Piping News Report
An informative newsletter for Piping Designers and Engineers

Basics of Steam System Design

by W. M. (Bill) Huitt

Introduction

This section is intended for people involved in the design, operation and/or maintenance of steam systems. It covers some basic terminology, principles, steam generation in general, and the design of steam distribution and condensate collection piping. Also covered are the sizing, specifying and installation of steam traps.

Terminology

Absolute Pressure:

> The theoretical pressureless state of a perfect vacuum is "absolute zero". Absolute pressure is, therefore, the pressure above absolute zero. At sea level, for instance, the pressure exerted by the atmosphere is 14.7 PSI absolute. Absolute pressure, when measured as pounds per square inch, is indicated as "PSIA". It is also commonly measured in millimeters of mercury, or "mm Hg".

Gauge Pressure:

> Gauge pressure is the internal pressure, as indicated on a gauge, of a sealed system. Such as a tank or piping system. Gauge pressure measures the pressure above atmospheric pressure where zero pounds gauge equals approximately 14.7 PSIA. Below zero pounds gauge a vacuum exists which is often expressed in either inches of mercury (Hg) or inches of water (H_20). Gauge pressure indication is shown as PSIG.

Enthalpy:

> Enthalpy is the total energy, due to both pressure and temperature, of a fluid or vapor at any given time or condition.

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The basic unit of measurement for all types of energy is the British Thermal Unit (BTU).

Specific Enthalpy:

This is the enthalpy of a unit mass (1 pound), generally expressed in BTU/lb.

Specific Heat Capacity:

A measure of the ability of a substance to absorb heat. It is the amount of energy (BTU's) required to raise 1 pound of water 1° F. Thus specific heat capacity is expressed in BTU/lb/° F.

The specific heat capacity of water is 1 BTU/lb/° F. This means that an increase in enthalpy of 1 BTU will raise the temperature of 1 pound of water by 1° F.

Heat:

Heat is a form of energy and as such is part of the enthalpy of a liquid or gas.

Heat Transfer:

Heat transfer is the flow of enthalpy from matter at a high temperature to matter at a lower temperature when brought into contact.

Heat of the Liquid (Enthalpy of Saturated Water):

Expressed in BTU's, this is the amount of heat required to raise the temperature of 1 pound of water from 32° F to the boiling point of a given pressure/temperature correlation. Also referred to as Sensible Heat.

EXAMPLE

Let us assume that 50° F water is available as feedwater to a boiler at atmospheric pressure. The water will begin to boil at 212° F. 1 BTU will be required to raise each pound of water 1° F. Therefore, for each pound of water in the boiler, the increase in enthalpy required to raise the temperature from 50° F to 212° F is:

$$(212 - 50) \times 1 = 162 \text{ BTU}$$

If the boiler holds 22000 pounds mass (2638 gallons) the increase in enthalpy to bring the total mass of water to it's boiling point is therefore:

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162 BTU/lb x 22000 lb or 3,564,000 BTU.

It must be remembered, this figure is not the sensible heat, but merely the increase in sensible heat required to raise the temperature from 50° F to 212° F. The datum point of the steam tables is water at 32° F, which is assumed to have a heat content of zero for our purposes. (The absolute heat content clearly would be considerable, if measured from absolute zero at minus 459° F). The total sensible heat of water at 212° F is therefore:

(212 - 32) x 1 = 180 BTU/lb.

Latent Heat of Evaporation (Enthalpy of Evaporation):

> Expressed in BTU's, this is the amount of heat required to change 1 pound of boiling water to 1 pound of steam. This same amount of heat is released when a pound of steam is condensed back to a pound of water. The quantity of latent heat will vary with the pressure and/or temperature of a closed system.

Total Heat of Steam (Enthalpy of Saturated Steam):

> The sum of the Heat of the Liquid and Latent Heat of Evaporation, also expressed in BTU's.

Flash Steam:

> When hot condensate, under pressure, is released to a lower pressure, a percentage of it is re-evaporated into flash steam. Depending on various economics, this can certainly be a viable source of low pressure steam.

Steam

This is the oldest and most widely used form of energy in industry. Yet in most plants and engineering offices it is still not understood to a large degree. It is neglected in an even larger degree. In a larger sense, plant operations quite often can’t see the forest for the trees. That is, until there’s a forest fire.

What is meant by that is that steam is used in such a wide spread manner in most heavy industrial manufacturing plants that it’s taken for granted until something fails. A good example of that is the care and maintenance of strainers installed upstream of steam traps in order to prevent the trap from plugging up from various deposits of pipe scale and chemical residue. That same thinking would logically conclude that if scale and residue are anticipated then it would stand to reason that the strainers should be blown down and/or cleaned out periodically.

It is surprising the number of times a systems failure or reduction in efficiency can be attributed to something as simple as the impacted build-up of scale in strainers; something that could have been averted with a little planned preventative maintenance. In conjunction with that is the periodic testing of the steam traps, all too often steam systems are installed and forgotten. Until, as mentioned earlier, a system breaks down or gradually becomes more and more inefficient.

Inefficiency translates into additional operating costs. In older plants the savings potential is enormous. Not only from a fiscal standpoint, but also from an environmental standpoint.

A good way to find out how current a plant is with it's steam system maintenance is to find out if there is a steam distribution list, a location plan or any kind of a record accounting for all of the steam traps throughout the plant. If any of these items do exist, when was the last time they were checked or updated? And, are they actually used as a preventative maintenance tool?

In a surprising number of plants the answer is that there is no formal record of any kind for steam traps. If this is true in your plant then this is an indication that the potential for both cost savings and increased production is a likely possibility.

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Steam – The Basics

Steam is the gas phase of water. It is created when heat energy is added to the water until, at some corresponding point of pressure and temperature, the water can no longer remain as a liquid. This is called the "Saturation" point and any additional heat energy added to the water at this point, will cause some of the water to boil off as steam. This action is referred to as evaporation. The energy in the evaporated state is referred to as the "Latent Heat of Evaporation".

The amount of energy required for evaporation, at lower pressures, is significantly higher than the amount of energy needed to bring that same water to its saturation point. The condensing process of steam takes place when the steam is allowed to give up its latent heat content and condense back to a liquid state.

The latent heat content of the steam is the energy that is given up so readily for heat transfer. In fact the steam is so willing to give up this heat energy that we have to take various measures, sometimes costly, just to contain it long enough to get it to where we need to use it. Some of these measures include insulation to contain the heat, steam traps and separators to keep condensate from accumulating in the distribution piping and proper line sizing to reduce friction loss. These measures will be covered in more detail later.

Steam can exist in either a saturated condition or a superheated condition. The temperature of saturated steam is in constant correlation with its pressure. If you know what the temperature of saturated steam is you then know what the pressure is. Inversely, if you know what the pressure is you then know what the temperature is. To illustrate; if we were to take a container with a volume of 26.8 cubic feet and pour in 1 pound of water at the temperature of melting ice, 32° F, in all practicality the heat content, or enthalpy, would be zero. We would then begin adding heat to the water, which is at atmospheric pressure, until it reached 212° F. The heat content of that water will then be 180.2 BTU's. By adding additional heat, or enthalpy, to the water it will begin to change state and evaporate into steam. The heat content of this latent heat of evaporation is 970.6 BTU's. At atmospheric conditions these numbers will always be constant. Water at it's saturation point will contain 180.2 BTU/lb and any steam that is formed will contain 970.6 BTU/lb.

By sealing our container, and continuing to heat the water until it has been completely evaporated, that 1 pound of water, with a volume of .016 cubic feet, has been transformed into 1 pound of steam with a volume of 26.8 cubic feet. That is an increase in volume of 1675 times. This is indeed one of the considerations in designing for steam distribution, mass volume.

At atmospheric pressure we now know that the constants are: 212° F at saturation point, heat content of the liquid is 180.2 BTU/lb, heat content of the steam is 970.6 BTU/lb and the specific volume of the steam is 26.8 cubic feet.

Now we will remove the seal on our container and install a vertical piston, one that doesn't leak around the edges. The only thing holding down the piston initially is atmospheric pressure at 14.7 P.S.I.A. or zero P.S.I.G. Consequently the container still has a volume of 26.8 cubic feet and the steam is still at atmospheric pressure. By adding a 10 P.S.I. weight on top of the piston, two things happen. The steam condenses back to water and the volume in the container is reduced. However the heat of the liquid still remains at 180.2 BTU's. With 10 P.S.I. of compression on the hot water in the container it will take more energy to release the molecules into steam. At 10 PSIG the saturation or boiling point of water is 239.4° F at which point the heat of the liquid will be 207.9 BTU/lb. By adding an additional 27.7 BTU to the water we can bring it to it's saturation point of 239.4° F. If we continue to add heat the water will begin to boil off to steam. If we boil off the entire pound of water under the 10 P.S.I. pressure the latent heat of the steam will then be 952.9 BTU's and occupy 16.5 cubic ft.

Under 10 PSI pressure the constants are: 239.4° F saturation point, heat content of the liquid is 207.9 BTU/lb, heat content of the steam is 952.9 BTU/lb and specific volume of the steam is 16.5 cubic feet.

Superheated steam, on the other hand, does not have a pressure/temperature correlation. Saturated steam and water exist at the same pressure and temperature. If heat transfer continues into the steam after saturation and evaporation has been accomplished, the steam temperature will continue to raise. Once it has exceeded the temperature of saturated steam at a corresponding pressure it becomes superheated. This superheat condition can be manufactured to achieve a certain amount of superheat for use in mechanical work such as steam turbines or for transmission over long distances (this will be discussed later). It can also be the unneeded result at pressure reducing stations and other cases where there is a turndown of steam pressure to effect temperature control or pressure control, the amount of superheat depends on the turndown ratio.

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Steam Tables

All of the information just discussed has fortunately been compiled into steam tables. One table referred to is "Properties of Saturated Steam and Saturated Water"; the other is "Properties of Superheated Steam". These tables show us the correlation of this data for various pressures and temperatures. By referring to the steam tables we find the constants that were discussed plus a column for "Total Heat of Steam" and "Specific Volume of Liquid". The "Heat of the Liquid" is also referred to as "Sensible Heat". It should also be noted that in the "Properties of Superheated Steam" there is no liquid heat content since liquid does not exist in superheat. There are also various temperatures at any given pressure because there is no correlation in superheated steam between pressure and temperature.

The information given in these steam tables is as follows:

Gauge Pressure:

> Indicated in Inches of Hg for vacuum below atmospheric pressure, and P.S.I.G. for pressures above atmospheric.

Absolute Pressure:

> Indicated in P.S.I.A. with zero being absolute zero or 14.7 P.S.I. below atmospheric pressure.

Temperature:

> In degrees F for the pressure indicated.

Heat of the Liquid (sensible heat):

> The figures in this column represent the heat content, in BTU/lb, for water at the indicated pressure/ temperature.

Latent Heat of Evaporation:

> Indicated in BTU/lb, this is the amount of heat content in every pound of steam at the indicated pressure/temperature.

Total Heat of Steam:

> Indicated in BTU/lb, this is the sum of both the Heat of the Liquid and the Latent Heat of Evaporation or the total heat content of the steam at the indicated pressure/ temperature.

Specific Volume:

Indicated in cubic ft./lb, this is the space occupied by 1 pound of water or 1 lb of steam at the indicated pressure/temperature.

Pressure		Temperature	Heat of	Latent Heat	Total Heat	Specific Volume	
Lbs. Per sq. in.			the	of	of Steam		
Absolute	Gage	*t*	Liquid	Evaporation	*hg*	Water	Steam
*P*1	*P*	Deg. F	Btu/lb	Btu/lb	Btu/lb	Ft.^{3}/lb.	Ft.^{3}/lb.
14.696	0.0	212.00	180.17	970.3	1150.5	0.016719	26.799
15.00	0.3	213.03	181.21	969.7	1150.9	0.016726	26.290
16.00	1.3	216.32	184.52	967.6	1152.1	0.016749	24.750
17.00	2.3	219.44	187.66	965.6	1153.2	0.016771	23.385
18.00	3.3	222.41	190.66	963.7	1154.3	0.016793	22.168
19.00	4.3	225.24	193.52	961.8	1155.3	0.016814	21.074
20.00	5.3	227.96	196.27	960.1	1156.3	0.016834	20.087
21.00	6.3	230.57	198.90	958.4	1157.3	0.016854	19.190
22.00	7.3	233.07	201.44	956.7	1158.1	0.016873	18.373
23.00	8.3	235.49	203.88	955.1	1159.0	0.016891	17.624
24.00	9.3	237.82	206.24	953.6	1159.8	0.016909	16.936
25.00	10.3	240.07	208.52	952.1	1160.6	0.016927	16.301
26.00	11.3	242.25	210.7	950.6	1161.4	0.016944	15.7138
27.00	12.3	244.36	212.9	949.2	1162.1	0.016961	15.1684
28.00	13.3	246.41	214.9	947.9	1162.8	0.016977	14.6607
29.00	14.3	248.40	217.0	946.5	1163.5	0.016993	14.1869
30.00	15.3	250.34	218.9	945.2	1164.1	0.017009	13.7436
31.00	16.3	252.22	220.8	943.9	1164.8	0.017024	13.3280
32.00	17.3	254.05	222.7	942.7	1165.4	0.017039	12.9376
33.00	18.3	255.84	224.5	941.5	1166.0	0.017054	12.5700
34.00	19.3	257.58	226.3	940.3	1166.6	0.017069	12.2234
35.00	20.3	259.29	228.0	939.1	1167.1	0.017083	11.8959

Table 1
Typical Steam Table

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Steam Generation

In order to put into action what we have just discussed we are going to require a steam generator or boiler, which is normally fired by coal, gas or oil. As was mentioned earlier, the sensible heat of water at 32° F is taken as zero. The specific heat capacity of water is 1.0 BTU/lb/° F. Therefore raising the temperature of 1 pound of water from 32° F to 212° F will require a sensible heat of:

$$(212-32) \times 1 = 180 \text{ BTU}$$

If there is 2000 pounds of water in the boiler brought to 212° F, the total sensible heat will be:

$$2000 \times 180 = 360{,}000 \text{ BTU}$$

But, if the water in the boiler was already at 70° F, the enthalpy required to bring the mass to saturation would be:

$$(212-70) \times 1 \times 2000 = 284{,}000 \text{ BTU}$$

It must be remembered that the total heat of the liquid is still 360,000 BTU. But, since the water at 70° F already had some heat content, we only had to make up the difference to get it to saturation. And that difference is 284,000 BTU.

As we did with the 1 pound container, we will close off the boiler and continue adding heat, allowing the water to evaporate to steam. As the volume of steam increases the pressure inside the boiler will continue to increase until the supply of heat to the water is stopped or the pressure inside the boiler released. Since we want to generate steam at a specific pressure/temperature, say 150 PSIG, then we need to continue adding heat and water to the boiler and releasing the steam that is formed, at 150 PSIG. By referring to the steam tables we can see that at 150 PSIG the temperature is 366° F, sensible heat is 338.6 BTU/lb, latent heat is 848 BTU/lb and specific volume of the steam is 2.76 cubic ft./lb.

Let us take into account the peak winter demands as well as the lighter summer requirements. We can average out those demands and assume a useage rate for a moderate size plant at 20,000 lb/hr of 150 PSIG saturated steam. The actual demand for this plant, in terms of energy requirements, would therefore be:

20,000 lb/hr x 338.6 BTU/lb = 6,772,000 BTU/hr

If all of the services and equipment requiring steam had the same steam pressure requirements of 150 PSIG, it would make life a little simpler. But life and the varying requirements of steam are never simple. When determining a specific steam pressure for a user; economics, efficiency, equipment design limits, production demands and other criteria will dictate steam pressure qualifications, not simply what steam pressure is available.

