# Brining in Liquid Overfeed Evaporators

Impacts Upon Top-Feed & Bottom-Feed Configurations

Key words: refrigeration; brining; subcool; evaporators

## *Abstract*

A mechanically pumped, bottom-fed liquid overfeed evaporator and its 45 foot recirculated return riser are discussed. What makes this evaporator unique is that the initial temperature difference (TD) is a variable with a declining value over time. This is typical of a blast cell. The causes behind the inevitable riser liquid stacking are discussed; impacts upon both bottom-fed and top-fed evaporators are illustrated.

This particular blast cell meets capacity for approximately one hour (after loading in warm boxed product on carts and closing the door). But after this, cell air temperatures begin to rise unexpectedly. The reasons governing this phenomenon are presented and discussed.

## 1 *Introduction*

### 1.1 *CPR_Fed Evaporators Excluded*

I should point out that this discussion covers mechanically pumped evaporators – namely, an electric-drive pump(s), normally centrifugal, feeds refrigerant liquid through a network of piping to one or more evaporators. Frequently, the static head component of pump TDH becomes the major force. Elsewhere on my website, you'll find a discussion of another type of overfed evaporator, namely that fed via a controlled pressure receiver (CPR-fed). These behave differently than mechanically pumped evaporators because these are fed from a higher temperature liquid source.

The pumped evaporators covered in this discussion are fed from an accumulator, normally thought of as holding the same liquid temperature as saturated liquid within an evaporator. This is not completely true. Oftentimes, this deviation from saturation plays havoc with evaporator performance (known as *brining*), particularly at low initial ($T_{\text{inlet air,db}} - T_{\text{sat,evap}}$) temperature differences. Top-fed evaporators are far more sensitive to this phenomenon than are bottom-fed as will be illustrated.

### 1.2 *Evaporator Brining – Its Causes*

Brining in a refrigerant evaporator can occur at either end of a span of temperatures. An example of a high temperature brining condition would be a heat transfer process with a 100 ºF initial temperature difference. Liquid ammonia simply doesn't come into contact with metal – it's too hot. At the other end, we have a condition wherein cold liquid is forced into individual evaporator circuits at a temperature $T_{ref,supply} < T_{sat,evap}$, the refrigerant's local boiling temperature. This deviation below saturation is called *liquid subcooling* and can be caused by any number of factors having one common denominator: they are all resistances to flow and are normally in series. The following list is not necessarily all-inclusive.

- Overly-long (> 10 ft, 3 m) recirculated return (suction) risers
- Back pressure regulators
- Too small or too large pipe diameters
- Partially closed valves
- Too many valves in series
- Oil accumulation
- Dirt, junk, 2 x 4's, shop rags, lunch boxes (you name it)
- Pressure losses through accumulators (very rare)

These also have another commonality: they all occur in the B to C line shown in a figure I'll present shortly.

The example chosen for this discussion is an evaporator meeting the ever-declining loads inside a blast cell. Here, the term TD is a variable. The owner wanted an efficient system so a 6" diameter riser was chosen because "Bigger is better, right? Lower pressure drop!" So how does one go about selecting an evaporator in a case like this? Often, these are selected based upon some stipulated mean air velocity over product. Then evaporator air flow (cfm) is found from:

$$cfm_{evap} = (A_{room} - A_{product})\ V_{air} \qquad (1)$$

Often, designers omit product from Eq 1 and simply use somewhere between 400 fpm to 500 fpm as mean room (cell) cross section air velocity. When selecting an evaporator to match room cfm, the evaporator face velocity must be considered. Wet coils are limited to 600 feet per minute face air velocity to avoid water carry-over. Frosted coils can accommodate higher face velocities (up to 750 fpm), provided the defrost sequence includes a "pre-chill" during which fans are off, liquid solenoid is open and suction stop valve is open. This interval only requires a few seconds (~15), during which any remaining water droplets adhering to tubes and fins are refrozen in-place.

