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Liquid Overfed Evaporators & Pressure Regulators – Problems & Solutions

When coupled with evaporator pressure regulators, mechanically-pumped liquid overfed evaporators can experience operating difficulties. Sometimes these difficulties can become quite perplexing and often result in less than ideal evaporator performance. For example, in your facility you may have noticed that some evaporators seem not to work particularly well. In some severe cases, evaporators simply refuse to meet capacity, falling far short of what was expected. What makes those poor performing evaporators different from others that seem to work better? This chapter examines this phenomenon in detail and presents some solutions that you can undertake to possibly remedy the performance of poorly performing evaporators.

1.1 An Overview of the Problem

Heat transfer from liquid overfed air-cooling evaporators can become severely impaired when connected with evaporator pressure regulators, particularly top-fed evaporators. Because evaporator pressure regulators are widely applied, their adverse impact on refrigeration system performance is often overlooked and frequently misunderstood. The issues arising in applying regulators on overfed evaporators is complex and one not generally considered by many refrigeration system designers. So we decided to dedicate this technote to develop both your better understanding of the pitfalls of regulators in this application and to explain this rather complicated subject in a way that you can hopefully understand. However this topic does involve a thorough understanding of the pressure- temperature relationships of ammonia, particularly the term *liquid subcooling*. A copy of the Little Red Book, published by the University of Wisconsin Department of Engineering Professional Development comes in handy to readily look up pressures and temperatures as well as subcooling and superheated fluid properties.

1.1.1 Achieving Good Evaporator Heat Transfer

Some text from ammonia refrigeration classes we teach at U/W bears repeating here:

> *What is an evaporator? It is a heat exchanger wherein a boiling refrigerant removes heat from a circulating fluid (air, water, glycol, etc).*

Note the key word here is *boiling*. Why boiling? Well, for several reasons. First, liquid ammonia at its saturation temperature is the state point that the evaporator manufacturer assumed when designing the evaporator. Second, when a refrigerant (ammonia in this case) is boiling, it removes a lot of heat from the fluid being cooled (about 550 – 600) Btu for each lb of ammonia liquid evaporated). Third, the heat transfer rate is highest when the refrigerant is undergoing a change in phase from a liquid to a vapor. Keep in mind that whenever ammonia inside your evaporator isn't boiling, its ability to absorb heat from a space or a process can become vastly impaired.

1.1.2 The Two Forms of Heat Transfer

The change in energy content of a refrigerant as it moves through an evaporator takes place in these two forms:

- Sensible (i.e. the refrigerant temperature increases but does not change state), and
- Latent (i.e. refrigerant temperature is constant but the added thermal energy is changing the state of the refrigerant from a liquid to a vapor)

It is highly desirable that *the need for* sensible heating of liquid inside an evaporator be as limited as possible (less is better). Conversely, the more heat transfer taking place in the latent form is much desired (more is better here).

In a *sensible* heat transfer process, a 'primary' fluid (be it water, glycol, or other working fluid – such as a subcooled refrigerant) is pumped though a heat exchanger absorbing energy by increasing its temperature as it exchanges heat with a space or a process. During a *sensible* energy change, the working fluid does not undergo a change in phase (stated another way, it neither boils nor condenses). This type of process is common to a brine or water coil for either cooling a 'secondary' fluid, like air.

In contrast, a *latent* heat transfer process relies on the change in phase between a liquid and a vapor. Remember that during a change in phase (i.e. an evaporation process), the temperature of the refrigerant remains constant for all pure fluids, like ammonia. The advantage to the latent energy change process is that each pound of refrigerant we evaporate in a heat exchanger has the ability to carry away considerably more heat than if it only relied on sensible energy change. In the case of ammonia, 1 lb of liquid refrigerant that is boiled in an evaporator removes approximately 550 – 600 Btu of load. To remove the equivalent amount of energy with a sensible heat exchange process and a rise in temperature of 20 F, we would need 27.5 lb of fluid!

If we were to solely rely on a sensible heat transfer process to exchange heat, the primary fluid (or refrigerant) will have to be circulated through a heat exchanger at a much greater flow rate to achieve the same design capacity as a latent energy change process. As our example, let's use a heat exchanger rated for 25 tons of cooling capacity and see how this heat exchanger would perform under these two very different modes of heat transfer.

