Armstrong
Machine Works

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	40 psig
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 throughout 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 above 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:

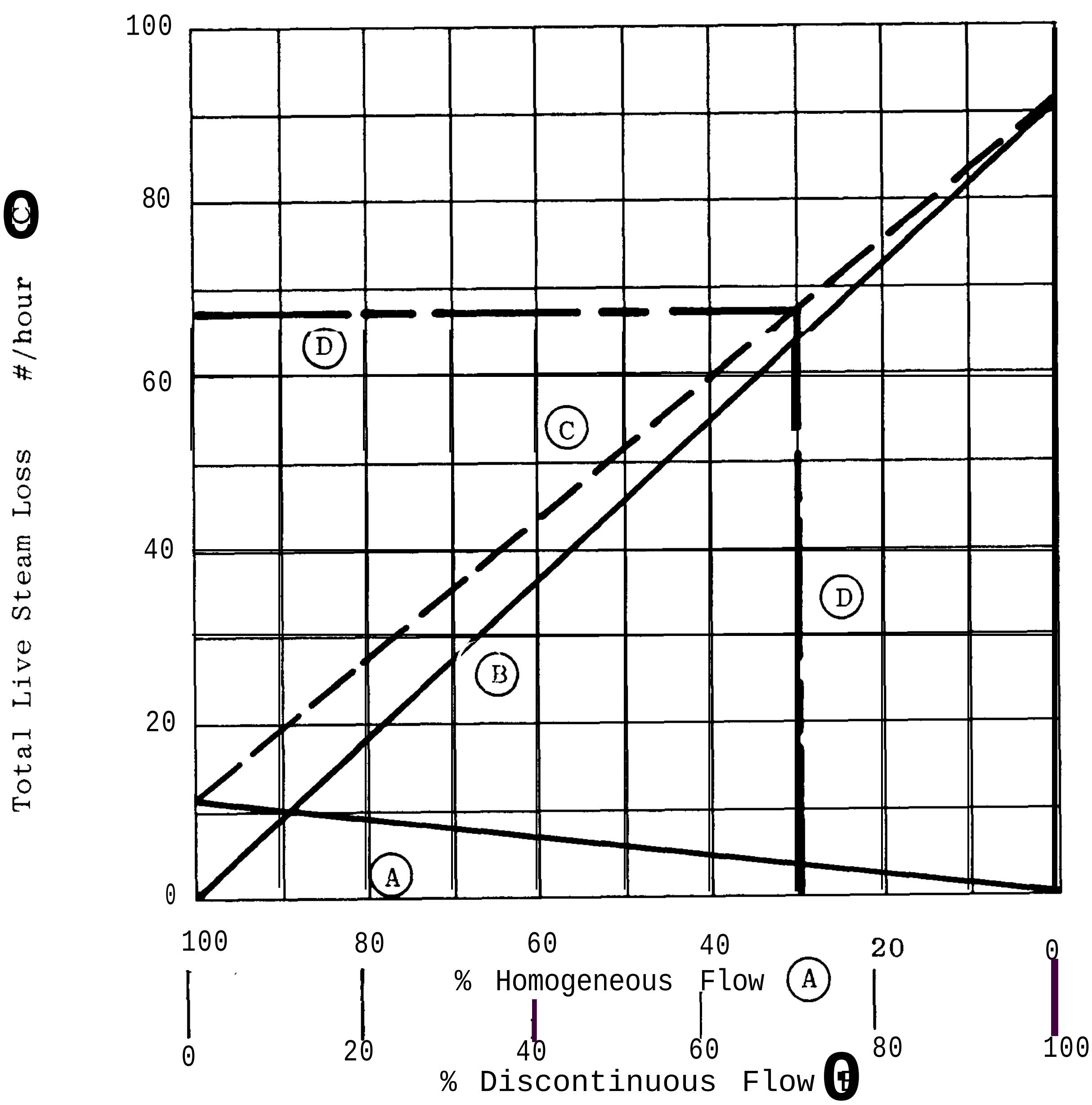
VARIATION OF LIVE STEAM LOSS THROUGH AN ORIFICE TRAP

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 30% homogeneous flow and 70% discontinuous flow results in 67#/hour live steam loss.

