Armstrong

HIGH CAPACITY STEAM TRAPPING

Any trap must be sized correctly, large or small, but going to an L or M or multiples of M with the resulting costs can be a difficult selling (or convincing) experience. We have safety factors we recommend, but often the customer or consulting engineer thinks we are gold-plating the job. How can we convince him otherwise?

Perhaps some of you remember the article, The Right Trapping Saves These Coils on Modulating Steam Supply in TRAP magazine No. 1 for 1979. We'd like to go one step further.

Consider the heat transfer equation: $Q = UA(\Delta T)$. "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. Delta 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.) The only variables then are "Q" and "AT", and that means that the heat transferred is directly proportional to the temperature differential.

In order to simplify the analysis, we will consider the arithmetic temperature difference instead of the log mean temperature difference which would be more accurate. (The resulting error is on the conservative side.)

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. According to M-101, page 22, we can determine the design condensate load by the following:

42,500 cfm x 1.08 x 80°F./(919 Btu/#) = 4000#/hr.

Again acording to page 22, the trap for this job should have a capacity of 12,000#/hr. (3 x 4000) at 20 psi differential (40/2). This would dictate a 75J8.

Every heating coil has a pressure drop from inlet to outlet. Let's assume ours has a drop of 5 psi. This means I would have a pressure at the steam trap of about 35 psi at design conditions. (Since the pressure goes down but the specific volume of the steam goes up as the load falls off, we assume the 5 psi pressure drop is the same for all steam pressures down to 0 psig.)

Let's consider what we know so far: At full capacity, we are heating 42,500 cfm air from -10°F. to 70°F. (30° average air temperature through coil), and the condensate load is 4000#/hr. Let's plot this information on the accompanying graph.

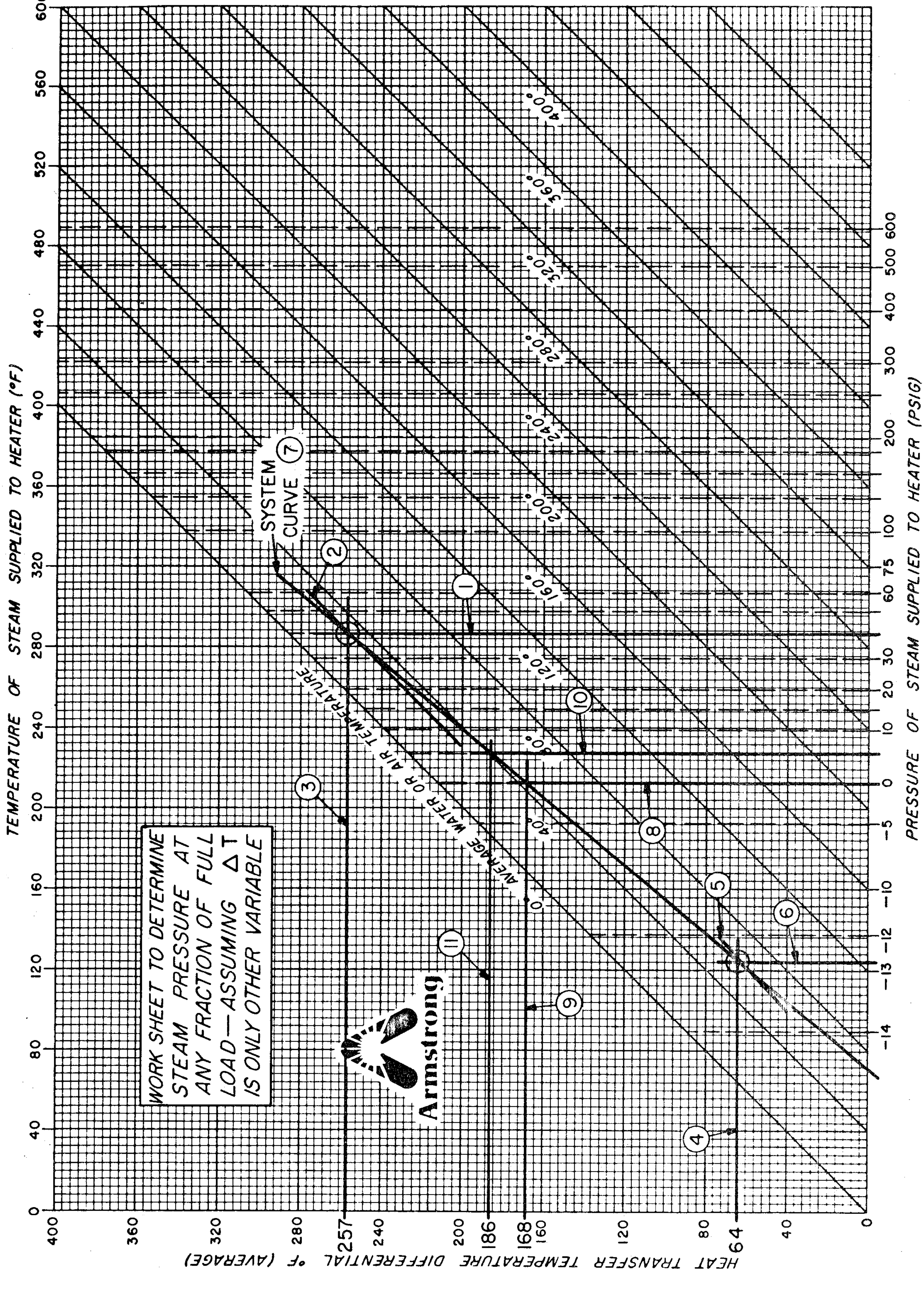
Draw in the 40 psi steam pressure line, 1. Draw in the 30' average air temperature line, 2. From the intersection of 1 and 2 draw a line horizontally to the left scale, 3, and read 257°F. average heat transfer differential. It's-at this 257°F. differential that the maximum load of 4000#/hr. occurs.