1.1.2.1 25 Ton Heat Exchanger, Sensible Heat Transfer

Some facilities may have a secondary loop cooling system that uses a mixture of glycol/water or calcium chloride/water to serve heat exchangers in an effort to eliminate ammonia from the working environment. Let's assume that our heat exchanger uses propylene glycol as its primary heat transfer fluid. How much of this propylene glycol solution would we have to circulate in order to meet a 25 ton cooling load? The answer to this question depends on the entering and leaving fluid temperatures across the heat exchanger, but let's assume our glycol changes temperature by 5 degrees Fahrenheit (it's leaving temperature is 5°F warmer than its supply) inside the heat exchanger. If this becomes our primary fluid temperature rise, we would need to pump 135 gallons per minute in order to achieve 25 tons of cooling capacity.

1.1.2.2 25 Ton Heat Exchanger, Latent Heat Transfer

Now let's use pumped recirculated ammonia to achieve 25 tons of cooling capacity by changing its phase from liquid to vapor. How much would we need? Using a 4:1 recirculation rate[1], our

[1] A 4:1 recirculation means that we need to pump 4 times the quantity of ammonia that actually boils inside an evaporator so that we completely wet the entire interior piping surface end-to-end with boiling liquid. This is considered by most engineers to be satisfactory for a pumped overfed application.

circulating pump would have to push 6.25 gallons per minute into the heat exchanger (evaporator), roughly 1/20th of that needed for the 25 ton sensible heat transfer process considered above. What does this tell us?

1.1.2.3 What Happens If the Ammonia Doesn't *Boil?*

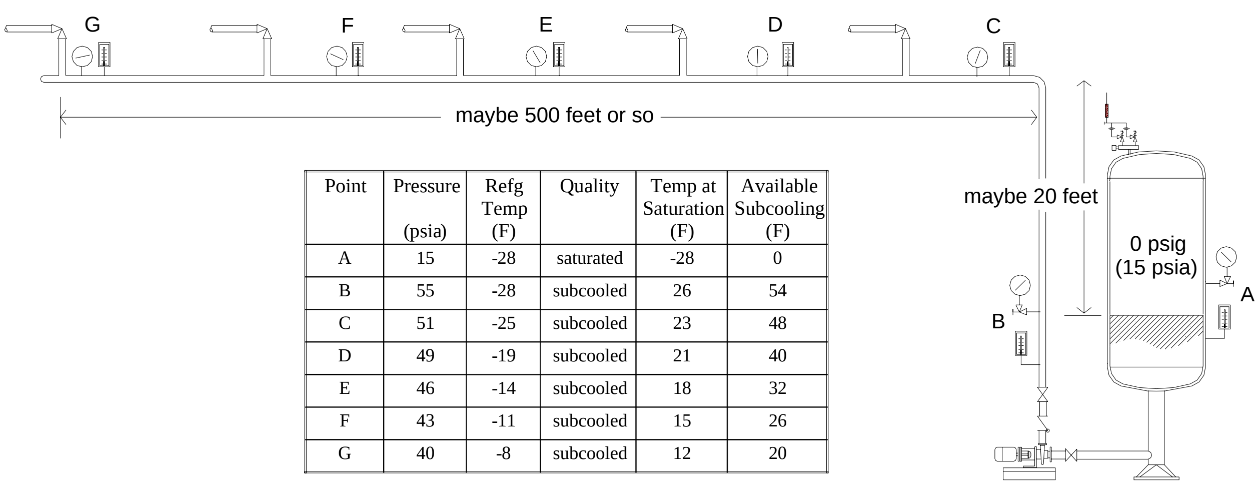
Sensibly heated liquid ammonia will carry away a little more heat per gallon than a propylene glycol solution but not by much (maybe 10% - 12% more). The take-home message: if our circulated ammonia doesn't boil inside our evaporator and it is merely heated sensibly, we are going to come up short of our 25 tons. *Really short*. Do you have any evaporators matching this description? Many plants do.

1.1.2.4 What Prevents Ammonia from Boiling?

This question can be answered in one word: *subcooling*. So if subcooled liquid ammonia doesn't boil inside an evaporator, how did it become subcooled in the first place? For us to understand this condition, we have to stand back and look at the Big Picture.