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

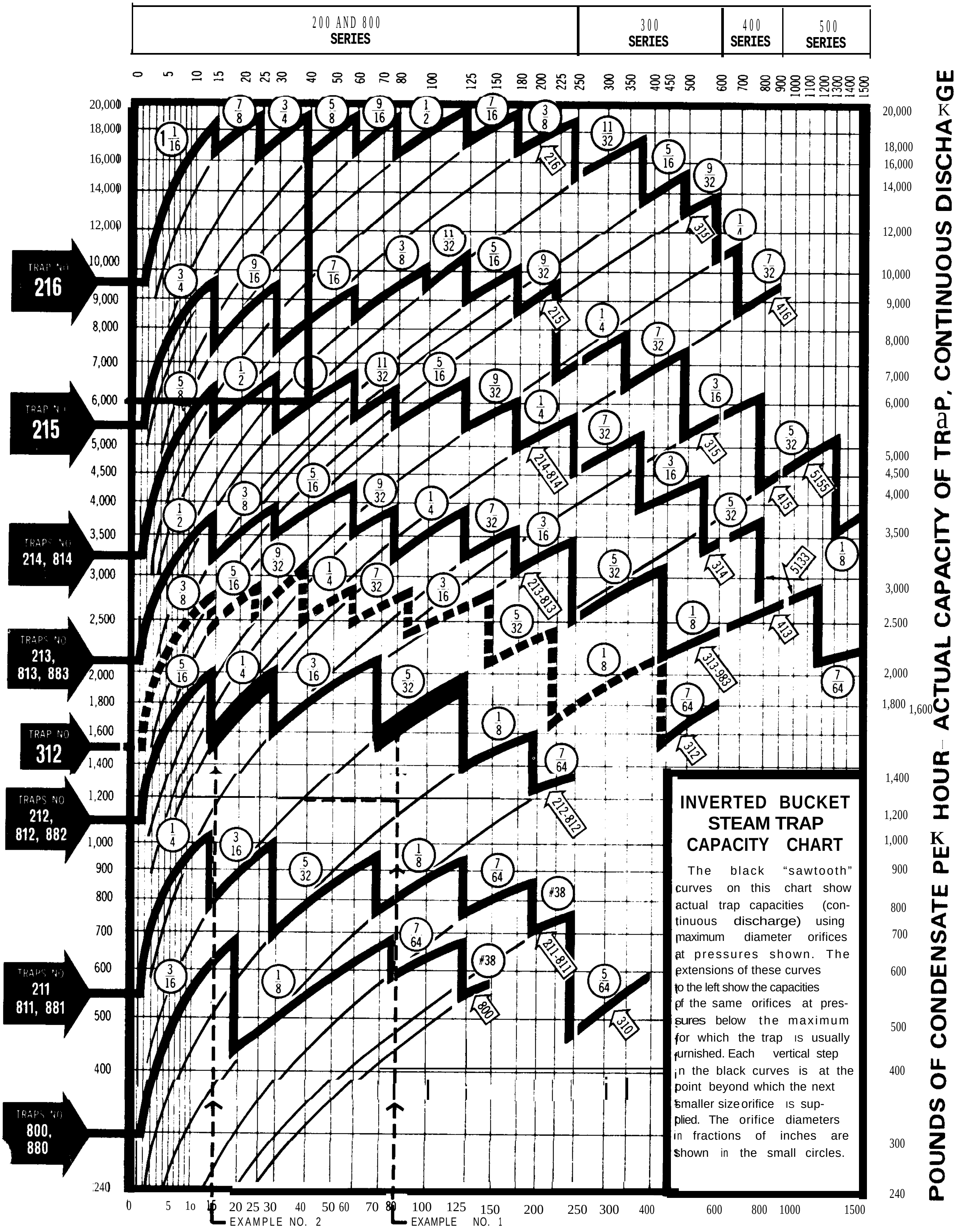
What we are saying is: There is a variation of live steam loss even when hourly rate of condensate load and upstream and downstream pressures are constant, depending on the nature of the instantaneous flow of condensate and live steam to the orifice.

