FUNDAMENTALS OF STEAM HEATING SYSTEMS

A discussion of the physics of two-phase flow as related to steam heating systems and the fundamentals that can assist design engineers in a redirection of their efforts in this area

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Few will deny that there have been unprecedented advances in building technology during the last half of this century. The typical building of the 1930s was basically a static structure with very few moving parts and consuming little energy. The moving parts comprising the energy systems were: 1) lighting, which was quite minimal since daylighting was the main source of light; 2) heating, which was usually standing cast-iron radiation, and 3) an occasional fan for summer comfort or, in limited cases, mechanical ventilation. In resource terms, the heating system was the major energy consumer. In those buildings, the most common method for providing the heat for the radiators was steam. Some buildings used gravity circulation hot water heating systems, but the overwhelmingly favored system was steam.

Following World War II, as the United States turned its industrial might to the civilian economy, buildings underwent a transfiguration that changed them from static structures to gigantic machines. This change took place guided not by some master plan but rather by rapid evolution based on technology. Unlike a consumer product that would be developed and then brought to market with an advertising campaign to create a need, the changes in building technology grew a step at a time through a complex network of machinery, materials, and economics. There were times when even the most skilled practitioners in the building industry were totally surprised by where they suddenly found themselves. One striking example of this situation is the impact that ASHRAE Standard 90-75, *Energy Conservation in New Building Design,* has had on today's buildings. Section 4 of that standard established minimum thermal requirements for the building envelope that when incorporated into building codes, made the building design community conscious, for the first time, of the need to consider thermal qualities of building envelopes from a standpoint other than simplified economics. Once the trend started, the true spinoff economics were revealed. The better envelopes resulted in smaller heating and cooling systems, improved comfort, etc.

TABLE 1 — Comparison of characteristics of two-phase (steam) and single-phase (water) systems.

Steam *(two-phase)*	**Water** *(single-phase)*
Source independence	Source interdependence
Constant temperature heat transfer	Variable heat transfer
Nondirectional distribution	Directional distribution
Temperature-pressure dependance	Temperature independent of pressure*
Minimum shaft energy required	Comparatively significant shaft energy required
Rapid fluid migration to low temperature area of system	Minor migration of fluid due to temperature Differentials extremely geometric dependent

**Except as a limit.*

In this evolutionary process, steam radiation was replaced by forced circulation hot water. The cast-iron radiator was replaced with a nonferrous product such as baseboard radiation, finned-tube radiation, or a convector. And with the improved envelopes, perhaps no radiation at all. As this change took place, the building HVAC system designers were busy learning a host of new skills to keep themselves current with the rapidly changing needs, and in many cases, the science of steam heating was lost completely.

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Many of the most commonly used textbooks on HVAC design used in undergraduate courses in our engineering schools today do not even have a chapter on steam system design. This may appear perfectly logical since so few new buildings are being designed with steam heat. It would be unfortunate, however, if the lack of understanding of steam dynamics were the driving force in the continuing decline of the use of steam. As an intermediate thermal fluid (that is, a fluid that transfers heat over distance from a high-temperature source to a lower-temperature receiver), steam has many advantages over a single-phase fluid such as water or air.

The purpose of this article is to present a discussion of the physics of two-phase flow specifically as it relates to steam heating systems. Some of the concepts presented will be inconsistent with past practices in steam system design simply because they have been updated from publications and concepts based upon 1930s technology.

Why use steam?

The first question that should be answered is why should an HVAC engineer concern himself or herself with steam system technology. The answer is twofold.

First, there is a need. A large portion of the activity in the building industry currently and in the future will be in retrofit and renovation of existing buildings. In the Northeast and Midwest regions, a majority of the existing buildings have steam heating systems (either total or partial). When these buildings are renovated, the option of upgrading those systems could in many cases be economically feasible. However, if the engineer hasn't the knowledge, this option is not available.

Furthermore, many buildings are served with steam from a central plant, such as on a university campus, or by a steam utility system. Virtually all of the older principal cities north of the thirty-eighth parallel have these utilities. Thus, even though the radiation may not be steam (if indeed there is radiation), the designer must have an understanding of steam technology to use the steam that is available for either direct heating of air or heating of water in a heat exchanger.

Another case of need is in an industrial facility or perhaps a health care facility where steam is needed for something other than space heat. The engineer has no choice but to become involved in the design of a steam system.

The second reason for the designer to have a knowledge of two-phase technology is that there are inherent advantages in two-phase as opposed to single-phase systems. Table 1 shows a side-by-side comparison of operating characteristics of two- and single-phase systems.

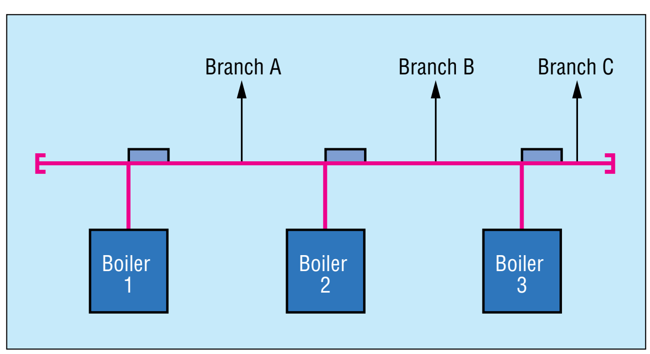
Some of the characteristics of Table 1 deserve some discussion since most of these comparisons have seldom been addressed in publications on the subject.