1.2 Pumped Liquid Recirculation Systems – The Big Picture

Consider the hypothetical pumped accumulator and liquid piping system shown in Figure 1.



Point	Pressure (psia)	Refg Temp (F)	Quality	Temp at Saturation (F)	Available Subcooling (F)
A	15	-28	saturated	-28	0
B	55	-28	subcooled	26	54
C	51	-25	subcooled	23	48
D	49	-19	subcooled	21	40
E	46	-14	subcooled	18	32
F	43	-11	subcooled	15	26
G	40	-8	subcooled	12	20

Temperatures have been rounded to nearest degree

Figure 1: Pumped accumulator & liquid line with defective insulation.

The figure above shows a pumped liquid accumulator operating at 0 psig. We'll assume for purposes of discussion that the liquid line insulation is in need of replacement (with lots of frost clearly visible). If the liquid line has visible frost, this is a clear sign that heat is passing through the insulation, thus warming the liquid as it flows through the piping. We'll also assume that we have some means for checking both the temperature and pressure of the liquid (with gauges and thermometers where shown). Each 'point' in this system has been indicated with a letter designation (first column of the table above).

At point 'A' the liquid in our accumulator is at saturation pressure and temperature of 0 psig (15 psia) and –28 F respectively. As the liquid passes through the circulating pump, its pressure increases – our example assumes the pump raises liquid pressure by 40 psi (shown as point 'B'). This pumped liquid is now subcooled (also known as 'pressurized' liquid).

1.2.1 Achieving Liquid Subcooling

There are two ways we can subcool liquid ammonia (or any refrigerant). These are:

- Take heat away from saturated liquid refrigerant with no change in pressure (in a heat exchanger), or
- Increase its saturation temperature by increasing its pressure without increasing the temperature of the liquid (in a mechanical pump)

Figure 2 graphically illustrates these two methods of obtaining subcooling. The first method listed above is represented by the line from Z to B. If we had 55 psia saturated liquid (point Z), we could cool it in a heat exchanger to -28°F and we would reach the subcooled liquid state shown at point B. The second listed way is by increasing the pressure of the refrigerant as shown from line A to B (corresponding to the state points in *Figure 1*). In the case shown in Figures 1 and 2, we begin with saturated liquid at 15 psia and we increase its pressure through a mechanical pump with a negligible change in the liquid's temperature[2] to obtain subcooled liquid at point B. Both of these methods result in an identical ending state point.

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[2] A negligible temperature rise assumes the pump is not cavitating; a slight temperature rise will be observed if a pump is cavitating.