In order to determine what the initial steam pressure should be generated at and how much, in pounds per hour, will be required, some preliminary work will have to be done.

One of the first steps in this process is to determine what the main users will be. Process and mechanical engineering will have this information. Through the exercise of determining the process requirements and most efficient and cost effective mechanical specifications, the equipment will be sized and the steam requirements established.

Lets assume that the process and mechanical engineers have established that 150 PSIG saturated steam will be the highest pressure required. Therefore our primary steam loop will be 150 PSIG. There will also be a requirement for steam tracing, unit heaters, air handling units and utility stations. These will be supplied, through a let-down or pressure reducing station, with 50 PSIG steam from a secondary steam loop. Other users will require steam at pressures between 150 and 50 PSIG and below 50 PSIG. The individual pressure demand for these users will be met with pressure reducing valves (or let-down stations) also.

Now that the basic pressure requirements of the steam are known we need to calculate the quantity of steam that will be needed. In order to do this the required number of BTU/hr for all steam users will have to be calculated. This is done by multiplying the lb/hr rate of the amount of steam required times the latent heat of the steam at that pressure. If one of the users required 120 PSIG steam at a rate of 300 lb/hr the demand would be:

300lb/hr x 871.5 BTU/lb = 261,450 BTU/hr(latent heat of evaporation)

By calculating the requirements of all users in this manner, a total latent heat requirement of 17,160,000 BTU is arrived at. To determine the lb/hr rate of steam that will have to be generated at 150 PSIG it would be:

21,450,000 BTU ÷ 858 BTU = 25,000 lb/hr (winter)

12,870,000 BTU ÷ 858 BTU = 15,000 lb/hr (summer)

17,160,000 BTU ÷ 858 BTU = 20,000 lb/hr (average)

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In trying to keep this fairly basic there are a few things that need to be touched on without going into great detail. In distributing steam throughout the plant, and particularly larger plants that may be generating 100,000 lb/hr, 1,000,000 lb/hr or more, line pressure loss is going to be a factor. In determining the steam generating pressure at the boiler you will need to calculate the pressure drop that will occur from the boiler to the most remote user. The result will be a nominal steam pressure somewhere in mid plant between where the steam is generated and the end user. In doing this we may be giving up a pressure loss of 5%. With a larger plant and higher production needs lets say the steam pressure required now is 600 PSIG. In order to meet the demands we just discussed we will generate steam at 615 PSIG so that mid plant would see 600 PSIG and the remote users would see 585 PSIG. There will also be line losses in the intermediate pressures such as 250 PSIG, 100 PSIG and 50 PSIG. However, since the distribution from the reducing station to the users will be more localized the runs should be shorter and pressure drop through friction loss less of a concern. With the boiler house running at a rate of 100,000 lb/hr or more, at 5%, the loss in energy is in thousands of pounds and millions of BTU's per hour.

One way to reduce this loss is in the use of superheated steam for transmission and distribution. If steam turbines are in use in the plant then superheated steam will be required in any case. The pressure and temperature of the superheated steam is chosen so that the moisture content of the steam, in the last row of turbine blades, is less than 10%. This is to prevent the erosion of the blades due to accumulated water particles. Superheated steam does not readily give up it's latent heat as does saturated steam. Although a pipeline pressure drop will still occur there won't be a significant loss in BTU's.

Another cause of pressure drop in pipelines is size. This is where the pressure of the steam has a major effect on the pipe size. If you will notice in the steam tables in the column for "Specific Volume of Steam" there is quite a difference in the volume of steam throughout the pressure range. The specific volume for 150 PSIG saturated steam is almost 4 times greater than that of 600 PSIG saturated steam. This has a significant effect on pipe sizing and operating pressure. In the design of the distribution system it should be considered that higher pressure steam can move more volume through smaller pipe size than lower pressure steam. Sometimes it is more economical to increase the initial steam pressure in order to reduce the pipe size.

What has to be considered here is the reduction of latent heat by volume as the pressure of the steam increases. As the latent heat content decreases with increases in pressure the volume of steam will have to increase in order to provide the same amount of latent heat.

A determination will have to be made about what pressure this stops being economically feasible. You will have to take into account the turn-down ratio on the pressure reducing valves, additional fuel costs for generating more steam, potentially higher cost in material and a higher cost in insulation. All of these and possibly more, are governing factors in sizing a steam system.

In laying out a distribution system the single objective is to get steam to the far reaches of the plant at as close to the generated steam pressure as possible. Doing so will require attention to line size, insulation, configuration, moisture separation and steam traps.

Line sizing is based on either pressure drop per 100 ft. or velocity in ft/sec or ft/min. Design parameters will sometime vary from plant to plant but, as a rule, allowable pressure drop is .5 lb/100 ft for runs 300 ft and less. For pipe runs over 300 ft long, .1 lb/100 ft is acceptable. Velocity is normally held at 6,000 to 10,000 ft/min or 100 to 167 ft/sec for saturated steam. Superheated steam can operate at higher velocities, 7000 to 20000 ft/min.

You have to maintain a balance between operability and economics. The larger the pipe the less pressure drop but also a higher cost for the pipe and insulation in addition to more heat transfer area for losing BTU's. Depending on the valving and pipe configuration, higher velocity rates may increase noise. And even though that might be tolerated, erosion to valve trim and piping becomes a definite consideration. This becomes even more of a concern where there is a high moisture content in the steam. With water droplets traveling at speeds of perhaps 12,000 ft/min they become very detrimental to the pipe internals.

Insulation is too broad a factor to go into with this report. It should be noted however, that this too hinges on economics. The cost to achieve a 100% insulation efficiency factor is both cost prohibitive and physically restrictive. Good design practice will usually apply a 75% efficiency factor when calculating insulation thickness.

In getting steam from the boiler to the user there is never a straight line. High points and low points are poor points when it comes to steam. As mentioned earlier, saturated steam is always on the verge of condensing back to water. And no matter what is done within the realm of economic restrictions some steam is going to condense. It's up to the designers to minimize the rate at which it does condense and to handle the condensate when it does. With that in mind lets take a look at what happens when steam does condense.

The feedwater used in generating steam will, of course, contain oxygen. It can also contain bicarbonate and carbonate alkalinities which, when broken down due to high temperatures, will produce $C0_2$. These two gases, O_2 and CO_2, alone or combined, when disolved in condensate are very corrosive. The oxygen causes oxygen pitting while the carbon dioxide, in solution with the condensate, forms carbonic acid. When combined, the oxygen accelerates the corrosive effects of the acid.

Deaerating the feedwater removes almost all of these gases. In general, as the feedwater enters the deaerator low pressure steam, typically 5 PSIG, is used to break up the water into a spray continuing across the spray carrying off the gases. There are also tray type deaerators and combinations of both the spray and tray types. They will typically have a residual deaerated feedwater storage tank combined with the deaerator.

Since deaeration does not remove all of the oxygen, an oxygen scavenger in the form of sulfites or hydrazine is employed. These scavengers are added to the already deaerated feedwater.

Even with the pretreatment of the feedwater there will still be carryover in the steam. Additionally there will be air in the system from start-ups and batch operations. The problem occurs when steam condenses. As it condenses these gases will separate out which will then have to be dispatched along with the condensate. Both the condensate and the gases need to be removed as quickly and as efficiently as possible. The oxygen and carbon dioxide are corrosive agents, the air acts as an insulator and the condensate can diminish heat transfer, create flow restriction, cause water hammer, moisturize the steam and corrode pipe and equipment. There are ways to alleviate a lot of the potential problems by using air vents, separators and steam traps in addition to good design practice.

Good design practice, in terms of configuration, should include, among other things, the following: proper slope, the elimination of pockets, proper trapping of condensate when pockets do occur, strategic location of steam traps and a configuration that integrates flexibility to keep the system piping itself within allowable stress ranges during expansion and contraction cycles.

An air film 0.04" thick has the same resistance to heat transfer as water 1" thick, iron 4.3" thick or copper 43 feet thick. As a film it acts as an insulator, in solution with steam it deprives the steam of its full heating potential. In other words, air will assume a part of the total volume or pressure that is available.

In gas mixtures, each gas assumes a part of the total volume or pressure. This is referred to as partial pressure. The partial pressure of each gas is dependent upon its proportion of the total mixture.

If we were to have a total steam line pressure of 50 PSIA consisting of 80% steam and 20% air the total steam pressure would be:

$$.80 \times 50 = 40.0 \text{ PSIA}$$

and the total air pressure would be:

$$.20 \times 50 - 10.0 \text{ PSIA}$$

As a result the steam would effectively be 40 PSIA steam in a 50 PSIA line. In checking a thermometer in the line at that location we would find the temperature to be 267° F for the 40 lb steam and not 281° F for the 50 lb steam we would expect to find. That is a 14° F difference between the two pressures. In addition there will also be a change in BTU's. Although the 40 lb steam has more enthalpy of latent heat per weight than does the 50 lb steam it has less by volume. As a comparison, inside the pipeline, a 1 square foot area of the 50 lb line would contain 108.7 BTU; a 1 square foot area of the 40 lb line would contain 88.9 BTU. That is a difference of 19.8 BTU/sq ft.

In effect the air has displaced a portion of the enthalpy needed, by displacing a portion of the steam. In order to make the distribution system as efficient as possible it becomes necessary to remove any air before it can effect heat transfer by filming or becoming mixed with the steam.

Knowing that it is virtually impossible to keep air, oxygen and carbon dioxide from getting into a system, lets deal with getting them out of a system. As mentioned earlier, these gases become free when the steam condenses. All steam traps will pass air and other gases along with the condensate, as it forms. The only difference is that some will handle a good deal more air than others. Without getting into discussions of the various types of traps, which will be discussed at length a bit later, we will assume a worse case which would be a batch operation or a modulating steam control valve with a vacuum breaker downstream of the control valve. When the control valve shuts off steam to the heating unit, the steam between the control valve and the steam trap will begin to collapse due to condensing. As the steam begins to collapse a vacuum begins to form. When the vacuum overcomes the set point of the vacuum breaker it will open, allowing air to replace the volume of steam. Now, in the heating unit, we have a volume of air which we have to evacuate as quickly as possible when steam is introduced again.

Depending upon the configuration of the heating unit, it may require only a steam trap or it may require two or more additional air vents. If it's a piping coil of some sort it may only require a float and thermostatic trap. As the steam is supplied to the unit again it will purge the air ahead of it and out the thermostatic vent in the trap. When the steam reaches the trap the vent will close. In this circumstance, the air normally does not combine with the steam. The air is located in a dead leg separate from the steam. When steam is introduced again it will move the air out ahead of it. Only if the air can not be purged fast enough would it be forced to combine with the steam causing inefficient heat transfer.

If the heating unit has a large cavity, such as the shell side of a shell and tube heat exchanger, a pocket of air could develop that would require venting. A thermostatic air vent should be located at this point. The air vent will allow the air to be removed from the steam chamber before it has a chance to combine with the steam or film up inside the heat exchanger or piping.

A peculiar side issue to air vents and even the ambient sensing valves used to drain condensate lines automatically before they freeze up, is the fact that they are sometimes misinterpreted. Keeping in mind that the pocket of air in a steam line will be hot and humid when released to the atmosphere, what will happen when it is winter and 0° F outside?

The moisture content in the escaping air will have a tendency to freeze upon contact with the cold discharge piping. After several discharges the formation of ice will become apparent. This presents the illusion that the air trap is leaking. When in essence it is doing its job.

The same holds true for the ambient sensing valve. This valve senses the internal temperature of the pipe contents. When the temperature of the fluid inside the piping drops to a predetermined set point, about 8° F above freezing, the valve will open. This allows the fluid to be drained from the piping before it has a chance to freeze up. In doing it's job this valve has a tendency to be misinterpreted also. If it is 0° F outside and the valve discharges, some of the fluid will freeze upon contact with the cold discharge piping. With repeated discharges there is an excellent likelihood that a good sized icicle will begin to form off the end of the discharge piping. Again this gives the appearance of a leaking valve when, in actuality, the valve is functioning properly.

Now back to distribution design. As mentioned earlier, moisture in steam is both inefficient, in regard to heat transfer, and detrimental to piping and equipment. In identifying the fact that moisture is going to condense out of saturated steam we have to determine why it does and how best to control it.

The two main factors attributable to the formation of condensate prior to reaching the various users are; heat transfer and pressure drop. As stated earlier, it is not economically feasible, in most cases, to achieve 100% efficiency in insulation design. We therefore have to design more realistically for 75% efficiency. Other insulation inefficiencies involve poor design standards, poor installation and, after a period of time, poor maintenance upkeep.

Any time a fluid moves through a pipeline there will be a pressure drop. Just how significant that pressure drop is depends on the characteristics of the fluid, the condition of the internal walls of the pipe, the configuration and length of the piping.

The designer can't do anything about the fluid, which is steam at a predetermined pressure. Neither can the designer do anything about the internal roughness of the pipe wall. What the designer can do is determine the piping configuration and research the best type insulation for the job. Installing and maintaining the insulation will be the responsibility of plant maintenance.

In routing the main steam supply line a constant slope should exist in all horizontal runs, sloping down in the direction of the steam flow. A good rule of thumb, for slope, is 1 inch in 20 feet, although this is certainly open to variation. Using a rule of thumb in general is fine, only after that rule of thumb has been determined to work and provided the designer can recognize the marginal areas where that rule of thumb may not apply. In order to do that we have to understand the reason why and the basis for, sloping the lines in the first place.

As condensate forms on the inside pipe walls it will run down the walls to the bottom inside of the pipe. When the condensate gets there it has to be able to flow easily to a point where it can be removed; a drip leg. Knowing that it is going to be easier for the condensate to flow with the steam we will need to slope the line down in the direction of the flow. If the flow of condensate was going against the flow of steam the following would

Occur: With the steam moving at a velocity of 6000 to 10000 fpm the condensate would tend to eddy into pools trying to overcome the friction of the steam on its flow to the low point upstream. As the condensate builds up in this manner the velocity of the steam will cause it to be picked up and introduced into the steam as a mist. Essentially humidifying the steam. As the moisture level of the steam rises the heat content lowers. It is apparent by this that you never want to consider having to slope the steam line down toward the flow.

In determining whether or not our rule of thumb, 1" in 20 feet, will be sufficient slope we need to know the reason for the slope. What is behind this is the fact that we need to keep the condensate flowing so it can be trapped and carried off before it has a chance to accumulate. If the steam line was perfectly level then the condensate would accumulate at the inside bottom of the pipe and, with steam as it's motive force, flow in the same direction as the steam. But in reality, horizontal runs of pipe will deflect or sag btween supports. The amount of deflection dependent upon the pipe size, schedule and the span between supports.

Lets use, as an example, a 4 inch sch. 40 steam line that is supported every 20 feet. The deflection in this line would be approximately 3/16". If this line was supported at the same elevation at each support there would be a series of 3/16" pockets that would accumulate condensate. In order to alleviate the problem the pipe will have to slope in excess of 3/16" in 20 feet. So in this case the 1" in 20 feet would be sufficient.

Trapping condensate in supply headers is much like the difference between using a dress glove to catch a line drive baseball or using a fielders mitt; which would you prefer? Aside from the pain issue the obvious selection would be the fielders mitt. The larger catching area provides more opportunity to catch the ball. In most cases a ½" trap and condensate return line is all that will be required to handle the condensate at a drip leg. But if you are trapping a l0" or 12" header and run that ½" trap line directly off of the header, the chance that the condensate, all of it at any rate, will find that ½" opening is very slim. In order to try and capture all of the condensate, a branch connection large enough to allow the condensate to drain in to it while moving down the header, needs to be provided. Refer to Table 1 for recommended drip leg sizes

RECOMMENDED DRIP LEG SIZES	
HEADER SIZE	**DRIP LEG SIZE**
1/2" Thru 4"	Same as header
6"	4"
8"	4"
10"	4"
12"	6"
14"	6"
16"	8"
18"	8"
20"	10"
24"	12"

Table 2
Relative Drip Leg Size

If there is a situation where a drip leg is placed at the end of a horizontal run and there is a riser at that point always use a full size tee. For two very good reasons; using an o-let fitting, a half coupling or stubbing into the line will have to be done at the tangent of the elbow in order to install the drip leg at the lowest point. In doing this it may mean encroaching on the buttweld of the elbow, which, in most cases is not a good practice. Also, when a full size tee is installed it is more cost effective to install a full size section of pipe with a cap. Rather than install a reducer, a section of pipe and a cap in an effort to comply with a drip leg sizing schedule like the one shown above.

