supply and return sides of the system. In a two-pipe air-vent system, as illustrated in Fig. 6–8, a water column is also present to force the return water into the boiler. The height of water column needed to balance the unit pressure can be selected from the following tabulation:

$$\begin{aligned} 1 \text{ oz per sq in.} &= 1.73 \text{ in. of water at } 70 \text{ F} \\ &= 1.78 \text{ in. of water at } 180 \text{ F} \\ &= 1.80 \text{ in. of water at } 200 \text{ F} \\ 1 \text{ psi} &= 28.80 \text{ in. of water at } 200 \text{ F} = 2.4 \text{ ft} \end{aligned}$$

Thus for a steam-system pressure loss of 4 oz per sq in., the water level in the return drop pipe (with 180 F water) would be 4×1.78, or 7.12 in. higher than that in the boiler.

Systems in which the equivalent length of run does not exceed 200 ft should be sized as follows:

1. For the steam main and dripped runouts to risers where the steam and condensate flow in the same direction, use $\frac{1}{16}$-psi drop (col. D).
2. Where the riser runouts are not dripped and the steam and condensation flow in opposite directions, and also in radiator runouts where the same condition occurs, use column L.
3. For upfeed steam risers carrying condensation back from the radiators, use column J.

TABLE 7–6

STEAM-PIPE CAPACITIES FOR LOW-PRESSURE SYSTEMS*

Pipe Size (in.)	Capacities of Steam Mains and Risers								Special Capacities for One-Pipe Systems Only		
	Direction of Condensate Flow in Pipe Line										
	With the Steam, in One-Pipe and Two-Pipe Systems						Against the Steam, Two-Pipe Only		Supply Risers, Upfeed	Radiator Valves and Vertical Connections	Radiator and Riser Runouts
	1/32-PSI or 1/2-Oz Drop	1/16-PSI or 3/4-Oz Drop	1/8-PSI or 1-Oz Drop	1/4-PSI or 2-Oz Drop	1/2-PSI or 4-Oz Drop	1-PSI or 8-Oz Drop	Vertical	Horizontal			
A	B	C	D	E	F	G	H†	I‡	J§	K	L‡
	Capacity Expressed in EDR of 240 Btuh										
3/4	.....	.....	30	.....	.....	.....	30	....	25	...	...
1	39	46	56	79	111	157	56	34	45	28	28
1 1/4	87	100	122	173	245	346	122	75	98	62	62
1 1/2	134	155	190	269	380	538	190	108	152	93	93
2	273	315	386	546	771	1,091	386	195	288	169	169
2 1/2	449	518	635	898	1,270	1,800	635	395	464	....	260
3	822	948	1,160	1,650	2,330	3,290	1,130	700	800	....	475
3 1/2	1,230	1,420	1,740	2,460	3,470	4,910	1,550	1,150	1,140	....	745
4	1,740	2,010	2,460	3,480	4,910	6,950	2,040	1,700	1,520	....	1,110
5	3,210	3,710	4,550	6,430	9,090	12,900	4,200	3,150	....	....	2,180
6	5,280	6,100	7,460	10,550	14,900	21,100	7,200	5,600	....	....	....
8	11,000	12,700	15,500	21,970	31,070	43,900	15,000	12,000	....	....	....
10	20,000	23,100	28,300	40,100	56,700	80,200	28,000	23,000	....	....	....
12	32,000	37,100	45,500	64,300	91,000	129,000	46,000	38,000	....	....	....
16	61,000	69,700	84,800	121,000	170,000	242,000	88,000	76,000	....	....	....
	Capacity Expressed in Pounds per Hour										
3/4	.....	.....	8	.....	.....	.....	8	.....	6	.....	7
1	10	12	14	20	28	40	14	9	11	7	7
1 1/4	22	25	31	43	61	87	31	19	20	16	16
1 1/2	34	39	48	67	95	135	48	27	38	23	23
2	68	79	97	137	193	273	97	49	72	42	42
2 1/2	112	130	159	225	318	449	159	99	116	.....	65
3	206	237	291	411	581	822	282	175	200	.....	119
3 1/2	307	355	434	614	869	1,230	387	288	286	.....	186
4	435	503	614	869	1,230	1,740	511	425	380	.....	278
5	806	928	1,140	1,610	2,270	3,210	1,050	788	.....	.....	545
6	1,320	1,520	1,870	2,640	3,730	5,280	1,800	1,400	.....	.....	.....
8	2,750	3,170	3,880	5,490	7,770	11,000	3,750	3,000	.....	.....	.....
10	5,010	5,790	7,090	10,000	14,200	20,000	7,000	5,700	.....	.....	.....
12	8,040	9,290	11,400	16,100	22,700	32,200	11,500	9,500	.....	.....	.....
16	15,100	17,400	21,200	30,300	42,400	60,500	22,000	19,000	.....	.....	.....
	All Horizontal Mains and Downfeed Risers						Upfeed Risers	Mains and Undripped Runouts	Upfeed Risers	Radiator Connections	Runouts Not Dripped

* Table based on pipe-size data developed through research investigations of the American Society of Heating and Ventilating Engineers. Steam at an average pressure of 1 psig is used as a basis for calculating capacities. All drops shown are in psi per 100 ft of equivalent run—based on pipe properly reamed.

† Do not use column H for drops of 1/64 or 1/32 psi; substitute column C or column B as required.

‡ Pitch of horizontal runouts to risers and radiators should be not less than 1/2 in. per ft. Where this pitch cannot be obtained, runouts over 8 ft in length should be one pipe size larger than called for in the table.

§ Do not use column J for 1/32 psi drop except on sizes 3 in. and over; below 3 in. substitute column B.

4. For downfeed systems the main risers of which do not carry any radiator condensation, use column H.
5. For the radiator valve size and the stub connection, use column K.
6. For the dry-return main, use column U.
7. For the wet-return main, use column T.

On systems the longest circuit of which exceeds an equivalent length of 200 ft, size as follows:

The total drop should not be over $\frac{1}{4}$ psi. The return-pipe sizes should correspond with those for the drop used on the steam side of the system. Thus, when $\frac{1}{24}$-psi drop is being used, the steam main and dripped runouts would be sized from column C; radiator runouts and undripped riser runouts, from column L; upfeed risers, from column J; the main riser on a downfeed system, from column C (it will be noted that if column H is used the drop would exceed the limit of $\frac{1}{24}$ psi); the dry return, from column R; and the wet return, from column Q.