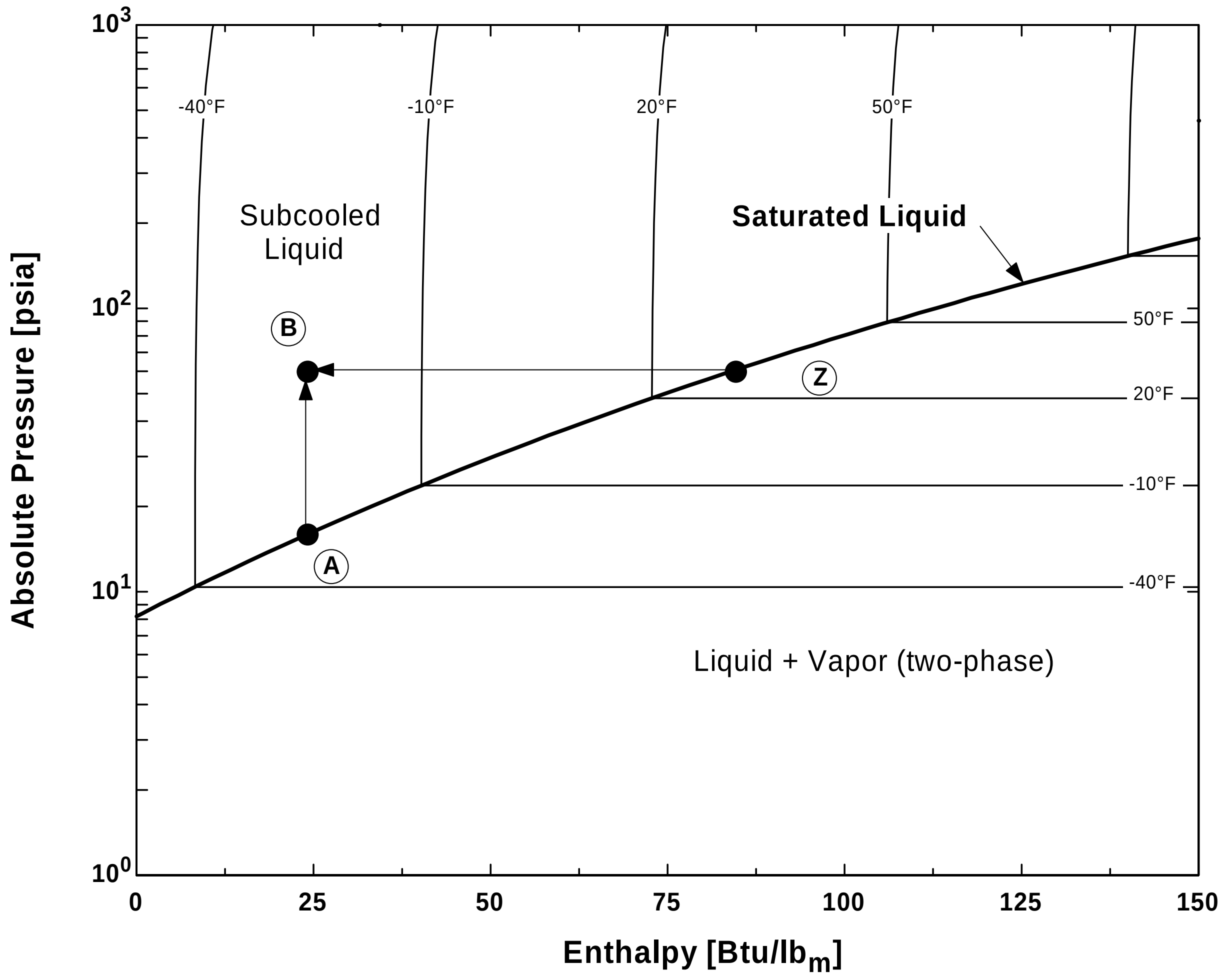

Figure 2: Two ways of subcooling liquid on a p-H diagram.

Liquid refrigerant circulating pumps accomplish subcooling by increasing the pressure of the saturated liquid with a negligible change in the liquid's temperature (similar to the process in moving from point A to B in *Figure 2*). How much liquid subcooling[3] (expressed in degrees Fahrenheit) does the liquid possess at point 'B'? The answer is determined by subtracting the saturation temperature of the refrigerant at the vessel pressure from the saturation temperature of the liquid refrigerant at the pump outlet pressure. The degree of subcooling is shown in the right-hand column of the table included in *Figure 1*: 54°F. Stated another way, the liquid at the pump

[3] Subcooling is the difference in temperature between the refrigerant's saturation temperature and its current temperature – a measure of how many degrees cooler the refrigerant is below its saturation temperature. For example, consider a glass of ice water. If its temperature is 32º F, it has 180º F subcooling under a pressure of one standard atmosphere (14.696 psia).

discharge would have to be heated by 54 degrees Fahrenheit in order for it to reach a temperature at which it would begin to boil (evaporate).

Beyond the pump outlet, the liquid line runs up to the roof which is about 20 feet above the liquid level in the accumulator for the example shown in Figure 1 – and its pressure drops to 51 psia (point ‘C’), due to the decrease in liquid static head and piping friction losses. At this point, its temperature has simultaneously risen to –25°F, due to solar and ambient heat gain. The combination of pressure loss and temperature increase results in a decrease in available subcooling for the process – the liquid inside the liquid line is now subcooled by 48°F.

From here, the liquid line continues to run across the roof, branching off to serve evaporators at points D, E, F and G. Our example in *Figure 1* assumes the total run of piping across the roof is about 500 feet, rather typical for a large facility. At each point, liquid pressure progressively decreases while the parasitic heat gains simultaneously raise the liquid temperature due to faulty insulation. However our pump is keeping us out of trouble because, for the entire 500 foot run, liquid is maintained in a subcooled condition. In other words, we don’t have a problem with liquid flashing to vapor in the liquid feed piping which would significantly restrict the flow carrying capability of the pipe. By the time liquid reaches the end of this long piping run, we still have 20 degrees of liquid subcooling (at point ‘G’). If our piping insulation were to deteriorate further or if our pump didn’t produce 40 psi of pressure “lift”, we would probably experience problems with flash gas forming somewhere inside the liquid piping. The formation of flash vapor in any significant quantity would restrict liquid flow to the evaporators furthest away from the pump i.e. “tail-end Charlie” at point G.

