Related Commercial Resources

CHAPTER 10

STEAM SYSTEMS

STEAM systems use the vapor phase of water to supply heat or kinetic energy through a piping system. As a source of heat, steam can heat a conditioned space with suitable terminal heat transfer equipment such as fan-coil units, unit heaters, radiators, and convectors (finned tube or cast iron), or steam can heat through a heat exchanger that supplies hot water or some other heat transfer medium to the terminal units. In addition, steam is commonly used in heat exchangers (shell-and-tube, plate, or coil types) to heat domestic hot water and supply heat for industrial and commercial processes such as in laundries and kitchens. Steam is also used as a heat source for certain cooling processes such as single-stage and two-stage absorption refrigeration machines.

ADVANTAGES

Steam offers the following advantages:

- Steam flows through the system unaided by external energy sources such as pumps.
- Because of its low density, steam can be used in tall buildings where water systems create excessive pressure.
- Terminal units can be added or removed without making basic changes to the design.
- Steam components can be repaired or replaced by closing the steam supply, without the difficulties associated with draining and refilling a water system.
- Steam is pressure-temperature dependent; therefore, the system temperature can be controlled by varying either steam pressure or temperature.
- Steam can be distributed throughout a heating system with little change in temperature.

In view of these advantages, steam is applicable to the following facilities:

- Where heat is required for process and comfort heating, such as in industrial plants, hospitals, restaurants, dry-cleaning plants, laundries, and commercial buildings
- Where the heating medium must travel great distances, such as in facilities with scattered building locations, or where the building height would result in excessive pressure in a water system
- Where intermittent changes in heat load occur

FUNDAMENTALS

Steam is the vapor phase of water and is generated by adding more heat than required to maintain its liquid phase at a given pressure, causing the liquid to change to vapor without any further increase in temperature. Table 1 illustrates the pressure-temperature relationship and various other properties of steam.

Temperature is the thermal state of both liquid and vapor at any given pressure. The values shown in Table 1 are for dry saturated steam. The vapor temperature can be raised by adding more heat, resulting in superheated steam, which is used (1) where higher temperatures are required, (2) in large distribution systems to compensate for heat losses and to ensure that steam is delivered at the desired saturated pressure and temperature, and (3) to ensure that the steam is dry and contains no entrained liquid that could damage some turbine-driven equipment.

Enthalpy of the liquid h_f (sensible heat) is the amount of heat in Btu required to raise the temperature of a pound of water from 32°F to the boiling point at the pressure indicated.

Enthalpy of evaporation h_{fg} (latent heat of vaporization) is the amount of heat required to change a pound of boiling water at a given pressure to a pound of steam at the same pressure. This same amount of heat is released when the vapor is condensed back to a liquid.

Enthalpy of the steam h_g (total heat) is the combined enthalpy of liquid and vapor and represents the total heat above 32°F in the steam.

Specific volume, the reciprocal of density, is the volume of unit mass and indicates the volumetric space that 1 lb of steam or water occupies.

An understanding of the above helps explain some of the following unique properties and advantages of steam:

- Most of the heat content of steam is stored as latent heat, which permits large quantities of heat to be transmitted efficiently with little change in temperature. Because the temperature of saturated steam is pressure-dependent, a negligible temperature reduction occurs from the reduction in pressure caused by pipe friction losses as steam flows through the system. This occurs regardless of the insulation efficiency, as long as the boiler maintains the initial pressure and the steam traps remove the condensate. Conversely, in a hydronic system, inadequate insulation can significantly reduce fluid temperature.
- Steam, as all fluids, flows from areas of high pressure to areas of low pressure and is able to move throughout a system without an external energy source. Heat dissipation causes the vapor to condense, which creates a reduction in pressure caused by the dramatic change in specific volume (1600:1 at atmospheric pressure).
- As steam gives up its latent heat at the terminal equipment, the condensate that forms is initially at the same pressure and temperature as the steam. When this condensate is discharged to a lower

The preparation of this chapter is assigned to TC 6.1, Hydronic and Steam Equipment and Systems.

Table 1 Properties of Saturated Steam

Pressure, psi		Saturation Temperature, °F	Specific Volume, ft^3/lb		Enthalpy, Btu/lb		
Gage	Absolute		Liquid v_f	Steam v_g	Liquid h_f	Evaporation h_{fg}	Steam h_g
25 in. Hg vac.	2.47	134	0.0163	142.2	101	1018	1119
9.6 in. Hg vac.	10.0	193	0.0166	38.4	161	982	1143
0	14.7	212	0.0167	26.8	180	970	1150
2	16.7	218	0.0168	23.8	187	966	1153
5	19.7	227	0.0168	20.4	195	961	1156
15	29.7	250	0.0170	13.9	218	946	1164
50	64.7	298	0.0174	6.7	267	912	1179
100	114.7	338	0.0179	3.9	309	881	1190
150	164.7	366	0.0182	2.8	339	857	1196
200	214.7	388	0.0185	2.1	362	837	1200

Note: Values are rounded off or approximated to illustrate various properties discussed in text. For calculation and design, use values of thermodynamic properties of water shown in Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals* or a similar table.

pressure (as when a steam trap passes condensate to the return system), the condensate contains more heat than necessary to maintain the liquid phase at the lower pressure; this excess heat causes some of the liquid to vaporize or "flash" to steam at the lower pressure. The amount of liquid that flashes to steam can be calculated as follows:

$$\% \text{ Flash Steam } = \frac{100(h_{f1} - h_{f2})}{h_{fg2}} \tag{1}$$

where

h_{f1} = enthalpy of liquid at pressure p_1
h_{f2} = enthalpy of liquid at pressure p_2
h_{fg2} = latent heat of vaporization at pressure p_2

Flash steam contains significant and useful heat energy that can be recovered and used (see the section on Heat Recovery). This reevaporation of condensate can be controlled (minimized) by subcooling the condensate within the terminal equipment before it discharges into the return piping. The volume of condensate that is subcooling should not be so large as to cause a significant loss of heat transfer (condensing) surface.

EFFECTS OF WATER, AIR, AND GASES

The enthalpies shown in Table 1 are for dry saturated steam. Most systems operate near these theoretically available values, but the presence of water and gases can affect enthalpy, as well as have other adverse operating effects.

Dry saturated steam is pure vapor without entrained water droplets. However, some amount of water usually carries over as condensate forms because of heat losses in the distribution system. **Steam quality** describes the amount of water present and can be determined by calorimeter tests.

While steam quality might not have a significant effect on the heat transfer capabilities of the terminal equipment, the backing up or presence of condensate can be significant because the enthalpy of condensate h_f is negligible compared with the enthalpy of evaporation h_{fg}. If condensate does not drain properly from pipes and coils, the rapidly flowing steam can push a slug of condensate through the system. This can cause water hammer and result in objectionable noise and damage to piping and system components.

The presence of air also reduces steam temperature. Air reduces heat transfer because it migrates to and insulates heat transfer surfaces. Further, oxygen in the system causes pitting of iron and steel surfaces. Carbon dioxide (CO_2) traveling with steam dissolves in condensate, forming **carbonic acid**, which is extremely corrosive to steam heating pipes and heat transfer equipment.

The combined adverse effects of water, air, and CO_2 necessitate their prompt and efficient removal.

HEAT TRANSFER

The quantity of steam that must be supplied to a heat exchanger to transfer a specific amount of heat is a function of (1) the steam temperature and quality, (2) the character and entering and leaving temperatures of the medium to be heated, and (3) the heat exchanger design. For a more detailed discussion of heat transfer, see Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals*.

BASIC STEAM SYSTEM DESIGN

Because of the various codes and regulations governing the design and operation of boilers, pressure vessels, and systems, steam systems are classified according to operating pressure. **Low-pressure systems** operate up to 15 psig, and **high-pressure systems** operate over 15 psig. There are many subclassifications within these broad classifications, especially for heating systems such as one- and two-pipe, gravity, vacuum, or variable vacuum return systems. However, these subclassifications relate to the distribution system or temperature-control method. Regardless of classification, all steam systems include a source of steam, a distribution system, and terminal equipment, where steam is used as the source of power or heat.

STEAM SOURCE

Steam can be generated directly by boilers using oil, gas, coal, wood, or waste as a fuel source, or by solar, nuclear, or electrical energy as a heat source. Steam can be generated indirectly by recovering heat from processes or equipment such as gas turbines and diesel or gas engines. The cogeneration of electricity and steam should always be considered for facilities that have year-round steam requirements. Where steam is used as a power source (such as in turbine-driven equipment), the exhaust steam may be used in heat transfer equipment for process and space heating.

Steam can be provided by a facility's own boiler or cogeneration plant or can be purchased from a central utility serving a city or specific geographic area. This distinction can be very important. A facility with its own boiler plant usually has a closed-loop system and requires the condensate to be as hot as possible when it returns to the boiler. Conversely, condensate return pumps require a few degrees of subcooling to prevent cavitation or flashing of condensate to vapor at the suction eye of pump impellers. The degree of subcooling varies, depending on the hydraulic design or characteristics of the pump in use. [See the section on Pump Suction Characteristics (NPSH) in Chapter 43.]

Central utilities often do not take back condensate, so it is discharged by the using facility and results in an open-loop system. If a utility does take back condensate, it rarely gives credit for its heat content. If condensate is returned at 180°F, and a heat recovery system reduces this temperature to 80°F, the heat remaining in the condensate represents 10 to 15% of the heat purchased from the utility. Using this heat effectively can reduce steam and heating costs by 10% or more (see the section on Heat Recovery).

Boilers

Fired and waste heat boilers are usually constructed and labeled according to the ASME *Boiler and Pressure Vessel Code* because pressures normally exceed 15 psig. Details on design, construction, and application of boilers can be found in Chapter 31. Boiler selection is based on the combined loads, including heating processes and equipment that use steam, hot water generation, piping losses, and pickup allowance.

The Hydronics Institute standards (HYDI 1989) are used to test and rate most low-pressure heating boilers that have net and gross ratings. In smaller systems, selection is based on a net rating. Larger system selection is made on a gross load basis. The occurrence and nature of the load components, with respect to the total load, determine the number of boilers used in an installation.

Heat Recovery and Waste Heat Boilers

Steam can be generated by waste heat, such as exhaust from fuel-fired engines and turbines. Figure 1 schematically shows a typical exhaust boiler and heat recovery system used for diesel engines. A portion of the water used to cool the engine block is diverted as preheated makeup water to the exhaust heat boiler to obtain maximum heat and energy efficiency. Where the quantity of steam generated by the waste heat boiler is not steady or ample enough to satisfy the facility's steam requirements, a conventional boiler must generate supplemental steam.