With $\frac{1}{32}$-psi drop the sizing would be the same as for $\frac{1}{24}$ psi except that the steam main and dripped runouts would be sized from column B; the main riser on a downfeed system, from column B; the dry return, from column O; and the wet return, from column N.

One procedure to follow in designing low-pressure systems is to allocate not more than one-half the initial gage pressure in computing the design pressure drop.

It is desirable for the main to pitch in the direction of flow at a rate of not less than $\frac{1}{4}$ in. in 10 ft, and preferably $\frac{1}{2}$ in. The runouts to radiators and risers should pitch toward the main at not less than $\frac{1}{2}$ in. in 10 ft. If such a pitch is not practicable, or if the runout is more than 8 ft long, pipe one size larger should be used for the runout. Further, to prevent sagging between supports, it is recommended that the main should not be smaller than nominal 2-in. pipe.

Where the cost of a two-pipe system can be justified, it may be desirable to select a vapor or vacuum-type system instead of using a conventional air-type design.

Example 7-8. A one-pipe low-pressure air-vent steam-heating system is similar in its basic layout to Fig. 6-7 except that risers run off the main and connect to radiators on upper floors. The initial pressure is 1 psig and the equivalent length of run from the boiler to the most distant radiator is estimated at 180 ft. A pressure drop of 1 oz per sq in. is allowed for each 100 ft, and the total load on the system is 800 EDR. (a) What should be the size of the main? (b) If each riser supplies 200 EDR, what is the size of the risers? (c) What should be the size of the dry-return main for the 800 EDR?

Solution: (a) From Table 7-6, column D, the next-larger capacity above 800 is 1160, which corresponds to a 3-in. main. *Ans.*

b) From column J, the next-larger capacity above 200 is 288, corresponding to a 2-in. supply riser. *Ans.*

c) From column U, a $1\frac{1}{2}$-in. dry return main is indicated. However, a main less than 2 in. should not be used and the run is so short that 3 in. may be carried throughout. *Ans.*

TABLE 7-7. RETURN-PIPE CAPACITIES FOR LOW-PRESSURE SYSTEMS*

In Square Feet of Equivalent Direct Radiation

Pipe Size (in.)	Capacity of Return Mains and Risers																	
	Mains																	
	3/32-PSI or 1/2-Oz Drop per 100 Ft			1/4-PSI or 1/2-Oz Drop per 100 Ft			1/8-PSI or 1-Oz Drop per 100 Ft			1/2-PSI or 2-Oz Drop per 100 Ft			1/4-PSI or 4-Oz Drop per 100 Ft			1/2-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.
M	N	O	P	Q	R	S	T	U	V	W	X	Y	Z	AA	BB	CC	DD	EE
3/4	....	....	....	....	....	326	....	....	400	....	....	568	....	....	800	....	....	1,130
1	500	248	....	580	285	570	700	320	700	1,000	412	994	1,400	460	1,400	....	....	1,980
1 1/4	850	520	....	990	595	976	1,200	670	1,200	1,700	868	1,700	2,400	962	2,400	....	....	3,390
1 1/2	1,350	822	....	1,570	943	1,550	1,900	1,060	1,900	2,700	1,360	2,700	3,800	1,510	3,800	....	....	5,370
2	2,800	1,880	....	3,240	2,140	3,260	4,000	2,300	4,000	5,600	2,960	5,680	8,000	3,300	8,000	....	....	11,300
2 1/2	4,700	3,040	....	5,300	3,470	5,450	6,700	3,800	6,700	9,400	4,900	9,510	13,400	5,450	13,400	....	....	18,900
3	7,500	5,840	....	8,500	6,250	8,710	10,700	7,000	10,700	15,000	9,000	15,200	21,400	10,000	21,400	....	....	30,200
3 1/2	11,000	7,880	....	13,200	8,800	13,000	16,000	10,000	16,000	22,000	12,900	22,700	32,000	14,300	32,000	....	....	45,200
4	15,500	11,700	....	18,300	13,400	18,000	22,000	15,000	22,000	31,000	19,300	31,200	44,000	21,500	44,000	....	....	62,190
5	....	....	....	....	....	31,500	....	....	38,700	....	....	54,900	....	....	77,400	....	....	109,000
6	....	....	....	....	....	50,450	....	....	62,000	....	....	88,000	....	....	124,000	....	....	175,000
	Risers																	
3/4	....	190	....	....	190	570	....	190	700	....	190	994	....	190	1,400	....	....	1,980
1	....	450	....	....	450	976	....	450	1,200	....	450	1,700	....	450	2,400	....	....	3,390
1 1/4	....	990	....	....	990	1,550	....	990	1,900	....	990	2,700	....	990	3,800	....	....	5,370
1 1/2	....	1,500	....	....	1,500	3,260	....	1,500	4,000	....	1,500	5,680	....	1,500	8,000	....	....	11,300
2	....	3,000	....	....	3,000	5,450	....	3,000	6,700	....	3,000	9,510	....	3,000	13,400	....	....	18,900
2 1/2	....	....	....	....	....	8,710	....	....	10,700	....	....	15,200	....	....	21,400	....	....	30,200
3	....	....	....	....	....	13,000	....	....	16,000	....	....	22,700	....	....	32,000	....	....	45,200
3 1/2	....	....	....	....	....	17,900	....	....	22,000	....	....	31,200	....	....	44,000	....	....	62,200
4	....	....	....	....	....	31,500	....	....	38,700	....	....	54,900	....	....	77,400	....	....	109,000
5	....	....	....	....	....	50,500	....	....	62,000	....	....	88,000	....	....	124,000	....	....	175,000

* Table based on pipe-size data developed through research investigations of American Society of Heating and Ventilating Engineers.