▲ **Source independence** —This characteristic results primarily in a higher degree of probable reliability with steam systems than with water systems. Generally, a sudden rupture and resulting loss of fluid at some point in the system will not affect the ability of the system to continue to operate due to the expandability of steam. This feature surfaced when internal combustion engines were used for cogeneration systems. It became evident, for example, that a rupture of a flexible pipe connector on an engine jacket connection of one engine (either feedwater or steam) did not affect the continued operation of the other engines if they were connected as independent steam generators, whereas with single-phase coolant systems, such a failure would result in a complete plant shutdown.

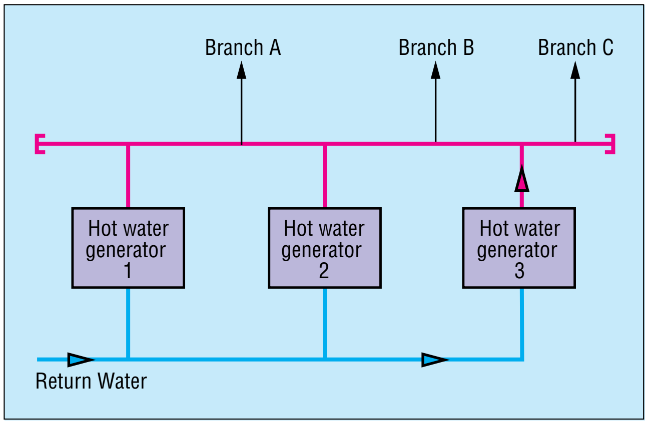
▲ **Constant temperature heat transfer** —The feature of constant temperature heat transfer vs. variable temperature has little significance in affecting the choice of system in most heating applications. However, in some process applications, the constant temperature has definite advantages.

For example, if one wishes to operate an absorption refrigeration unit at a generator temperature of 240 F, this is readily accomplished with 240 F or 10 psig steam. If water is used at a 20 F drop in temperature, to achieve an average of 240 F, the entering temperature would have to be 250 F, which is the upper limit of the ASME low-pressure code, thus almost impossible to achieve in practice with low-pressure class equipment.

▲ **Nondirectional distribution** —This characteristic is neither an advantage nor disadvantage for either of the systems compared. It is, however, a feature that design engineers must recognize to avoid costly errors in designing water systems. The concept of piping headers was developed with steam systems. Referring to Fig. 1, note that with a steam header, steam can enter from any boiler and go to any load, regardless of which boilers are operating or which loads are calling for flow. However, this is not true with a water system. In a water system with multiple boilers, if it is piped as shown in Fig. 2 and boilers and load flow are equal, the water from Boiler 1 flows through Branch



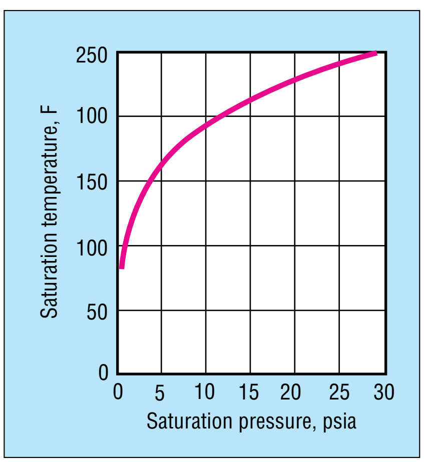
1 Typical steam header piping schematic.



2 Hot water system with header. Note: A water system (hot or chilled) should not be piped in this manner.

A, the water from Boiler 2 flows through Branch B, and the water from Boiler 3 flows through Branch C. Then, if Boiler 1 cycles off, Load A receives return water temperature, etc. Thus, in the piping of water header, all of the supplies from the source devices (boilers) should connect, mix, and then branch to loads (*e.g.,* sources at one end, loads at the other). Again, this is a subtle but extremely important difference for designers to keep in mind. The header approach has been used (or misused) with catastrophic results in both heating and cooling systems.

▲ **Temperature-pressure dependence** —Fig. 3 is a saturation curve (boiling or condensing temperature vs. pressure) for steam in the operating range of steam heating systems. This is the characteristic difference between steam and water systems that perhaps more than any other is responsible for the preference of water over steam in building space heating systems. The advantage was first identified with standing radiation systems. The most direct method of reducing the capacity of a heating device at part-load conditions is to reduce the temperature difference between the fluid and the air (the ΔT or the *LMTD*). With water, this can be done by several methods, two of which are reducing the supply water temperature and slowing down the water flow—both quite simple to achieve. However, a steam system operating at 5 psi, because of the temperature-pressure dependence, will only allow a capacity reduction from 100 to 92 percent before going into the subatmospheric pressure range. Although it is quite easy to achieve pressure reductions well into the subatmospheric range in the heating device (simply throttle the valve), the problems resulting from doing so can be overwhelming and can require expensive machinery and installations.