Now let's explore what happens at quarter-load. At quarter-load, the average heat transfer differential (AT) 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 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. That's all very interesting, but so what? Well, we'll show you "so what?". Draw a line through the two points.we have located. This line is the curve for the air heating system we just described. If we rise from 0 psi on the bottom scale, we intersect the "system curve" at the level of 168° average heat transfer differential. The maximum load of 4000#/hr. occured 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 availabe, to push the condensate through the trap. Checking our selection, the 75J8, we see it has a capacity of 3100#/hr. with a ½ psi differential. So if there is a 15" drop from the coil to the level of the float and thermostatic trap orifice, there is sufficient hydraulic head to keep the coil free of condensate under all conditions. Thirty inches of 212°F. condensate exert a pressure of 1 psi. (Don't forget though there must be a low differential vacuum breaker at the coil outlet to assure this drainage. See previously referenced magazine article.) Complete drainage is necessary to prevent coil damage by freezing, corrosion and/or water hammer.

Let's consider one size smaller trap for this application, the 75A8. It's capacity at ½ psi differential is about 1175#/hr. It requires about 5 psi steam pressure to handle the 2600#/hr., "no steam pressure load". This would require a hydraulic head of 12½ ft.

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. Draw in the 5 psi steam line and determine the system load at 5 psi. Here the average heat transfer differential is 186°. This means 72% of the maximum load (186-247) or 2880#/hr. The safety drain must handle this at 5 psi differential as well as the previously calculated 2600#/hr. at ½ psi differential. A 15B8 would do nicely. Our rule of thumb which by-passes these calculations, says use the same model trap as the primary trap, but with the suitable low pressure orifice. This holds true, a 2" F & T trap in both cases.

Please bear in mind that the foregoing deals with a more or less ideal situation. Any safety factors applied to heat exchange


WORK SHEET TO DETERMINE STEAM PRESSURE AT ANY FRACTION OF FULL LOAD—ASSUMING ΔT IS ONLY OTHER VARIABLE
TEMPERATURE OF STEAM SUPPLIED TO HEATER (°F)
PRESSURE OF STEAM SUPPLIED TO HEATER (PSIG)
HEAT TRANSFER TEMPERATURE DIFFERENTIAL °F (AVERAGE)
AVERAGE WATER OR AIR TEMPERATURE
SYSTEM CURVE
Armstrong
0°
40°
80°
120°
160°
200°
240°
280°
320°
360°
400°
0
40
80
120
160
200
240
280
320
360
400
440
480
520
560
600
-14
-13
-12
-10
-5
0
10
20
30
60
75
100
200
300
400
500
600
64
168
186
257
1
2
3
4
5
6
7
8
9
10
11

equipment will increase the pressure drop and aggravate *the condensate drainage problem. The coil's actual capacity must be considered in drawing the curve, not the anticipated load. We have had some of these grids printed up for you. We hope you can put them to good use with your "analytical customers".*

Functionally, there are no minimum load requirements for the "L" series trap. Since the "M" cannot be counted on for tight closure its load should never be less than about 200#/hr. Visualize the construction. The trap is assembled cold and that's when the **valves** *are adjusted to be shut. Expansion with steam temperature results in unsealing one or the other valve resulting in some leakage when the float goes down to the closed position.*

We have not experienced wire drawing or valves or seats in "L"'s or "M"'s because the traps **are** *used where loads are relatively large and just cracking the valve does not occur. Should very low loads be encountered, the trap tends to intermit. The wire drawing experienced with inverted bucket traps usually happens on high pressure, 600 psi and above, and where there is super heat. Neither of these conditions is encountered by an "L" or "M".*

We have had some wire drawing of cap extensions due to corrosive condensate, where a thermostatic element failed open and also on condensate controllers. These occurred in cast iron. The problem was corrected by going to the stainless steel cap extensions. Some cap erosion has occurred where the cap extension was not bolted tightly and evenly to the cap, resulting in a leaking gasket.