There should be a drip leg located at all low points or pockets**.** In long straight runs there should be a drip leg located about every 300 feet. This assumes a properly sized and insulated system is installed. If the piping is sized too small undue friction loss will create additional amounts of condensate. Likewise, with insulation that is not sufficiently thick enough. It will permit an additional amount of condensate to form due to added radiant heat loss.

Piping News Report
An informative newsletter for Piping Designers and Engineers

Basics of Steam System Design

by W. M. (Bill) Huitt

Steam Traps

Having gone to a lot of expense in generating steam and installing a distribution system we now need to get it to it's various users around the plant as efficiently as possible. Efficiency translates into getting it to the users with a minimal loss in latent energy at a reasonable cost. This is where the steam trap comes in. Without steam traps unabated condensate would form in distribution piping, creating a wide range of problems. In addition there would be no essential control at the users. Steam would enter a set of tubes or a coil at one end and come out the other as either steam, condensate or a two-phase mixture of the two; very dangerous, damaging and wasteful.

By installing steam traps in strategic locations throughout the distribution system we can alleviate those problems. A steam trap on the outlet side of a heat exchanger allows the steam to reside there until all of the latent heat energy is transferred and the accumulated condensate is carried off. With proper placement and specification of steam traps for these purposes we can create and maintain an efficient, cost effective steam supply and distribution system. In knowing this, the next step is to determine the best trap to use for a given application. So lets take a look at the various types of traps and what each is or is not suited for.

In identifying steam traps we can break them down into three main groups: Thermodynamic, Thermostatic and Mechanical.

THERMODYNAMIC: In addition to downstream flash steam assist, this type of trap operates on the difference in velocity or kenetic energy between steam and condensate passing through a fixed or modulating orifice.

THERMOSTATIC: This type of trap operates on the principle of expanding liquids and metals used to drive a valve into or back it away from a seat.

MECHANICAL: This trap is made up of mechanical apparatus that are driven by the density of the condensate to operate a float or a bucket.

On a generic basis, let us take a look at the various types of traps within each group. Generic is used because the various manufacturers have several different designs of the same basic trapping principle. What we will discuss here are the primary trap designs in each category. For instance, within the Thermodynamic category of traps there are the orifice, disc, impulse or piston and labyrinth types. We will only discuss the orifice and disc types because of their predominance.

THERMODYNAMIC STEAM TRAPS

Orifice Type - Description

This is nothing more than a steel plate with a hole. Some are styled after a pancake blind and others are made as an integral part of a union. The basic principal behind this trap is that the expanded volume of steam compared to condensate has a throttling effect at the orifice. With a properly sized orifice, condensate, at its lower specific volume will pass through the opening at a comparatively slow velocity. As steam begins to reach the orifice plate the condensate will begin to expand. As the condensate expands, the velocity through the orifice will increase and the throttling action will start to take place. According to the laws of physics, you can have mass flow or you can have increased velocity through a fixed opening but you cannot have both.

Consequently, once steam reaches the orifice the velocity has increased to such a point that the pressure drop across the orifice creates the throttling effect. If you were to watch this in action it would appear that there is, indeed, steam passing through the orifice. What you would actually be seeing is flash steam. This occurs when condensate, at a higher pressure, is passed to a lower pressure. A portion of the sensible heat content of the condensate at the higher pressure will become latent heat at the lower pressure and flash to steam. We will discuss flash steam in more detail a little later.

About the only good thing that can be said for this type of trap is that it has no working parts to fail and it requires very little space for installation.

Disadvantages to this trap are its limited capacity range, its inability to discharge a large volume of air at startup and its tendency to be nothing more than a leak source when live steam reaches the orifice. If live steam is allowed to blow through the orifice for an extended period the orifice can become enlarged through erosion. This, over time, provides a larger, more erratic, opening. Which, in turn, provides an increasingly larger and costlier leak source.

Orifice Type - Applications

The orifice trap, although not practical as an in-service steam trap, does have its place. Where temporary steam trapping is required for a short period of time the orifice trap is an inexpensive and practical solution.

In the event that a trap has failed and a permanent replacement is not available, a calculated orifice can be temporarily installed with reduced down time and minimal expense.

If a permanent by-pass has been installed around the trap an orifice can be installed in that by-pass along with a throttling valve. This alleviates the need to blind throttle the steam when the by-pass is placed into service. The calculated orifice will automatically throttle the by-pass eliminating the need to guess at how many turns to open and set the throttling valve. Should the orifice be oversized (a probable indication would be water hammer in the receiving condensate line) the installed throttling valve can serve as backup to correct it.

Disc Type - Description

Aside from the orifice trap this is probably the simplest trap on the market and yet is the most widely used. The disc trap is made up of three primary components: the body, the cap and the disc. Like the orifice trap, the operation of the disc trap utilizes the difference in specific volume between steam and condensate. With flow moving through a fixed orifice this translates into a difference in velocity between steam and condensate. Operation of the disc trap also utilizes flash steam as an operating force to work in conjunction with the velocity of the steam.

In order to understand the operation of the disc itself we have to know the principle under which it operates. Bernoulli's Principle, simply stated: THE PRESSURE OF A FLUID (LIQUID OR GAS) DECREASES AT POINTS WHERE THE SPEED OF THE FLUID INCREASES.

Applying this to the disc trap we are, in fact, creating a low pressure zone between the disc and seats whenever we increase the velocity of the steam or condensate flowing through this zone. In addition we are providing a small chamber for the accumulation of flash steam above the disc. Fig. 1 shows a simple disc trap. As flow, in the form of condensate, moves into the trap and through the inlet orifice it forces the disc to lift, allowing the condensate to pass through and out the outlet. As the temperature of the condensate reaches it's saturation point a percentage of that condensate will flash as it exits the inlet orifice. When steam reaches the inlet orifice two things will immediately happen: the velocity will increase sufficiently to create a low pressure zone between disc and seats pulling the disc down upon the seats. At the same time flash steam will have formed behind the disc and, with the exit orifice sealed off, the pressure induced by this non-escaping flash steam will hold the disc in place. The disc will remain in place until the flash steam has condensed, thus allowing the disc to open again.

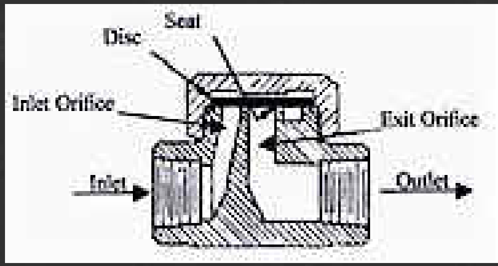


Fig. 1 - Thermodynamic Disc Trap

You can see by this that the governing factor behind the operation of the disc trap is the flash steam. The amount of condensate that accumulates upstream of the trap is dependent upon the amount of time the flash steam remains in that state. This, in essence, is what regulates operation of the trap.

Since the flash steam is contained in the cavity under the cap the length of time that the flash steam exists as such depends on the amount of heat transfer through that cap. What regulates the trap is the ambient temperature the trap is subjected to. If we were to watch an uninsulated disc trap, open to grade, on an outside installation we could observe the change in the frequency of cycles the trap would go through from summer to winter. Providing that the only change in heat transfer rate is caused by the change in ambient temperatures the cycles in the winter would be more frequent than in the summer owing to the additional heat loss in the colder temperature. This would, at the same time, compensate for the added heat transfer load that the unit being trapped would realize.

Unlike the mechanical traps that follow, the cycle of this type of trap depends on the condensing rate of the flash steam that keeps the disc closed. By performing a simple test, of pouring water over the cap of a trap discharging to atmosphere, we can see abruptly, the change in the cycle of the trap as it quickens. Inversely we can put an insulation cap on the trap and see the cycles slow. It is obvious that the discharge of condensate is dependent upon the resident time of the flash steam in the disc chamber and not on the build-up of the condensate itself.

With that in mind, allowances regarding the upstream length (or more appropriately, the capacity) of the piping will have to be made for the installation of the disc trap. The trap itself is very compact, which adds to its popularity, but does not allow it to store condensate. Any build-up in condensate will take place in the piping upstream of the trap. Consequently the smaller the pipe and the more condensate that accumulates between cycles, the further upstream the condensate will backup. If the trap is inadvertently insulated it will compound the situation by not allowing the flash steam, holding down the disc, to condense more readily allowing the condensate to pass.

In too many cases the disc trap has been installed as close as possible to the drip leg which puts it in close proximity to the unit it is trapping. If you have access to equipment, particularly unit heaters, that are using a disc trap mounted within approximately 6" of the drip leg and they have been installed in this manner for several years please inspect them. What you are likely to find is a corrosion problem that will be apparent by leaking at pipe joints and within the lower section of the unit heater coils. With a little research you may even find that in some of the older locations, units have been replaced or repaired one or more times.

The cause of this is the continual backup of condensate, under pressure, into the piping and equipment. As mentioned earlier this condensate can be very corrosive if not removed quickly. Allowing the condensate to repeatedly backup into the piping and equipment is essentially the same as allowing it to reside there. In addition, by allowing the condensate to backup into the equipment the heat transfer efficiency of the unit will fall off proportionately to the area that is flooded.

In most cases placing the trap 12" to 18" from the drip leg will provide enough of a reservoir to prevent condensate from backing up into the equipment. If there are space constraints keep in mind, the distance from drip leg to trap does not have to be in a strait line.

Disc Type - Application

This type of trap is an excellent general service trap that can be specified for both saturated or super heated steam. Although it operates well throughout a wide pressure range please verify its low pressure limitations with the manufacturers literature. This trap will have a low pressure limit that it will need to exceed in order to pass condensate. Nor should it be used where steam throttling could fluctuate pressure at the trap to near atmospheric pressures. Within its capacity range this is an excellent trap for a wide range of applications.

The disc trap is lightweight, compact, easy to install, easy to maintain, withstands waterhammer and is relatively less expensive than other types of traps. When a slow warm up is part of the start-up procedure it is an excellent trap for automatic start-ups. If, during start-up, steam is introduced suddenly to a system, without benefit of a warm up cycle, the rush of air preceding the condensate and steam to the trap will create enough velocity across the trap seating surface to force the disc to close. Disabling the system in this manner is referred to as air binding. Since there is no flash steam to keep the disc closed the disc will have a tendency to flutter, or chatter. Although it will continue to discharge air it will prevent the trap from discharging it at the necessary rate.

In the case where steam is introduced without warm-up it is recommended to either provide additional valving for blowdown during start-up or specify a different type trap altogether.

There is a tendency to consider the disc trap as self-draining. In the event that the steam supply is shut off to the trap, the trap installed in the vertical and the discharge down and open to atmosphere, it is suggested it will drain any remaining condensate. This of course is a major consideration when designing for freeze protection.

What actually occurs when the supply of steam is shut off is the following: the steam remaining in that isolated section of pipe will continue to condense. Initially the disc in the disc trap will remain open. With some residual pressure still in the line condensate will continue to drain through the trap. As the steam condenses and its volume collapses it will attempt to draw in outside air to replace that lost volume. When this occurs the displacement air will be pulled in through the trap orifice, its only path in. As the displacement rate increases the velocity of air will force the disc down onto the seat to close off that section of pipe or tubing between the steam shut-off valve and the trap. With that section of piping closed off the condensing steam will create a vacuum essentially locking the disc in its closed position. With residual condensate remaining in the line that section now has the potential to freeze.

There are three general ways of preventing this. One is to install a vacuum break in a high point between the shut-off and the trap. The second is to install an ambient sensing valve in a low point between the shut-off and the trap. Both of those methods will control the situation automatically. The third is to install a low point, manual drain valve between the shut-off and the trap.

There is also a fourth method to be considered. Armstrong manufactures a valve that operates as an automatic low point drain. It operates essentially like a vacuum breaker with a set point at 5 PSIG. It is simply a ball check in which the ball remains in its seat as long as the steam line is under pressure. The ball is under 5 pounds tension from a spring that is attempting to push the ball off of its seat. If steam pressure is shut off to the line and the in-line pressure drops to 5 PSIG or less the valve will open to drain the accumulated condensate.

THERMOSTATIC STEAM TRAPS

Balanced Pressure Type - Description

The Balanced Pressure type trap operates on the principal of liquid expansion due to an increase in temperature. The liquid is contained in a bellows (Fig. 2) internal to the steam trap and fixed at one end. Integral to the bellows is a valve attached to its free end.

The liquid in the partially filled bellows can be as simple as distilled water under vacuum or an alcohol combination to reduce its vapor point to a lower degree than water. At ambient conditions the bellows is contracted with the valve away from the seat. When steam, or condensate near it's saturation point, comes in contact with the bellows the liquid inside the bellows expands and drives the valve into the seat closing off steam flow.

As the steam condenses, collects and cools, the bellows will cool and contract, backing the valve away from the seat and allowing the accumulated condensate to pass. As the condensate passes through the trap and is replaced by steam the bellows heats up again. As steam comes in contact with the bellows it expands and closes the valve, shutting off flow. As you can see this trap is not governed not only by the pressure differential but rather by the temperature of the steam and condensate.

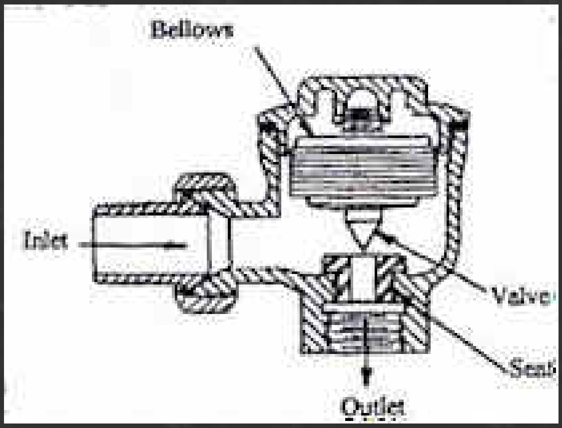


Fig. 2 - Balanced Pressure Thermostatic Trap

Balanced Pressure Type - Application

This trap is excellent on start-ups, automatic or manual. It remains in its full open position allowing for the removal of air and condensate until steam comes in contact with the bellows. Additionally it is a relatively small, lightweight trap that handles a wide range of condensate loads over a wide range of pressures. One concern with this trap is its susceptibility to water hammer. When specifying this type of trap, consider the potential for water hammer if the trap discharges into a common header or sub-header. If that potential exists specify a check valve to be installed downstream of the trap and prior to its connection to a common header.

Although there are designs of this type of trap that will operate in super heated conditions it is generally not recommended for that service. If you feel the need to use this trap in super heated service check the manufacturers recommendations before specifying.

Applications include drip legs, heating coils, steam tracers and various process requirements. Consideration, as in the disc trap, must be given to providing sufficient upstream piping capacity for the accumulation of condensate. As with the disc trap, the designer should provide an upstream reservoir sufficient enough to contain the residual condensate preventing it from backing up into equipment. A 12" to 18" run of pipe, in most cases, should be sufficient. If the possibility exists for the flow rate of condensate to fluctuate to a point at or above the capacity of the trap a longer or larger diameter section of pipe may be required in order to contain the condensate between the trap and equipment.