Heat Exchangers

Heat exchangers are used in most steam systems. Steam-to-water heat exchangers (sometimes called converters or storage tanks with steam heating elements) are used to heat domestic hot water and to supply the terminal equipment. These heat exchangers are the plate type or the shell-and-tube type, where the steam is admitted to the shell and the water is heated as it circulates through the tubes. Condensate coolers (water-to-water) are sometimes used to subcool the condensate while reclaiming the heat energy.

Water-to-steam heat exchangers (steam generators) are used in high-temperature water (HTW) systems to provide process steam. Such heat exchangers generally consist of a U-tube bundle, through which the HTW circulates, installed in a tank or pressure vessel.

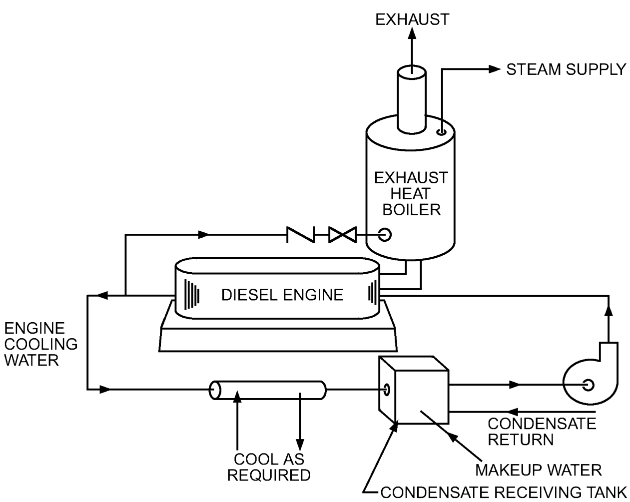

Fig. 1 Exhaust Heat Boiler

All heat exchangers should be constructed and labeled according to the applicable ASME *Boiler and Pressure Vessel Code*. Many jurisdictions require double-wall construction in shell-and-tube heat exchangers between the steam and potable water. Chapter 43 discusses heat exchangers in detail.

BOILER CONNECTIONS

Figure 2 shows recommended boiler connections for pumped and gravity return systems; local codes should be checked for specific legal requirements.

Supply Piping

Small boilers usually have one steam outlet connection sized to reduce steam velocity to minimize carryover of water into supply lines. Large boilers can have several outlets that minimize boiler water entrainment. The boiler manufacturer's recommendations concerning near-boiler piping should be followed because this piping may act as a steam/liquid separator for the boiler.

Figure 2 shows piping connections to the steam header. Although some engineers prefer to use an enlarged steam header for additional storage space, if there is no sudden demand for steam except during the warm-up period, an oversized header may be a disadvantage. The boiler header can be the same size as the boiler connection or the pipe used on the steam main. The horizontal runouts from the boiler(s) to the header should be sized by calculating the heaviest load that will be placed on the boiler(s). The runouts should be sized on the same basis as the building mains. Any change in size after the vertical uptakes should be made by reducing elbows.

Return Piping

Cast-iron boilers have return tappings on both sides, and steel boilers may have one or two return tappings. Where two tappings are provided, both should be used to effect proper circulation through the boiler. Condensate in boilers can be returned by a pump or a gravity return system. Return connections shown in Figure 2 for a multiple-boiler gravity return installation may not always maintain the correct water level in all boilers. Extra controls or accessories may be required.

Recommended return piping connections for systems using gravity return are detailed in Figure 3. Dimension A must be at least 28 in. for each 1 psig maintained at the boiler to provide the pressure required to return the condensate to the boiler. To provide a reasonable safety factor, make dimension A at least 14 in. for small systems with piping sized for a pressure drop of 1/8 psi, and at least 28 in. for larger systems with piping sized for a pressure drop of 0.5 psi. The **Hartford loop** protects against a low water condition, which can occur if a leak develops in the wet return portion of the piping. The Hartford loop takes the place of a check valve on the wet return; however, certain local codes require check valves. Because of hydraulic pressure limitations, gravity return systems are only suitable for systems operating at a boiler pressure between 0.5 and 1 psig. However, because these systems have minimum mechanical equipment and low initial installed cost, they are appropriate for many small systems. Kremers (1982) and Stamper and Koral (1979) provide additional design information on piping for gravity return systems.

Recommended piping connections for steam boilers with pump-returned condensate are shown in Figure 2. Common practice provides an individual condensate or boiler feedwater pump for each boiler. Pump operation is controlled by the boiler water level control on each boiler. However, one pump may be connected to supply the water to each boiler from a single manifold by using feedwater control valves regulated by the individual boiler water level controllers. When such systems are used, the condensate return pump runs continuously to pressurize the return header.

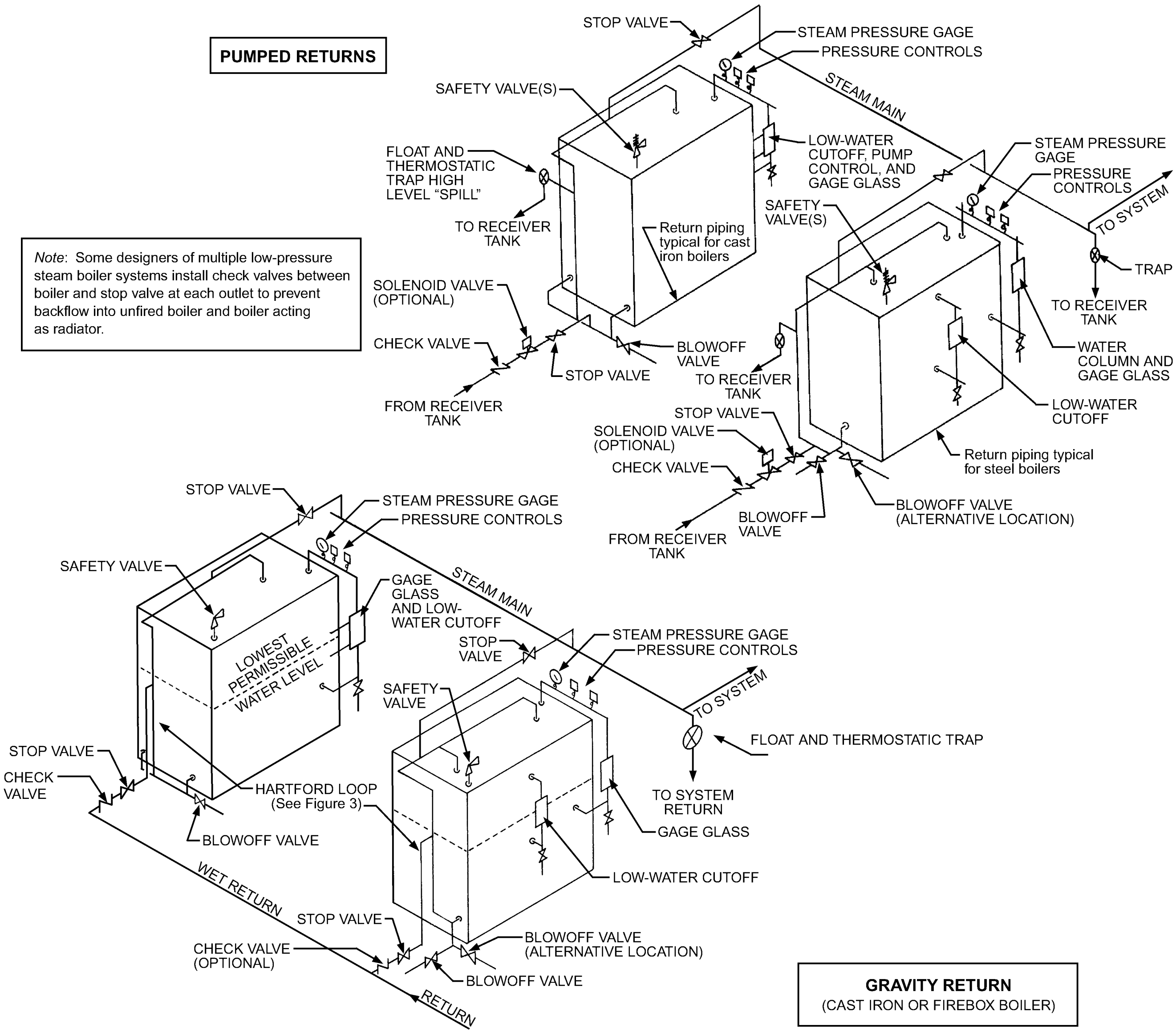

Fig. 2 Typical Boiler Connections

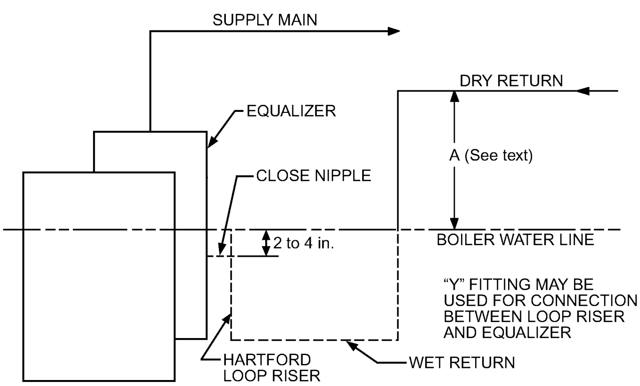

Fig. 3 Boiler with Gravity Return

Return piping should be sized based on total load. The line between the pump and boiler should be sized for a very small pressure drop and the maximum pump discharge flow rate.

DESIGN STEAM PRESSURE

One of the most important decisions in the design of a steam system is the selection of the generating, distribution, and utilization pressures. Considering investment cost, energy efficiency, and control stability, the pressure should be held to the minimum values above atmospheric pressure that accomplish the required heating task, unless detailed economic analysis indicates advantages in higher pressure generation and distribution.

The first step in selecting pressures is to analyze the load requirements. Space heating and domestic water heating can best be served, directly or indirectly, with low-pressure steam less than 15 psig or 250°F. Other systems that can be served with low-pressure steam include single-stage absorption units (10 psig), cooking, warming, dishwashing, and snow melting heat exchangers. Thus, from the standpoint of load requirements, high-pressure steam (above 15 psig) is required only for loads such as dryers, presses, molding dies, power drives, and other processing, manufacturing, and power requirements. The load requirement establishes the pressure requirement.

When the source is close to the load(s), the generation pressure should be high enough to provide the (1) load design pressure, (2) friction losses between the generator and the load, and (3) control range. Losses are caused by flow through the piping, fittings, control valves, and strainers. If the generator(s) is located remote from the loads, there could be some economic advantage in distributing the steam at a higher pressure to reduce pipe size. When this is considered, the economic analysis should include the additional investment and operating costs associated with a higher pressure generation system. When an increase in the generating pressure requires a change from below to above 15 psig, the generating system equipment changes from low-pressure class to high-pressure class and there are significant increases in both investment and operating cost.

Where steam is provided from a nonfired device or prime mover such as a diesel engine cooling jacket, the source device can have an inherent pressure limitation.