The thermostatic element in the "L"'s and "M"'s has a 3/16" orifice which is satisfactory for running loads. If extra fast start-up is wanted, an external thermostatic vent can be used such **as** *a TTF-1.*

In some cases, the condensate controller configuration with its purging ability is superior. These cases would include:

1) *header type heat exchangers (as compared* to *serpentine coils);*
2) *heat exchangers with large cavities filled with steam;*
3) *where syphon lifts occur resulting in flash steam being produced ahead of the trap.*

In 1) and 2) above, air pockets can *exist undisturbed if drainage is sluggish, and there is no positive flow of vapor. The purging of the condensate controller helps to eliminate this air. In 3) above, the flash steam is purged which would otherwise prevent the float mechanism from efficiently evacuating the condensate. In most cases 1), 2) and 3), the 1/8" orifice* **of** *the factory-built condensate controller is inadequate to produce the desired purge. The preferred approach is to order an "L" or "M" liquid drainer and then pipe a by-pass from the top of the body to the return line with a 1¼" plug valve in it.*

Armstrong large capacity traps compare favorably with the **Sarco** *and the Fisher large capacity traps. Our tests have shown that the traps of the competition have insufficient buoyant force in their mechanisms to hold them steadily open on large loads. There is a strong pull to close the'valves of these traps due to the high velocity between the valves and seats. Armstrong is the only one we know of whose traps can function satisfactorily on these heavy 'loads.*

In single orifice traps, if we can overcome the force holding the valve on the seat, we usually can hold the'valve open against the "flow forces" which tend to close it. With a dual orifice trap, it's different. A dual orifice trap is designed so that. the internal pressure seeks to hold the valve shut as well as to open it. Since both valves are of the same area, the force to hold closed is equal to the force to open. On the surface, it seems that a relatively small float on a relatively short lever would be adequate to open these dual orifices. However, the high velocity of the discharge flow between valve and seat results in a significant pressure drop here which tends to close the valves. These forces must be overcome by'the float and its lever arm. Armstrong's "M" and "MS" series are the only dual orifice drainers we know of that can do this.' When dual orifice drainers are assembled, the mechanism is adjusted so that both valves close tight simultanelusly. On cold service, this adjustment holds. However, on steam service because of unequal expansion of the various parts, this "tight adjustment" is not maintained. A leak rate of 50 to 100 pounds per hour is to be anticipated.