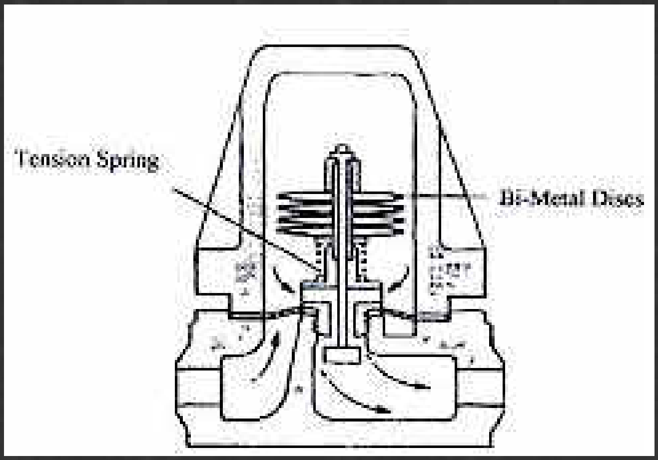


Fig. 3 - Bi-Metallic Thermostatic Trap

Bi-Metal Type – Description

Like the Balanced Pressure trap, the Bi-Metallic trap is also governed by temperature variations. However, as the name suggests it does so by utilizing the differing expansion rates of metals when exposed to temperatures above or below ambient.

By laminating two dissimilar metal strips and exposing the resulting composite to elevated temperatures the differing expansion rates of the composite metal strip will cause it to bow. The higher the temperature the more extreme the bow.

The reaction of the Bi-Metal composite is utilized in several different forms with various valve and seat arrangements. The two most widely used designs are variations of the bellows style. One uses Bi-Metal circular discs. The two sets of Bi-Metal laminates are joined at the perimeter with the metal layer of each Bi-Metal disc having the lower rate of expansion facing each other. Several sets of these joined discs may be stacked to increase the force applied when they expand.

Through the center of the stacked discs is a rod, which is attached to the upper most disc. The rod runs through the sets of discs then through a seated orifice. At the end of the rod is a valve. In the relaxed or ambient condition the discs are flat against one another. In their hot condition each set expands against itself causing the bellows to expand. As the bellows expands it draws the valve into the seat of the orifice blocking off the flow of steam.

Bi-Metal Type – Application

Like its Balanced Pressure counterpart, this trap is excellent on start-ups, automatic or manual. It remains in its full open position allowing for the removal of air and condensate until steam comes in contact with the Bi-Metal element. Additionally it is a relatively small, lightweight trap that handles a wide range of condensate loads over a wide range of pressures. The Bi-Metal trap is more resistant to water hammer than the Balanced Pressure trap. This makes it a good alternative when a thermostatic trap is the first choice and the potential for water hammer exists.

Applications include drip legs, heating coils, steam tracers and various process and utility requirements. Consideration, as in the disc trap, must be given to providing sufficient upstream piping capacity for the accumulation of condensate. As with the disc trap and the Balanced Pressure trap, the designer should provide an upstream reservoir sufficient enough to contain the residual condensate preventing it from backing up into equipment. A 12" to 18" run of pipe, in most cases, should be sufficient. If the possibility exists for the flow rate of condensate to fluctuate to a point at or above the capacity of the trap a longer or larger diameter section of pipe may be required in order to contain the condensate between the trap and equipment.

Liquid Expansion Type – Description

This type of trap, like the other thermostatic traps, is designed to respond and control the release of condensate relative to the temperature of that condensate. This trap (Fig. 4) contains an oil-filled cylinder encasing a sealed bellows with an enclosed piston driven rod with a valve on one end. When the temperature of the condensate, flowing through the trap housing, begins to rise the oil in the cylinder begins to expand in response. As the oil expands it drives the rod assembly toward a valve seat located in the inlet of the trap body. As the condensate surrounding the cylinder cools down the oil contracts allowing the rod assembly to retract from the valve seat. Condensate begins to flow until it heats sufficiently to close the valve again.

The Liquid Expansion Trap also has an adjustable nut on one end. This allows the set temperature to be adjusted within the operating range of the trap.

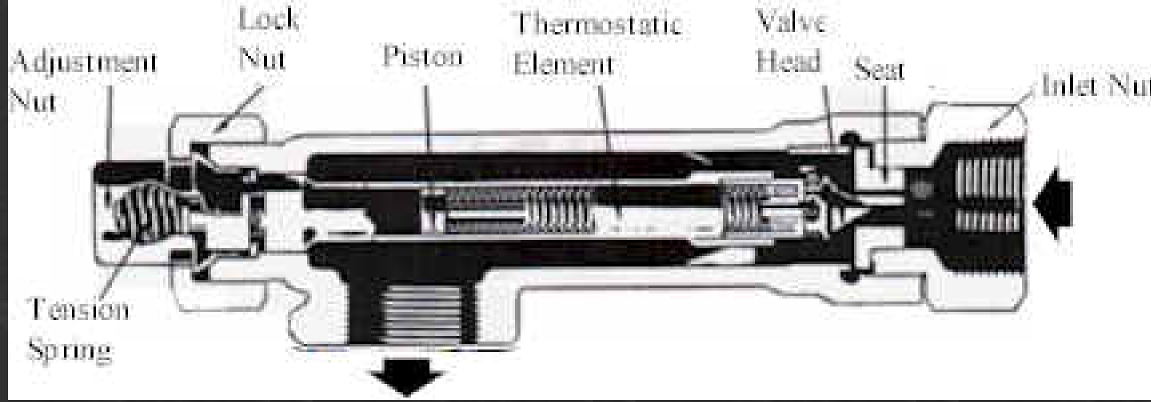


Fig. 4 - Liquid Expansion Trap

Liquid Expansion Type – Application

The Liquid Expansion Trap has limited use in trapping steam. It releases condensate at or below an adjustable set point of 212° F. Normal operation subjects the piping and equipment upstream of the trap to condensate flooding. In some situations this is acceptable. This trap cannot respond to a wide fluctuation in steam and condensate temperatures either. The type of service this trap is most suited for would be something on the order of maintaining a storage tank temperature. A condition that is constant and does not require response to sudden changes in heat transfer rates or continual wide shifts in temperature.

MECHANICAL STEAM TRAPS

Inverted Bucket Type – Description

Next to the Disc Trap this is the most widely used trap in the industry. It can arguably be said that this trap is overused. It has such a wide use range that it is probably specified out of misunderstanding in a large number of situations. These aren't necessarily situations where this type of trap wouldn't work but rather situations where a less expensive, smaller, possibly longer lasting type of trap could have been applied.

This trap (Fig. 5) operates on the principal of an inverted water glass (the component referred to as the bucket). The air and CO_2 entrapped in the inverted bucket provides buoyancy keeping the inverted bucket in its raised position. Extending partially inside this inverted bucket is a dip tube, which is where the condensate and steam enter. In the top of the inverted bucket is a vent hole. This allows air and CO_2 to continually discharge. Attached to the top of the inverted bucket is a lever and valve assembly, which is attached to a valve seat insert.

On start-up, the initial surge of condensate, which precedes the flow of steam, will provide the liquid prime needed to make the Inverted Bucket trap work. That prime is what seals the bottom open portion of the bucket. Without it, steam would enter the bucket through the dip tube flow out the bottom of the inverted bucket and pass through the outlet. At times, due to transient surges, super heat conditions or a transient vacuum condition, a trap may lose its prime. An obvious indication of this is a sudden water hammer problem in the condensate return system due to the trap allowing steam to blow through because the water seal is not there to retain the steam.

The trap operates by entrapping steam in the prime sealed inverted bucket as it enters through the dip tube. As the steam resides in the inverted bucket it condenses. At this point there are two issues that have to be resolved in order for the trap to function properly: 1. As the steam condenses inside the bucket, a portion of the air and CO_2 dissolved within the condensate has to be released. If the CO_2 is not released from the condensate, and remains dissolved, it forms carbonic acid having a detrimental effect on carbon steel, cast iron and bronze material; 2. If the released air and CO_2 are not evacuated they will accumulate until they bind the trap, preventing it from functioning.

In order to evacuate the released air and CO_2 a vent hole is provided in the top of the inverted bucket. The hole is sized to allow the air and CO_2 to escape the bucket and flow out through the discharge at a rate that will insure that a residual amount remains to help maintain buoyancy for the bucket.

A proportional volume of air and CO2 inside the bucket keeps the bucket raised, keeping the valve closed. As the condensate accumulates and displaces the volume of air and CO_2 it reaches a point at which the bucket can no longer sustain its buoyancy. At that point the bucket drops pulling open the valve and allowing condensate to flow out through the discharge.

This trap is specified based on the differential pressure between the inlet and outlet pressures of the trap. With the length of the valve lever fixed the differential pressure is used to determine the weight of the bucket. The result allows the bucket to lift and reset the valve after dumping its condensate. The calculated weight of the bucket also allows the bucket to drop against the upstream pressure when it's full of condensate.

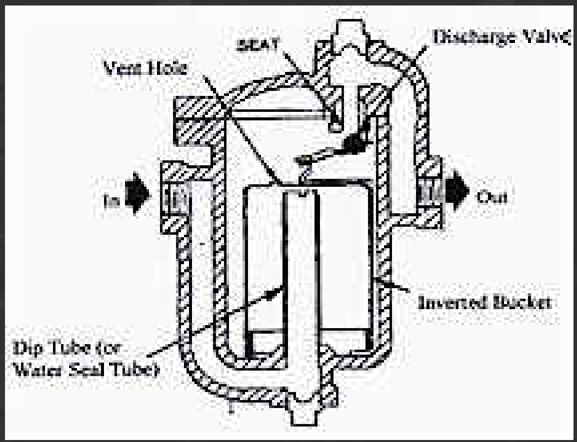


Fig. 5 - Inverted Bucket Trap

Inverted Bucket Type – Application

As indicated earlier, the Inverted Bucket Trap is used for a wide range of applications. It has such a general use range that designers, engineers, mechanical contractors and maintenance personnel tend to gravitate to the

Inverted Bucket as a reflex rather than through a determining process. Because of this there is money being wasted not to mention misapplication in size and function.

In services where a 1/2" Disc Trap could have been specified 1 1/2" Inverted Bucket Traps have been installed. In too many cases like that it is apparent that whoever specified those steam traps didn't know what they were doing. If you don't know what you are doing or your not sure, contact one or more steam trap manufacturing representatives. But, as always, when working with someone who is trying to sell you something, DO YOUR HOMEWORK. Be ready to both ask questions as well as provide them with proper, concise information. They would much rather help you size and match the trap for the application than repeatedly get called back for something that doesn't work properly. They're in business to make money and they can't do that by spending time at your facility replacing poorly specified traps or by loosing customers due to ineffective engineering assistance.

The Inverted Bucket Trap can be used over a wide range of pressures and temperatures to trap process and utility equipment, as a drip trap and in selected HVAC service. It can be installed inside or outside. However, if an Inverted Bucket Trap is specified for outside service in a freezing environment, a stainless steel type would be highly recommended. These encapsulated traps, like Armstrong's model 1010, have the capacity, due to their material of construction, to withstand inadvertent freeze ups. When condensate inside the trap freezes, the trap expands like a balloon. Once thawed it goes right back into service.

Should this occur it is always a safe bet to replace the trap at the next opportune time. It isn't suggested that these traps be placed into a potential below freezing environment without freeze protection. On the contrary, if the possibility of freeze up exists then the designer should take steps to configure the installation, heat trace the trap and piping or otherwise design an installation that protects the trap from freeze up. However, these stainless steel encapsulated traps are an excellent final protection against having to shut down a line or system because several steam traps ruptured and have to be replaced.

Float & Thermostatic Type – Description

As the name implies, the Float & Thermostatic Trap utilizes two individual mechanisms that operate in conjunction with one another. The float operates a valve (Fig. 6) that controls the discharge of condensate. The thermostatic element controls the release of air and CO_2. The float itself, which is normally a ball type, is located in the lower portion of the trap body. It is attached to a rod which is, in turn, attached to the body of the trap in such a way that it is free to pivot about that point, allowing the float the freedom to move vertically. Near the end where the rod is attached to the body a valve is attached to the rod. The valve is positioned so that when the float is at rest the valve is seated in the outlet of the trap.

The thermostatic element is located in the upper part of the trap body. One end of the element is fixed allowing the opposite end with an attached valve to move in and out of a seated vent discharge opening located in the body of the trap. That vent discharge is connected to the discharge for the condensate. In its relaxed position the valve is pulled away from the seat.

With the trap out of service the float rests in its bottomed out position, with the valve closed. The thermostatic vent element is in its contracted position with its valve open. As steam is introduced to the system and begins to move through the piping it will force the volume of air in the piping out ahead of it. This air is forced out of the system through the opened thermostatic vent. Depending upon the length of the piping the steam will initially condense in the piping before it gets to the trap. As the condensate enters the trap and begins filling the trap body the float will rise pulling the valve off of the discharge seat. This allows the condensate to be removed immediately.

With the thermostatic vent and the condensate discharge both in their open positions steam begins to enter the trap. As this happens the heat of the steam causes the thermostatic element to expand closing off the vent orifice. However, as long as there is a sufficient condensate level in the trap the condensate outlet orifice will remain open. This trap is self-regulating. It does not go through the fill/ discharge cycles like most other traps. If the condensate enters at a constant rate it will discharge at a continuous and constant rate.

As the steam condenses the released air and CO_2 will accumulate at the top of the trap, around the thermostatic vent. This provides an insulation barrier from the heat of the steam allowing the thermostatic vent to cool down sufficiently to contract, opening the vent discharge orifice. As the air and CO2 are released steam replaces that volume and again comes in contact with the thermostatic element causing it to expand, closing off the discharge orifice.

When using an ultrasonic listening device it is difficult to determine if this type of trap is working properly. Since there is no fill & dump cycle, which is easily recognizable and is used on most traps to determine condition, only a trained ear can tell whether or not a Float & Thermostatic Trap is operating properly.

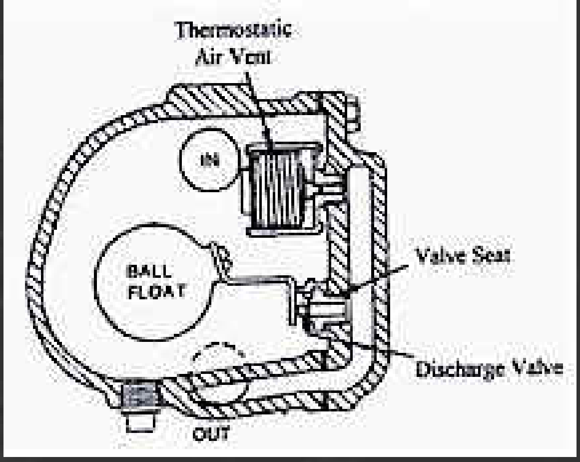


Fig. 6 - Float & Thermostatic Trap

Float & Thermostatic Type - Application

This trap can be used in Process and utility applications as well as HVAC service. It is a good choice for high capacity requirements. However, it does not stand up well in conditions where the trap may be subjected to water hammer and/or freeze up. The float and thermostatic element can be damaged by water hammer and the float can be damaged by freeze up. If selected, given those considerations, this is an excellent multi-use trap

Piping News Report

An informative newsletter for Piping Designers and Engineers

Basics of Steam System Design

by W. M. (Bill) Huitt

Steam Trap Application

In order to effectively design a steam trap installation the designer needs to profile each trap requirement. Keeping in mind that the vast majority of trap requirements on a new project will be typical this reduces the amount of time involved by a significant amount. Most trap requirements on a project will pertain to drip traps and steam tracer traps (if the project is indeed using steam as its freeze protection medium). Consequently when you profile either of those requirements you are doing so for a group of traps. The same applies for typical process and utility equipment that requires steam trapping.