If the condensate flow increases above 4000#/hour (upstream and downstream pressures 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 and 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 a back pressure of 0 psig. Referring to the Armstrong saw-tooth curve, we see that a 3/8" orifice in a #814 trap fills the bill.



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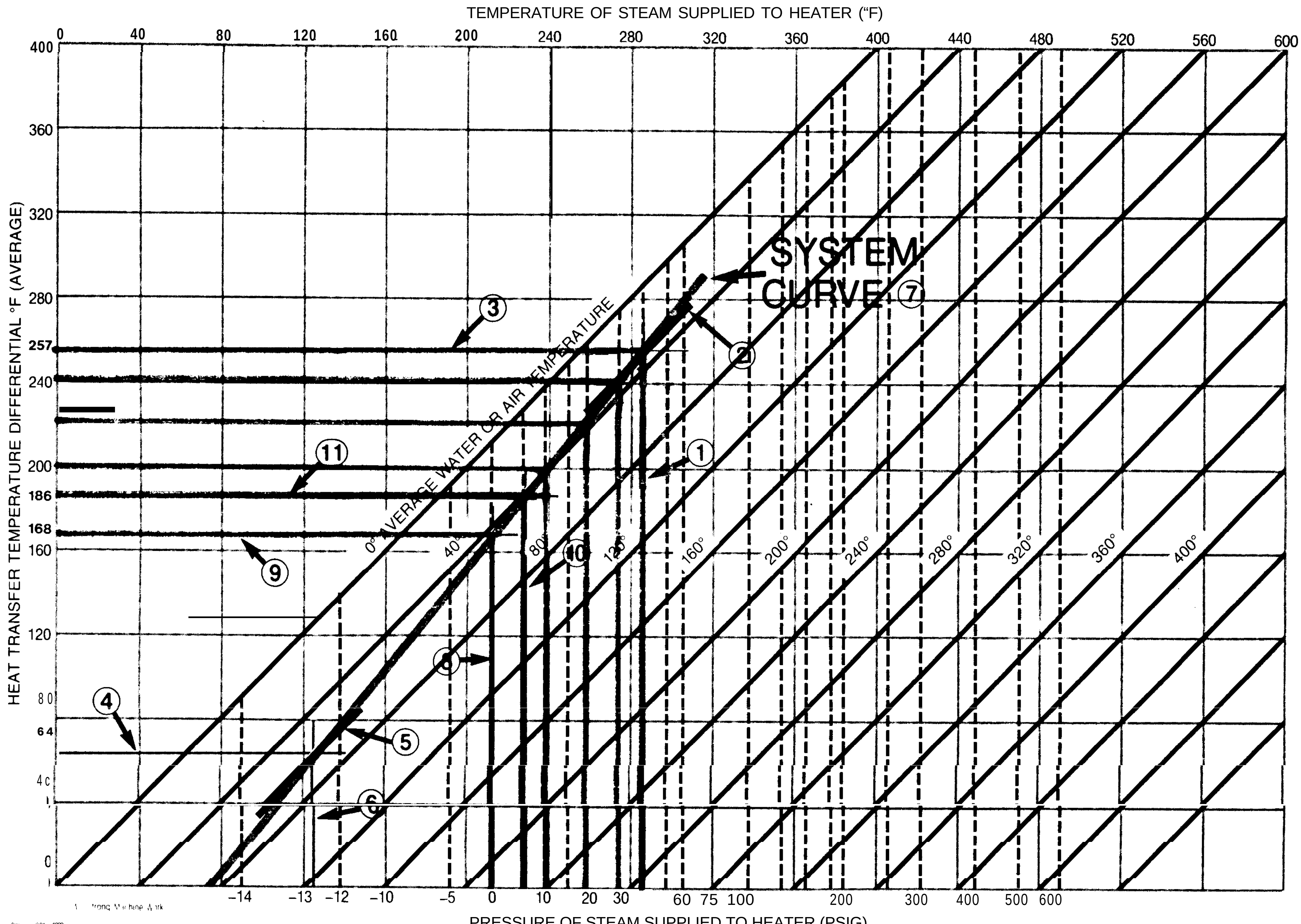
Note Above capacity chart does not Include all models available Refer to specific page of trap required for capacities not covered above