PIPING

The piping system distributes the steam, returns the condensate, and removes air and noncondensable gases. In steam heating systems, it is important that the piping distribute steam, not only at full design load, but at partial loads and excess loads that can occur on system warm-up. The usual average winter steam demand is less than half the demand at the lowest outdoor design temperature. However, when the system is warming up, the load on the steam mains and returns can exceed the maximum operating load for the coldest design day, even in moderate weather. This load comes from raising the temperature of the piping to the steam temperature and that of the building to the indoor design temperature. Supply and return piping should be sized according to Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*.

Supply Piping Design Considerations

1. Size pipe according to Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*, taking into consideration pressure drop and steam velocity.
2. Pitch piping uniformly down in the direction of steam flow at 0.25 in. per 10 ft. If piping cannot be pitched down in the direction of the steam flow, refer to Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals* for rules on pipe sizing and pitch.
3. Insulate piping well to avoid unnecessary heat loss (see Chapters 23 and 24 of the 2005 *ASHRAE Handbook—Fundamentals*).
4. Condensate from unavoidable heat loss in the distribution system must be removed promptly to eliminate water hammer and degradation of steam quality and heat transfer capability. Install drip legs at all low points and natural drainage points in the system, such as at the ends of mains and the bottoms of risers, and ahead of pressure regulators, control valves, isolation valves, pipe bends, and expansion joints. On straight horizontal runs with no natural drainage points, space drip legs at intervals not exceeding 300 ft when the pipe is pitched down in the direction of the steam flow and at a maximum of 150 ft when the pipe is pitched up, so that condensate flow is opposite of steam flow. These distances apply to systems where valves are opened manually to remove air and excess condensate that forms during warm-up conditions. Reduce these distances by about half in systems that are warmed up automatically.
5. Where horizontal piping must be reduced in size, use eccentric reducers that permit the continuance of uniform pitch along the bottom of piping (in downward pitched systems). Avoid concentric reducers on horizontal piping, because they can cause water hammer.
6. Take off all branch lines from the top of the steam mains, preferably at a 45° angle, although vertical 90° connections are acceptable.
7. Where the length of a branch takeoff is less than 10 ft, the branch line can be pitched back 0.5 in. per 10 ft, providing drip legs as described previously in (4).
8. Size drip legs properly to separate and collect the condensate. Drip legs at vertical risers should be full-size and extend beyond the riser, as shown in Figure 4. Drip legs at other locations should be the same diameter as the main. In steam mains 6 in. and over, this can be reduced to half the diameter of the main, but to no less than 4 in. Where warm-up is supervised, the length of the collecting leg is not critical. However, the recommended length is 1.5 times the pipe diameter and not less than 8 in. For automatic warm-up, collecting legs should always be the same size as the main and should be at least 28 in. long to provide the hydraulic pressure differential necessary for the trap to discharge before a positive pressure is built up in the steam main.
9. Condensate should flow by gravity from the trap to the return piping system. Where the steam trap is located below the return line, the condensate must be lifted. In systems operating above 40 psig, the trap discharge can usually be piped directly to the return system (Figure 5). However, back pressure at the trap discharge (return line pressure plus hydraulic pressure created by height of lift) must not exceed steam main pressure, and the trap must be sized after considering back pressure. A collecting leg must be used and the trap discharge must flow by gravity to a vented condensate receiver, from which it is pumped to the overhead return in systems (1) operating under 40 psig, (2) where the temperature is regulated by modulating the steam control valves,

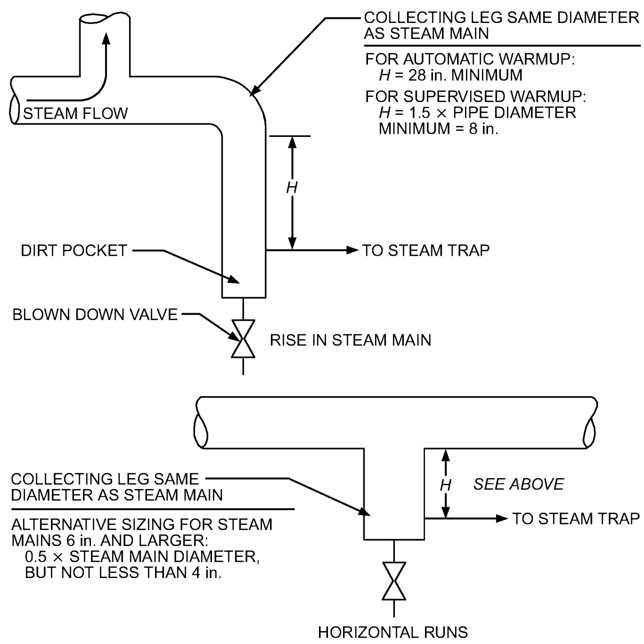

Fig. 4 Method of Dripping Steam Mains

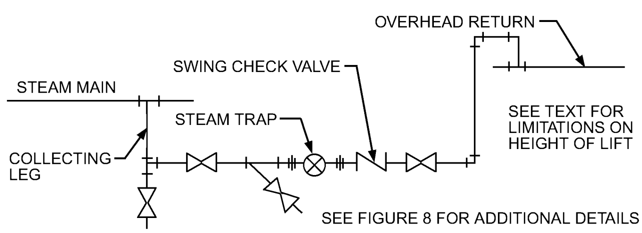

Fig. 5 Trap Discharging to Overhead Return

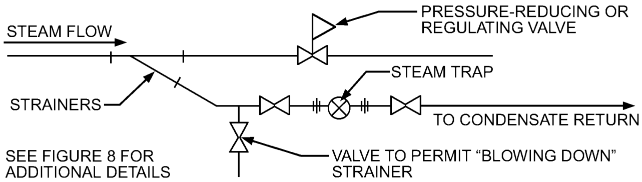

Fig. 6 Trapping Strainers

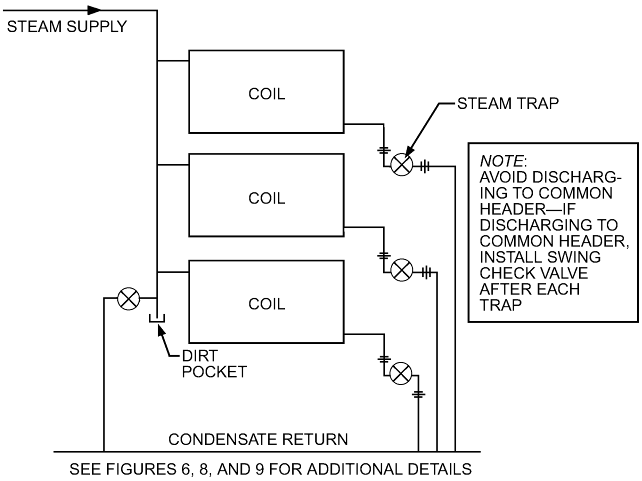

Fig. 7 Trapping Multiple Coils

or (3) where the back pressure at the trap is close to system pressure. The trap discharge should flow by gravity to a vented condensate receiver from which it is pumped to the overhead return.

10. Strainers installed before the pressure-reducing and control valves are a natural water collection point. Since water carryover can erode the valve seat, install a trap at the strainer blowdown connection (Figure 6).

Terminal Equipment Piping Design Considerations

1. Size piping the same as the supply and return connections of the terminal equipment.
2. Keep equipment and piping accessible for inspection and maintenance of the steam traps and control valves.
3. Minimize strain caused by expansion and contraction with pipe bends, loops, or three elbow swings to take advantage of piping flexibility, or with expansion joints or flexible pipe connectors.
4. In multiple-coil applications, separately trap each coil for proper drainage (Figure 7). Piping two or more coils to a common header served by a single trap can cause condensate backup, improper heat transfer, and inadequate temperature control.
5. Terminal equipment, where temperature is regulated by a modulating steam control valve, requires special consideration. Refer to the section on Condensate Removal from Temperature-Regulated Equipment.

Return Piping Design Considerations

1. Flow in the return line is two-phase, consisting of steam and condensate. See Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals* for sizing considerations.
2. Pitch return lines downward in the direction of the condensate flow at 0.5 in. per 10 ft to ensure prompt condensate removal.
3. Insulate the return line well, especially where the condensate is returned to the boiler or the condensate enthalpy is recovered. In vacuum systems, the return lines are not insulated since condensate subcooling is required.

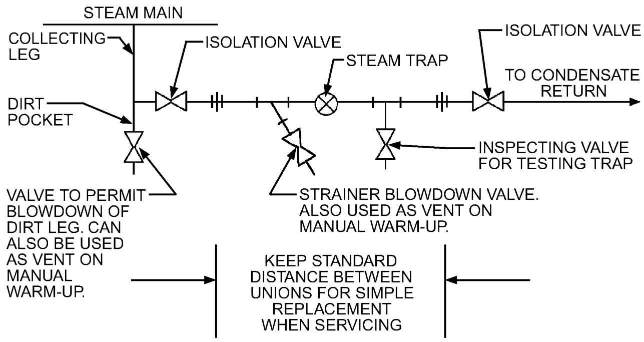

Fig. 8 Recommended Steam Trap Piping

4. Where possible and practical, use heat recovery systems to recover the condensate enthalpy. See the section on Heat Recovery. In vacuum systems, the return lines are not insulated since condensate subcooling is required.
5. Equip dirt pockets of the drip legs and strainer blowdowns with valves to remove dirt and scale.
6. Install steam traps close to drip legs and make them accessible for inspection and repair. Servicing is simplified by making the pipe sizes and configuration identical for a given type and size of trap. The piping arrangement in Figure 8 facilitates inspection and maintenance of steam traps.
7. When elevating condensate to an overhead return, consider the pressure at the trap inlet and the fact that it requires approximately 1 psi to elevate condensate 2 ft. See (9) in the section on Supply Piping Design Considerations for a complete discussion.

CONDENSATE REMOVAL FROM TEMPERATURE-REGULATED EQUIPMENT

When air, water, or another product is heated, the temperature or heat transfer rate can be regulated by a modulating steam pressure control valve. Because pressure and temperature do not vary at the same rate as load, the steam trap capacity, which is determined by the pressure differential between the trap inlet and outlet, may be adequate at full load, but not at some lesser load.

Analysis shows that steam pressure must be reduced dramatically to achieve a slight lowering of temperature. In most applications, this can result in subatmospheric pressure in the coil, while as much as 75% of the full condensate load has to be handled by the steam trap. This is especially important for coils exposed to outside air, because subatmospheric conditions can occur in the coil at outside temperatures below 32°F, and the coil will freeze if the condensate is not removed.