3 Saturation temperature-pressure curve for low-pressure steam.

There are cases, however, in which the pressure-temperature dependence can be used to advantage once the dynamics are understood. It is this phenomenon that will be discussed extensively below.

▲ **Minimum shaft energy required** —This assumes a comparison between modern pumped type systems for both the steam and the water. This statement relates to two more fundamental characteristics—heat carrying capacity and the migration phenomenon. Each lb of steam at, say, 5 psig contains a heating capacity in latent heat of 960 Btu per lb, which with 40 F of subcooling would be 1000 Btu per lb. Thus, if the return were pumped, for each 1000 Btu, 1 lb of water would have to be pumped against a 5 psi pressure difference. It is a bit more complex in the single-phase or water system. First, the heat carrying capacity is a function of temperature differential. If the temperature differential were 20 F, for each 1000 Btu, 50 lb of water would have to be pumped. The pressure would be determined by the dynamic friction losses in the system, which generally range from 10 to 40 psi in larger systems. At 25 psi (60 ft), the water would thus require 250 times more shaft energy to move or transport the same quantity of heat as the steam!

▲ **Rapid fluid migration to low-temperature area of system** — Unlike the gravity flow hot water system, the phase change that occurs as steam gives up its heat and condenses is accompanied by a significant change in volume. If the condensation occurs at 5 psig, the specific volume changes from approximately 20 to 0.017 cu ft per lb (a decrease of 1200 to 1) as the steam condenses. This change creates a low-pressure zone into which the available steam molecules move. With single-phase water systems, for a temperature range of 20 F from 220 F, the volume change is closer to 1.01 to 1, or approximately 1 percent as compared to 1200 percent for steam. The following discussion explores the methods in which the design of steam heating systems utilizes this characteristic, combined with the temperature-pressure dependence, to achieve capacity control.

System dynamics

Fig. 4 is a simplified diagram of a steam heating device operating at 5 psig and heating air from 60 to 80 F. The amount of heat dissipated to the air can be expressed by the equation:

$q = UA\,(LMTD)$

The *LMTD* in this example is 154 F. The capacity so calculated could be called the "design" capacity of this device. In heating service and in many other type systems, such as the heating of domestic hot water, the design capacity is required during a small portion of the operating hours of the device. The equation shows that there are three characteristics that can be varied

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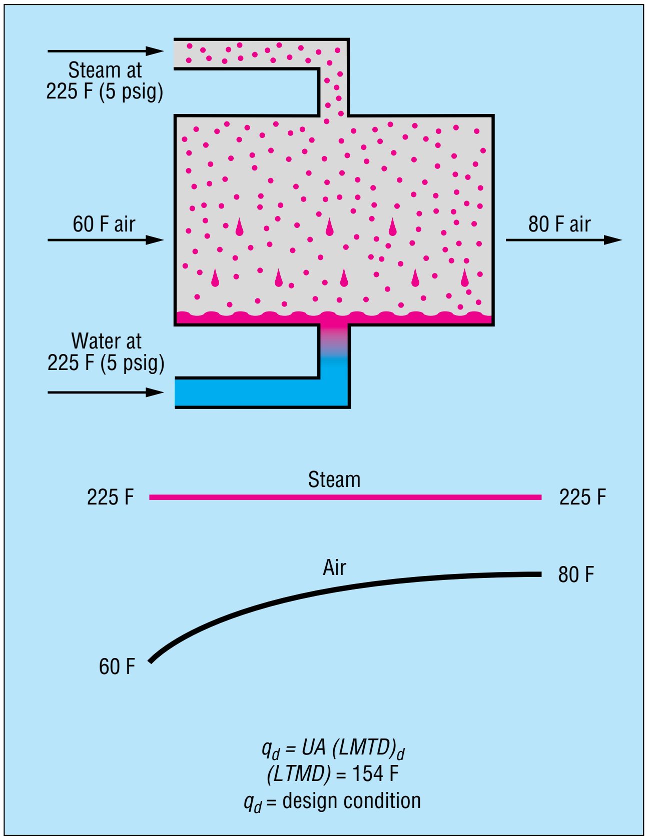
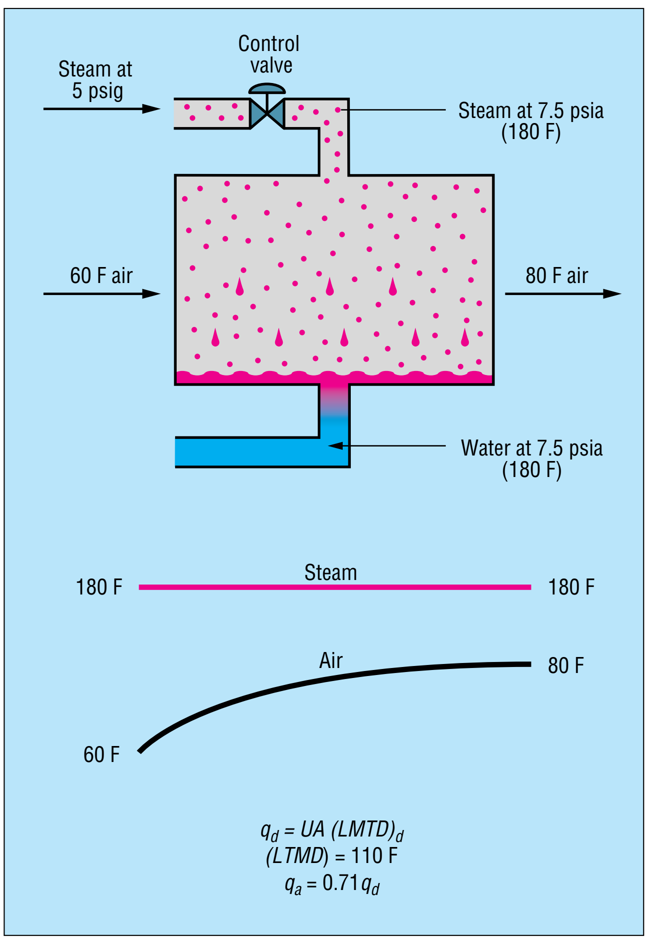