TABLE 7-8. RETURN-PIPE CAPACITIES FOR LOW-PRESSURE SYSTEMS*

In Pounds per Hour

Pipe Size (in.)	Capacity of Return Mains and Risers																	
	Mains																	
	1/32-PSI or 1/2-Oz Drop per 100 Ft			1/24-PSI or 3/4-Oz Drop per 100 Ft			1/16-PSI or 1-Oz Drop per 100 Ft			1/8-PSI or 2-Oz Drop per 100 Ft			1/4-PSI or 4-Oz Drop per 100 Ft			1/2-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.
M	N	O	P	Q	R	S	T	U	V	W	X	Y	Z	AA	BB	CC	DD	EE
3/4	....	....	....	....	....	42	....	....	100	....	....	142	....	....	200	....	....	283
1	125	62	....	145	71	143	175	80	175	250	103	249	350	115	350	....	....	494
1 1/4	213	130	....	248	149	244	300	168	300	425	217	426	600	241	600	....	....	848
1 1/2	338	206	....	393	236	388	475	265	475	675	340	674	950	378	950	....	....	1,340
2	700	470	....	810	535	815	1,000	575	1,000	1,400	740	1,420	2,000	825	2,000	....	....	2,830
2 1/2	1,180	760	....	1,580	868	1,360	1,680	950	1,680	2,350	1,230	2,380	3,350	1,360	3,350	....	....	4,730
3	1,880	1,460	....	2,130	1,560	2,180	2,680	1,750	2,680	3,750	2,250	3,800	5,350	2,500	5,350	....	....	7,560
3 1/2	2,750	1,970	....	3,300	2,200	3,250	4,000	2,500	4,000	5,500	3,230	5,680	8,000	3,580	8,000	....	....	11,300
4	3,880	2,930	....	4,580	3,350	4,500	5,500	3,750	5,500	7,750	4,830	7,810	11,000	5,380	11,000	....	....	15,500
5	....	....	....	....	....	7,880	....	....	9,680	....	....	13,700	....	....	19,400	....	....	27,300
6	....	....	....	....	....	12,600	....	....	15,500	....	....	22,000	....	....	31,000	....	....	43,800
	Risers																	
3/4	....	48	....	....	48	143	....	48	175	....	48	249	....	48	350	....	....	494
1	....	113	....	....	113	244	....	113	300	....	113	426	....	113	600	....	....	848
1 1/4	....	248	....	....	248	388	....	248	475	....	248	674	....	248	950	....	....	1,340
1 1/2	....	375	....	....	375	815	....	375	1,000	....	375	1,420	....	375	2,000	....	....	2,830
2	....	750	....	....	750	1,360	....	750	1,680	....	750	2,380	....	750	3,350	....	....	4,730
2 1/2	....	....	....	....	....	2,180	....	....	2,680	....	....	3,800	....	....	5,350	....	....	7,560
3	....	....	....	....	....	3,250	....	....	4,000	....	....	5,680	....	....	8,000	....	....	11,300
3 1/2	....	....	....	....	....	4,480	....	....	5,500	....	....	7,810	....	....	11,000	....	....	15,500
4	....	....	....	....	....	7,880	....	....	9,680	....	....	13,700	....	....	19,400	....	....	27,300
5	....	....	....	....	....	12,600	....	....	15,500	....	....	22,000	....	....	31,000	....	....	43,800

* Table based on pipe-size data developed through research investigations of American Society of Heating and Ventilating Engineers.

Example 7–9. If one of the radiators in the system referred to in example 7–8 has a capacity equal to 50 EDR, what must be the size of the valve and the vertical pipe connecting it with the runout?

Solution: In Table 7–6, column K, the next-larger radiator surface above 50 is 62, corresponding to a $1\frac{1}{4}$-in. valve. The vertical connection is of the same size, $1\frac{1}{4}$ in. *Ans.*

Example 7–10. If the required valve at the radiator is $1\frac{1}{4}$ in. in diameter, as determined by Table 7–6, what should be the size of the horizontal runout from the main to the riser serving the radiator?

Solution: The size is taken from column L, and in this case is $1\frac{1}{4}$ in. *Ans.*

7–6. PIPE SIZES FOR VAPOR HEATING SYSTEMS

In design of vapor heating systems the pipe sizes can be obtained by use of tables 7–6, 7–7, and 7–8. The least pressure drop consistent with reasonable pipe sizing should be employed. It is most desirable that the condensate be returned to the boiler by gravity, and to obtain a uniform distribution of steam throughout the system it is particularly desirable to have almost the same pressure throughout. Excessive variations in pressure throughout the system would make less effective the temperature control it is possible to obtain by setting the graduated valves on the radiators.

Where the equivalent length of run does not exceed 200 ft, the sizes of mains and runouts to risers which are dripped may be selected from Table 7–6, column D, and undripped runouts and radiator branches may be taken from column I. Upfeed supply risers are sized in column H. Return mains and risers should be sized in column U, Table 7–7. If the supply mains are overhead, making a downfeed system, the main upfeed riser size is selected from column H of Table 7–6 and the downfeed risers are selected from column D.

If the equivalent length of run is more than 200 ft, the pressure drop used in selecting the pipe sizes should not exceed $\frac{1}{8}$ to $\frac{1}{4}$ psi. Thus for a 400-ft run the drop per 100 ft might be $\frac{1}{8}$ psi divided by 4, or $\frac{1}{32}$ psi, and in this case the supply-main size would be selected from column B, Table 7–6; the radiator and undripped runout size, from column I; and the risers, from column H. However, it may be desirable also to select the risers from column B because the design pressure in column H exceeds $\frac{1}{32}$ psi. With downfeed systems, column B should be used both for the main risers and for the smaller risers feeding the radiators. The return risers should be sized from column O, Table 7–7, while wet returns should be sized from column N. The same pressure drop should be used for both the steam and return sides of the system.

In addition to the general comments on the use of tables 7–6, 7–7, and 7–8, it should also be noted that the mains should pitch at not less than $\frac{1}{4}$ in. in 10 ft. Horizontal runouts in risers and radiators should be at least $\frac{1}{2}$ in. per ft, or, where this pitch cannot be obtained, runouts over 8 ft in

length should be one pipe size larger than that called for in the table. Supply mains less than 2 in. should not, in general, be used. Supply mains, supply risers, and runouts to supply risers should be dripped separately into a wet return, or should be connected into a dry return with a thermostatic trap.

