APPLICATION GUIDELINES

ORIFICE TRAPS

The flow of saturated condensate and/or live steam through an orifice is not totally predictable, nor is it measurable by usual means. Given constant upstream and downstream pressures and a constant amount of steam being condensed hourly, a relatively wide span of varying live steam losses are usually experienced.

Yes, even though the pressures and condensing rate remain constant.

For example, if the following conditions prevail:

Inlet pressure to orifice 40psig
Outlet pressure from orifice .... 0 psig
Condensate flow .... 4000#/hr.
Orifice sized with 1.5 safety factor .... 3/8” dia.

A variation of live steam loss from 11 to 91#/hour is possible.

If a homogeneous mix of condensate and steam prevails through the discharge, 11#/hour will be the loss of live steam. If no mixing occurs, but instead a slug of condensate only, is followed by a flow of steam only, is followed by a slug of condensate,...a live steam loss of 91#/hour would be expected. Of course, neither of the happens exclusively, but rather a combination of both. It depends on the manner of flow to the orifice: surges, steady dribbles, etc.

Consider the following calculated graph.

Variation Of Live Steam Loss Through An Orifice Trap

40 psig constant inlet pressure

0 psig constant outlet pressure

4000#/hour condensate load

3/8” orifice selected on basis of 1.5 safety factor

Example:

See curve D. 30% homogenous flow and 70% discontinuous flow results in 67#/hour love steam loss.

(Based on modified Darcy equation and assuming flashing fluids behave similar to expanding fluids. Supported by test.)

What we are saying is: There is a variation of live steam loss even when hourly rate of condensate load and upstream and downstream pressures are constant, depending on the nature of the instantaneous flow of condensate and live steam to the **orifice**.

If the condensate flow increases above 4000#/hour (upstream and downstream pressure constant), the loss of live steam will reduce but there is a danger of backing condensate into the heat transfer apparatus and limiting its output. If the condensate flow goes below 4000#/hour (upstream and downstream pressures constant), the loss of live steam will increase.

Similarly, if the condensate load stays constant the pressure differential increases, the live steam loss will increase. And if the differential decreases, back up of condensate can be expected.

Modulating Steam Supply

Let’s consider the example given on the enclosed Nomogram involving a modulating steam supply. (These are the same values that were used in the example on page 1.) If we have a maximum condensate load of 4000#/hour and use a safety factor of 1.5 :1.0, we would look for an orifice with a capacity of 6000#/hour at 40 psig inlet steam pressure and back pressure of 0 psig. Referring to the Armstrong sawtooth curve, we see that a 3/8” orifice in a #814 trap fills the bill.

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INVERTED BUCKET TRAP CAPACITY CHART

CHART

Inverted Bucket Trap Capacity Chart

Sizing a steam trap for maximum capacity at maximum pressure is not a difficult task, but because steam pressure diminishes much more rapidly than steam temperature, it ia difficult to assure sufficient trap capacity at reduced loads.

Consider the heat transfer equation: Q=UA (ΔT).

"Q" represents the heat transferred in an hour: Btu/hr.

"U" is the heat transfer co-efficient which varies with the materials of construction, the media from which the energy flows, and the media to which it flows: Btu/hr.- sq. ft.-°F.

"A" represents the area of the heat transfer surface: sq. ft.

ΔT represents the temperature differential across the heat transfer surface: Fahrenheit degrees.

In a given heat transfer apparatus, the area of heat transfer surface does not vary, so "A" is a constant. Since the media and materials of construction do not change "U" is also pretty much a constant. (It varies some with the velocity of liquid or gas). The only heat variables then are "Q" and "ΔT", and that means that the heat transferred is directly proportional to the temperature differential.

EXAMPLE: Air heater operation at full load

Let's consider an air heater selected to function on full load 40 psig (287°F saturated). Let's say further, that the air is being heated from 10°F to 70°F at design conditions. The air flow is 42,500 cfm.

$$42{,}500\ \frac{\text{cubic feet}}{\text{min.}} \times 60\ \frac{\text{min.}}{\text{hr.}} \times 0.018\ \frac{\text{Btu.}}{\text{°F - cubic foot}} \times 80\text{F.}/919\ \frac{\text{Btu.}}{\#} = 4000\#/\text{Hr.}$$

Draw in the 40 psi steam pressure line,①. Draw in the 30°F average air temperature line,②. From the intersection of ① and ② draw a line horizontally to the left scale, ③, and read 257°F. average heat transfer differential. It's at this 257 differential that the maximum load of 4000#/hr. occurs.