4 Simplified diagram of steam heating device.



5 Simplified diagram of steam heating device with control valve (at reduced load q_a).

Steam heating basics

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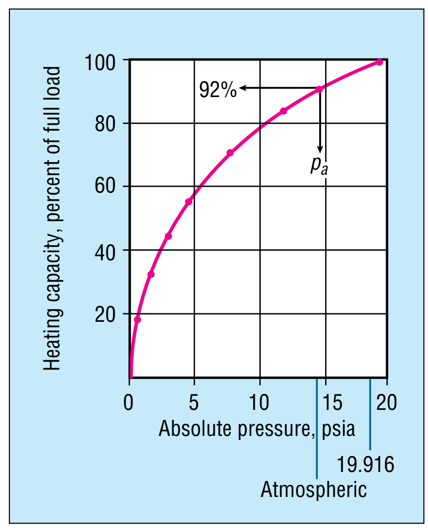
if the heating capacity is to be changed—they are *U, A,* and *LMTD*. The most common method employed is to reduce the *LMTD*, and the most common method of achieving this is to reduce the steam temperature by reducing the pressure.

If a control valve is added to the exchanger as shown in Fig. 5, as the valve closes, the pressure of the condensing steam within the unit is reduced. The phenomenon that reduces the pressure is simply that the throttled opening of the valve does not allow the steam molecules to enter the exchanger as quickly as they are being condensed—thus, fewer molecules, less pressure. The reducing pressure is accompanied by a reducing temperature (Fig. 3), which reduces the *LMTD*. If, for example, the system load is 71 percent of design load as shown, the valve will position to create a 180 F exchanger at a condensing pressure of 7$^1/_2$ psia or approximately 15 in. Hg vacuum. If the capacity for this exchanger expressed in percent of full load capacity is plotted against the absolute pressure to establish the range of subatmospheric operation, the curve appears as shown in Fig. 6.

In Fig. 6, it can be seen that at any operating point below 92 percent of design, the exchanger's steam side pressure is subatmospheric. (Notice that at 50 percent load the pressure is approximately 3$^1/_2$ psia or 23 in. Hg vacuum!) Needless to say, this is a very high vacuum and difficult to maintain with most commercial piping systems.



6 Heating capacity vs. absolute pressure.

This subatmospheric performance creates numerous problems. In most actual systems, the heat exchanger is provided with some type of steam trap as shown in Fig. 7. Traps are devices that are designed to hold steam in the exchanger and allow liquid condensate and noncondensable gases to pass into the condensate system. As the steam condenses, the liquid drains to the trap, the trap opens, and a pressure difference between the exchanger and the condensate line causes the water to flow through the trap.

However, when the system load condition is below 92 percent of design load, there is no pressure difference if the condensate system is vented to atmosphere. Thus, the only way the system can work in this mode is to have the return system connected to a high vacuum pump. Although this is a conceptually valid method of solving the problem, it is not the most commonly used method for several reasons two of which are:

▲ Most commercial grade piping systems will not hold a high vacuum.

▲ Such systems are vulnerable to total system failure because of the failure of a single component such as a drip trap.

Although there are systems available, for the above reasons, they are not in common use today.

The most common solution to this problem as practiced in the design profession today is that of preventing subatmospheric pressures by the use of vacuum breakers. The two methods of connecting these vacuum breakers are either to atmosphere as shown in Fig. 8 or into the condensate line as shown in Fig. 9. Consider first the technique of breaking the vacuum with the atmosphere.

Fig. 6 reveals that if the load is 71 percent of design, the steam temperature will be 180 F, the saturation pressure for which is 7.5 psia. This condition is shown in Fig. 8 with the vacuum breaker added and a *total* pressure of 15 psia. Now consider the dynamics of the problem:

▲ The steam valve responding to the load tries to position to allow the correct number of steam molecules through to hold the necessary partial pressure.