The strong force in any heat transfer process is temperature, namely the *difference* between the warm source (entering air) flowing over a tube to the *saturated temperature* of liquid ammonia flowing inside. The assumption regarding the statepoint of liquid (saturated) is crucial to attaining rated evaporator capacity. If the liquid supply isn't saturated, then it won't boil because it hasn't yet reached its boiling temperature. To make this happen requires both temperature *difference* and surface area. Some portion of an evaporator's surface area becomes dedicated to adding this additional sensible heat (about 1.05 Btu/lb-ºF), therefore latent capacity falls. Many of today's evaporators, especially multi-fan, have sufficiently long face tubes to address up to a 10 ºF temperature difference between $T_{ref,supply}$ and $T_{sat,evap}$ when the air-to-refrigerant TD ≥ 10 ºF, however overall heat transfer capacity from rated (listed) will suffer. In a four-pass arrangement, with 10 ºF degrees of entering liquid subcooling, ~25% of total evaporator surface area becomes dedicated for sensibly heating liquid to saturation in a bottom-fed evaporator. A top-fed evaporator can typically see a 50% reduction.

But less than 10 ºF, all four wheels fall off the cart. If the air TD is less than 6 ºF, and the evaporator is top-fed, its ability to transfer heat can become so severely compromised that it simply ceases to function. Looking at an evaporator in this condition, one would see narrow stripes of frost running along bottoms of all tubing. Cold liquid flows in, cold liquid flow out and nothing seems to happen. A client once asked "My liquid doesn't gas-off. Why?" When observing a tube, you'll see that some frost will form if liquid is cold enough but it doesn't thicken and grow out onto fin surfaces. The entire brining problem is further compounded by its high latent heat of vaporization ($h_{fg}$=590 Btu/$lb_m$ at-30 ºF). Less liquid is needed for identical capacities, therefore even at a 4:1 recirculation ratio, liquid flows into each circuit at such a low velocity that its inlet condition in a given tube results in laminar flow ($N_{Re}$ <2000). During laminar, heat transfer between liquid ammonia and tubing wall is at its least effective compared with that same tube when it sees a boiling liquid and a high Reynolds Number, $N_{Re}$>3000.

# 2 Technical Background

## 2.1 Energy & Mass Balances

Examine Figure 1, a diagram of a recirculator package supplying pumped liquid to an evaporator. Both bottom and top-fed liquid feed methods are illustrated. The letters A, B and C designate:

A: vapor pressure at evaporator inlet (actually occurs deep within each circuit), $lb_f/in^2_{evap,inlet}$

B: vapor pressure at evaporator outlet, $lb_f/in^2_{evap,outlet}$

C: vapor pressure above liquid inside accumulator, $lb_f/in^2_{accum}$

Component mass and energy balances for this recirculator pump and evaporator are:

Mass: $$\dot{m}_{C\to A} - \dot{m}_{B\to C} = 0 \quad (2)$$

Energy: $$Q_{evap} - W_{pump} + (\dot{m} h_A - \dot{m} h_B) = 0 \quad (3)$$

Rate: $$\dot{Q}_{evap} \approx U \cdot surface_{evap}(T_{inlet,air,db} - T_{sat,evap.B})_{evap} \quad (4)$$

The term *W* in energy equation 3 is negligible, therefore it's normally neglected.