Armers (1985) provides detailed methods for determining condensate load under various operating conditions. However, in most cases, this load need not be calculated if the coils are piped as shown in Figure 9 and this procedure is followed:

1. Place the steam trap 1 to 3 ft below the bottom of the steam coil to provide a pressure of approximately 0.5 to 1.5 psig. Locating the trap at less than 12 in. usually results in improper drainage and operating difficulties.
2. Install vacuum breakers between the coil and trap inlet to ensure that the pressure can drain the coil when it is atmospheric or subatmospheric. The vacuum breaker should respond to a differential pressure of no greater than 3 in. of water. For atmospheric returns, the vacuum breaker should be opened to the atmosphere, and the return system must be designed to ensure no pressurization of the return line. In vacuum return systems, the vacuum breaker should be piped to the return line.
3. Discharge from the trap must flow by gravity, without any lifts in the piping, to the return system, which must be vented properly

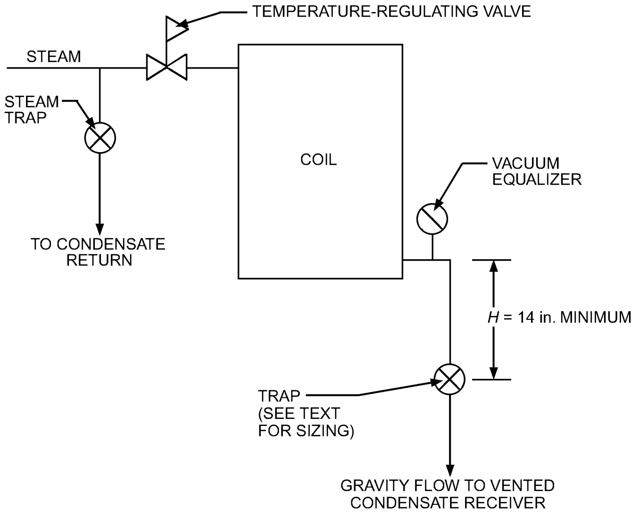

Fig. 9 Trapping Temperature-Regulated Coils

to the atmosphere to eliminate any back pressure that could prevent the trap from draining the coil. Where the return main is overhead, the trap discharge should flow by gravity to a vented receiver, from which it is then pumped to the overhead return.

4. Design traps to operate at maximum pressure at the control valve inlet and size them to handle the full condensate load at a pressure differential equal to the hydraulic pressure between the trap and coil. The actual condensate load can vary from the theoretical design load because of the safety factors used in coil selection and the fact that condensate does not always form at a uniform steady rate; therefore, size steam traps according to the following:

   For an actual steam pressure p_1 in the coil at full condensate load w, the proportion X of full load needing atmospheric pressure at the coil is

$$X = \frac{212 - t_c}{t_s - t_c} \quad (2)$$

   where

   t_c = control temperature, °F
   t_s = steam temperature at p_1, °F

   Then, the steam trap must be sized both to pass the full load w at a differential pressure equal to p_1 and to pass $X \cdot w$ (the proportion of full load) at 0.5 psi.
5. To reduce the possibility of a steam coil freezing, the temperature-regulating valve is often left wide open, and the leaving air temperature is controlled by a face-and-bypass damper on the steam coil.
6. For air temperatures below freezing, traps selected for draining steam coils should fail open (e.g., bucket traps) or two traps in parallel should be used.

STEAM TRAPS

Steam traps are an essential part of all steam systems, except one-pipe steam heating systems. Traps discharge condensate, which forms as steam gives up some of its heat, and direct the air and noncondensable gases to a point of removal. Condensate forms in steam mains and distribution piping because of unavoidable heat losses through less-than-perfect insulation, as well as in terminal equipment such as radiators, convectors, fan-coil units, and heat exchangers, where steam gives up heat during normal operation. Condensate must always be removed from the system as soon as it accumulates for the following reasons:

- Although condensate contains some valuable heat, using this heat by holding the condensate in the terminal equipment reduces the heat transfer surface. It also causes other operating problems because it retains air, which further reduces heat transfer, and noncondensable gases such as CO_2, which cause corrosion. As discussed in the section on Steam Source, recovering condensate heat is usually only desirable when the condensate is not returned to the boiler. Methods for this are discussed in the section on Heat Recovery.
- Steam moves rapidly in mains and supply piping, so when condensate accumulates to the point where the steam can push a slug of it, serious damage can occur from the resulting water hammer.

Ideally, the steam trap should remove all condensate promptly, along with air and noncondensable gases that might be in the system, with little or no loss of live steam. A steam trap is an automatic valve that can distinguish between steam and condensate or other fluids. Traps are classified as follows:

- **Thermostatic traps** react to the difference in the temperatures of steam and condensate.
- **Mechanical traps** depend on the difference in the densities of steam and condensate.
- **Kinetic traps** rely on the difference in the flow characteristics of steam and condensate.

The following points apply to all steam traps:

- No single type of steam trap is best suited to all applications, and most systems require more than one type of trap.
- Steam traps, regardless of type, should be carefully sized for the application and condensate load to be handled, because both undersizing and oversizing can cause serious problems. Undersizing can result in undesirable condensate backup and excessive cycling, which can lead to premature failure. Oversizing might appear to solve this problem and make selection much easier because fewer different sizes are required, but if the trap fails, excessive steam can be lost.

Steam traps should be installed between two unions to facilitate maintenance and/or replacement.

Thermostatic Traps

In thermostatic traps, a bellows or bimetallic element operates a valve that opens in the presence of subcooled condensate and closes in the presence of steam (Figure 10). Because condensate is initially at the same temperature as the steam from which it was condensed, the thermostatic element must be designed and calibrated to open at a temperature below the steam temperature; otherwise, the trap would blow live steam continuously. Therefore, the condensate is subcooled by allowing it to back up in the trap and in a portion of the upstream drip leg piping, both of which are left uninsulated. Some thermostatic traps operate with a continuous water leg behind the trap so there is no steam loss; however, this prohibits the discharge of air and noncondensable gases, and can cause excessive condensate to back up into the mains or terminal equipment, thereby resulting in operating problems. Devices that operate without significant backup can lose steam before the trap closes.

Although both bellows and bimetallic traps are temperature-sensitive, their operations are significantly different. The **bellows thermostatic trap** has a fluid with a lower boiling point than water. When the trap is cold, the element is contracted and the discharge port is open. As hot condensate enters the trap, it causes the contents of the bellows to boil and vaporize before the condensate temperature rises to steam temperature. Because the contents of the bellows boil at a lower temperature than water, the vapor pressure inside the bellows element is greater than the steam pressure outside, causing the element to expand and close the discharge port.

Assuming the contained liquid has a pressure-temperature relationship similar to that of water, the balance of forces acting on the bellows element remains relatively constant, no matter how the steam pressure varies. Therefore, this is a balanced pressure device that can be used at any pressure within the operating range of the device. However, this device should not be used where superheated steam is present, because the temperature is no longer in step with the pressure and damage or rupture of the bellows element can occur.

Bellows thermostatic traps are best suited for steady light loads on low-pressure service. They are most widely used in radiators and convectors in HVAC applications.

The **bimetallic thermostatic trap** has an element made from metals with different expansion coefficients. Heat causes the element to change shape, permitting the valve port to open or close. Because a bimetallic element responds only to temperature, most traps have the valve on the outlet so that steam pressure is trying to open the valve. Therefore, by properly designing the bimetallic element, the trap can operate on a pressure-temperature curve approaching the steam saturation curve, although not as closely as a balanced pressure bellows element.

Unlike the bellows thermostatic trap, bimetallic thermostatic traps are not adversely affected by superheated steam or subject to damage by water hammer, so they can be readily used for high-pressure applications. They are best suited for steam tracers, jacketed piping, and heat transfer equipment, where some condensate backup is tolerable. If they are used on steam main drip legs, the element should not back up condensate.

Mechanical Traps

Mechanical traps are buoyancy operated, depending on the difference in density between steam and condensate. The **float and thermostatic trap** (Figure 10) is commonly called the F&T trap and is actually a combination of two types of traps in a single trap body: (1) a bellows thermostatic element operating on temperature difference, which provides automatic venting, and (2) a float portion, which is buoyancy operated. Float traps without automatic venting should not be used in steam systems.

On start-up, the float valve is closed and the thermostatic element is open for rapid air venting, permitting the system or equipment to rapidly fill with steam. When steam enters the trap body, the thermostatic element closes and, as condensate enters, the float rises and the condensate discharges. The float regulates the valve opening so it continuously discharges the condensate at the rate at which it reaches the trap.

The F&T trap has large venting capabilities, continuously discharges condensate without backup, handles intermittent loads very well, and can operate at extremely low pressure differentials. Float and thermostatic traps are suited for use with temperature-regulated steam coils. They also are well suited for steam main and riser drip legs on low-pressure steam heating systems. Although F&T traps are available for pressures to 250 psig or higher, they are susceptible to water hammer, so other traps are usually a better choice for high-pressure applications.

Bucket traps operate on buoyancy, but they use a bucket that is either open at the top or inverted instead of a closed float. Initially, the bucket in an open bucket trap is empty and rests on the bottom of the trap body with the discharge vented, and, as condensate enters the trap, the bucket floats up and closes the discharge port. Additional condensate overflows into the bucket, causing it to sink and open the discharge port, which allows steam pressure to force the condensate out of the bucket. At the same time, it seals the bottom of the discharge tube, prohibiting air passage. Therefore, to prevent air binding, this device has an automatic air vent, as does the F&T trap.

Inverted bucket traps eliminate the size and venting problems associated with open bucket traps. Steam or air entering the submerged inverted bucket causes it to float and close the discharge port. As more condensate enters the trap, it forces air and steam out of the vent on top of the inverted bucket into the trap body where the steam condenses by cooling. When the mass of the bucket exceeds the buoyancy effect, the bucket drops, opening the discharge port, and steam pressure forces the condensate out, and the cycle repeats.