▲ The vacuum breaker follows the steam valve to allow the air to make up the difference between the saturated steam pressure and atmospheric pressure.

▲ The trap follows the control of the steam valve and air valve (vacuum breaker), theoretically allowing the liquid and air to leave the system at the rate of entry.

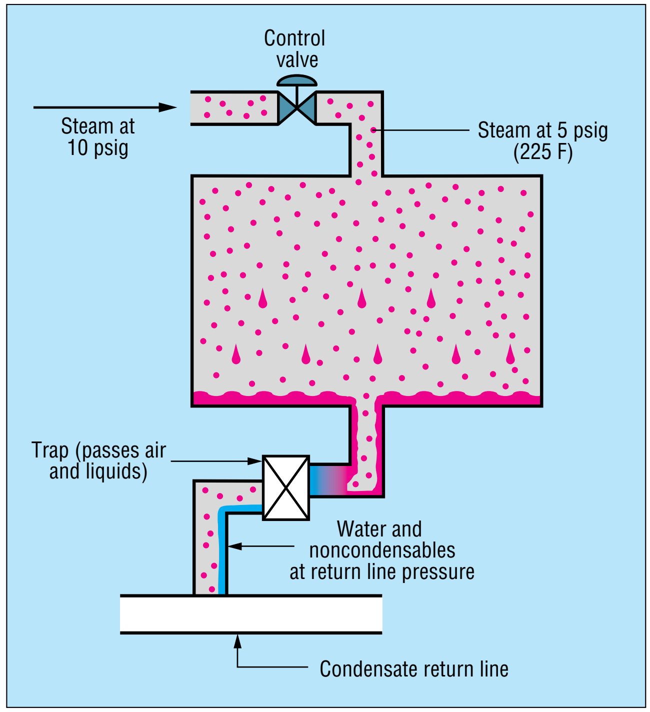
Thus, there is not only one control device but three—the steam valve, the air valve (vacuum breaker), and the trap. With this system, the following problems are presented:

▲ Under this mode (at all capacity conditions below 92 percent of design), and assuming that the condensate return line is at atmospheric pressure, there is no pressure differential across the trap. Thus, the only differential that can exist is a fluid head buildup in the outlet line leaving the heat exchange device.

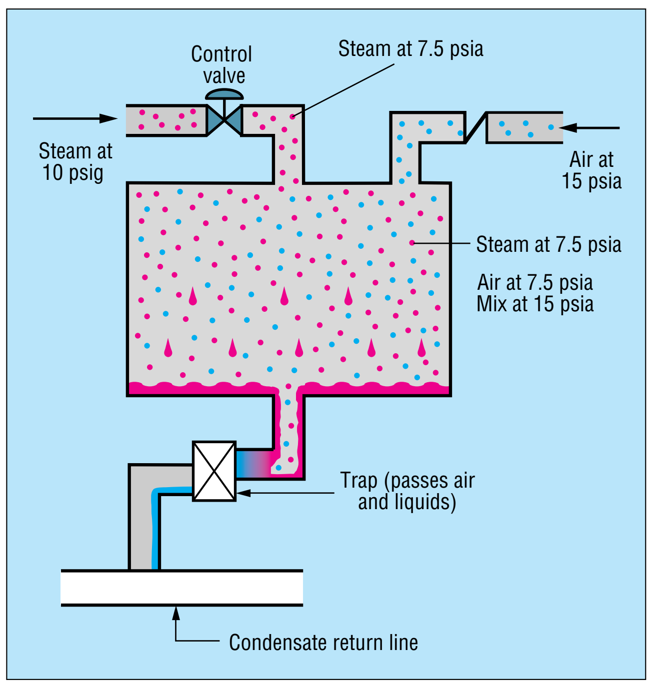
▲ If there is a thermostatic element in the trap, and if there is a liquid leg above the trap, none of the air can get to the thermostatic element and be released.

▲ If there is a positive pressure in the return line, which could result from another trap such as a drip trap feeding into it, then whatever fluid (liquid, air, or steam) that is in the upper part of the return line will move backward through the trap into the exchanger when the trap opens.

This scenario of dynamic response presents another interesting problem. With the mixture of air and steam in the heat exchanger, do the two gases mix or stratify? If it is assumed that the air and steam mix in



7 Steam heating unit showing trap connection.



8 Steam load unit with atmospheric vacuum breaker.

a psychrometric type of system as was done here, a potentially significant energy burden on steam radiation type systems, or any steam system with a thermostatic trap element, is readily revealed.

Fig. 10 shows a typical connection for a steam radiation device with the often used steam control valve. Notice the radiator trap. Most, if not all, thermostatic traps are designed to operate at a saturation pressure equal to the steam saturation pressure at a temperature about 20 F below the steam temperature. Thus, under the condition shown, with 180 F fluid mixture temperature, the pressure in the trap is equal to the saturation pressure of steam at 180 F + 20 F or 200 F, which is 11.52 psia (Fig. 3), but since the 180 F mix is at 15 psia, the trap will open and remain open as long as the reduced load condition continues!