We also recall:

$$Q_{evap,ref} - Q_{air\ stream} = 0 \quad (5)$$

and $$\dot{Q}_{air\ stream} \approx 1.1\ cfm\,(T_{inlet\ air,db} - T_{outlet\ air,db}) \quad (6)$$

## 2.2 Changes in Tube Velocity

Up until liquid reaches saturation, tube fluid velocities stay very low; zero vapor exists in the lowermost passes of a bottom-fed circuit. Therefore, 100% of its tube diameter becomes flooded, making it available for sensibly heating incoming supply liquid. This isn't true in a top-fed evaporator. If we consider face tubing on the air entering side, that entire face may see a zero vapor velocity (known as *slip* velocity) flowing over the surface of liquid lying in the bottom of these particular tubes. Very little of the total tube surface becomes exposed to liquid, therefore tube heat transfer becomes a function of the percent surface in actual contact with cold liquid; often this is <10%. The balance of that tube only sees vapor and worse, it is stationary – it isn't moving, $N_{Re} = 0$. Heat transfer through this particular tube is zip, nada, zilch.

Now consider that same evaporator supplied with liquid at saturation. No messin' around here – liquid begins boiling *immediately*. Vapor begins to form – and lots of it. At -30 ºF saturated, liquid ammonia changes volume (at constant pressure) by slightly over 800 times its volume in a liquid state. Since the cross sectional area of a tube is a constant, a rapid increase in vapor velocity takes place in that section of a given tube. When the air-to-refrigerant TD is high (say 25 ºF), tube slip velocity reaches the point where plug flow occurs quickly, doing so within a few inches of the entrance. Then the entire tube circumference becomes wetted with liquid droplets and heat transfer increases by 10 times or more of what it was before liquid started boiling.

But when the TD is low, say less than 6 ºF, plug flow occurs much deeper into the coil bundle and the loss of HX capacity accelerates. But the key is: it *does* occur *if* the coil designer took low heat exchanger TD into account by increasing the number of passes. Bottom-fed evaporators respond more favorably under these conditions than their top-fed counterparts. Lesson learned: it doesn't take much to mess up a top-fed evaporator performance.

### *2.3 Seemingly Minute Pressure Differences Mess Up Top-Fed Coils*

A top-fed evaporator coil bundle is susceptible to very small liquid temperature deviations, on the order of 1 ºF with a 10 ºF TD (in my opinion). This one degree Fahrenheit difference at -30 ºF ($T_{ref,supply}$=-31 F, $T_{sat,evap}$=-30F) seems insignificant when viewing their respective pressures at saturation. Example: 13.9 psia – 13.5 psia equals 0.4 psid. It's difficult to read this deviation on a psi format pressure gauge. It's *impossible* to read on a gauge graduated in bar, therefore it goes unnoticed. "I don't see any pressure differences, Herr Denkmann. What are you talking about?" It's because bar pressure gauges are incapable of showing a change = 2.759E-2 bar. Now imagine what happens to an evaporator when its entering liquid has been subcooled by 20 degrees! You would just look at those little frost strips hugging the bottoms of tubing and wonder WTF? Now you know.

Recall the typical method of supplying liquid ammonia into our subject evaporator: a single, "set it once and forget it" liquid balancing valve and solenoid. We also know that a blast cell, being a batch-fed operation, sees a falling mean product surface temperature over elapsed time. Therefore, as $T_{product,surface}$ falls, the evaporator inlet air temperature, $T_{inlet\ air,db}$ also falls. Meanwhile, mass refrigerant fed to this evaporator, $m_{ref,supply}$ doesn't change – it's a constant. In the example, the liquid balancing hand valve was initially set for a a 1:1 overfeed rate at 25 ºF TD. But as the hours lapse and product temperatures fall, this initial TD falls to 1.5 ºF TD after 72 hours (see test data in Table 1).

While the mass flow of refrigerant in and out of the evaporator (Eq 1) is a constant, the overfeed ratio becomes a variable.

Therefore, when $$m_{ref,liq} + m_{ref,vap} = \text{constant} \qquad (7)$$

it follows $m_{ref,liq}$ must rise as $m_{ref,vap}$ falls. As seen in Table 1, the overfeed ratio has now risen to 66:1 after 72 hours. Over this same time interval, the superficial vapor velocity in the recirculated return riser has fallen from 4,535 fpm to 273 fpm. The riser has now become filled with bubbly liquid.