1.2.2 Top-Fed Evaporators

Now let’s roll two evaporators into this scenario – the evaporator closest to the pump at point C and the one at point G – tail-end Charlie - and see what’s happening. Figure 3 shows the evaporators at both ends of the pumped recirculation liquid line that were previously shown in Figure 1. In coordinating these two figures, we’ve labeled the evaporators to coincide with the points shown in Figure 1.

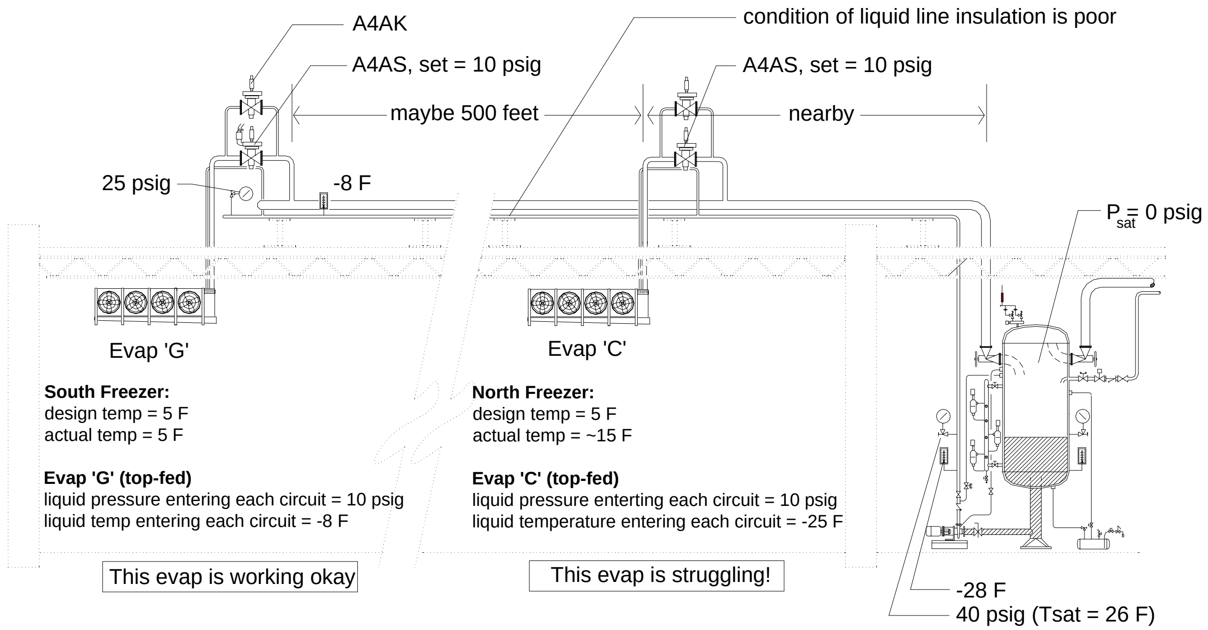

Figure 3 Top-fed evaporators with evaporator pressure regulators

1.2.2.1 Evaporator 'G'

Just before liquid enters the upstream liquid line isolation valve feeding evaporator 'G', its quality is subcooled (by 20°F – see the table in Figure 1). Then the liquid passes through a globe, a strainer, a solenoid, a check valve and a hand balancing valve, finally dropping down through the roof where it splits off through the evaporator's liquid header into a multitude of parallel button orifices at the entrance of each liquid circuit in the evaporator. After passing through these orifices, the liquid pressure drops to 10 psig – the set point of the evaporator pressure regulator. At this point, it's temperature is also at saturation i.e. –8°F[4]. Because the refrigerant is all liquid at its boiling point, it will begin evaporating immediately upon entering each circuit as it absorbs heat from the coil surface. As Figure 2 indicates, "This evaporator is working okay" – we're able to meet our 5°F freezer temperature. Now let's look at evaporator 'C' and see what could possibly be troubling this unit.