The other possibility is that the air and steam stratify as shown in Fig. 11. Since the air density under the most favorable of conditions is approximately 1.6 times that of the steam, the stratified condition *always* finds the steam on top and the air on the bottom. Under this mode, the control system becomes a two-position system working in cycles.

Let's return to the basic heat transfer equation. With the stratified condition, if it can be assumed that the air temperature within the exchanger is the same as the air being heated and saturated steam is above the air, this phenomenon satisfies the heat transfer equation by reducing the area. Thus, part of the heat exchanger has been flooded with a noncondensable gas, causing an effective reduction of heat transfer area.

If the flow of condensate through the trap is not great enough to prevent the air from escaping, the trap continually tries to purge the radiator of air until either the steam hits the trap and closes it or the liquid fills the volume of the trap and seals the air in. It is virtually impossible to anticipate or predict the nature of these unstable and unsteady happenings since published information is not available that gives the rate of condensate flow with zero pressure differential. The fact that has been examined here is that under most part load operating conditions, in most heating applications, the trap operates with no pressure differential and thus functions more like a weir than an orifice.

Pressurized return line

The possibility of the return line being at a higher pressure than the heat exchanger was previously discussed. This would be the case if the return line were pressurized with flash steam from a nonsubcooled load or a drip trap. If this were to occur, the use of the vacuum breaker would be ineffective in assuring that the entering trap pressure were at least as great as the outlet pressure. One method that has been used for addressing this problem is to connect the vacuum breaker into the condensate return system as illustrated in Fig. 9. With this arrangement, the designer is assured that there will never be an inverse pressure gradient across the trap. This concept, if properly applied, however, must recognize the following two facts:

▲ The system will operate at zero pressure differential across the trap at most load conditions.

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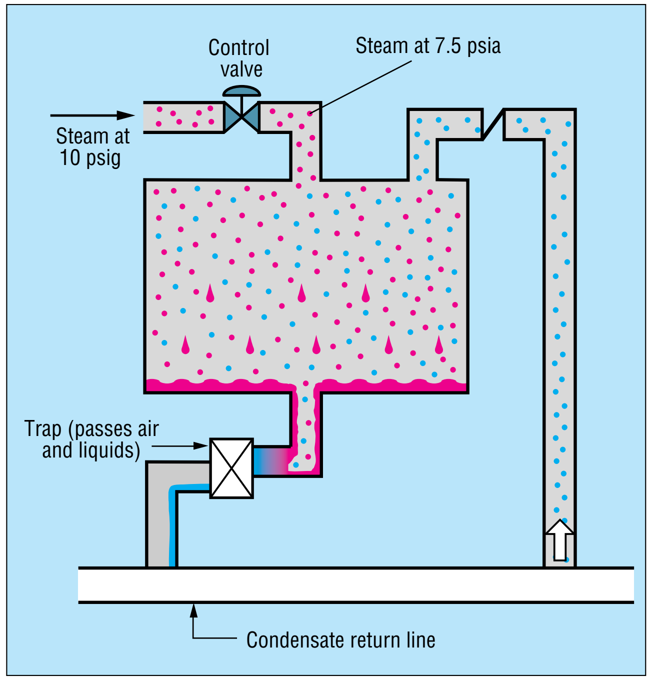
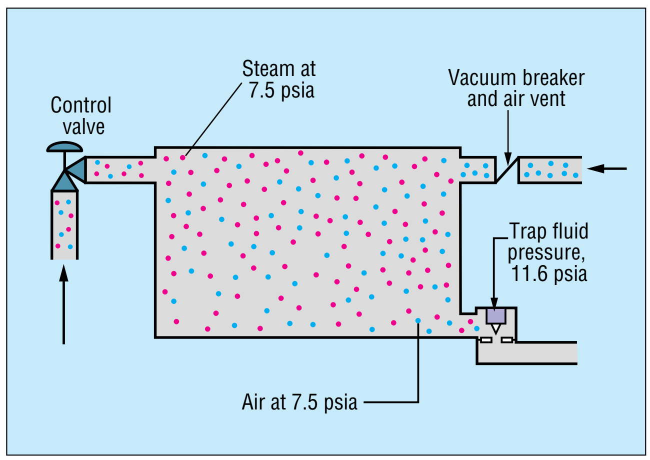
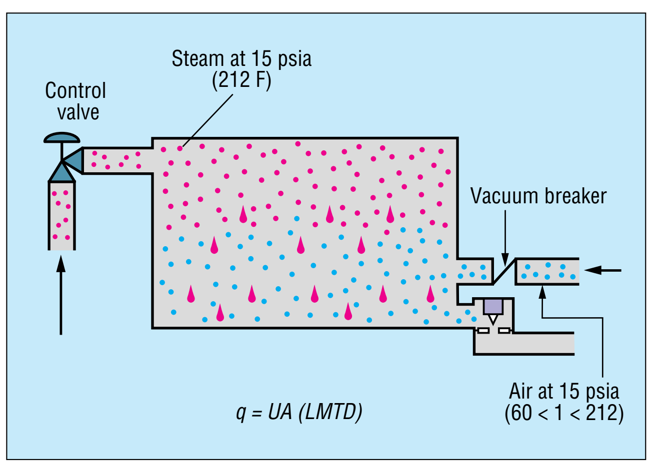