In laying out two-pipe systems, the complete piping circuit should first be sketched on the architectural plan. For vapor and return-trap systems in particular, it is desirable for the supply mains to connect the convectors and other radiation with piping circuits as short and direct as possible from the boiler. With complex circuits a reversed-return system may be desirable. In such a circuit the convectors which are first supplied with steam are connected into the far end of the return main so that, although the steam gets to these convectors first, it has a long circuit to travel before returning to the boiler. By use of a reversed-circuit arrangement in a complex system, the air travel during the warm-up period is about the same for both the close and most distant radiation. Moreover, in a reversed-return arrangement, both the mains and the returns pitch in the same direction, which gives a much neater appearance than is the case with a direct-return arrangement. Where possible, return mains should pitch more steeply than supply mains, 1 in. in 10 ft being desirable. On full vacuum systems, return mains need not pitch more than $\frac{1}{2}$ in. in 10 ft. Figures 6–11 and 7–6 both show reversed-return circuits.

To illustrate a representative vapor-system layout, the residence pictured in Figs. 7–6, 7–7, and 7–8 has been designed for a vapor system, with Fig. 7–6 presenting the basement layout for the vapor system. In this design, although it would be possible to employ a single main looping around the building, two circuits were used to give direct service to each side of the building. This also illustrates how two dry returns can lead into a common wet return. In this design, since inadequate headroom is available in the basement to return the water into the boiler properly, a return trap (alternating receiver) is used.

Details of the computations will not be given in the text, with the exception of selecting the mains. The length of the north steam-supply main measures 52 ft plus a 4-ft rise at the boiler, and there are 6 elbows and 1 full diversion tee. By Table 7–4, there are $6 + (1)(1.8) = 7.8$ elbow equivalents, and for an assumed pipe size of 2 in. this amounts to $(7.8)25 \times 2 \times \frac{1}{12} = 33$ ft. Thus the equivalent length is $52 + 4 + 33 = 89$ ft. From this main, $38.5 + 18 + 23 + 22.5 + 28 + 32$, or 180 EDR, must be supplied. The pressure drop for a vapor system in a building of this size may reasonably be around $\frac{1}{8}$ psi total, which would indicate (say) $\frac{1}{16}$ psi per 100 ft of run. Thus column D, Table 7–6, as before indicated, should be used. While 190 EDR would be allowed by this table for a $1\frac{1}{2}$-in. main, a 2-in. main is recommended for structural reasons. For the

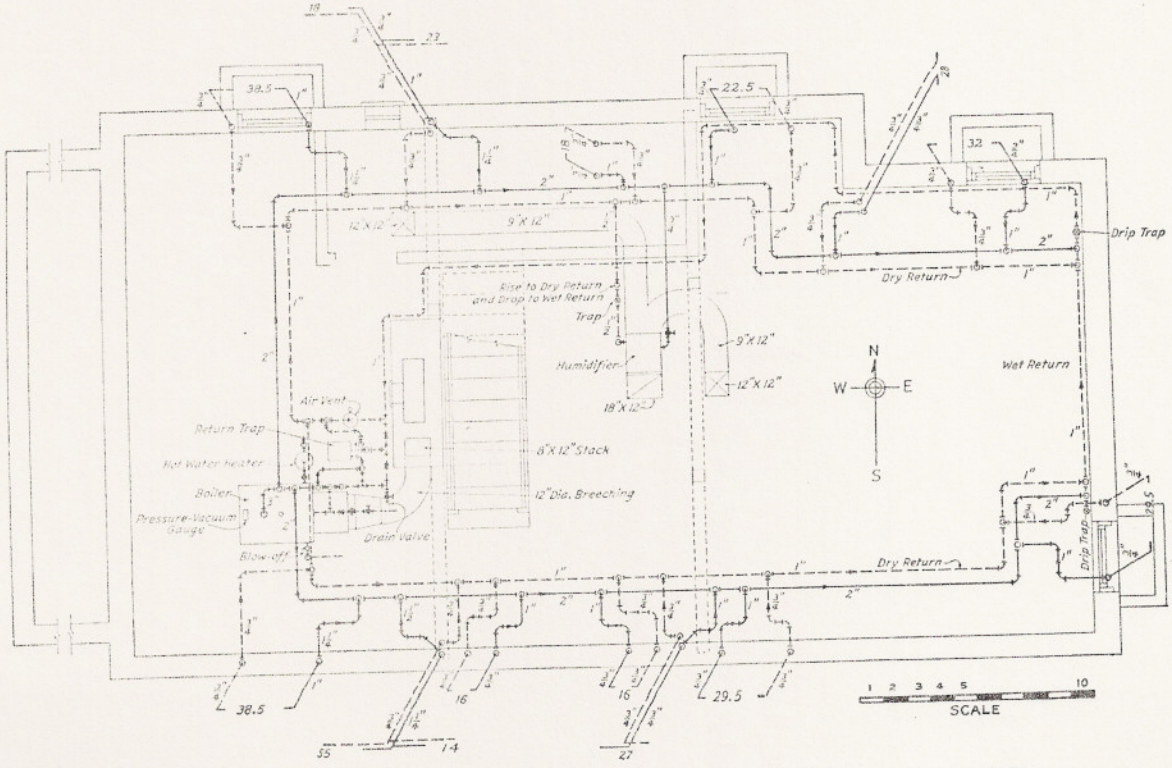

FIG. 7–6. Basement plan, vapor steam-heating system. Radiation expressed in EDR.