When developing a *Steam Trap Profile Sheet provide space to list all steam traps, by identification number, that pertain to that particular profile. In other words, one sheet should be dedicated to one typical profile. All traps intended to match that profile should be identified on that sheet.

Following is information needed to determine a steam trap specification:

1. What are the steam & condensate conditions at the user (pressure, temperature and volume)?
   a. One of the primary considerations in steam trap sizing is the differential conditions between the steam on the upstream side of the trap and the condensate on the downstream side.
2. Is there an automatic throttling valve upsteam of the trap?
   a. An automatic throttling valve upstream of the trap may cause a wide fluctuation in both the upstream pressure and the condensate load.
3. Will the condensate have to rise from the trap are will it drain down from the trap?
   a. If the condensate has to be lifted then the water column creating back pressure on the trap has to be considered.
4. Will the condensate have to be lifted to the trap?
   a. Lifting condensate to a trap that is elevated above the equipment it is trapping (as in a submerged coil) requires special consideration. As condensate is lifted to the trap it has a tendency to flash back to steam and bind the trap so that it doesn't operate.
5. Is operation of the user continuous, batch or intermittent?
   a. When trapping an item considered to be a main component in a process system which is under continuous operation, down time becomes an issue and duplication becomes a consideration. In a batch operation there may be

sufficient time between batches to allow for a single trap assembly. An intermittent process requirement would indicate unpredictive intervals between use. In a situation such as that a dual trap assembly might be called for.

6. Is the installation in an environmentally controlled area or outside?
   a. An outside installation might require freeze protection.
7. What is the pressure differential across the trap with no restrictions to the steam?
   a. Required information in sizing or specifying the trap.
8. What is the pressure differential across the trap during maximum heat transfer in the user?
   a. During a maximum heat transfer period (as in air handling unit coils) the pressure differential can get too low to move condensate through certain types of traps or lift the condensate to a return header above the trap.
9. What is the status of the equipment being trapped (major, secondary, process, utility, redundant, storage, manufacturing?
   a. Applying a ranking or an order of importance to equipment being trapped helps determine their need for single or dual trap assemblies.
10. How important is access?
    a. Knowing the order of importance of equipment being trapped allows the designer to quantify the importance of accessibility for future maintenance.
11. Safety Factor.
    a. Due to the many variances in heat transfer conditions a steam trap for a particular application could experience a wide range of condensate loads; loads that may exceed those calculated. Based on past experience certain applications have a wider load variation beyond that calculated. Steam trap manufacturers provide this information in the form of a safety factor. This is a multiplier used to increase the calculated condensate load in order to compensate for variations in actual condensate load the trap may experience.

* A Steam Trap Profile Sheet is one of the many forms that Piping News will provide on its web site.

By resolving these questions prior to specifying a trap or designing an installation the designer is much better prepared to do the job more effectively, more efficiently and correctly. For the engineer it means saving money by doing the job more efficiently at the design stage. For the owner it means saving money because the proper installation reduces steam waste, reduces the chance for having to make a correction during start-up and minimizes repairs and replacements after the system is on-line and the plant is in operation.

Anyone that performs maintenance or steam trap surveys has had the opportunity to witness carelessness, disinterest and lack of knowledge at work. When design complacency or lack of understanding does occur it's the owner that usually gets stuck with the cost of having to correct the problem installation. As with most project design and engineering issues the owner needs to safeguard their interest by doing their homework.

A large number of these design mistakes make it through start-up because they don't have an initial adverse effect on the operation that is obvious. After the design firm is long gone and the contractor has everything signed off and hits the road, these mistakes linger until, at some point later, they're discovered by design or by accident. These are conditions that usually don't get caught for months, or even years. However, until they are discovered these problems can cost the owner money.

The London based EMEA acts as the focal point for the new European regulatory system. This site contains their guidelines, press releases and other information.

European Drug Regulatory Agencies Network (EudraNet) is a joint Euopean undertaking of several Euopean organizations. Their website contains links to documents and information as well as addresses and links to the various EU agencies.

In such cases the owner isn't aware the problem exists until, in most instances, it's inadvertently discovered during a maintenance check or a trap survey. Once the problem is discovered and corrected the water hammer problem, that may have existed, disappears the system operates more efficiently and steam demand drops. These latent problems, when left undetected, can cost the owner thousands, tens of thousands and even hundreds of thousands of dollars. And by the time these problems are detected and corrected the owner usually has no recourse but to pay for the mistake out of their own pocket and go on.

There is a way for the owner to reduce and possibly even eliminate these and many other mistakes from slipping through the checkout and start-up process. This will be covered in a later article. Right now we want to focus on determining the correct steam trap and installation design for various applications.

For the infinite types and variations of steam heated equipment and their installation there are that many and more variations of steam trap applications. What this section will do is work with the primary steam trap applications and their basic installation criteria. Those applications are:

1. Boiler Main
2. Steam Distribution Main
3. Steam Tracing
4. Steam Separators
5. HVAC Coils
6. Humidifiers
7. Unit Heaters
8. Submerged Coils
9. Heat Exchangers
10. Storage Tanks
11. Plate Coils
12. Steam Jackets

Before getting into specifics with the installation design of the above applications there are three issues that need to be mentioned. As a general rule it is poor practice to trap two or more equipment items with a single trap. In doing this, the potential exists for short circuiting one of the equipment items.

A short circuit occurs when one of the equipment items has more of a steam or condensate load than another equipment item connected to the same trap. The equipment with the higher load, and this can be a very slight difference, will create a primary path to the single trap essentially blocking condensate flow from the equipment with the lighter condensate load. This effect can create a backup of condensate in the blocked equipment causing low heat transfer efficiency as well as damage to the equipment that is being short-circuited.

Secondly, and these are not mentioned in order of importance, is consideration for the individuals that will be required to work on these installations after they are in operation. Their personal safety should be uppermost in the engineer's or designer's mind when designing any installation. The ease of access should be a definite consideration. This needs to be weighed against how frequent access is going to be required based on what service the trap will see and whether or not the trap is servicing a major piece of equipment.

In particular, traps that will be servicing major pieces of equipment operating on a continuous basis should get the highest degree of consideration. Anyone who has had to peel back the insulation on a high pressure steam line in preparation for pulling a malfunctioning steam trap, while the system is still operating, understands the need for providing the necessary valving, pipe configuration and access to make this job as safe as possible.

This is not only the responsibility of the engineering firm, but also the plant owners own project team. As a basic concept the owner needs to have maintenance personnel involved in reviewing installation drawings while the drawings are in design, not after the fact. This makes sense on two levels: 1. From a safety standpoint, plant maintenance won't inadvertently be placed in a situation where they have to deal with a potentially hazardous circumstance when maintenance is called for, 2. The cost involved in correcting a poor installation design after the fact is going to be relatively more expensive than the initial installation cost. It is more cost effective to put it through the proper reviews and get it right the first time before final approval, fabrication and installation.

There will be further articles regarding project teams, planning, coordinating and execution. These issues go beyond the scope of the design of steam and condensate systems and the application of steam traps.

And the third point of discussion is a maintenance program for steam trap stations. As an example, if a moderately sized plant produces an average summer/winter steam rate of 50,000 lbs/hr at a cost of $7.00/1000 lbs. the math tells us it is costing the plant $350.00/hr or $8400.00/day to keep this plant operating. At $3,066,000.00/yr the manufacture of steam in this plant is a major overhead. With that much associated cost operating the steam system and its users at a high degree of efficiency should be a top priority. At least you would think so. Too often plants are not aware of their steam trap population, how many traps they have, where they are located or the last time they were tested. Simple questions, but all too often unanswered questions. At least until something goes wrong.

After discussing the various design applications we will expand on the profile questions and perform a sample trap specification routine.

Boiler Main

The following criteria is based on the assumption that the piping involved has been designed based on good and basic pipe design practices. As an example, the boiler main piping is normally contained within the boiler house itself with the plant distribution piping branching off of the boiler main inside the boiler house and then running outside to distribute steam to the plant. The purpose of the example is to make clear that in determining the number of traps and their placement on the boiler main, pipe run length should not be a factor.

The boiler main (ref. Fig. 7) is the piping that connects directly to the boiler or boilers and feeds steam to the distribution piping. Steam distribution piping for the plant, or facility, feeds off of this boiler main piping to distribute steam throughout the plant. A portion of the boiler main piping should be at a low point of the piping system between the boiler connections and the plant distribution headers. Any carryover from the boilers will initially wind up at that low point. This low point is the section of piping where the traps should be located.

If the steam in the boiler main, whether it is supplied from a single boiler or multiple boilers, flows in one continuous direction to its termination point only one drip trap station will be required and it should be located at the termination point of the header. If the steam in the boiler main is routed in multiple directions in order to arrange the distribution headers in a more efficient manner then a drip trap station will be needed at the termination of each of those boiler main runs.

Referring to the Profile List, the boiler main should be considered a major utility in continuous service. This would suggest redundant traps and possible in-service trap repair/replacement. The installation design should reflect these requirements in a way that provides consideration for the safety of the individuals who’s job it will be to perform the repair while the system remains in service.

Another consideration is possible water hammer (hydraulic shock) and variations in condensate load due to boiler carryover. The selected steam trap should be able to handle both conditions. The carryover, which could be a combination of condensate and entrained solids, has to be removed immediately before it gets out into the system. Because the specified traps may be required to handle these potential upsets the sizing will have to reflect that.

At this point the developing trap station profile looks like this:

1. Locate trap assemblies at the terminus of either single or multiple boiler mains.
2. Provide redundant trap assemblies.
3. Configuration of trap assembly should consider in-service repair or replacement.
4. Because of possible in-service repair or replacement, unobstructed access should be considered.
5. Trap should withstand hydraulic shock.
6. Trap should be able to accommodate transient condensate loads above what is normally expected.

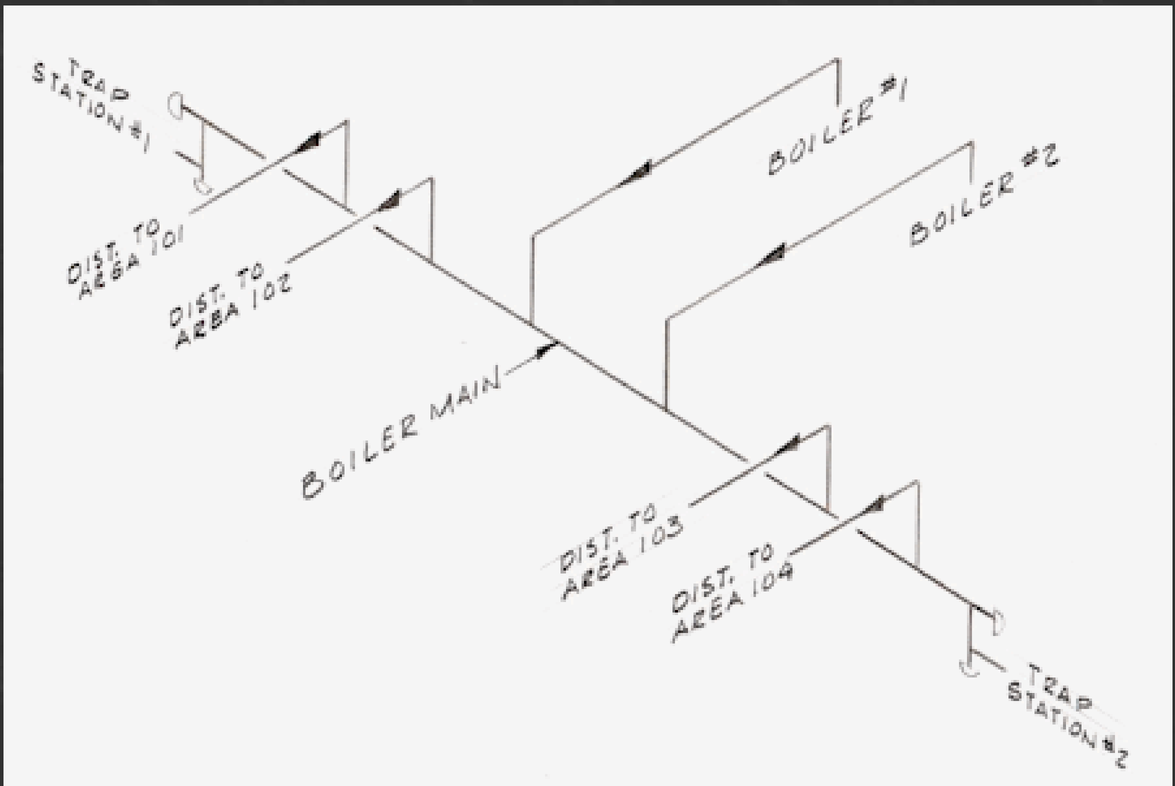
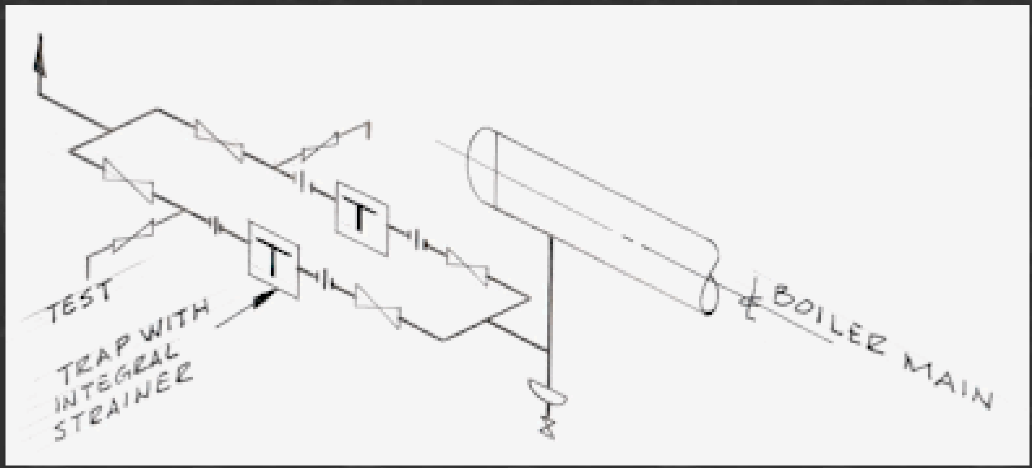


Fig. 7 - Boiler Main Trap Station Location



Gig. 8 - Boiler Main Trap Station Detail

Referring to Figure 7 you will notice that the distribution segment of the boiler main is located in a pocketed section of the piping. The two boiler feed lines drop down into this section while the plant distribution mains feed up. Notice also, in Figure 8, that two trap assemblies are indicated at each location. This includes the necessary valving required to isolate each trap and strainer to allow for in-service repair or replacement. Since the frequency of preventative maintenance, or testing, would be on the order of once a year and the potential of repair/replacement would be on the order of three to five years plus, unassisted access is not necessarily required; only clear access.

Unassisted access is a condition in which personnel can gain full access to a trap station simply by walking up to it at floor level.

Unobstructed access is a condition in which personnel would require scaffolding or a portable platform to gain full and unobstructed access to the trap station.

Obstructed access is a condition in which the trap station is in an elevated position with an obstruction such as piping, a cable tray, conduit, etc. between the station, the access floor or from the approach direction.

It is not against good practice to locate the trap station itself a short distance from the drip leg if it facilitates access and simplifies support without creating additional problems. This applies to most applications. The concern involved with locating the trap station away from the drip leg is the added loss of radiant heat due to the added length of pipe.