[4] Confirm that we are at saturation conditions for a pressure of 10 psig by checking to see what the saturation temperature corresponding to 10 psig in your *Little Red Book* of ammonia properties.

1.2.2.2 Evaporator 'C'

Liquid entering the valve train serving evaporator 'C' is subcooled by 48°F. After passing through a similar liquid valve train and button orifices, liquid enters each circuit at –25°F under a pressure of 10 psig (25 psia). When we check our ammonia properties book, we find that this liquid still possesses 17°F subcooling. In other words, the liquid refrigerant in this case will not immediately begin boiling when it enters this evaporator – all of the liquid has to first sensibly warm up by 17°F (to reach its saturation temperature) before any will begin to evaporate. Furthermore, because these coils are (liquid) top-fed, very little of the internal piping surface comes into contact with liquid because gravity is acting upon it, pulling it down into the lower tube passes, deep inside the evaporator. The result: the liquid refrigerant is not effectively being heated to its saturation temperature by the airstream passing over the evaporator fins and tubing. This delays the boiling process, degrading evaporator capacity as shown in the next *Figure 4*.

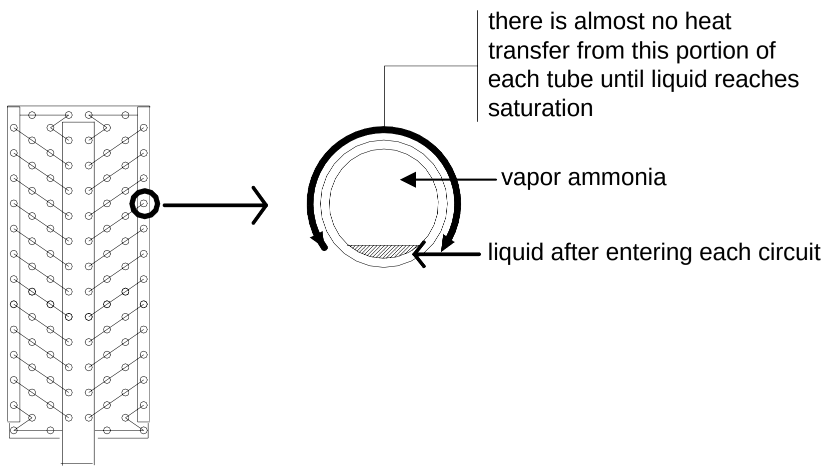

Figure 4: Top-fed evaporator feed with subcooled liquid.

1.2.2.3 What Creates This Subcooled Liquid?

Note the large-scale tube shown in Figure 4 and how liquid simply 'lays low' at the entrance of each circuit. This is due to its subcooled state. You can visually see how this could occur when you pour cool water into a pan and put the pan on the stove and crank on the heat. Does it boil? No. Instead, it has to be heated to its boiling point first. Upon reaching its boiling temperature (its saturation temperature), it becomes vigorously agitated. An identical process occurs inside an evaporator – when liquid becomes agitated, it splashes up, coating the inside periphery of each tube, completely wetting it. This completely wetted condition is needed to achieve rated heat transfer (our 25 tons).

Liquid fed from a 0 psig accumulator becomes subcooled in evaporators by the presence of evaporator pressure regulators. Evaporator pressure regulators ‘artificially’ hold liquid back pressure to whatever pressure the EPR has been set, in this case 10 psig (25 psia). The saturation temperature of ammonia at 25 psia is –8°F while liquid entering the pump (at saturation) is considerably colder, –28°F. In the case of evaporator ‘G’ (Figure 2), both the liquid line pressure drop and the parasitic heat gain from the sun and the warm ambient provide sufficient sensible heating to raise the liquid saturation temperature outside of the evaporator just enough so it will boil immediately upon entering each circuit within the evaporator. But in the case of evaporator ‘C’, the liquid still possesses too much subcooling for it to immediately boil, even after passing through the valve train and button orifices. The result? The refrigerating capacity of this top-fed evaporator drops significantly.