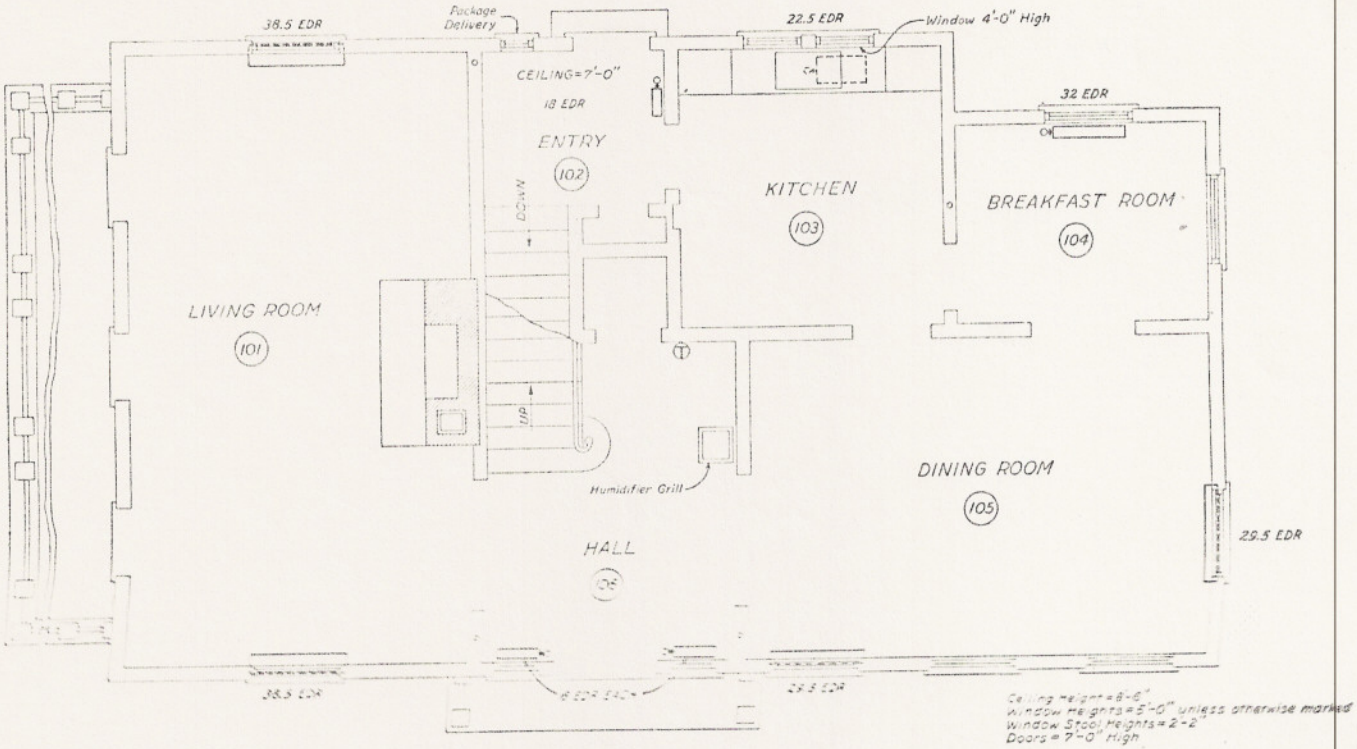

FIG. 7-7. First-story plan, vapor steam-heating system.

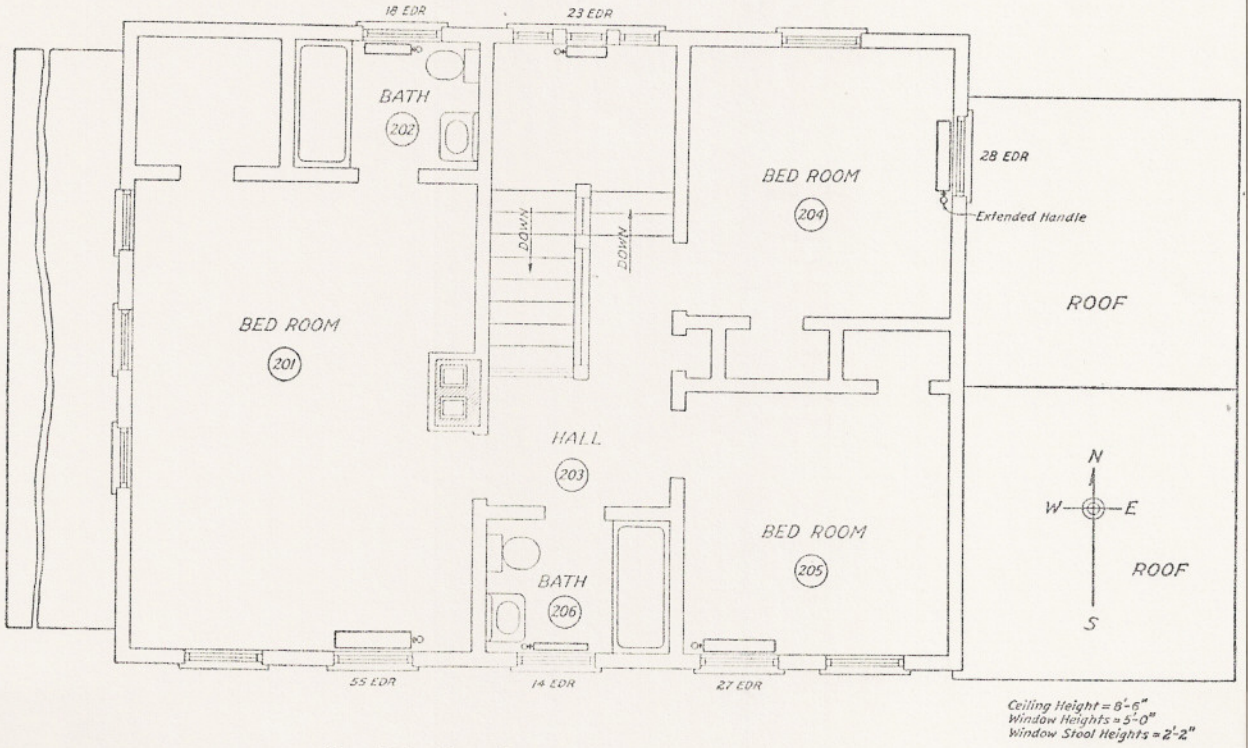

FIG. 7-8. Second-story plan, vapor steam-heating system.

return mains of a vapor system, smaller pipes are permitted than on the supply side.

For the south main, with a run of 67 ft and a rise of 4 ft at the boiler, and with 6 elbows and 1 full diversion tee, for a 2-in. assumed pipe size the equivalent length is 104 ft. For the 225.5 EDR, again by use of column D in Table 7–6, it is seen that a 2-in. pipe should be used. The return pipe lengths are closely the same as the supply mains, and their sizes can be found in Table 7–7, using the same pressure drop per 100-ft run as for the supply mains. The dry-return pipe size for a vapor system is given in column U. Evidently a 1-in. main is ample for each return.

7–7. VACUUM STEAM-HEATING SYSTEMS

Vacuum steam-heating systems are of two-pipe design and use mechanical means for removing air and condensate. Vacuum systems are normally used in large buildings because with them it is possible to obtain both close control of temperature and well-balanced steam circulation, and also, smaller pipe sizes can be used. The differential pressure in the vacuum system consists of the suction produced by the vacuum pump on the return side, and in addition, such positive (gage) pressure as may exist on the supply side.