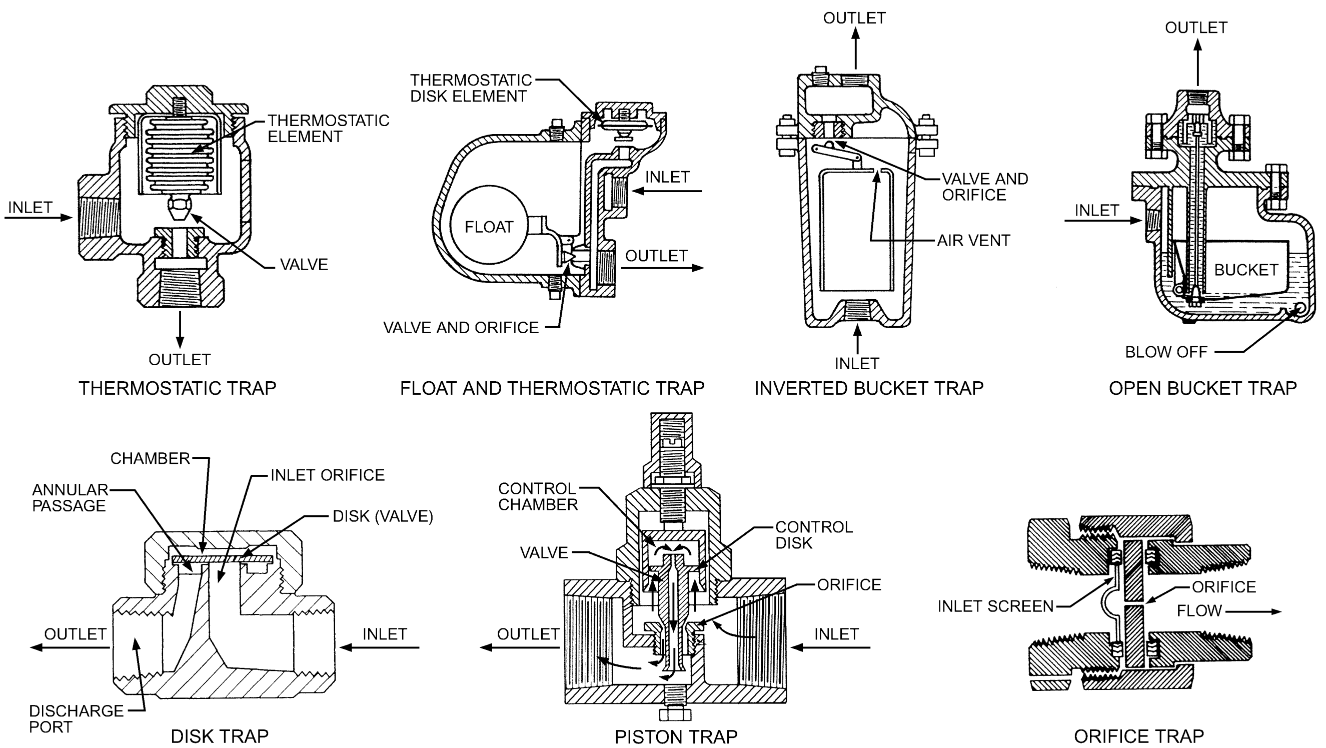

Fig. 10 Thermostatic Traps

Unlike most cycling-type traps, the inverted bucket trap continuously vents air and noncondensable gases. Although it discharges condensate intermittently, there is no condensate backup in a properly sized trap. Inverted bucket traps are made for all pressure ranges and are well suited for steam main drip legs and most HVAC applications. Although inverted bucket traps can be used for temperature-regulated steam coils, the F&T trap is usually better because it has the high venting capability desirable for such applications.

Kinetic Traps

Numerous devices operate on the difference between the flow characteristics of steam and condensate and on the fact that condensate discharging to a lower pressure contains more heat than necessary to maintain the liquid phase. This excess heat causes some of the condensate to flash to steam at the lower pressure.

Thermodynamic traps or **disk traps** are simple devices with only one moving part. When air or condensate enters the trap on start-up, it lifts the disk off its seat and is discharged. When steam or hot condensate (some of which "flashes" to steam upon exposure to a lower pressure) enters the trap, the increased velocity of this vapor flow decreases the pressure on the underside of the disk and increases the pressure above the disk, causing it to snap shut. Pressure is then equalized above and below the disk, but because the area exposed to pressure is greater above than below it, the disk remains shut until the pressure above is reduced by condensing or bleeding, thus permitting the disk to snap open and repeat the cycle. This device does not cycle open and shut as a function of condensate load, it is a time-cycle device that opens and shuts at fixed intervals as a function of how fast the steam above the disk condenses. Because disk traps require a significant pressure differential to operate properly, they are not well suited for low-pressure systems or for systems with significant back pressure. They are best suited for high-pressure systems and are widely applied to steam main drip legs.

Impulse traps or **piston traps** continuously pass a small amount of steam or condensate through a "bleed" orifice, changing the pressure positions within the piston. When live steam or very hot condensate that flashes to steam enters the control chamber, the increased pressure closes the piston valve port. When cooler condensate enters, the pressure decreases, permitting the valve port to open. Most impulse traps cycle open and shut intermittently, but some modulate to a position that passes condensate continuously.

Impulse traps can be used for the same applications as disk traps; however, because they have a small "bleed" orifice and close piston tolerances, they can stick or clog if dirt is present in the system.

Orifice traps have no moving parts. All other traps have discharge ports or orifices, but in the traps described previously, this opening is oversized, and some type of closing mechanism controls the flow of condensate and prevents the loss of live steam.

The orifice trap has no such closing mechanism, and the flow of steam and condensate is controlled by two-phase flow through an orifice. A simple explanation of this theory is that an orifice of any size has a much greater capacity for condensate than it does for steam because of the significant differences in their densities and because "flashing" condensate tends to choke the orifice. An orifice is selected larger than required for the actual condensate load; therefore, it continuously passes all condensate along with the air and noncondensable gases, plus a small controlled amount of steam. The steam loss is usually comparable to that of most cycling-type traps.

Orifice traps must be sized more carefully than cycling-type traps. On light condensate loads, the orifice size is small and, like impulse traps, tends to clog. Orifice traps are suitable for all system pressures and can operate against any back pressure. They are best suited for steady pressure and load conditions such as steam main drip legs.

PRESSURE-REDUCING VALVES

Where steam is supplied at pressures higher than required, one or more pressure-reducing valves (pressure regulators) are required. The pressure-reducing valve reduces pressure to a safe point and regulates pressure to that required by the equipment. The district heating industry refers to valves according to their functional use. There are two classes of service: (1) the steam must be shut off completely (dead-end valves) to prevent buildup of pressure on the low-pressure side during no load (single-seated valves should be used) and (2) the low-pressure lines condense enough steam to prevent buildup of pressure from valve leakage (double-seated valves can be used). Valves available for either service are direct-operated, spring-loaded, weight-loaded, air-loaded, or pilot-controlled, using either the flowing steam or auxiliary air or water as the operating medium. The direct-operated, double-seated valve is less affected by varying inlet steam pressure than the direct-operated, single-seated valve. Pilot-controlled valves, either single- or double-seated, tend to eliminate the effect of variable inlet pressures.

Installation

Pressure-reducing valves should be readily accessible for inspection and repair. There should be a bypass around each reducing valve equal to the area of the reducing-valve seat ring. The globe valve in a bypass line should have plug disk construction and must have an absolutely tight shutoff. Steam pressure gages, graduated up to the initial pressure, should be installed on the low-pressure side and on the high-pressure side. The low-pressure gage should be ahead of the shutoff valve because the reducing valve can be adjusted with the shutoff valve closed. A similar gage should be installed downstream from the shutoff valve for use during manual operation. Typical service connections are shown in Figure 11 for low-pressure service and Figure 12 for high-pressure service. In the smaller sizes, the standby pressure-regulating valve can be removed, a filler installed, and the inlet stop valves used for manual pressure regulation until repairs are made.

Strainers should be installed on the inlet of the primary pressure-reducing valve and before the second-stage reduction if there is considerable piping between the two stages. If a two-stage reduction is made, it is advisable to install a pressure gage immediately before the reducing valve of the second-stage reduction to set and check the operation of the first valve. A drip trap should be installed before the two reducing valves.

Where pressure-reducing valves are used, one or more relief devices or safety valves must be provided, and the equipment on the low-pressure side must meet the requirements for the full initial pressure. The relief or safety devices are adjoining or as close as possible to the reducing valve. The combined relieving capacity must be adequate to avoid exceeding the design pressure of the low-pressure system if the reducing valve does not open. In most

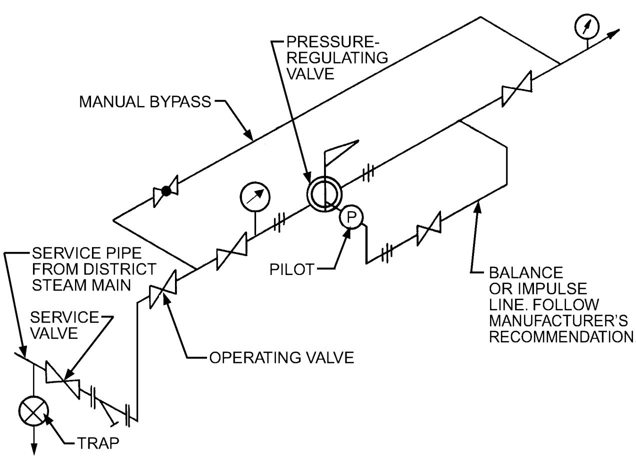

Fig. 11 Pressure-Reducing Valve Connections—Low Pressure

areas, local codes dictate the safety relief valve installation requirements.

Safety valves should be set at least 5 psi higher than the reduced pressure if the reduced pressure is under 35 psig and at least 10 psi higher than the reduced pressure if the reduced pressure is above 35 psig or the first-stage reduction of a double reduction. The outlet from relief valves should not be piped to a location where the discharge can jeopardize persons or property or is a violation of local codes.

Figure 13 shows a typical service installation, with a separate line to various heating zones and process equipment. If the initial pressure is below 50 psig, the first-stage pressure-reducing valve can be omitted. In Assembly A (Figure 13), the single-stage pressure-reducing valve is also the pressure-regulating valve.

When making a two-stage reduction (e.g., 150 to 50 psig and then 50 to 2 psig), allow for the expansion of steam on the low-pressure side of each reducing valve by increasing the pipe area to about double the area of a pipe the size of the reducing valve. This also allows steam to flow at a more uniform velocity. It is recommended that the valves be separated by a distance of at least 20 ft to

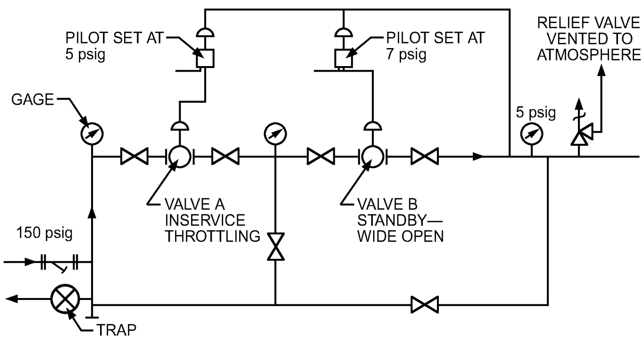

Fig. 12 Pressure-Reducing Valve Connections—High Pressure

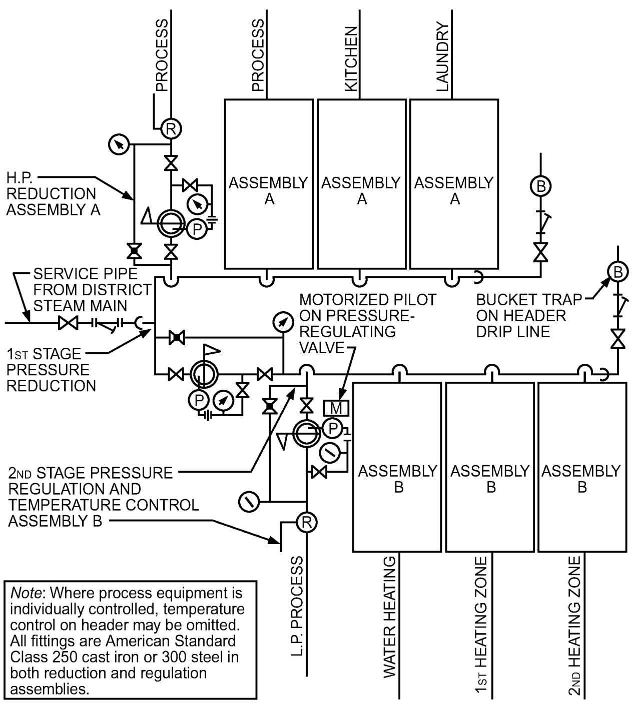

Fig. 13 Steam Supply

reduce excessive hunting action of the first valve, although this should be checked with the valve manufacturer.