9 Steam load unit with condensate line vacuum breaker.



10 Steam load device with radiator trap and vacuum breaker.



11 Steam load device with steam/air stratification.

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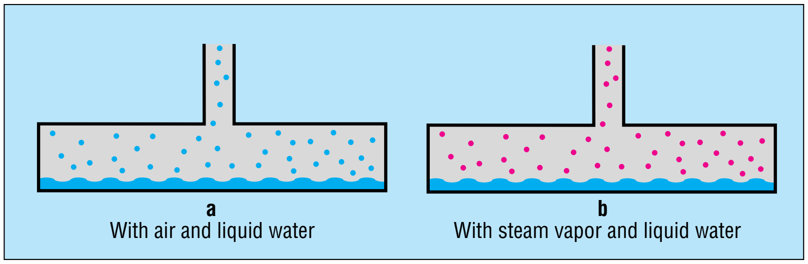
▲ Predictable performance of the system becomes a function of the design of the condensate return system.

A so-called dry condensate return line is a line with fluids in two different phases—liquid and gaseous.

If the gas phase fluid is air (Fig. 12a), then the system functions very much as it did with the vacuum breaker connected to the atmosphere except that the lack of an inverse pressure gradient is assured. If the gas phase fluid is water vapor (steam) as illustrated in Fig. 12b, then the vacuum breaker has simply become a second steam source for the load device.



12 Dry return lines in a vented system.

What has been done in practice to help designers achieve stable system control is to assure the lack of superatmospheric pressure *and* presence of air in the dry return by venting the return line system to atmosphere. When this is done effectively, flash steam is vented off through the vents, and as vapor is condensed, air is drawn in. This vented concept has been used in the vast majority of building heating steam systems over the past 20 to 30 years. It must be recognized that with such systems the flow of the condensate is not pressure motivated but rather is motivated purely by gravity as illustrated in Fig. 13.

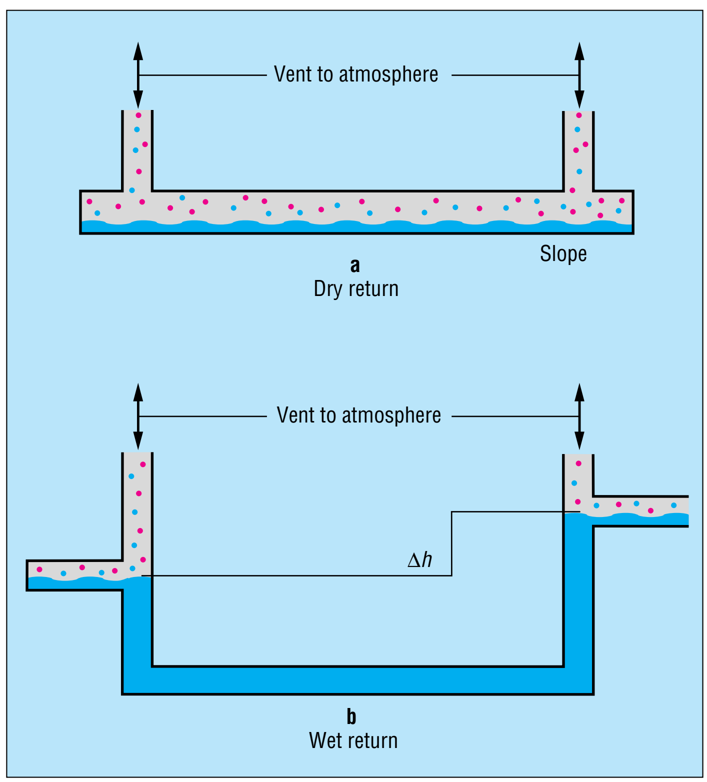
Return line flow

There are two phenomena that describe the flow in a vented condensate system. If it is a dry return (Fig. 13a), the liquid flows essentially like a sewer line in an open channel. If there is a flooded section of the line, called a wet return (Fig. 13b), the flow phenomenon is described by the Darcy Weisbach equation. The open channel flow is described by the Manning equation, which does not contain a pressure differential term; this is understandable since the flow is motivated by gravity. In the flooded or wet return sections of a vented return system, the only available pressure differential is created by a head differential at the two ends of the piping section.

Alternative control concepts

Referring again to Fig. 5, remember that the above discussions assumed that the method for reducing the heating capacity was to reduce the *LMTD*. Two other options, however, are available—the reduction of either *U* or *A*. If we assume *U* to be an essentially dependent variable, the only other alternative to reducing the *LMTD* is to reduce or vary the heat transfer area. This technique has been used in the refrigeration industry to control head pressure in refrigeration condensers. The way the area has been varied in those systems has been to flood part of the condenser with liquid refrigerant. In steam systems, this can be accomplished by several means, all of which result in holding a constant steam pressure on the exchanger and moving the control of flow to the leaving or condensate side. This method of control has been used and has proved to provide extremely good control, but the technology is not without problems and still needs to be improved.