Good design practice calls for not more than $\frac{1}{8}$-psi drop per 100 ft of equivalent run and not more than 1 psi total pressure drop in a system. Pitch of main should not be less than $\frac{1}{4}$ in. in 10 ft. Supply mains smaller than 2 in. are undesirable. Connection should not be made between the steam and return sides of a vacuum system except through steam traps, as it is necessary to prevent steam in quantity from entering the return line.

Lifts for raising condensate from higher to lower levels should be avoided wherever possible, even though lift fittings (Fig. 6–19) are reasonably effective. Lift fittings for low-lying radiators are particularly undesirable. They must not be used in orifice and atmospheric systems.

Although pressure drops up to 1 psi are occasionally employed, pressure drops of $\frac{1}{4}$ to $\frac{1}{2}$ psi represent better design. For systems with equivalent lengths not in excess of 200 ft, the $\frac{1}{4}$ psi is preferable. For systems of greater length, $\frac{1}{2}$ psi total drop may be desirable because of the smaller pipe sizes which might be used. For example, in a 1200-ft equivalent-length system, with a drop of $\frac{1}{2}$ psi allowed, the drop per 100 ft would be $\frac{1}{24}$ psi. For this condition, the steam main could be sized from column C, Table 7–6, and the risers also could be selected from column C. Column H might also be used for the risers, but in this case the $\frac{1}{24}$ psi design figure would be exceeded. If for a system as long as this the design were based on $\frac{1}{4}$ in., the sizing tables would not be adequate. Riser runouts use column C if dripped, or column I if undripped. For return risers, column S in Table 7–7 should be employed.

7–8. ORIFICES IN STEAM SYSTEMS

With vacuum systems, in particular, it is possible to use orifices for basic adjustment of the load and also for control. An orifice is usually supplied as a drilled hole in a plate made to be inserted in the radiator valve

TABLE 7–9

ORIFICE CAPACITIES FOR LOW-PRESSURE STEAM SYSTEMS*

(Lb per Hr)

Orifice Diameter		Differential Pressure across Orifice (in. Hg)				
In 64ths of an Inch	In Inches	6	5	4	2	1
7	0.109	4.5– 5.8	4.0– 5.3	3.8– 4.8	2.5– 3.3	.....
8	0.125	5.8– 7.3	5.3– 6.8	4.8– 6.3	3.3– 4.3	2.0– 2.8
10	0.156	9.0–11.0	8.3–10.0	7.5– 9.3	5.3– 6.5	3.5– 4.3
12	0.188	13.0–15.5	12.0–14.3	11.0–12.8	7.8– 9.3	5.0– 6.0
14	0.219	18.0–20.8	16.5–19.0	14.8–16.8	10.8–12.3	7.0– 8.0
16	0.250	23.5–26.5	21.5–24.3	19.0–21.5	14.0–16.0	9.3–10.5
18	0.281	29.8–33.3	27.3–30.5	24.3–27.0	18.0–20.0	11.8–13.0
19	0.297	33.3–37.0	30.5–33.8	27.0–30.0	20.0–22.0	13.0–14.5
20	0.313	37.0–40.8	33.8–37.3	30.0–33.3	22.0–24.5	14.5–16.0
21	0.328	40.8–44.8	37.3–41.0	33.3–36.3	24.5–26.8	16.0–17.8

NOTE. The radiator orifice plates recommended in this table are made of brass stampings 0.023 in. thick, cup-shaped to be inserted in radiator valve unions.

* Based on test data by Sanford and Sprenger, *Trans. ASHVE*, Vol. 37 (1931), p. 371. To convert data in this table to EDR, multiply by 4.

fitting. Table 7–9 gives data on representative sizes and also shows the capacity that might be expected from them under various differential pressures.

The piping in a system which employs orifices is similar to that which might be found in any well-designed vacuum system. The steam flow through the orifice for the small pressure ranges involved varies as the square root of the pressure differential across the orifice. In an orifice system, if the steam pressure is dropped from (say) 1 psig to 0.2 psig, with the pressure on the vacuum side remaining unchanged at −0.2 psig, the weight of steam flowing in the radiator would be reduced to

$$\frac{\sqrt{0.2 - (-0.2)}}{\sqrt{1 - (-0.2)}} = \sqrt{\frac{0.4}{1.2}} = 0.58$$

That is, only 58 per cent as much steam would flow under the changed condition as before. As the heat produced is a function of the steam flowing and condensed, it is evident immediately that the capacity can be changed through wide limits by varying the inlet pressure in an orifice system. Control for the system can be arranged to take place either by

varying the inlet pressure into the steam mains feeding the various radiators, or by varying the vacuum in the return lines. For some vacuum systems, it has been observed that with the use of radiator orifice plates it is possible to eliminate traps on the discharge sides of each radiator. In long or complex systems, where there is necessarily an appreciable pressure difference between the supply and remote end of the steam main, it is possible to provide larger orifices for the more distant radiation, and smaller orifices for the radiation close to the supply source.

7–9. HIGH-PRESSURE STEAM SYSTEMS

For heating of industrial space, high-pressure steam may be used directly in suitable radiation. However, the pressure is frequently dropped through reducing valves and the low-pressure steam is used in conventional systems. Where high-pressure steam is used directly, the load-carrying capacity of steam and condensate return pipes is given in tables 7–10 and 7–11 for 30-psig systems, and in tables 7–12 and 7–13 for 150-psig systems.[1] With high-pressure steam, higher pressure drops can be used to force the steam throughout the system. In the 30-psig system, for example, total pressure drops from 5 to 10 psi are customary, while in a 150-psig system, 25- to 30-psi drops are employed. High-pressure mains should pitch at least $\frac{1}{4}$ in. in 10 ft, and horizontal runouts to risers and heaters should pitch at least $\frac{1}{2}$ in. per foot.