In calculating the condensate load requirement for the boiler main traps two factors need to be considered: radiant heat loss and transient carryover volume. Radiant heat loss can be calculated as follows:

$$C = \frac{A \times U \times (t_1 - t_2)E}{H} \times L$$

Where:
C = Condensate in lbs/hr/foot
A = External area of pipe in square feet/foot of pipe
U = BTU/sq ft/degree temperature difference/hr
t_1 = Steam temperature in degrees F

t_2 = Air temperature in degrees F

E = 1 minus efficiency of insulation (1-.75=.25)
H = Latent heat of steam
L = Total Length of pipe and fittings in feet

It is recommended that the transient upset condensate load be added to the radiant heat loss condensate load when sizing the trap. When factored in, the upset load is calculated based upon a percentage of the steam production. However, this percentage has to be tempered with the integrity and quality of the boiler system. If steam is produced at the rate of 20,000 lbs./hr and an upset factor of 10% is used, the additional load for the trap would be: 20,000 x .10 = 2000 lbs/hr.

Based on saturated steam at 600 PSIG running through a 16"ϕ pipe, 80 feet long, inside a heated facility the calculated condensate load for one of the trap stations would be:

$$C = \frac{4.2 \text{ x } 3.6 \text{ x } (487 - 70).25}{740} \text{ x } 80 = 170.4 \text{lbs/hr}$$

By using a safety factor of 2:1 (safety factors will be discussed later) the calculated condensate load would then be 340.8 or 341 lbs/hr. That gives us a total combined condensate load of 2000 lbs/hr (upset load) + 341 lbs/hr. (calculated load) = 2341 lbs/hr.

If we elect to go with an inverted bucket trap one of the selections would be an Armstrong 3/4" model #983 with an 1/8" orifice and a capacity of 2425 lbs/hr. Another choice would be a 3/4" Spirax/Sarco disc trap model #TD42 with a capacity of 2200 lbs/hr.

In sizing the trap for this application we are able to accommodate the added transient upset load by only marginally increasing the trap size that would have been selected had the upset load not been considered. However, this could change if steam production was 50,000 or 100,000 lbs/hr.

Although the condensate load from radiant heat loss would remain the same, the increased production value would proportionately increase the volume of the upset load to 5,000 and 10,000 lbs/hr respectively. The determination as to whether or not to maintain that 10% factor or to make a compromise by reducing that factor has to be an educated one, a decision that should involve a discussion with the boiler manufacturer.

Steam Distribution Mains

Steam distribution mains are the segment of a piping system used to transport steam from the boilers to the various areas or departments throughout the plant. It is the designer's job to design a system that will accomplish this as efficiently as possible.

The configuration for a distribution main will usually consist of risers, pockets and long distances with expansion loops. These are all conditions that are conducive to the formation and retention of condensate. In such conditions it is the designers responsibility to incorporate the use and placement of steam traps in strategic locations. In doing so it will prevent the accumulation of unwanted condensate by capturing and returning it.

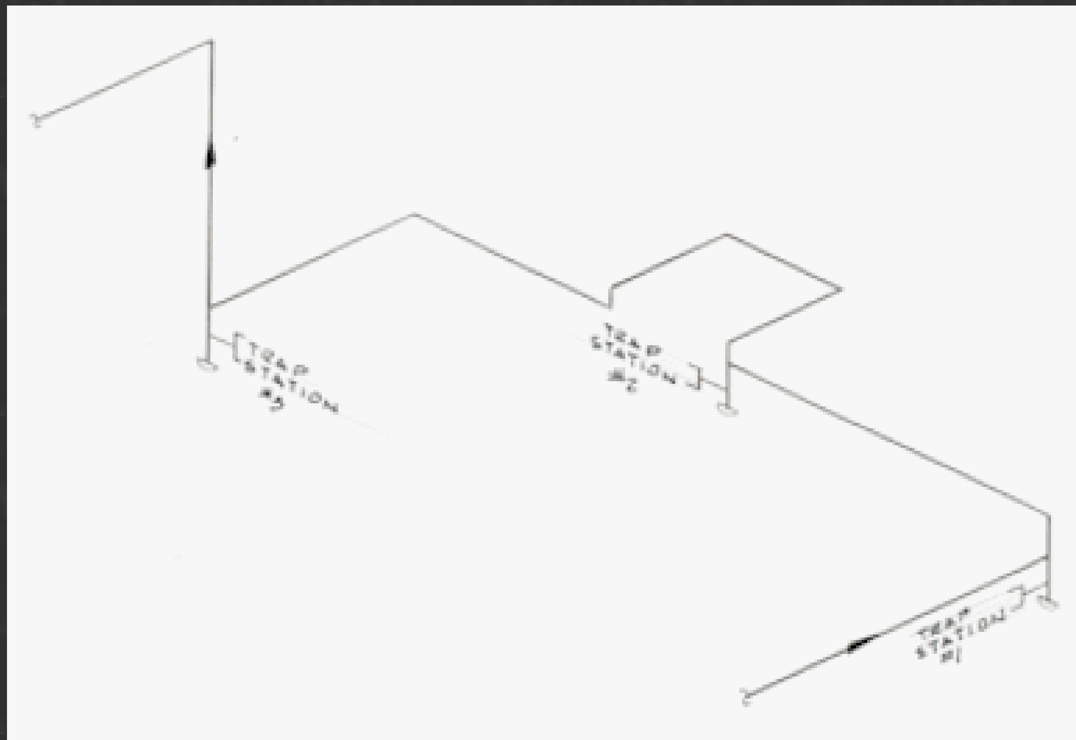


Fig. 9 - Steam Distribution Main - Trap Locations

Referring to Fig. 9, there should be a trap station located at each riser and on the upstream side of each expansion loop. There should also be a trap station approximately every three to four hundred feet on long horizontal runs containing no risers or expansion loops.

Steam traps will be sized based on the amount of radiant heat loss occurring in a specific segment of the distribution main. Start-up conditions do not necessarily effect these traps. In starting up a steam distribution system there are two things you will need to consider in regard to steam traps. One is the start-up condensate load and the second is the displacement of air residing in the pipeline.

The start-up load is short-lived and not that much in excess of the normal load. However, the necessity to displace air in the system along with the warm up of pipe and components as well as the additional condensate load at start-up demands specific consideration. Start-up of a system needs to be done in a very planned and controlled manner. This start-up consideration should be a separate issue from the trap station design itself. Which is based on the system operating parameters and not on the momentary conditions seen at start-up.

This is another good example of a design situation that should include, whenever possible, discussions and input from the operations and/or maintenance personnel; the group that will ultimately be operating the facility. They may have an idea or two about how they would like to shut down and start up the system. What this type of input hopefully provides, when working with an existing plant site, is continuity and applied lessons learned. Continuity as to the methods and procedures used to perform typical plant activities from department to department. And the practicality of taking the lessons learned from previous plant installations and integrating them into current design.

Although there are situations where automatic shutdown and start-up design is feasible the vast majority of main distribution systems are designed for manual, or supervised, shutdown and start-up. A typical automated start-up and shutdown system might be a steam tracing distribution system. This type of system would start-up and shut down based on the operation of an ambient sensing valve.

With all sub-header and supply line valves closed, all drip leg drain valves closed and all drip traps valved off, steam is slowly bled into the system through the main header valve. *(A future segment of the design series will cover check-outs and start-ups in greater detail. This segment on steam will cover only a few basics for clarification.)* This can be done by cracking open the main valve or by providing a small, 3/4" or 1" by-pass around the main valve. This can either be an external or integral by-pass to the main valve.

As steam fills the piping system the initial volume of steam will condensate at a higher rate due to the relatively cold pipe. However, as the supply of steam continues to move through the piping it will cause a large portion of the forming condensate to flash back to steam. This will be apparent by, what should be, a small amount of internal crackling and banging that takes place during this process.

With the system now flooded with steam and the main valve fully open the closed system contains partial pressures of steam and air. The air that resided in the system before the introduction of steam is still contained in the closed system under a partial pressure.

The theory of partial pressures is based on Dalton's Law of Partial Pressures. John Dalton observed that the Total Pressure of a gas mixture was the sum of the Partial Pressure of each gas, or P total = P1 + P2 + P3 + ...Pn. The Partial Pressure is further defined as the pressure of a single gas in the mixture as if that gas alone occupied the container. It is therefore determined that, in our case, steam, as a partial pressure, will have the same characteristics as its partial pressure.

Let us assume that the entrapped air, contained within the closed system, now makes up 10% of the volume within that closed system and the newly introduced steam makes up the 90% balance of the volume. With an absolute pressure of 300 PSIA, the pressure of the air would be 30 PSIA and the pressure of the steam would be 270 PSIA. With an absolute pressure of 270 PSIA the steam's characteristics would be a temperature of 407.8 ^{o}F with a latent heat content of 818.3 BTU's and a specific volume of 1.71 ft^3/lb. Instead of the 417.4 ^{o}F, 808.9 BTU's and 1.547 ft^3/lb it would be at its original 300PSIA.

Not only does the air reduce the overall heat content of the steam by volume but once the air is in the system it will have a tendency to plate out, or collect, on heat transfer surfaces further effecting the overall heating efficiency of the steam. It is therefore essential that the air be removed from the closed system before the sub-header valves are opened, compromising the efficiency of the entire system. Incidentally, this same process will be repeated with each sub-header. The overall checkout and start-up procedure should be outlined and accomplished in such a way that virtually all air is purged from the system before the final block valve, at each user, is opened.

With the main system still closed off from the sub-headers, personnel, in a controlled procedure, will go to each drip leg drain valve and slowly open it. It is recommended to use a multi-turn valve like a gate valve rather than a quarter turn valve like a ball valve in steam service. The multi-turn valve provides much more control when opening and closing, unlike a quarter turn valve.

The sequence of performing this blow-down should be in the direction of flow, starting with the furthest upstream drip leg. And due to the inherent dangers of steam, much caution should be exercised while performing this procedure.

Each drain valve only needs to remain open for a brief period. With the valve fully open for approximately one minute this should be sufficient enough time to purge the steam containing air from each particular segment of the piping system. Once the purge is completed for a specific segment close off the drain valve and open the valves on both sides of the steam trap. Continue this process in the direction of flow.

That basically touched on the high points of starting up a steam system. There is much more to the process that will be covered in a later section on check-outs and start-ups. These highlights were presented at this time to show that, when considering start-up loads and conditions, there are no special needs. And please do not take that as a general statement. There are certainly cases where a particular system may require that special consideration be given to the start-up and shutdown process of a system.

Getting back to the placement of trap stations; In the process of locating the strategic positions of each trap the designer has to determine what segment of pipe each trap will be assigned to. In the case of the trap located at the base of the riser in Fig. 9, the designer should calculate the height of the riser plus the distance from the upstream side of the expansion loop (at Trap Station #2). This will be the segment of pipe assigned to Trap Station #3 at the base of the riser.

As in the previous calculation for radiant heat loss:

$$C = \frac{A \times U \times (t_1 - t_2)E}{H} \times L$$

Let us assume that the riser has a length of 80'-0" and the lineal distance from the upstream side of the expansion loop to the bottom of the riser is 160'-0". That gives us a total length of 240'-0" that trap station #3 has to be sized for. Based on 300 PSIA saturated steam running through a 10"ϕ header in an outside pipe rack the calculation would be:

$$C = \frac{2.81 \times 3.9 \times (417 - 0).25}{808.9} \times 240 = 338.97 \text{ lbs/hr}$$

By using a 2:1 safety factor the condensate load would then be 677.94 or 678 lbs/hr. A practical trap for this application would be a thermodynamic or thermostatic type.

One further note when designing trap stations for this application; address the issue of preventative trap maintenance. If you're in plant maintenance or operations and you have the opportunity to provide input during the design phase of a project, make this one of your discussion points. If you’re a designer, address this with the owner's team.

There are two considerations we need to address with this particular application (actually it can be made to apply to virtually any application): the first is the need to facilitate testing of the trap for preventative maintenance, and the second is periodic strainer blow-down.

In the first consideration, when designing trap stations for steam distribution mains a typical trap station for a drip leg would usually be installed in a pipe rack. This places the trap station in an elevated and very inaccessible location.

When you consider the possibility of several hundred traps throughout a plant requiring lifts, ladders, scaffolding and tie-offs, just to gain access to, it can be a costly and time consuming exercise. What ultimately occurs is the trap stations are not maintained properly and in too many cases are even forgotten.

The second consideration is an all too frequent problem. Strainers, whether a separate item or an integral part of the trap, are there to capture errant, entrained particles in the steam before they can reach and plug the trap orifice. Just the nature of the intended purpose of the strainer indicates the fact that particles will be trapped and will accumulate on the strainer mesh until they are removed. As they accumulate they progressively block off flow through the mesh until, if not periodically cleaned, they completely block the flow of condensate to the trap.

Inaccessible trap stations are frequently ignored or forgotten. Dropping a trap station to grade or access level is one simple way to avoid this and should be a design consideration. In considering this option we need to develop a cost profile for the intended alternate configuration. This will determine what the initial installed cost for this alternative design might be and what the additional running or ongoing operating cost might be. This will provide management with the necessary information to help determine whether or not to make the change.

This application we will use as an example will be an inside installation with a controlled environment. The initial installation differential will include an additional length of pipe, insulation, added fittings and labor. For this case we will assume the pipe support cost to be a trade off.

In order to locate the trap station at an accessible level it will have to drop 15'-0" requiring 30'-0" of additional pipe, (2) 90B Elbows and 21'-0" of insulation. Using 3/4"φ , sch. 80 c.s. pipe with 3000 lb socketweld fittings and 21'-0" of 1 1/2" thick fiberglass insulation. The condensate return leg from the trap station will only be insulated up to 7'-0" above the access level for personnel protection. If this installation was to be outside, the additional cost for heat tracing the condensate return riser along with insulating the balance of that return line would have to be added into the initial cost.

Pipe, Fittings & Labor $458.00

Insulation Material & Labor $147.00
Installed cost differential $605.00

Additional cost consideration comes from any added operational costs due to relocating the trap station. In this case it would consist only of the added radiant heat loss in the additional 15'-0" drop to the trap station. Because the 15'-0" return section of piping is carrying condensate it is not a part of the heat loss calculation. The radiant heat loss can be calculated using the preceding radiant heat loss calculation as follows:

$$C = \frac{.275 \times 3.9 \times (417 - 0).25}{808.9} \times 15 = 2.07 \text{ lbs/hr}$$

If we base this on a system that operates 24 hrs/day, 7 days/week the total annual, additional steam loss in that 15'-0" drop would be: 8760 hrs. x 2.07 lbs/hr = 18133.2 lbs/yr. Assuming an approximate cost of $7.00/1000 lbs to manufacture the steam, the total annual cost incurred by making the trap station more accessible would be: 18133.2 lbs/yr x $7.00/1000 lbs = $126.93. You can see by these figures that the ongoing operating cost differential incurred in providing improved access amounts to less than $11.00/mo.

The advantage in doing this, however, will more than offset this added cost. By making these trap stations more accessible, and again this does not only apply to steam main applications, it makes the preventative maintenance effort much more efficient. Strainers can be blown down and traps can be tested in a much safer, less time consuming and a more controlled manner. This is an issue that needs to be tabled and discussed early in the design stages of a project. If agreement can be reached at an early stage it will save a great deal of time in trying to redesign a system to accommodate such changes.

Steam Tracing

Very frequently there is a gap between an engineers perception of priorities and the priority perception of the plant maintenance and operations personnel. In order to understand the importance of a well thought-out and well designed steam tracing system from the plant personnel's viewpoint a design engineer needs to get a phone call at 3:00 AM, with outside temperatures hovering around -20B F and winds gusting at 30 MPH. The maintenance shift supervisor says a steam tracer in a pipe rack froze and ruptured, creating a potential freeze-up of the pipeline it was protecting.