Another method for reducing the area is to flood part of the exchanger with air—a scheme that has been extensively employed in the past but usually by accident! Research has shown that with the vacuum breaker located as shown in Fig. 10, the steam and air molecules mix in the exchanger in a sort of a psychrometric system with the mixed fluid temperature at the saturation temperature of the partial pressure of the steam. If the stratified performance is desired (*i.e.,* flooding part of the exchanger with air), the vacuum breaker can simply be relocated to the outlet location as shown in Fig. 11.



13 Flow motivation in vented return system.

Conclusion

A lack of understanding of the above concepts relating to the physics of steam systems has resulted in performance problems and failures in many steam systems. These failures manifest themselves in several different ways, some relatively minor but some that have been catastrophic. The result is that a large sector of the industry has simply avoided using steam and thereby has lost the inherent benefits available. Some of these problems have been:

▲ Steam hammer causing undesirable noise or damage to piping.

▲ Instability in control of temperature.

▲ Inability to control at low loads, resulting in overheating.

▲ Excessive energy cost.

▲ Damage to equipment and property due to freezing of condensate in steam coils, piping, and devices.

▲ Implosion destruction in steam-to-water heat exchanger tubes.

▲ Excessive corrosion in heating devices and return systems.

However, in the design of low-pressure steam systems for heating service, engineers are encouraged to keep in mind the advantages of two-phase systems when they offer the best alternative and apply the physical principles discussed above utilizing the products that are available for these systems. Some suggested guidelines are:

▲ **Consider corrosion** —Unfortunately, there will always be a corrosive atmosphere in a low-pressure steam system. For this reason, it is necessary either to use materials that are resistive to corrosion failure or to assure a chemical treatment program that includes corrosion inhibitors for both the steam side of the system and the condensate side. The condensate side tends to be the more corrosive because of the presence of air and water. *Caution!* If steam from the system is used by direct injection for humidification, care must be taken to use chemicals that are compatible with air quality requirements.

▲ **Pitch steam lines** —Pitch all steam lines (preferably in direction of flow) to low points and drain the low points through drip traps. Drip traps can be sized at the pressure differential available between the steam system and the return system.

▲ **Install vacuum breaker** —If there is valve control on a load device, install a vacuum breaker or the condensate will flood the exchanger under virtually all conditions of reduced load. Remember, if the vacuum breaker is on the entering steam side of the exchanger, the steam and air will tend to mix, varying the temperature of the steam air mix; if the vacuum breaker is on the leaving side, it will tend to flood the lower sections of the exchanger with air, varying the effective heat exchanger area.

▲ **Vent return line** —The return line should be liberally vented to atmosphere to prevent any pressure buildup.

▲ **Connect vacuum breaker into return** —As an additional precaution to assure drainage of condensate, the vacuum breakers can be connected into the return line as in Fig. 9, but if the return line is not adequately vented, this can cause control problems.

▲ **Size trap for open channel flow** —There is *no pressure differential* available to cause the liquid to flow through the trap of a load device at reduced loads unless a fluid head is allowed to build up ahead of the trap.

▲ **Do not use modulating control valve on preheat coils** —Whenever a steam heat exchanger is subjected to below freezing temperatures, such as an outdoor air heating coil, valve control should not be used unless the traps are liberally sized for open channel flow and the vacuum breaker is connected into the return line. The suggested method of control for this service is to assure full available steam pressure on the coil at all times and to handle the capacity control on the air side (such as with face and bypass dampers). Control valves can be used in sequence with face and bypass control if they are controlled to assure fully open position at or below 32 F outdoors.

▲ **Pitch coils and return lines** —The only force that motivates condensate flow in either the load device with valve control or in the return line is gravity. All passages of all coils must be pitched to the outlet, and the return line must be pitched to its collection points. Vented return lines should be sized as open channel flow.

▲ **Address dynamics of isolated components** —Whenever a section of a steam system is or can be isolated, the isolated section tends to reach the temperature of the surroundings. As this occurs, the steam pressure becomes the saturation pressure at that temperature (Fig. 3). Any such sections should be provided with vacuum breakers to prevent damage from air leakage that would certainly occur. An example of this phenomenon is the piping of multiple boilers, in which there are two options:

◆ If check valves are installed on the boiler outlet, a vacuum breaker should be installed on the boiler side of the check valve.

◆ If check valves are not installed, an overflow trap should be installed on the boiler just above the normal high operating water level to prevent excessive level resulting from condensation in the off boiler.

In conclusion, the physics of two-phase thermal fluid systems is such that in numerous applications steam provides certain intrinsic benefits over the hot water alternative. However, because the characteristics of these systems have been somewhat overshadowed by the other pressures of technological change over the past half century while little attention has been given to updating the technology of two-phase flow, the design community has tended to overlook these benefits. This article has attempted to provide the fundamentals necessary to assist design engineers in a redirection of their efforts in this regard. Ω