Figure 14 shows a typical double-reduction installation where the pressure in the district steam main is higher than can safely be applied to building heating systems. The first or pressure-reducing valve effects the initial pressure reduction. The pressure-regulating valve regulates the steam to the desired final pressure.

Pilot-controlled or air-loaded direct-operated reducing valves can be used without limitation for all reduced-pressure settings for all heating, process, laundry, and hot water services. Spring-loaded direct-operated valves can be used for reduced pressures up to 50 psig, providing they can pass the required steam flow without excessive deviation in reduced pressure.

Weight-loaded valves may be used for reduced pressures below 15 psig and for moderate steam flows.

Pressure-equalizing or impulse lines must be connected to serve the type of valve selected. With direct-operated diaphragm valves having rubber-like diaphragms, the impulse line should be connected to the reduced-pressure steam line to allow for maximum condensate on the diaphragm and in the impulse or equalizing line. If it is connected to the top of the steam line, a condensate accumulator should be used to reduce variations in the pressure of condensate on the diaphragm. The impulse line should not be connected to the bottom of the reduced pressure line since it could become clogged. Equalizing or impulse lines for pilot-controlled and direct-operated reducing valves using metal diaphragms should be connected into the expanded outlet piping approximately 2 to 4 ft from the reducing valve and pitched away from the reducing valve to prevent condensate accumulation. Pressure impulse lines for externally pilot-controlled reducing valves using compressed air or fresh water should be installed according to manufacturer's recommendations.

Valve Size Selection

Pressure-regulating valves should be sized to supply the maximum steam requirements of the heating system or equipment. Consideration should be given to rangeability, speed of load changes, and regulation accuracy required to meet system needs, especially with temperature control systems using intermittent steam flow to heat the building.

The reducing valve should be selected carefully. The manufacturer should be consulted. Piping to and from the reducing valve should be adequate to pass the desired amount of steam at the desired maximum velocity. A common error is to make the size of the reducing valve the same as the service or outlet pipe size; this makes the reducing valve oversized and causes wiredrawing or erosion of the valve and seat because of the high-velocity flow caused by the small lift of the valve.

On installations where the steam requirements are large and variable, wiredrawing and cycling control can occur during mild

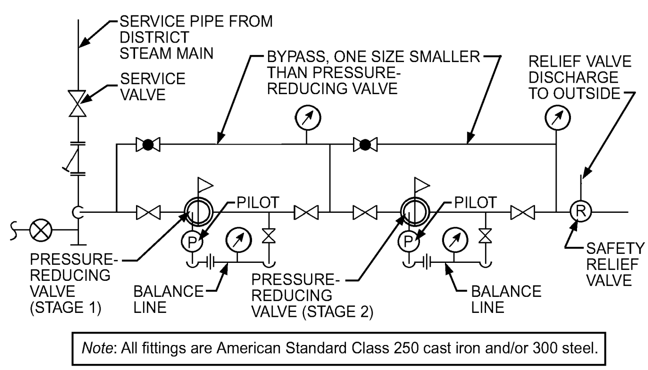

Fig. 14 Two-Stage Pressure-Regulating Valve
(Used where high-pressure steam is supplied for low-pressure requirements)

weather or during reduced demand periods. To overcome this condition, two reducing valves are installed in parallel, with the sizes selected on a 70 and 30% proportion of maximum flow. For example, if 10,000 lb of steam per hour are required, the size of one valve is based on 7000 lb of steam per hour, and the other is based on 3000 lb of steam per hour. During mild weather (spring and fall), the larger valve is set for slightly lower reduced pressure than the smaller one and remains closed as long as the smaller one can meet the demand. During the remainder of the heating season, the valve settings are reversed to keep the smaller one closed, except when the larger one is unable to meet the demand.

TERMINAL EQUIPMENT

A variety of terminal units are used in steam heating systems. All are suited for use on low-pressure systems. Terminal units used on high-pressure systems have heavier construction, but are otherwise similar to those on low-pressure systems. Terminal units are usually classified as follows:

1. **Natural convection units** transfer most heat by convection and some heat by radiation. The equipment includes cast-iron radiators, finned-tube convectors, and cabinet and baseboard units with convection-type elements (see Chapter 35).
2. **Forced-convection units** employ a forced air movement to increase heat transfer and distribute the heated air within the space. The devices include unit heaters, unit ventilators, induction units, fan-coil units, the heating coils of central air-conditioning units, and many process heat exchangers. When such units are used for both heating and cooling, there is a steam coil for heating and a separate chilled water or refrigerant coil for cooling. See Chapters 1, 3, 4, 5, 22, 26, and 27.
3. **Radiant panel systems** transfer some heat by convection. Because of the low-temperature and high-vacuum requirements, this type of unit is rarely used on steam systems.

Selection

The primary consideration in selecting terminal equipment is comfortable heat distribution. The following briefly describes suitable applications for specific types of steam terminal units.

Natural Convection Units

Radiators, convectors, convection-type cabinet units, and baseboard convectors are commonly used for (1) facilities that require heating only, rather than both heating and cooling, and (2) in conjunction with central air conditioning as a source of perimeter heating or for localized heating in spaces such as corridors, entrances, halls, toilets, kitchens, and storage areas.

Forced-Convection Units

Forced-convection units can be used for the same types of applications as natural convection units but are primarily used for facilities that require both heating and cooling, as well as spaces that require localized heating.

Unit heaters are often used as the primary source of heat in factories, warehouses, and garages and as supplemental freeze protection for loading ramps, entrances, equipment rooms, and fresh air plenums.

Unit ventilators are forced-convection units with dampers that introduce controlled amounts of outside air. They are used in spaces with ventilation requirements not met by other system components.

Cabinet heaters are often used in entranceways and vestibules that can have high intermittent heat loads.

Induction units are similar to fan-coil units, but the air is supplied by a central air system rather than individual fans in each unit. Induction units are most commonly used as perimeter heating for facilities with central systems.

Fan-coil units are designed for heating and cooling. On water systems, a single coil can be supplied with hot water for heating and chilled water for cooling. On steam systems, single- or dual-coil units can be used. Single-coil units require steam-to-water heat exchangers to provide hot water to meet heating requirements. Dual-coil units have a steam coil for heating and a separate coil for cooling. The cooling media for dual-coil units can be chilled water from a central chiller or refrigerant provided by a self-contained compressor. Single-coil units require the entire system to be either in a heating or cooling mode and do not permit simultaneous heating and cooling to satisfy individual space requirements. Dual-coil units can eliminate this problem.

Central air-handling units are used for most larger facilities. These units have a fan, a heating and cooling coil, and a compressor if chilled water is not available for cooling. Multizone units are arranged so that each air outlet has separate controls for individual space heating or cooling requirements. Large central systems distribute either warm or cold conditioned air through duct systems and employ separate terminal equipment such as mixing boxes, reheat coils, or variable volume controls to control the temperature to satisfy each space. Central air-handling systems can be factory assembled, field erected, or built at the job site from individual components.

CONVECTION STEAM HEATING

Any system that uses steam as the heat transfer medium can be considered a steam heating system; however, the term "steam heating system" is most commonly applied to convection systems using radiators or convectors as terminal equipment. Other types of steam heating systems use forced convection, in which a fan or air-handling system is used with a convector or steam coil such as unit heaters, fan-coil units and central air-conditioning and heating systems.

Convection-type steam heating systems are used in facilities that have a heating requirement only, such as factories, warehouses, and apartment buildings. They are often used in conjunction with central air-conditioning systems to heat the perimeter of the building. Also, steam is commonly used with incremental units that are designed for cooling and heating and have a self-contained air-conditioning compressor.

Steam heating systems are classified as one-pipe or two-pipe systems, according to the piping arrangement that supplies steam to and returns condensate from the terminal equipment. These systems can be further subdivided by (1) the method of condensate return (gravity flow or mechanical flow by means of condensate pump or vacuum pump) and (2) by the piping arrangement (upfeed or downfeed and parallel or counterflow for one-pipe systems).

One-Pipe Steam Heating Systems

The one-pipe system has a single pipe through which steam flows to and condensate is returned from the terminal equipment (Figure 15). These systems are designed as gravity return, although a condensate pump can be used where there is insufficient height above the boiler water level to develop enough pressure to return condensate directly to the boiler.

A one-pipe system with gravity return does not have steam traps; instead it has air vents at each terminal unit and at the ends of all supply mains to vent the air so the system can fill with steam. In a system with a condensate pump, there must be an air vent at each terminal unit and steam traps at the ends of each supply main.

The one-pipe system with gravity return has low initial cost and is simple to install, because it requires a minimum of mechanical equipment and piping. One-pipe systems are most commonly used in small facilities such as small apartment buildings and office buildings. In larger facilities, the larger pipe sizes required for two-phase flow, problems of distributing steam quickly and evenly throughout the system, the inability to zone the system, and

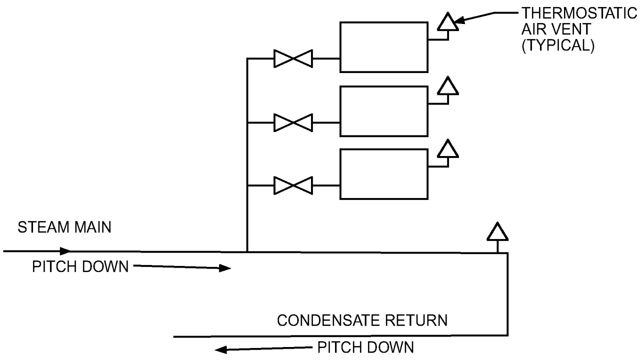

Fig. 15 One-Pipe System

difficulty in controlling the temperature make the one-pipe system less desirable than the two-pipe system.