1.2.3 Bottom-Fed Evaporators

Evaporators configured for bottom-feed do not experience the same capacity degradation as those configured for top-feed, although the impact of an EPR still takes its toll. In a bottom-feed configuration, liquid completely fills each tube after passing through the respective button orifices. When liquid comes into contact with the entire inside of the tube, heat from the incoming airstream warms the liquid more quickly with the result that it begins to boil sooner (rather than later as in a top-fed evaporator).

1.3 Steps You Can Take To Minimize or Eliminate This Problem

1) *Reduce Evaporator Liquid Feed*

   Excessive liquid feed to evaporators tops the list when *brining*[5] is suspect. Oftentimes when evaporators are initially set up and adjusted, a technician might feel that “about half-way” is the correct hand valve setting. This is usually wrong – more often than

[5] Whenever subcooled liquid is fed into an evaporator, fluid brining (single phase heat transfer) occurs up until the liquid has reached its temperature at saturation, at which point boiling heat transfer takes over. This is one form of brining. We will have more to say on the complex subject of brining in upcoming technotes.

not, you'll find that with today's oversized hand valves (aka "needle valves") setting the valve as little as 1/8th turn open is about right. We'll have more to say on this topic in future technotes and steer you to some software that will assist you in setting liquid feed hand valves on liquid overfeed evaporators.

2) *Reduce Setting of EPRs*

Reduce your EPR setpoints to the extent possible – (increasing EPR setpoint increases your liquid subcooling and delays liquid saturation even further – this only makes matters worse). The extent to which you are able to reduce your EPR setpoint will depend upon the refrigerated space. Can you tolerate a lower relative humidity here or will this dry out product? If product is wrapped and sealed, you can generally tolerate a somewhat lower evaporating pressure[6]. Care must be taken here to insure that you have adequate regulator capacity to regulate the increased load due to a larger coil temperature difference (TD) over a smaller pressure difference. In addition, the HEV may need to be adjusted more open (only slightly!) in order to maintain the correct overfeed rate.

3) *Eliminate EPRs & Raise Compressor Suction Pressure*

Of the three steps discussed, this is by far the most preferable. However the oil eliminator shell diameters of your connected screws compressors will govern the extent to which you will be able to raise your suction pressure without oil carryover.

Remove all EPRs and replace them with suction stop valves (CK-2, CK-5 or S9A) and raise your suction pressure to compensate for the operating pressure loss across an EPR. Doing so will increase compressor capacity – in the case discussed here, raising your suction from 0 psig to 10 psig would increase capacity of those compressors serving the accumulator by roughly 38%. This is an option which should only be undertaken with engineering assistance.

4) *Pre-Heat Liquid Feed*

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[6] Generally we would believe that a lower evaporating pressure will increase frost loading. But if your evaporator is suffering along with an EPR, it probably isn't loading too much frost to begin with.

As we saw in Figures 1 and 2, the liquid line is being heated by ambient and solar parasitics. So what if we replaced the faulty insulation – what then? Even ‘tail-end-Charlie’ would run into trouble with liquid subcooling. So what if we removed the liquid line insulation altogether? Would that work? Theoretically, yes – but only during hot sunny weather. However, the parasitics (our source of heat to warm up the liquid so that saturation is achieved inside each evaporator) aren’t always constant. Besides, we hardly think a frosty liquid line is a realistic logistical solution to this problem. How much heat would we need and how could this be done? This solution isn’t something you’d be able to do yourself – it would require some engineering and a contractor. A separate liquid line heater would have to be installed for each evaporator – a very costly undertaking indeed and one we really hesitate recommending due to the complexity involved. But, it *is* possible. As a source of heat, you could use vapor from your hot gas line and condense it in a small heat exchanger used to preheat your subcooled liquid by a few degrees. However, again – this option would be highly unusual, to say nothing of its cost.

If we knock this option out, then that leaves us with options 1, 2 or 3 with 3 being the most preferred. However, we have seen cases where sensibly heating the liquid line a few degrees with some electric resistance heaters had amazing results.

1.4 Closing Thoughts

1.4.1 Reinsulating Liquid Line Piping

One of the unfortunate aspects of this problem will manifest itself the day after your piping insulator finishes reinsulating your liquid line. Now, all of the evaporators connected to the branch line runouts in Figure 1 will bump into the same problem we discussed with Evap ‘C’ – liquid subcooling resulting in a loss of evaporating capacity. Even poor tail-end Charlie won’t be feeling too well. What steps can you take? Our recommendations would generally be to (and your mileage may vary):

Step 1 Raise your suction pressure to the extent possible. Again as we mentioned earlier, this will depend upon your engine room. We’ll assume your compressors are screws.