Instructions on How to Use
SIZING STEAM TRAPS FOR STEAM HEATING COILS- MODULATED SERVICE



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

Consider the heat transfer equation: Q=UA ($\triangle$T).

"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.

$\triangle$T 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 variables then are "Q" and "$\triangle$T", and that means that the heat transferred is directly proportional to the temperature differential.

EXAMPLE: Air heater operating at full load

Let's consider an air heater selected to function on full load at 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.

$$42{,}000\ \frac{\text{cubic feet}}{\text{min.}} \times 60\ \frac{\text{min.}}{\text{hr.}} \times 0.018\ \frac{\text{Btu.}}{\text{cubic foot}\ {}^\circ\text{F.}} \times 80^\circ\text{F.} / 919\ \frac{\text{Btu.}}{\#} = 4000\#/\text{Hr.}$$

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 ($\triangle$T) 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,@. Further, at one-fourth load, the air temperature rise will be 20%. instead of 80°F. at maximum load. The rise will be from 50" to 70° 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 -13 psi. 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 0 psi on the bottom scale to where it intersects the System Curve and then ⑨ move to the left from this intersection, we find that 168" is the average heat transfer differential. The maximum load of 4000#ihr. occurred at a differential of 257°, so 168 represents a load of 2600#/hr. ([168°/257°] x 4000) or 65% of full load. This means 2600#/hr. is the maximum load that must be handled by the trap with no steam pressure, only whatever hydraulic head is available, to push the condensate through the trap. We check our selection with a ½ 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 coil 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 from 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 5 psi 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* must be considered in drawing the curve, not the anticipated load.

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

Pressure Steam psig	Pressure Condensate inches W.C.	Steam Temp. of	Load #/hr. from Nomogram	3/8" orifice capacity #/hr. from saw-tooth
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

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

Let's assume there is a water column equivalent to ½ psig at the trap inlet (14-15"). At 212°F., coincident with the greatest condensate load occuring when there is no steam pressure, the condensate load is 2600#/hr.

Normally, we would select an inverted bucket or float and thermostatic steam trap with a capacity of 12,000#/hour (safety factor of 3:1). At one-half the maximum differential (40 x ½ = 20 psi), this would result in the selection of a trap having a 9/16" orifice, (a 7558 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 cycled to the varying load by the mechanism.

If a 9/16" orifice trap were installed to handle the 212° load at ½ psi differential, it would pass live steam at full load (40 psig) 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 BY-PASS DAMPERS

If this coil were equipped with face and by-pass 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 be somewhere between 11 and 91#/hour.

PRACTICAL LIMITATION OF ORIFICE SIZE

Apparently a 1.5:1 safety factor is applied 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 clean! To overcome this it is preceded by a strainer that has a smaller particle retention size than the orifice diameter. Keeping the 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 the vender 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 through 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 not 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, as with face and by-pass dampers;

4) Light load applications where losses are small, dictate the use of troublesome, super small orifices which are often drilled out to "practical" size 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.

JAK:bfs
9-82 Salesmen's List