Unit pressure drops employed in design of high-pressure systems range from $\frac{1}{2}$ to 1 psi per 100 ft of equivalent run. With high-pressure steam, particularly when it exceeds 100 psig, it should be realized that the surface temperature of the pipes and radiator surface is in excess of 330 F. Where physical contact is made with surfaces of this temperature, bad burns can result. Thus the pipes must be so placed as to prevent physical contact. A common arrangement with high-pressure steam is the use of blast coils, in which air is warmed by being passed over the steam coils and the space is then warmed by the air. Unit heaters such as those shown in Figs. 9–10 and 9–11 also can be employed. It is also possible to use finned-type pipe coil with suitable shielding. The same type of device is also satisfactory with low-pressure steam. In fact, it may be preferable to pipe the high-pressure steam to a reducing valve and drop the pressure in the reducing valve to the lower pressure and temperature ranges employed in conventional steam systems. Whenever the steam pressure is 100 psig or above, it is desirable to use two reducing valves in series to drop the pressure to, for example, 5 or 2 psig. In general, systems operating at 15 psig or less are called low-pressure systems, and those operating at pressures above 15 psig are called high-pressure systems.

[1] Tables 7–10, 11, 12, and 13 are reprinted, by permission, from *Heating Ventilating Air Conditioning Guide 1956*, Chapter 21.

TABLE 7–10

STEAM-PIPE CAPACITIES FOR 30-PSIG STEAM SYSTEMS*

(Lb per Hr)

(Steam and Condensate Flowing in Same Direction)

PIPE SIZE (IN.)	DROP IN PRESSURE (PSI PER 100-FT LENGTH)					
	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	1	2
$\frac{3}{4}$	15	22	31	38	45	63
1	31	46	63	77	89	125
$1\frac{1}{4}$	69	100	141	172	199	281
$1\frac{1}{2}$	107	154	219	267	309	437
2	217	313	444	543	627	886
$2\frac{1}{2}$	358	516	730	924	1,030	1,460
3	651	940	1,330	1,630	1,880	2,660
$3\frac{1}{2}$	979	1,410	2,000	2,450	2,830	4,000
4	1,390	2,000	2,830	3,460	4,000	5,660
6	4,210	6,030	8,590	10,400	12,100	17,200
8	8,750	12,600	17,900	21,900	25,300	35,100
10	16,300	23,500	33,200	40,600	46,900	66,400
12	25,600	36,900	52,300	64,000	74,000	104,500

* Steam at an average pressure of 30 psig used as basis for calculating the values in the table.

TABLE 7–11

RETURN-PIPE CAPACITIES FOR 30-PSIG STEAM SYSTEMS*

(Lb per Hr)

PIPE SIZE (IN.)	DROP IN PRESSURE (PSI PER 100-FT LENGTH)				
	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	1
$\frac{3}{4}$	115	170	245	308	365
1	230	340	490	615	730
$1\frac{1}{4}$	485	710	1,025	1,290	1,530
$1\frac{1}{2}$	790	1,160	1,670	2,100	2,500
2	1,580	2,360	3,400	4,300	5,050
$2\frac{1}{2}$	2,650	3,900	5,600	7,100	8,400
3	4,850	7,100	10,300	12,900	15,300
$3\frac{1}{2}$	7,200	10,600	15,300	19,200	22,800
4	10,200	15,000	21,600	27,000	32,300
6	31,000	45,500	65,500	83,000	98,000

* Table based on steam at pressures of 0 to 4 psig.

TABLE 7–12

STEAM-PIPE CAPACITIES FOR 150-PSIG STEAM SYSTEMS*

(Lb per Hr)

(Steam and Condensate Flowing in Same Direction)

PIPE SIZE (IN.)	DROP IN PRESSURE (PSI PER 100-FT LENGTH)						
	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	1	2	5
$\frac{3}{4}$	29	41	58	71	82	116	184
1	58	82	117	143	165	233	369
$1\frac{1}{4}$	130	185	262	320	370	523	827
$1\frac{1}{2}$	203	287	407	497	575	813	1,290
2	412	583	825	1,010	1,170	1,650	2,600
$2\frac{1}{2}$	683	959	1,360	1,650	1,920	2,710	4,290
3	1,240	1,750	2,480	3,020	3,500	4,940	7,820
$3\frac{1}{2}$	1,860	2,630	3,720	4,550	5,250	7,420	11,700
4	2,630	3,720	5,260	6,430	7,430	10,500	16,600
6	7,960	11,300	16,000	19,500	22,600	31,900	50,400
8	16,600	23,500	33,200	40,600	47,000	66,400	105,000
10	30,800	43,400	61,700	75,600	87,300	123,000	195,000
12	48,600	68,800	97,300	119,000	138,000	194,000	307,500

* Steam at average pressure of 150 psig used as basis for calculating the table.

TABLE 7–13

RETURN-PIPE CAPACITIES FOR 150-PSIG STEAM SYSTEMS*

(Lb per Hr)

PIPE SIZE (IN.)	DROP IN PRESSURE (PSI PER 100-FT LENGTH)					
	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	1	2
$\frac{3}{4}$	156	232	360	465	560	890
1	313	462	690	910	1,120	1,780
$1\frac{1}{4}$	650	960	1,500	1,950	2,330	3,700
$1\frac{1}{2}$	1,070	1,580	2,460	3,160	3,800	6,100
2	2,160	3,300	4,950	6,400	7,700	12,300
$2\frac{1}{2}$	3,600	5,350	8,200	10,700	12,800	20,400
3	6,500	9,600	15,000	19,500	23,300	37,200
$3\frac{1}{2}$	9,600	14,400	22,300	28,700	34,500	55,000
4	13,700	20,500	31,600	40,500	49,200	78,500
6	42,000	62,500	96,000	125,000	150,000	238,000

* Table based on steam at pressures of 1 to 20 psig.

PROBLEMS

7–1. A 4-in. steam main has an actual length of 90 ft. It contains four elbows and one gate valve. What is its equivalent length? *Ans.* 127.5 ft

7–2. What size of steel pipe is required to carry 6000 lb of steam per hour through a main having an equivalent length of 450 ft if the initial pressure is 100 psig and the final pressure is 95 psig? *Ans.* 4 in.

7–3. A two-pipe vapor system serving 2800 EDR has a total equivalent length of 200 ft, and the total pressure drop is to be 2 oz per sq in. (a) What size of main and dry return should be used? (b) To supply a radiator on the second floor containing 75 EDR, what size of riser and horizontal branch should be used?
Ans. (a) 5 in., $2\frac{1}{2}$ in.; (b) $1\frac{1}{4}$ in., $1\frac{1}{4}$ in.