The next thing you know, you're not in your warm bed any longer but are instead at the plant. You're 30'-0" up in a scissors lift in sub zero temperatures trying everything your experience allows to keep this line from freezing. Which, if you don't succeed, could ultimately shut down a production facility.

Personal inconvenience aside, the potential capital loss incurred from a shut down of operations because a weak link in the steam tracing system failed to protect some piping is very real. That reality exists because all too often steam tracing, when called for on a project, is treated as an afterthought instead of as an integral part of the design process.

At this time we will address only the trap station design for steam tracing. A later section in the design series will discuss the design of steam tracing systems in detail.

To facilitate shutdowns, start-ups and isolated repairs, individual tracer supplies should be grouped together in a manifold arrangement whenever possible. With this intent the designer should plan the routing of the tracing in an effort to allow the tubing to congregate and terminate at common points for trap manifolding whenever possible. This is something that is relatively simple to do in a pipe rack situation but requires additional forethought when the tracing is required away from common groupings of heat traced pipe like a pipe rack.

An identification system should be established in order to identify both the supply source and the termination of each common tracer. This will allow each tracer to be isolated safely for shutdown, start-up or repair when necessary. If this is not done it will not only cause unnecessary delays but could also place personnel at risk.

When traps are arranged in a manifold configuration each tracer line should consist of (in the direction flow) a block valve, a strainer upstream or integral to the trap, the steam trap, a test valve, a check valve and a block valve. The check valve is required when the manifold, or condensate return header, is sized for expected flow only. There are other manifold arrangements where condensate is released into a collection chamber. The chamber is usually nothing more than a section of oversized pipe that allows condensate to be collected then removed through a dip tube by siphon effect.

The tracer lines connecting to a collection chamber don't normally experience water hammer as a result of surges from other connecting lines. However, where a return header is sized for the flow load only, from several connecting lines using cycling traps, a discharge surge from one trap can create a reverse water hammer impact on the other connecting lines. The check valve installed downstream of the trap prevents and protects each trap from the hydraulic impact caused by the discharge of the other traps.

Steam Separators

A steam separator is an in-line piece of equipment that is used to remove entrained condensate particles. These condensate particles can originate from boiler carryover, or be the unwanted result of undersized pipe and equipment. A well-designed steam system minimizes the accumulation of condensate within the steam system and immediately carries off the condensate that does accumulate. With a properly sized and configured piping system, properly located drip legs and properly sized equipment a steam separator would not be required.

However, when a plant experiences reduced heat exchanger efficiency, erosion at pipe directional changes, erosion to in-line equipment and water hammer, the installation of a separator is a consideration. All are possible indicators that the presence of entrained condensate particles and the accumulation of condensate exists in the flow of steam. Unless a system is designed poorly to begin with, these symptoms are usually the result of poorly planned, multiple expansions and modifications to a steam system.

If a system is expanded to and operated at its capacity any weakness in the design and/or installation will begin to show. One of the first telltale signs that a steam system is at its capacity and/or has a weakness in its design is the above indications of condensate in the system. In order to alleviate the problem without going through the time and expense of resizing and replacing existing piping a separator could be installed. A strategically placed separator can't cure the problem but it can prevent erosion and other damage caused to pipe and in-line equipment due to the formation and build-up of condensate.

The separator is designed to work in-line and should be expected to remove approximately 95 - 98% of the entrained condensate particles. With various designs of the same theme the separator functions as an enlarged section in a pipeline. Internally there are baffle plates designed to either impinge the particles, force the particles to the outer perimeters of the housing or both.

As the steam, with the entrained condensate particles, enters the chamber of the separator it suddenly and momentarily looses some of its velocity due to the sudden enlargement of the separator chamber. The mass of the condensate particles propels them forward into the impingement baffles. Or, in the case of a cyclone type design, the particles will be forced by vanes or fins into a high velocity rotation inside the chamber. The particles will then be forced to accumulate either on the impingement baffle or the outer perimeter wall of the separator chamber and collect at a low point in the separator. Connected to the low point of the separator is piping that will lead to a steam trap where the condensate will then be carried off.

The need for a separator, aside from installing one to safeguard the system from possible boiler carryover, indicates a system problem. The separator is installed to control certain aspects of the problem not correct it. Depending on the severity and magnitude of the problem it may be more cost effective to research the problem itself, determine the cause and correct it.

Air Handling Unit Coils

The heating coils in air handling units have a demand range governed mainly by outside temperature fluctuation. This demand range requires that the designer give added consideration in the selection process of a steam trap for each particular coil. In some cases there may be multiple sets of coils servicing one air handling unit. Each coil in the set will require its own steam trap. An excellent trap for this service, because of the wide range in pressure and volume, is the float & thermostatic trap. This trap allows condensate at low pressures and low flows to free flow through its orifice while also accommodating and regulating the higher flows at higher pressures.

Two conditions can create a low pressure situation at the coil outlet. Depending upon what latitude an installation is located in the outside temperature could create a sufficient temperature differential across the heating coils to cause a coil outlet pressure of less than 1 pound. This can occur in either of two ways: 1. Either the demand across the coils is so great that the steam is condensing in the coils at a rate that exceeds the supply capability. Secondly the demand across the coils is so low that the steam control valve has throttled back to the point of providing less than 1 PSIG of steam. As an example, let us assume we are supplying a single coil with 15 PSIG steam. If this supply pressure were to be maintained through the coil and up to the trap it would have enough pressure to lift the outlet condensate 34.5 feet, or 15 PSIG x 2.3 (feet of lift per pound) = 34.5 ft.

If enough of a demand, or the lack of demand, was placed on the steam in the coil it could create a 14 PSIG pressure drop across that coil or across the steam control valve. That would leave a pressure residual of only 1 pound at the trap. This translates into enough pressure to lift the condensate only 2.3 feet. You can see why additional, and early, consideration must be given to this application. The issue with how to return the condensate under such potentially low pressure needs to be determined early in the design phase of a project due to the impact that may be caused to other disciplines.

In determining how the condensate is going to be returned there are basically two considerations: 1. Can the condensate return headers be run below (on a lower floor) the coil outlet trap for gravity drainage and 2. Does the condensate have to be lifted to overhead condensate return piping?

If it is possible to run the return header below the trap outlet this would be the more practical method in regard to the flow of condensate. Regarding other concerns, dropping the trap discharge piping down normally necessitates floor penetrations. At the very least, grating or steel plate penetrations. Floor penetrations will require coordination with the architectural and/or structural group. In some cases it would be easier to plan on core drilling. If you're working with an unstable design due to equipment drawings that are not approved, lack of information on in-line equipment and any one of a number of other problems, don't attempt to resolve the location of the penetrations. Doing so only creates confusion, construction delays and added cost. Core drilling allows the penetrations to be made after their locations have been confirmed without regard to the flooring schedule. This allows the floor to go in on schedule without having to wait for approved penetration drawings.

With the condensate dropping down from the trap discharge to the return header the designer doesn't have to be concerned with the lack of lift pressure. However, if there is no alternative but to return the condensate overhead then the designer is going to require a non-electric condensate pump. If it is at all possible combine the flow of two or more traps by routing a collection header and running it to a non-electric condensate pump.

The non-electric condensate pump is a mechanical equipment item that allows condensate or other liquids to accumulate, by gravity, in the pump chamber under low pressure. The condensate then gets pumped to its destination by steam, air or inert gas pressure.

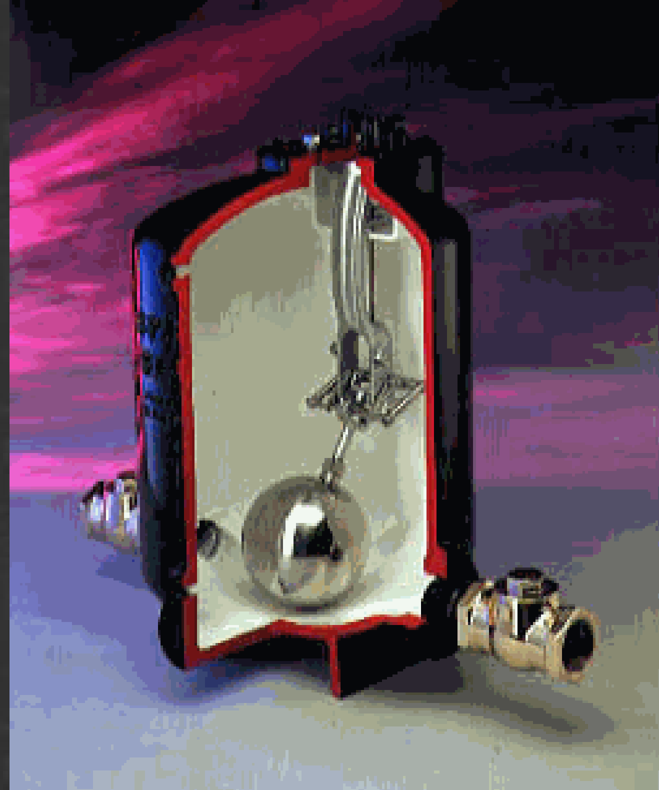


Fig. 10 - Spirax-Sarco's Pressure Powered Pumpt

Fig. 11 - Armstrong's Condensate Pumps

Figures 10 and 11 represent examples of condensate pumps by two of the largest manufacturers of steam traps and other related equipment. Even though these condensate pumps vary in style and manufacturer they operate under the same basic principal. There is a check valve on both the inlet side and the outlet side of the collection chamber. As you can see in the cutaway in Fig. 10 there is a ball float attached to a spring mechanism. This spring mechanism is designed to trip at a predetermined point as the ball float rises. The spring mechanism is, in turn, attached by rod to two valve assemblies. One valve for the pressure supply port and the other for the vent port.

In its empty, or idle, state the ball float is resting at its lowest point, the pressure side valve at the top of the unit is closed and the vent valve is in its open position.

As condensate enters through the inlet port and accumulates in the pump chamber air, non-condensables and some flash steam are vented through the vent port (discharge of the vent should be piped to the outside) and the ball float begins to rise. The check valve on the discharge port prevents condensate in the return system from back-flowing into the condensate pump chamber. As the volume in the pump chamber increases the ball float eventually reaches its predetermined set point height, its high level point, and the spring mechanism trips. This pulls the valve rod down closing the vent port and simultaneously opening the pressure port. When the higher pressure, in the form of steam, air or inert gas, enters the pump chamber through this port it forces the condensate through the outlet port. The check valve in the inlet port prevents the condensate from flowing back through the inlet.

As the level in the pump chamber lowers so does the ball float. When the ball float reaches the low level set point the spring mechanism trips pushing the valve rod up closing the pressure inlet port and opening the vent port. The pump is now ready for another cycle.

When returning condensate to a return header located below the trap discharge configuration of the installation would require a drip leg, strainer (possibly integral to the trap), steam trap, test valve, block valve and header connection. The block valve on the steam side of the coil will be used to block the upstream side if the assembly needs to be isolated for maintenance.

When returning condensate overhead with the assist of a non-electrical condensate pump the installation would require a drip leg, strainer (possibly integral to the trap), steam trap and test valve. If the condensate pump is servicing a single trap then the balance of the installation would include the condensate pump, block valve and header connection. If the condensate pump was servicing multiple traps with their discharge lines in a manifold configuration then the balance of the installation would include a block valve, sub-header connection, condensate pump, block valve and header connection.

Since these heating coils, and that includes pre-heat and re-heat coils, operate on a seasonal basis there is ample time for maintenance to clean out or blow down each drip leg and to drop out the strainer screens for cleaning. This should be a scheduled preventive maintenance item.

The trap for this application should be sized based on the specified BTU output of the coil. Let us assume that the specifications for the coil indicate a BTU output requirement of 400,000 BTU's/hr using 15 PSIG steam. To determine the anticipated condensate load divide the BTU output by the latent heat of evaporation for 15 PSIG steam. For this example the calculation would look like this:

$$C = \frac{400{,}000\ \mathrm{BTU/hr}}{945.3} = 423.2\ \mathrm{lbs/hr}$$

Using a safety factor of 3 multiply the anticipated condensate load as follows: 423.2 lbs/hr x 3 = 1270 lbs/hr.

Humidifiers

The humidifier is an equipment item that introduces steam either directly into an area or into the airflow of an air handling unit in order to maintain an environment with a stable relative humidity. If the relative humidity is too low it causes discomfort to personnel and creates the potential for static electricity. In an explosive process area this cannot be tolerated for obvious reasons. If the relative humidity is too high it causes discomfort to personnel and is detrimental to office machines and operating equipment.

The best engineered humidifier can be compromised by a poor installation design. It would be time well spent for the engineer and designer time to look into the installation and operation of humidifiers in some detail. This section will cover only the basics of the humidifier as a prerequisite to trapping the condensate.

The humidifier assembly consists of a steam inlet connection with a strainer, a jacketed distribution manifold, metering valve with operator, separation chamber and steam trap. Some manufacturers choose to integrate the metering valve with the separation body and others choose to make them independent items.

In regard to humidifiers with distribution manifolds and referring to Fig. 12, steam enters the assembly through the strainer, which removes most of the entrained particulate that may be carried by the steam. The flow of steam then enters the jacket of the distribution manifold. There may be multiple distribution manifolds determined by specific requirements. The humidifier is designed to remove moisture and sustain a dry steam up until the moment the steam is released into the area or airflow. The final protection against moisture build up is the jacketed distribution manifold. This prevents the airflow itself from condensing the steam prior to discharging from the manifold. After passing through the manifold jack the steam enters a separation chamber. Any condensate carried into the chamber by the steam is removed through the use of impingement baffles. The condensate flows to the bottom of the chamber and into a drip leg where it is trapped and returned to the condensate collection system.

With the condensate removed in the separation chamber the steam flows into a second separation or drying chamber then finally into the distribution manifold. At this point steam is distributed directly into the airflow or out into a control space.

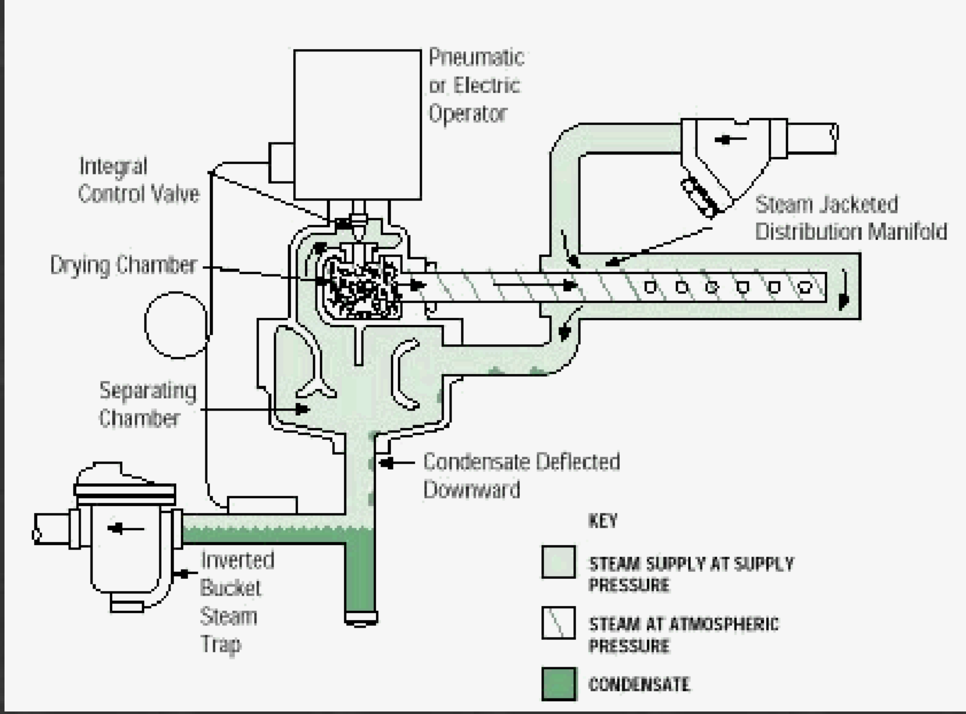


Fig. 12 - Humidifier
Compliments of Armstrong International, Inc.