The heat input to the system is controlled by cycling the steam on and off. In the past, temperature control in individual spaces has been a problem. Many systems have adjustable vents at each terminal unit to help balance the system, but these are seldom effective. A practical approach is to use a self-contained thermostatic valve in series with the air vent (as explained in the section on Temperature Control) that provides limited individual thermostatic control for each space.

Many designers do not favor one-pipe systems because of their distribution and control problems. However, when a self-contained thermostatic valve is used to eliminate the problems, one-pipe systems can be considered for small facilities, where initial cost and simple installation and operation are prime factors.

Most one-pipe gravity return systems are in facilities that have their own boiler, and, because returning condensate must overcome boiler pressure, these systems usually operate from a fraction of a psi to a maximum of 5 psig. The boiler hookup is critical and the Hartford loop (described in the section on Boiler Connections) is used to avoid problems that can occur with boiler low-water condition. Stamper and Koral (1979) and Hoffman give piping design information for one-pipe systems.

Two-Pipe Steam Heating Systems

The two-pipe system uses separate pipes to deliver the steam and return the condensate from each terminal unit (Figure 16). Thermostatic traps are installed at the outlet of each terminal unit to keep the steam in the unit until it gives up its latent heat, at which time the trap cycles open to pass the condensate and permits more steam to enter the radiator. If orifices are installed at the inlet to each terminal unit, as discussed in the sections on Steam Distribution and Temperature Control, and if the system pressure is precisely regulated so only the amount of steam is delivered to each unit that it is capable of condensing, the steam traps can be omitted. However, omitting steam traps is generally not recommended for initial design.

Two-pipe systems can have either gravity or mechanical returns; however, gravity returns are restricted to use in small systems and are generally outmoded. In larger systems that require higher steam pressures to distribute steam, some mechanical means, such as a condensate pump or vacuum pump, must return condensate to the boiler. A **vacuum return system** is used on larger systems and has the following advantages:

- The system fills quickly with steam. The steam in a gravity return system must push the air out of the system, resulting in delayed heat-up and condensate return that can cause low-water problems. A vacuum return system can eliminate these problems.
- The steam supply pressure can be lower, resulting in more efficient operation.

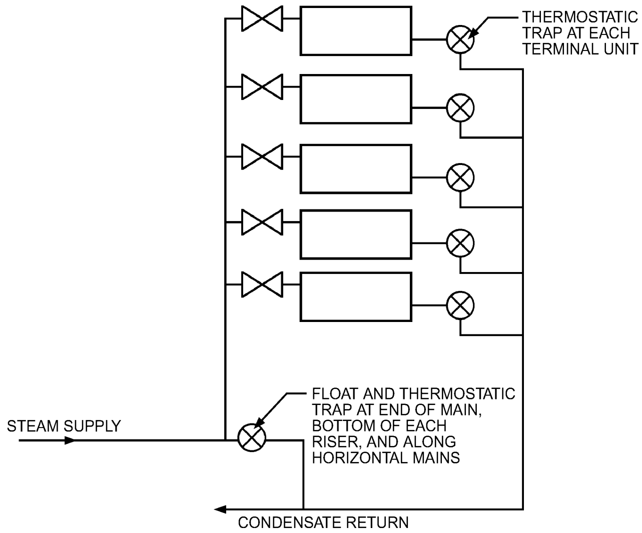

Fig. 16 Two-Pipe System

Variable vacuum or subatmospheric systems are a variation of the vacuum return system in which a controllable vacuum is maintained in both the supply and return sides. This permits using the lowest possible system temperature and prompt steam distribution to the terminal units. The primary purpose of variable vacuum systems is to control temperature as discussed in the section on Temperature Control.

Unlike one-pipe systems, two-pipe systems can be simply zoned where piping is arranged to supply heat to individual sections of the building that have similar heating requirements. Heat is supplied to meet the requirements of each section without overheating other sections. The heat also can be varied according to factors such as the hours of use, type of occupancy, and sun load.

STEAM DISTRIBUTION

Steam supply piping should be sized so that the pressure drops in all branches of the same supply main are nearly uniform. Return piping should be sized for the same pressure drop as supply piping for quick and even steam distribution. Because it is impossible to size piping so that pressure drops are exactly the same, the steam flows first to those units that can be reached with the least resistance, resulting in uneven heating. Units farthest from the source of steam will heat last, while other spaces are overheated. This problem is most evident when the system is filling with steam. It can be severe on systems in which temperature is controlled by cycling the steam on and off. The problem can be alleviated or eliminated by balancing valves or inlet orifices.

Balancing valves are installed at the unit inlet and contain an adjustable valve port to control the amount of steam delivered. The main problem with such devices is that they are seldom calibrated accurately, so that variations in orifice size as small as 0.003 in^2 can make a significant difference.

Inlet orifices are thin brass or copper plates installed in the unit inlet valve or pipe unions (Figure 17). Inlet orifices can solve distribution problems, because they can be drilled for appropriate size and changed easily to compensate for unusual conditions. Properly sized inlet orifices can compensate for oversized heating units, reduce energy waste and system control problems caused by excessive steam loss from defective steam traps, and provide a means of temperature control and balancing within individual zones. Manufacturers of orifice plates provide data for calculating the required sizes for most systems. Also, Sanford and Sprenger (1931) developed data for sizing orifices for low-pressure, gravity return systems

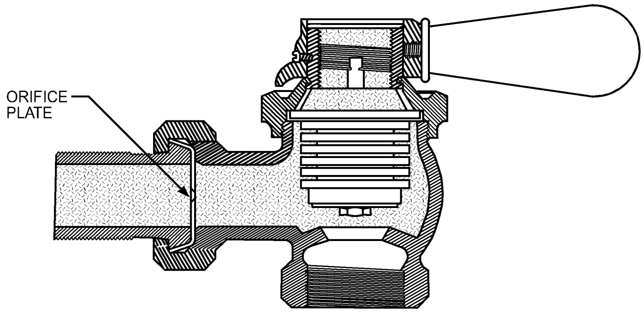

Fig. 17 Inlet Orifice

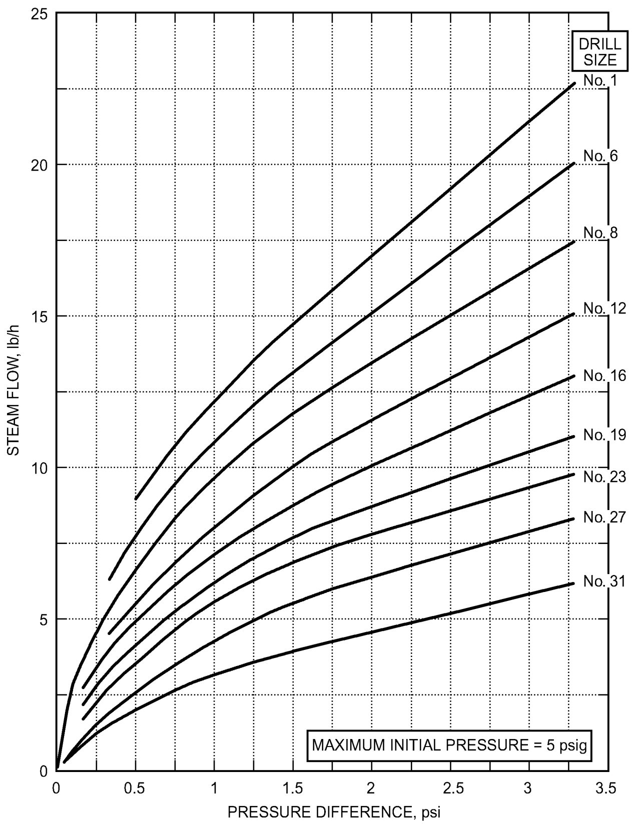

Fig. 18 Orifice Capacities for Different Pressure Differentials
(Schroeder 1950)

See Figure 18 for capacities of orifices at various pressure differentials.

If orifices are installed in valves or pipe unions that are conveniently accessible, minor rebalancing among individual zones can be accomplished through replacement of individual orifices.

TEMPERATURE CONTROL

All heating systems require some means of temperature control to achieve the desired comfort conditions and operating efficiencies. In convection-type steam heating systems, the temperature and resulting heat output of the terminal units must be increased or decreased. This can be done by (1) permitting the steam to enter the heating unit intermittently, (2) varying the steam temperature delivered to each unit, and (3) varying the amount of steam delivered to each unit.

There are two types of controls: those that control the temperature or heat input to the entire system and those that control the temperature of individual spaces or zones. Often, both types are used together for maximum control and operating efficiency.

Intermittent flow controls, commonly called **heat timers**, control the temperature of the system by cycling the steam on and off during a certain portion (or fraction) of each hour as a function of the outdoor and/or indoor temperature. Most of these devices provide for night setback, and computerized or electronic models optimize morning start-up as a function of outdoor and indoor temperature and make anticipatory adjustments. Used independently, these devices do not permit varying heat input to different parts of the building or spaces, so they should be used with zone control valves or individual thermostatic valves for maximum energy efficiency.

Zone control valves control the temperature of spaces with similar heating requirements as a function of outdoor and/or indoor temperature, as well as permit duty cycling. These intermittent flow control devices operate full open or full closed. They are controlled by an indoor thermostat and are used in conjunction with a heat timer, variable vacuum, or pressure differential control that controls heat input to the system.

Individual thermostatic valves installed at each terminal unit can provide the proper amount of heat to satisfy the individual requirements of each space and eliminate overheating. Valves can be actuated electrically, pneumatically, or by a self-contained mechanism that has a wax- or liquid-filled sensing element requiring no external power source.

Individual thermostatic valves can be and are often used as the only means of temperature control. However, these systems are always "on," resulting in inefficiency and no central control of heat input to the system or to the individual zones. Electronic, electric, and pneumatic operators allow centralized control to be built into the system. However, it is desirable to have a supplemental system control with self-contained thermostatic valves in the form of zone control valves, or a system that controls the heat input to the entire system, relying on self-contained valves as a local high-temperature shutoff only.

Self-contained thermostatic valves can also be used to control temperature on one-pipe systems that cycle on and off when installed in series with the unit air vent. On initial start-up, the air vent is open, and inherent system distribution problems still exist. However, on subsequent on-cycles where the room temperature satisfies the thermostatic element, the vent valve remains closed, preventing the unit from filling with steam.

Variable vacuum or subatmospheric systems control the temperature of the system by varying the steam temperature through pressure control. These systems differ from the regular vacuum return system in that the vacuum is maintained in both supply and return lines. Such systems usually operate at a pressure range from 2 psig to 25 in. Hg vacuum. Inlet orifices are installed at the terminal equipment for proper steam distribution during mild weather.

Design-day conditions are seldom encountered, so the system usually operates at a substantial vacuum. Because the specific volume of steam increases as the pressure decreases, it takes less steam to fill the system and operating efficiency results. The variable vacuum system can be used in conjunction with zone control valves or individual self-contained valves for increased operating efficiency.