7–4. A vacuum system having 9000 EDR has a total equivalent length of 800 ft. It is to be designed for a total pressure drop of 1 psi. What size of steam and return main should be used? *Ans.* 6 in., 3 in.

7–5. Refer to the vapor heating system illustrated in Fig. 6–11 and compute the total equivalent length of the right-hand circuit from the outlet of the boiler to the point where the circuit drops into the downpipe of the wet return. The linear length of this circuit is 166 ft, which includes the 6-ft rise from the boiler outlet to the highest point of the main. Consider the pipe that is run from the boiler to be 3-in. pipe, becoming 2 in. at the tee and finally running as $1\frac{1}{2}$-in. pipe from the last radiator connection to the drop leg which leads into the wet return. Consider the inlet tee to this circuit as diverting approximately one-half the flow. The scale of the drawing in Fig. 6–11 is approximately $\frac{1}{16}$ in. = 1 ft. *Ans.* 166 + 57 = 223 ft

7–6. Refer to the vapor heating system illustrated in Fig. 6–11 and compute the total equivalent length of the left-hand circuit from the outlet of the boiler to the point where the circuit drops into the downpipe of the wet return. The linear length of this circuit is 161 ft, including a 6-ft rise from the boiler outlet to the highest point of the main. Consider the main run to be 3-in. pipe which reduces to 2 in. at the tee, where the two loops separate. The loop then runs as 2 in. until past the last radiator connection, following which it is reduced to $1\frac{1}{2}$ in. The scale of the drawing in Fig. 6–11 is approximately $\frac{1}{16}$ in. = 1 ft. *Ans.* 161 + 39 = 200 ft

7–7. The right-hand circuit of the vapor heating system of Fig. 6–11 has a connected load of 270 EDR, while the left-hand circuit has a connected load of 360 EDR. In accordance with good practice, select the required pipe sizes of the main that is to serve the right-hand circuit. Data on the length of the circuit can be found in problem 7–5. Assume that the return right-hand circuit is of essentially the same length as the supply circuit. Find the size of dry-return main which should be employed.
Ans. 2-in. main, 1-in. return

7–8. The left-hand circuit of the vapor heating system of Fig. 6–11 has a connected load of 360 EDR, and the right-hand circuit has a connected load of 270 EDR. Select in accordance with good practice the required pipe sizes of the main to be specified to serve this left-hand circuit. Data on the length of the circuit can be found in problem 7–6. Assume that the return of this left-hand circuit is of essentially the same length as the supply circuit. Find the size of dry-return main which should be employed.

7–9. For the two-pipe, air-vent, gravity-return system of Fig. 6–8, considering the extreme right circuit as having the greatest length, compute the required size of

gain, return, risers, and runouts for this circuit. Each unit of radiation delivers 14,400 Btuh and requires approximately 14.4 lb of steam per hour, with a total boiler output of 86.4 lb of steam per hour.

7–10. For the two-pipe, air-vent, gravity-return system of Fig. 6–8, compute the size of the risers and runouts for the radiation which in the sketch appears to be roughly over the boiler. Refer to problem 7–9 for data regarding the system.

7–11. A five-story narrow building is heated by a one-pipe air-vent steam system supplied from a boiler located in an adjacent building. The run to the riser is 165 ft long and contains tees at the boiler and riser, and two elbows. The main continues, eventually dropping into a wet return back to the boiler, for a total return length of 175 ft. The return circuit contains five elbows up to the Hartford return loop at the boiler. The first-floor radiator take-offs from the riser are 5 ft above the supply main. The second floor and succeeding floors are each 10 ft above their respective lower floors. Each floor, except the top, has two 50-EDR take-offs, while the top floor with an associated sundeck has two 100-EDR take-offs. Make a rough diagrammatic sketch of the system as described, and compute the pipe sizes required for each part of the system—supply main, return main, riser, radiator branches, etc.

7–12. Consider that the underground steam line of problem 6–9 serves a single building with a load of 4000 EDR. The allowable pressure loss in the line is 40 psi, and the steam, on entering the building, passes through two reducing valves in series for distribution throughout the building at 5 psig. Find (a) the amount of saturated steam flowing, in pounds per hour, and (b) the closest standard pipe size required.

Ans. (a) 1000 lb per hr (approx.); (b) 12 in. [oversize; run 2500 ft at (say) 10-in. size, 2500 ft at 12-in. size]

7–13. A building employing a vacuum steam system is so arranged that overhead mains supply the greater part of the radiation. For this system, which has a total equivalent length of 1100 ft, a design pressure drop of $\frac{1}{2}$ psi can be used. (a) What is the size of the upfeed riser serving the whole system if 1200 EDR are installed? (b) What is the size of the individual downfeed risers if each serves 150 EDR? (c) For each radiator at 75 EDR, find the size of the runouts.

Ans. Use $\frac{1}{2}$-psi drop. (a) $3\frac{1}{2}$ in.; (b) $1\frac{1}{2}$ in.; (c) $1\frac{1}{4}$ in.

7–14. Rework problem 7–13 on the basis of first changing the heating requirements to pounds per hour and then employing appropriate tables.

Ans. Same as for prob. 7–13

7–15. A central blast coil heats the air supplied to an auditorium in winter. The coil capacity at design outdoor conditions amounts to 175,000 Btuh. Steam at 5 psig is available and a vacuum return line equipped with a pump takes care of condensate and air venting. Find (a) the maximum steam flow to the coil, in pounds per hour; (b) the size of the feed pipe from the pressure-reducing valve to the coil if its equivalent length is 200 ft and if $\frac{1}{2}$-psi pressure drop is allowed; (c) the return pipe size to the vacuum pump if $\frac{1}{4}$-psi drop is allowed for the 200-ft run. *Ans.* (a) 175 lb/hr; (b) 2 in; (c) 1 in.

7–16. Rework problem 7–13 with no change other than that the capacity is 2000 EDR. *Ans.* (a) 4 in.; (b) $1\frac{1}{2}$ in.; (c) $1\frac{1}{4}$ in.

7–17. Rework problem 7–15 under the assumption that the blast-coil capacity is 25,000 Btuh, with no other change in data. *Ans.* (a) 125 lb/hr; (b) 2 in.; (c) $\frac{3}{4}$ in.

7–18. Make a diagrammatic sketch showing what piping connections to the blast coil of problem 7–15 should be made, and how they should be made. Include the trap and a supply-line runout and trap.