1.4.1.1 Screw Compressor Oil Eliminator Shell Diameter

All twin screw compressors currently available from the marketplace require an oil eliminator. Oil eliminators come in standard shell diameters in 6 inch increments (30”, 36”, 42”, etc). The larger the shell diameter, the more oil coalescers that will fit inside. And as shell diameter increases, more vapor can flow through it while allowing for a sufficiently low gas velocity so that small oil particles will drop out into the sump (in other words, less oil carryover into your hot gas line). Two factors influence gas velocity through the shell, one of them is ‘strong’ while the other is ‘weak’:

- Suction pressure (the ‘strong’ influence)
- Discharge pressure (the ‘weak’ influence)

An increase in operating suction pressure is always desirable from an energy standpoint as we have discussed in our classes at U/W. But a limitation arises: how large are your oil eliminators? Can you effectively raise your suction pressure? Nearly always, the answer to this question is ‘yes’, but just *how much* we can’t say. Our recommendation is to first check with the screw compressor manufacturer, giving them your compressor package serial number(s). They will probably want to know how high you’d like to raise suction pressure and quite frankly at this point, you may not know. You also have another factor to consider and that being compressor motor sizes. When you raise suction pressure, will your motor amps exceed the factory setting for maximum operating amperage?

Our suggestion is to ask the manufacturer just *how far* you could raise suction pressure without either overloading your motor or carrying oil out the discharge line. This will give you a starting point.

Using the example in the preceding figures (air-cooling evaporators), about as high as you’d want to raise suction pressure (if you could) would be roughly 12 F – 15 F below the desired room temperature – in this case the room is 5 F, so subtracting 15 F from room temperature, you’d have a saturated suction temperature of –10 F with a 9 psig saturation pressure. If all of your regulators were set at an equal pressure (10 psi in Figure 3), then add this pressure to the existing suction pressure.

Step 2 We'll again assume the regulators in your evaporator suction lines are the model A4AS (and again, your mileage will vary, depending upon your type(s) of regulators). Remove these and install a gas-operated suction stop valve. We happen to prefer the normally-open CK-5 because of its delay-on-reopening feature in the event of a power failure.

Step 3 Rebuild each A4AK defrost regulator with a new, smaller seat and disk (a kit available from R/S) and convert your regulator to an A4A. Start by setting its pressure at 70 psig and see how this works for you, then readjust upwards or downwards later as you see necessary (lower is better, if you can do it – less evaporator 'steaming' during defrost).

After you've completed these three steps, we'd be surprised if you can't shut down a compressor or two. Your facility has now become more profitable as a return.

1.4.2 Feed Issues to Other Evaporators Having No EPRs (or EPRs at Different Setpoint Pressures)

Let's suppose for a moment you're feeding multiple evaporators from one pump, and each evaporator regulator has a different setpoint. Man – you got yourself a whale of a pup to try balancing!

Two methods:

- This is where liquid pressure regulators come in very handy. Have your contractor install an R/S model FFR-2 (or AFR-3) liquid pressure regulator in each liquid feed line. The FFR-2 has a fixed orifice and does not come with an internal check valve. The AFR-3 has an internal check – the AFR-3 takes the place of a hand (needle) valve and the liquid line check (usually a CK4). After installing these, your system will balance itself.

- Tough it out. If you have gauges downstream of the hand valves this facilitates the job. Set your hand (needle) valve for a pressure about 5 psi above the evaporating pressure (this assumes the evaporator manufacturer used a 5 pound loss across their button orifices (this is usually close). However if you don't have a gauge, your balancing job becomes tougher. Start by fully closing the hand valve. If your valve is larger than 1", you'll wind up throttling the valve almost to the closed position (maybe less than ¼ turn open). Have someone assist you while each of you have walkie-talkies. Immediately following defrost (clean, frost-free evaporator surface) gradually open your hand valve (1 full 360 degree turn per 30 minutes) and watch the frost growth. When it comes up to the top of the evaporator, stop. Then you might even ratchet your hand valve back a bit, especially if you're close to the refrigerant pump.

End Technote01