Pressure differential control (two-pipe orifice) systems provide centralized temperature control with any system that has properly sized inlet orifices at each heating unit. This method operates on the principle that flow through an orifice is a function of the pressure differential across the orifice plate. The pressure range is selected to fill each heating unit on the coldest design day; on warmer days, the supply pressure is lowered so that heating units are only partially filled with steam, thereby reducing their heat output. An advantage of this method is that it virtually eliminates all heating unit steam trap losses because the heating units are only partially filled with steam on all but the coldest design days.

Table 2 Pressure Differential Temperature Control

Outdoor Temperature, °F	Required Pressure Differential*, in. Hg.
0	6.0
10	4.5
20	3.2
30	2.1
40	1.2
50	0.5
60	0.1

*To maintain 70°F indoors for 0°F outdoor design.

The required system pressure differential can be achieved manually with throttling valves or with an automatic pressure differential controller. Table 2 shows the required pressure differentials to maintain 70°F indoors for 0°F outdoor design, with orifices according to Figure 18. Required pressure differential curves can be established for any combination of supply and return line pressures by sizing orifices to deliver the proper amount of steam for the coldest design day, calculating the amount of steam required for warmer days using Equation (3), and then selecting the pressure differential that will provide this flow rate.

$$Q_r = Q_d \frac{(t_i)_{design} - t_o}{(t_i)_{design} - (t_o)_{design}} \quad (3)$$

where

Q_r = required flow rate
Q_d = design day flow rate
$(t_i)_{design}$ = design indoor temperature
$(t_o)_{design}$ = design outdoor temperature
t_o = outside temperature

Pressure differential systems can be used with zone control valves or individual self-contained thermostatic valves to increase operating efficiency.

HEAT RECOVERY

Two methods are generally employed to recover heat from condensate: (1) the enthalpy of the liquid condensate (sensible heat) can be used to vaporize or "flash" some of the liquid to steam at a lower pressure, or (2) it can be used directly in a heat exchanger to heat air, fluid, or a process.

The particular methods used vary with the type of system. As explained in the section on Basic Steam System Design, facilities that purchase steam from a utility generally do not have to return condensate and, therefore, can recover heat to the maximum extent possible. On the other hand, facilities with their own boiler generally want the condensate to return to the boiler as hot as possible, limiting heat recovery because any heat removed from condensate has to be returned to the boiler to generate steam again.

Flash Steam

Flash steam is an effective use for the enthalpy of the liquid condensate. It can be used in any facility that has a requirement for steam at different pressures, regardless of whether steam is purchased or generated by a facility's own boiler. Flash steam can be used in any heat exchange device to heat air, water, or other liquids or directly in processes with lower pressure steam requirements. Equation (1) may be used to calculate the amount of flash steam generated, and Figure 19 provides a graph for calculating the amount of flash steam as a function of system pressures.

Although flash steam can be generated directly by discharging high-pressure condensate to a lower pressure system, most designers prefer a **flash tank** to control flashing. Flash tanks can be mounted either vertically or horizontally, but the vertical arrangement shown in Figure 21 is preferred because it provides better separation of steam and water, resulting in the highest possible steam quality.

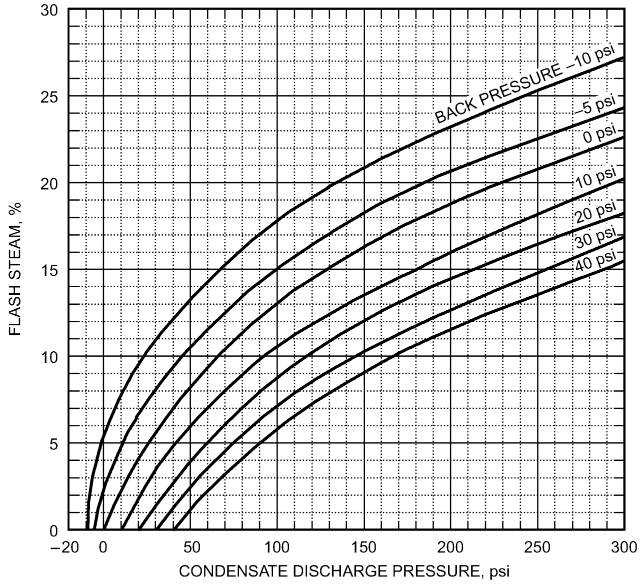

Fig. 19 Flash Steam

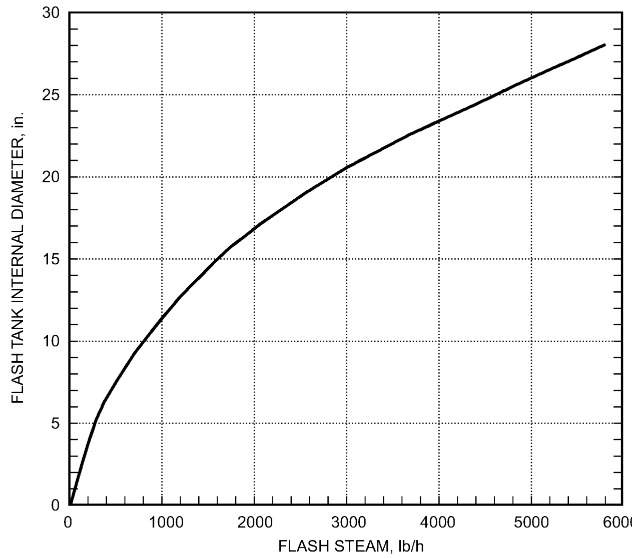

Fig. 20 Flash Tank Diameters

The most important dimension in the design of vertical flash tanks is the internal diameter, which must be large enough to ensure a low upward velocity of flash to minimize water carryover. If this velocity is low enough, the height of the tank is not important, but it is good practice to use a height of at least 2 to 3 ft. The graph in Figure 20 can be used to determine the internal diameter and is based on a steam velocity of 10 ft/s, which is the maximum velocity in most systems.

Installation is important for proper flash tank operation. Condensate lines should pitch towards the flash tank. If more than one condensate line discharges to the tank, each line should be equipped with a swing check valve to prevent backflow. Condensate lines and the flash tank should be well insulated to prevent any unnecessary heat loss. A thermostatic air vent should be installed at the top of the tank to vent any air that accumulates. The tank should be trapped at

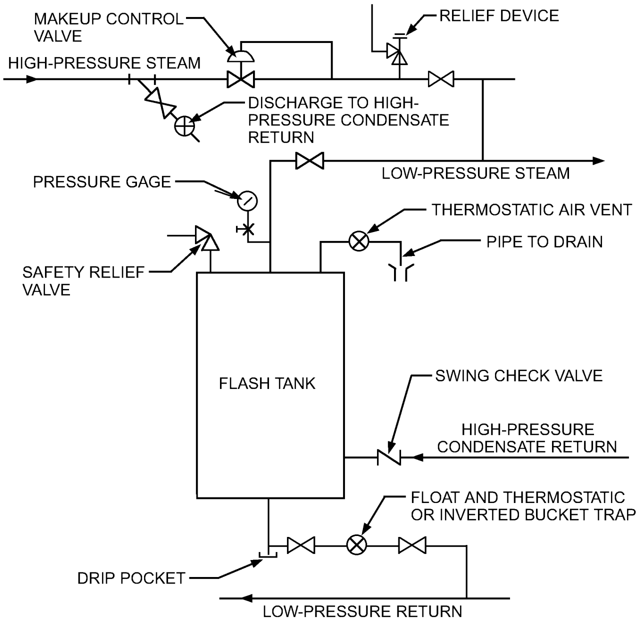

Fig. 21 Vertical Flash Tank

the bottom with an inverted bucket or float and thermostatic trap sized to triple the condensate load.

The demand load must always be greater than the amount of flash steam available to prevent the low-pressure system from becoming overpressurized. A safety relief valve should always be installed at the top of the flash tank to preclude such a condition.

Because the flash steam available is generally less than the demand for low-pressure steam, a makeup valve ensures that the low-pressure system maintains design pressure.

Flash tanks are considered pressure vessels and must be constructed in accordance with ASME and local codes.

Direct Heat Recovery

Direct heat recovery that uses the enthalpy of the liquid in some type of heat exchange device is appropriate when condensate is not returned to a facility's own boiler; any lowering of condensate temperature below saturation temperature requires reheating at the boiler to regenerate steam.

The enthalpy of the condensate can be used in fan-coil units, unit heaters, or convectors to heat spaces where temperature control is not critical such as garages, ramps, loading docks, and entrance halls or in a shell-and-tube heat exchanger to heat water or other fluids.

In most HVAC applications, the enthalpy of the liquid condensate may be used most effectively and efficiently to heat domestic hot water with a shell-and-tube or plate-type heat exchanger, commonly called an economizer. Many existing economizers do not use the enthalpy of the condensate effectively because they are designed only to preheat makeup water flowing directly through the heat exchanger at the time of water usage. Hot water use seldom coincides with condensate load, and most of the enthalpy is wasted in these preheat economizer systems.

A storage-type water heater can be effectively used for heat recovery. This can be a shell-and-tube heat exchanger with a condensate coil for heat recovery and a steam or electric coil when the condensate enthalpy cannot satisfy the demand. Another option is a storage-type heat exchanger incorporating only a coil for condensate with a supplemental heat exchanger to satisfy peak loads. Note that many areas require a double-wall heat exchanger between steam and the potable water.

Chapter 49 of the 2007 *ASHRAE Handbook—HVAC Applications* provides useful information on determining hot water loads for various facilities. In general, however, the following provide optimum heat recovery:

- Install the greatest storage capacity in the available space. Although all systems must have supplemental heaters for peak load conditions, with ample storage capacity these heaters may seldom function if all the necessary heat is provided by the enthalpy of the condensate.
- For maximum recovery, permit stored water to heat to 180°F or higher, using a mixing valve to temper the water to the proper delivery temperature.

COMBINED STEAM AND WATER SYSTEMS

Combined steam and water systems are often used to take advantage of the unique properties of steam, which are described in the sections on Advantages and Fundamentals.

Combined systems are used where a facility must generate steam to satisfy the heating requirements of certain processes or equipment. They are usually used where steam is available from a utility and economic considerations of local codes preclude the facility from operating its own boiler plant. There are two types of combined steam and water systems: (1) steam is used directly as a heating medium; the terminal equipment must have two separate coils, one for heating with steam and one for cooling with chilled water, and (2) steam is used indirectly and is piped to heat exchangers that generate the hot water for use at terminal equipment; the exchanger for terminal equipment may use either one coil or separate coils for heating and cooling.

Combined steam and water systems may be two-, three-, or four-pipe systems. Chapters 3 and 5 have further descriptions.

COMMISSIONING

After design and construction of a system, care should be taken to ensure correct performance of the system and that building personnel understand the operating and maintenance procedure required to maintain operating efficiency.

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