When the humidifier is installed in an air handling unit the outlets of the distribution manifold should be directed against the airflow. This allows for a quicker and more thorough disbursement of the steam throughout the cross sectional area of the airflow itself.

The type of trap used for this service could be an inverted bucket, a thermodynamic or thermostatic type. One of the key factors in cases where trap selection is a multiple choice is current plant spare parts inventory. Ideally what you want to do is minimize the additional new parts the plant is required to handle. If you can work with one of their current selections without compromising design integrity then this needs to be considered.

With a large plant this is usually not a problem. Most plants will have a wide variety of trap types installed from a wide variety of manufacturers. As mentioned earlier, the more practical thing to do is to consult with plant personnel through the project team. Their may be a certain manufacturer, manufacturers representative or product that they have had difficulty with in the past. Don't lose site of the fact that the plant personnel will continue to live with this design long after the engineering firm is gone. Allow them to be a part of the process by providing know-how from their prospective. This is something that needs to be considered early in the design process.

When sizing the trap the condensate load will be based on radiant heat loss. That is, the amount of heat loss that occurs through the jacket of the distribution manifold, and the upstream piping. There will be some heat loss in the body of the humidifier but we can factor that in by adding a foot to the length we calculate for the upstream piping.

In calculating the condensate load there will be two separate calculations: one for the insulated upstream piping and one for the uninsulated manifold jacket. For this example we will use 15 PSIG steam in 1" pipe with an upstream length of 20'-0" from the last trap to the distribution manifold jacket. The jacket on the distribution manifold is 48" long. We can simplify the calculation process by replacing the jacket configuration with standard pipe. With the jacket having dimensions of 1 1/4" wide by 1 7/8" deep and one side open we can use a 1 1/2" pipe to represent the jacket in our calculation. Since the air-handling unit is in a controlled area we will assume an air temperature of 70B F.

The calculation for the upstream piping would use the radiant heat loss calculation described earlier and would look like this:

$$C = \frac{.344 \text{ x } 2.9 \text{ x } (250 - 70).25}{945.3} \text{ x } 20 = 0.95 \text{ lbs/hr}$$

The calculation for the jacket would look like this:

$$C = \frac{.497 \text{ x } 2.9 \text{ x } (250 - 70) \text{ x } 1}{945.3} \text{ x } 4 = 1.10 \text{ lbs/hr}$$

One difference between the manifold jacket and the supply piping, aside from the jacket having no insulation, is the airflow across the manifold. Since the manifold is mounted inside the duct it will be in the mainstream of the airflow. With 70B F air moving across the manifold jacket at a velocity of 1000 feet per minute that particular aspect would normally be considered in the above calculation. However, the operation of the humidifier basically negates that aspect. By injecting the steam against the flow of air it forces the steam to rapidly reverse its direction and flow back over the manifold jacket. This action actually creates a hot barrier between the jacket and the 70B F flow of air. In creating this barrier it virtually negates the tentative effect the air flow would have on the manifold jacket.

In an attempt to simplify the calculation process while trying to maintain a balance between accuracy and efficiency we will allow the steam flow over the jacket to cancel out the airflow factor in the calculation. We have therefore omitted that factor from the calculation.

With an accumulated condensate load of 0.95 lbs/hr + 1.10 lbs/hr = 2.05 lbs/hr x 3 (safety factor) = 6.15lbs/hr a 1/2" or 3/4" disc trap or an inverted bucket would be a good selection.

Unit Heaters

Unlike a central heating system that distributes heat throughout a building the unit heater is used to maintain a desirable temperature in a single, confined space. Depending on the configuration and size of the space to be heated multiple unit heaters may be required.

Although standard convection heaters like baseboard heaters and radiators are considered unit heaters the only type we are concerned with in this section is the forced air type. This type of heater functions by forcing room air across coils heated by steam, hot water or an ethylene glycol solution. Radiant heat from the heater coils heats the air as it passes through the coils and out into the space to be heated.

A later section will discuss the method for determining requirements of a unit heater. In regard to steam traps, while other types of traps will work in this application there are two types that are recommended for this service: the float & thermostatic and the inverted bucket.

If a thermodynamic or thermostatic type trap is being considered for this application you need to consider whether or not the heater will be heating outside air. If the heater is positioned at an overhead door that is frequently opened, or remains opened for extended periods, that heater could see wide fluctuations in heat transfer requirements. Make certain to run the calculations on those variations to determine the load range that will be required of the trap at that location.

Additionally, if a thermodynamic or thermostatic trap is specified for this application ensure that the trap is located sufficiently far enough away from the heater fins to prevent condensate from backing up into them, creating corrosion. As described earlier, these types of traps do not have an integral reservoir to collect the condensate until it discharges. In this type of trap the condensate backs up into the piping until it is discharged.

Using the unit heater specifications to obtain the necessary information, the condensate load can be determined with the following calculation:

$$C = \frac{F \text{ x } Cp \text{ x } d \text{ x } 60 \text{ min/hr x } \Delta T}{H}$$

Where:

C = Condensate in lbs/hr

F = Cubic feet of air per minute

Cp = Specific heat of air in BTU/lb/B F

d = Dendity of air w .075 lbs/cu ft.

DT = Temperature rise in B F

H = Latent heat of steam

As an example we will assume 15psig steam heating 9,300 CFM of air from 60°F to 112°F. The calculation would then be:

$$C = \frac{9300 \text{ x } .24 \text{ x } .075 \text{ x } 60 \text{ x } 52}{945.3} = 552.5 \text{ lbs/hr}$$

To illustrate the variance in loads when a heater is located near an open overhead door we will use the same criteria except we will change the inlet air temperature to 0°F. The calculation would then be:

$$C = \frac{9300 \times .24 \times .075 \times 60 \times 112}{945.3} = 1190.0 \text{ lbs/hr}$$

This would indicate a condensate load for a unit heater, at an open door with sub-zero outside temperatures, at twice that of the units further inside the heated space. Using a safety factor of 3:1 the load range on the various units would be 1658 lbs/hr to 3570 lbs/hr. depending on their location.

The only block valves required would be located on the upstream side of the unit heater and on the downstream side of the steam trap. Include a drip leg directly out of the unit heater. The strainer can either be independent or integral to the steam trap.

Submerged Coils

Submerged coils are heating or cooling coils contained inside a vessel and are designed to come in direct contact with the fluid to be heated or cooled. The depth, diameter, pipe size and length of the coil is determined by several factors that include, but are certainly not limited to: characteristics of the fluid to be heated or cooled, volume of the fluid, whether the fluid is agitated or static in the vessel and its expected retention time. Given the same fluid characteristics and volume the variation in the condensate load will depend on whether the fluid is agitated or static and what its retention time

Depending on the type of vessel design coil configurations will vary. The inlet and outlet of the coil may be in the bottom out the bottom, in the top out the bottom, in the side out the side, in the side out the bottom, in the top out the side or in the top out the top.

Special consideration needs to be assessed with an in the top out the top configuration when using steam as a heating fluid. This configuration creates a pocket in which the forming condensate has to be lifted up to the trap. Condensate at its equilibrium temperature has the potential to flash back to steam if either its temperature is increased or its pressure is reduced. In the process of being lifted from a lower elevation to a higher elevation the pressure of the condensate will be reduced at a rate of 1 pound for every 2.3 feet of rise in elevation. A lift of 12 ft. would translate into a 5.2 psig drop in pressure.

The percent of flash steam created from a drop in pressure can be determined with the following calculation:

$$\% \text{ flash steam} = \frac{SH - SL}{H} \times 100$$

Where:

SH = sensible heat of the condensate (btu/lb) at the higher pressure (before lift)

SL = sensible heat of the condensate (btu/lb) at the lower pressure (after lift)

H = latent heat of the steam (btu/lb) at the lower pressure

To illustrate, lets assume we are supplying 50 psig steam to a top in, top out coil. The condensate that collects at the low point of the coil will be considered 50 psig condensate with a sensible heat content (heat of the liquid) of 267.6 btu/lb. The vertical distance from the inside bottom (invert) of the low point of the coil and the inlet to the steam trap is 10'-9". This is the height that the condensate has to be lifted.

That vertical distance is equal to 10'-9" ÷ 2.3 (2.3 ft. of lift equals 1 lb) = 4.7. So the difference in pressure between the low point of the coil and the inlet to the trap is 50 - 4.7 = 45.3 psig. The sensible heat content at 45.3 psig is 262.2 btu/lb and the latent heat at that pressure is 915.4 btu/lb.

Based on that information the calculation would be:

$$\%\text{ flash steam} = \frac{267.6 - 262.2}{915.4} \times 100 = 0.59\%$$

The accumulation of the flash steam will prevent the steam trap from operating properly and efficiently. As steam, the flash steam will prevent the trap from systematically collecting and discharging the condensate. It will bind the trap until the steam residing in the trap condenses. When the condensate contained in the trap is discharged additional condensate is lifted from the coil. As the new condensate is lifted a portion of it will flash. The trap will detect the flash steam and stop discharging before all of the condensate is removed from the coil.

This process of removing only a portion of the condensate allows a large portion of the condensate to remain in the coil. This reduces heat transfer efficiency and promotes corrosion in the coil.

There are designs that work effectively in trying to avoid these inherent problems. One such design utilizes a specifically designed steam trap by Armstrong. They call it their Automatic Condensate Controller as shown in Figure 13.

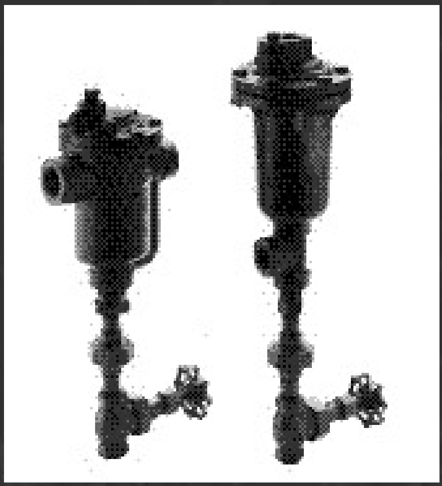
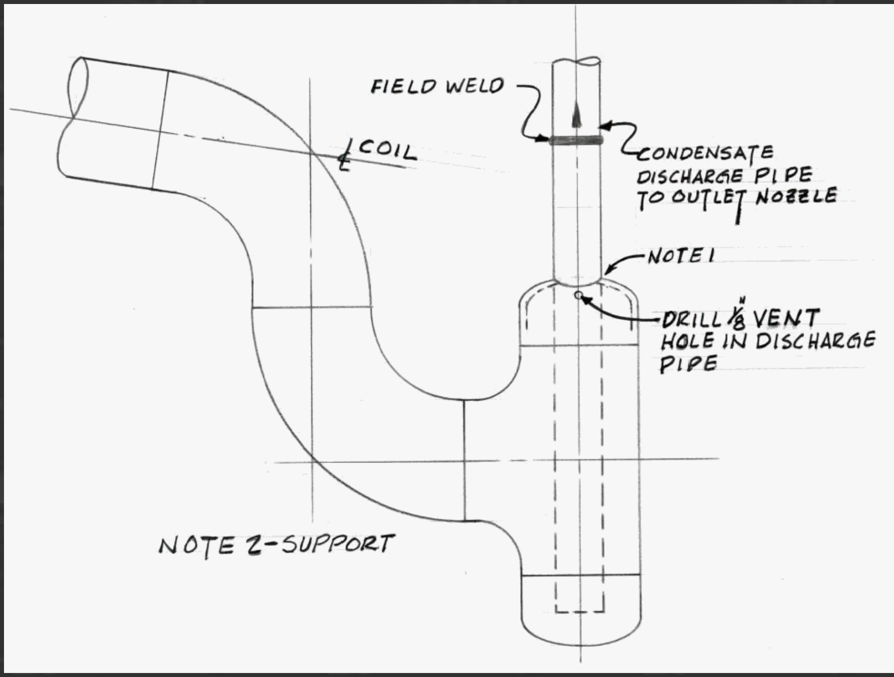


Fig. 13 - Armstrong's Automatic Condensate Controller

This is an inverted bucket trap with a modified assembly allowing flash steam to by-pass the trap preventing the trap from binding. By continuously moving the flash steam through the system and into the condensate return line it allows the trap to remove condensate in a more efficient manner.

Another method of removing condensate from this type of submerged coil involves the design of the discharge section of the submerged coil. This is the section of pipe at the end of the coil that rises straight up to the outlet nozzle. See Figure 14.



Notes: 1. Check for reinforcement requirements at penetration. 2. Support at larger bore coil not at small bore discharge pipe.

Fig. 14 - Coil & Discharge Dip Tube Detail

This design, as shown in Fig. 14, is only practical when the coil size, or the pocket size is 2" and above. It consists of a low point trap for condensate to accumulate in and a smaller dip tube to serve as a siphon type discharge for the accumulated condensate. The cooler temperature further up the discharge pipe represents lower pressures causing the condensate to migrate up the discharge pipe toward the steam trap. The tee, as shown in Fig. 14, can attach directly to the coil without the two additional elbows as represented in Fig. 14. However, the above design provides added retention volume away from the coil for the accumulating condensate as it forms.

Some coil designs extend a discharge pipe, the same size as the coil itself, vertically from the low point of the coil to the outlet nozzle. A dip tube is then extended down the length of the vertical discharge pipe to a location near the low point. Much like what is shown in Fig. 14. This type of design essentially creates a jacketed dip tube arrangement. The larger pipe will contain steam virtually the entire length of the dip tube. This effectively assists the condensate inside the smaller pipe to flash back to steam. We want to prevent flashing, or at the least not create a design that may assist the flashing effect as condensate moves up the discharge pipe. As mentioned, the lift alone will initiate flashing.

The design in Fig. 14 eliminates the jacketing effect and allows further cooling of the condensate as it moves up the discharge pipe toward the outlet nozzle. Since the discharge pipe is not encased in the larger pipe the condensate in the discharge pipe is cooled by the surrounding product or utility fluid contained in the tank. Depending on the temperature differential between the condensate being lifted out of the coil and the surrounding fluid in the tank it may be enough to keep the condensate from flashing.

If the design configuration in Fig. 14 is used then a standard inverted bucket or an F&T trap would be suitable for this application as well as the Automatic Differential Condensate Controller. In both discharge pipe configurations the situation exists for non-condensables to accumulate. To a greater degree where the discharge pipe remains the same size as the coil and extends to the outlet nozzle.

In order to allow these non-condensables to escape an 1/8" diameter hole should be provided in the dip tube near the point where it passes through the cap, as in Fig. 14, or where it passes through the flange at the nozzle (not represented). This build-up of non-condensables, if allowed to accumulate, could eventually bind up the coil effectively preventing full use of the coil's heat transfer area.

PREVIOUS

W. M. (Bill) Huitt
For the past 34 years Bill has been involved in mechanical design and engineering, as well as the construction, of heavy industrial plants and facilities. His involvement has included positions as design engineer, piping design instructor, project engineer, project supervisor, piping department supervisor, engineering manager and president of W. M. Huitt Co. founded in 1987. His experience covers both the engineering and construction fields and crosses industrial lines to include petroleum refining, chemical, petro-chemical, pharmaceutical, pulp & paper, nuclear power, and coal gasification. He has written numerous specifications and guidelines to insure that design and construction complies with code requirements, owner stipulations and good design practices. Bill has also been called on to perform steam and condensate system analysis to resolve operating problems with existing operating systems. The above article, and others to follow, are excerpts from his upcoming book, "Constructable Piping Design".