<u>*VACUUM BREAKERS*</u>

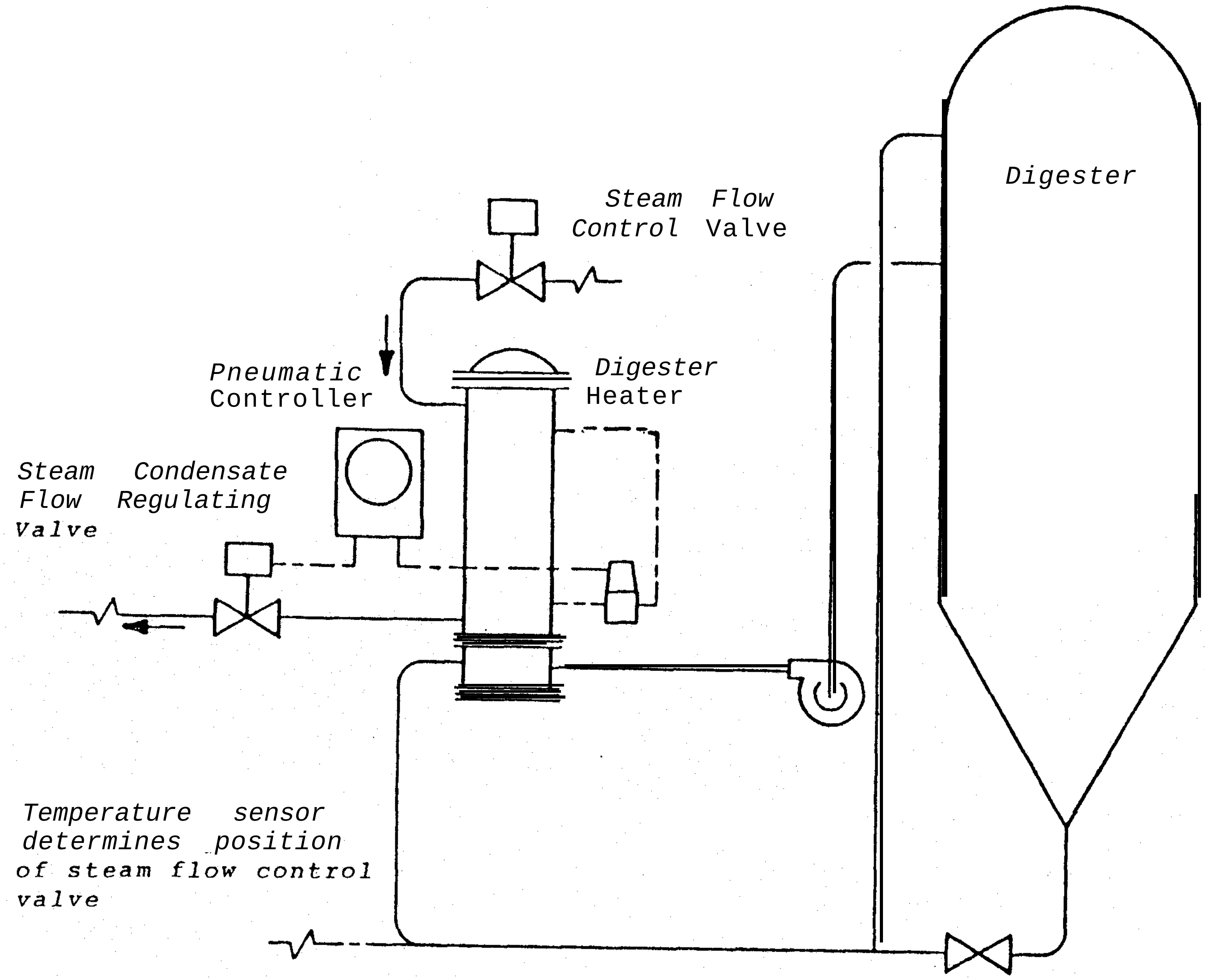
As with other steam traps, anytime a product, be it water, air or chemical, is heated to a temperature below 212°F., there is probably a need for a vacuum breaker. Large traps indicate large heat exchange equipment which indicate the need for vacuum relief in larger than usual capacities. To date, our integral vacuum breaker on an "L" or "M" trap has proven adequate.

<u>*PRESSURE LIMITATIONS*</u>

The cast iron "L" and "M" bodies are limited to use on 250 psi maximum pressure whether it's at 100°F. or 450°F. The steel "LS" and "MS" traps are limited to 450 psi regardless if the temperature is 100°F. or 650°F. In addition to the above limitations, those of accessories must be considered. The gauge glass is limited to 250 psi at 425°F. The vacuum breaker is limited to 150 psi, and the thermostatic element to 250 psi. If any of these accessories are incorporated in an assembly, their limitations will prevail over those of the vessel itself or its mechanism.

<u>*TRAPS & DRAINERS vs. LEVER CONTROL DEVICES*</u>

Very few instrument people are concerned with the use of steam traps for non-critical applications: low loads, space heating, main drips,



STEAM CONDENSATE DRAINAGE by MEANS
of LEVEL CONTROL INSTRUMENTATION

tracer drips, etc.; and therefore, don't think of them for condensate drainage on production reboilers and digester heaters, etc. They use level sensing instrumentation feeding a signal to a controller which in turn positions a control valve. (Perhaps this is partly due to erratic behavior of our competition's mechanisms.) At any rate, Armstrong high capacity traps should be considered because they are reliable, durable, simple and competitive in price.

Consider the accompanying schematic and see what is required to drain condensate by instrumentation. There is a differential pressure transmitter at about $700, a pheumatic controller at about $900 and a valve with operator (2") at $650. This totals $2,250 Net and does not include pressure taps and their installation. Sophisticated instrumentation personnel are required to service it. Compare this with an "M" series drainer at $1,432 List. (The above instrument prices reflect those of Taylor, Fischer & Porter, Honeywell, Foxboro, etc.) If a load dictated 3 or 4 "M"'s in parallel, we might not be competitive, but we're still reliable, durable and simple.

High capacity traps are used in chemical plants on evaporators, reboilers, reactors and towers. They are also used on process air heaters. Paper mills use them on digester heaters and make-up air heaters. Large office buildings, school buildings and hospitals use them on absorption chillers, water heaters (domestic 6 radiation) and air heaters. Power plants use them on condensers.

Sugar mill evaporators and juice heaters also require high capacity steam traps, but condensate controllers are almost always used. So I will defer sugar mill trapping to the condensate controller seminar of this school.

Long runs, vertical or horizontal or meandering should be avoided ahead of these traps. They should be coupled as close as possible to the heating coil. If the coil is up in the air and the trap is wanted at ground level for service accessibility, a back-vent may be necessary.

John Kremers
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