Air heater operating at one quarter load

Now let's explore what happens at quarter-load. At quarter-load, the average heat transfer differential (ΔT) will be one-fourth that at maximum load. One-fourth of 257° is 64°. So locate this on the left scale and draw a horizontal line to the right, ④. Further, at one-fourth load, the air temperature rise will be 20°F. instead of 80°F. at maximum load. The rise will be form 50°F to 70°F for an average air temperature of 60°F. so let's intersect line ④ with the average air line of 60°,⑤. Dropping down with line ⑥, we read a steam temperature of 124° which has an accompanying pressure just above -13psi. Now draw a line through the two points we have located ⑦. This line is *the curve for the air heating system* we just described. ⑧ If we rise from 0psi on the bottom scale to where it intersects the System Curve and then ⑨ move to the left form this intersection, we find that 168° is the average heat transfer differential. The maximum load of 4000#/hr. occurred at a differential of 257°, so 168 represents a load of 2600#/hr. ([168°/257°]x4000) or 65% of full load. This means 2600#/hr. is the maximum load that must be handled by a trap with no steam pressure, only whatever hydraulic head is available, to push the condensate through the trap. We check our selection with a 1/2 psi differential. So if there is a 15" drop from the coil to the level of the trap orifice, there is sufficient hydraulic head to keep the coil free of condensate under all conditions. (Don't forget , though, there must be a low differential vacuum breaker at *the coil outlet* to assure this drainage.) Complete drainage is necessary to prevent coiled damage by freezing, corrosion and/or water hammer. Let's consider another possibility, at 5 psi back pressure due to an overhead return line or simply back pressure due to flash. Now a safety drain is in order. ⑩ Draw in the 5 psi steam line to where it intersects the System Curve 11. Moving to the left form this intersection, we read that the average heat transfer differential is 186°.This means 72% of the maximum load (186°/257°) or 2900#/hr. The safety drain must handle this at 5psi differential as well as the previously calculated 2600#/hr. at 1/2 psi differential.

Please bear in mind that the foregoing deals with a more or less ideal situation. Any safety factors applied to heat exchange equipment will increase the pressure drop and aggravate the condensate drainage problem. Note: *The coil's actual capacity* much be considered in drawing the curve, not the anticipated load.

Inverted Bucket Trap Capacity Chart

Let's trace this variable load with a 3/8"orifice on Nomogram.

The trace indicates that from that from 40 psig to 8 psig, the 3/8" orifice is adequate (safety factor ignored). At pressure below this, it is inadequate.

Let's assume there is a water column equivalent to the 1/2 psig at the trap inlet (14-15"). At 212°F., coincident with he greatest condensate load occurring when there is no steam pressure, the condensate load is 2600#/hr.

Pressure		Steam Temperature of	Load #/hr. from Nomogram	3/8" Orifice Capacity #/hr. from Sawtooth
Steam psig	Condensate Inches W.C.			
40		287	4000	6000
30		274	3735	5400
20		259	3425	4500
10		240	3100	3400
5		228	2900	2500
0		212	2600	0
0	7	212	2600	830
0	14	212	2600	1175
0	28	212	2600	1600
0	168	212	2600	2600

Normally, we would select an inverted bucket or float and thermostatic steam trap with a capacity of 12,000#/hour (safety factor 3:1). At one-half the maximum differential (40 x 1/2 = 20psi), this would result in the selection of a trap having a 9/16" orifice, (a 75J8 float &thermostatic trap or a #216 inverted bucket trap. Of course, this orifice in a float & thermostatic or an inverted bucket trap would be modulated or cycle to the varying load by the mechanism.

If a 9/16" *orifice trap* were installed to handle the 212° load at 1/2 psi differential, it would pass live steam at full load (40psig) amounting to between 25 and 205#/hour. These values would be less at smaller loads. In order to efficiently trap this air heater at all loads, a trap with a brain is essential.

Steady Steam Pressure W/Face and Bypass Dampers

If this coil were equipped with face and bypass dampers to permit use of full steam pressure at all loads, the 3/8" orifice would pass approximately 324#/hour live steam on no load. As the load increased, the live steam loss would diminish until full load was achieved. Here again, the live steam loss would we somewhere between 11 and 91#/hour.

Practical Limitation or Orifice Size

Apparently a 1.5:1 safety factor is applied to most frequently to minimize live steam loss. Low loads such as on main drips and tracers often dictate orifices too small to be practical in use. The smallest we have heard of is 0.020 inch diameter. Although this might be the best choice possible to minimize steam loss, it is a "bear" to keep it clean! To overcome this it is preceded with a strainer that has a small particle retention size than the orifice diameter. Keeping th screen clean then becomes a problem, in an effort to make the tracer line or drip point work reliably, very often the orifice is drilled out to a "practical" size and then excessive steam loss occurs.

Selection of Orifice Traps

To order an orifice trap, a customer must describe his application fully to vendor so that the proper selection can be made. Selections cannot be made by the user, because we know of no instance of a customer receiving complete selection data.

In Summary

1) There is always a loss of live steam though a properly functioning orifice condensate drainage device and it varies over a wide range even when hourly load and upstream and downstream pressures remain constant;
2) This steam loss is no readily measured with ordinary instrumentation;
3) Live steam losses are especially excessive when an orifice trap is used on an application where there is a constant steam pressure and variable load, a with face and bypass dampers;
4) Light load applications where losses are small, dictate the use of troublesome, super small orifices which are often drilled out to "practical" side with resultant excess live steam loss;
5) Users cannot make their own orifice trap selections.

Finally, a properly selected and installed inverted bucket or float and thermostatic trap is the most efficient trap for immediate condensate drainage, maximum heat transfer and maximum life.

Supplement "A" to Orifice Traps

Since writing the initial commentary on orifice traps we have had the opportunity to test and compare; 1) orifice plates, 2) staged orifice and 3) short tube orifice traps.

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Supplement "A" to Orifice Traps

The data given here indicated the difference in steam waste between the short tube insert orifice and the plain orifice plate is not appreciable.

The data given here indicated the staged orifice trap wastes more steam than the plain orifice plate.

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