### 2.4 Threshold of Annular Flow

The superficial (excluding liquid) velocity threshold below which liquid stacking begins is 2,750 fpm at -30 F saturated vapor. This velocity ($J_v$) may be found from the following:

$$J_v \rho_v^{1/2} = 443.075 \cdot \left[ (\rho_l - \rho_v) \cdot g \cdot \sigma \right]^{1/4} \qquad (8)$$

At vapor velocities higher than this value, the required force to drag a column of mixed phase ammonia uphill inside a tube of a given diameter becomes greatly diminished. But when $J_v$ falls below this threshold, liquids stacking begins. When it does, top-fed evaporators loose most of their effectiveness. Bottom-fed evaporators, on the other hand, aren't as fussy. But anything greater than 10 ºF of liquid subcooling, even bottom-fed evaporators will become adversely affected as well.

By the time a riser 45 feet tall becomes filled with liquid (with some bubbles of vapor – see Figure 2, condition A), the differential force applied to the base of a column of -30 ºF liquid, rises by 11.7 $psi_f$, This value has been reflected in Table 1. It's the reason why the subject evaporator ceases to function properly.

**Table 1**
**Test Results of Bendemcrunch Evaporator in Blast Cell**

| **Min following Start** | Temperature Difference ºF | Tons | Overfeed Rate | Superficial Vapor Vel. fpm | $T_{ref,sat,\ evap}$ ºF | Fluid Static Head $psi_f$ |
|---|---|---|---|---|---|---|
| **15** | 25 | 142 | 1:1 | 4,535 | -30 | 0.90 |
| **4320** | 1.5 | 8.5 | 66:1 | 273 | -7 | 11.7 |

### 2.5 Brining In a Nutshell

From equation 2, we know mass input minus mass output equals zero for both the evaporator (A to B) and the accumulator (vapor out minus liquid in). From rate equation 4, we can also see that when the evaporating temperature ($T_{sat,evap}$) starts rising due to an ever-increasing back pressure, the air temperature coming off the evaporator ($T_{outlet\ air,\ db}$) must also rise by the same change. And it keeps doing so for hour after hour. The condition known as *brining* is occurring; you're probably scratching your head and wondering why.

The same holds true in the example I've selected (a project several years back): an evaporator suffering with a 45 ft recirculated suction riser. When that's coupled with a batch-fed chilling or freezing application, with its ever decreasing product temperature over time, it is certain that liquid stacking will occur as product temperature falls simultaneously with a rising evaporating temperature. These two temperatures cannot cross; instead they can only approach one another (Figure 3). You could run this blast cell until Hell froze over and never achieve the desired product mass temperature. After calling in an ol' fart consulting engineer, he convinces the owner to run some tests with his help.. The data shown in Table 1 summarizes their findings.

The owner has stated he desires his product to be frozen within a certain time interval (15 hours), however the subject blast cell falls far short of expectations as can be seen in Figure 3, a graph of evaporating and product surface temperatures over time and the results in Table 1. Resulting internal product temperatures are usually 15 ºF to 18 ºF higher than desired, and this after quadrupling the hours initially forecast by the designer. (Can you say lawyers?)

One possible solution might be to use dual liquid solenoids and incorporate a two-step liquid feed. A dual step liquid feed arrangement won't eliminate liquid stacking; that will occur regardless of liquid mass flow rates because of the riser diameter (6″). But reducing the quantity of liquid fed to this blast cell evaporator does make sense as is evidenced by Figure 3. This will be discussed further in a later paragraph.

### *3.4 How Liquid Becomes Subcooled*

Subcooling a stream of liquid refrigerant usually confuses many people. In one breath, we engineers will say that liquid subcooling is a good thing and then turn right around and say it's a bad thing. No wonder this subject seems confusing! Let me see if I can clear the smoke away. Subcooling is a good thing when we talk about cooling a stream of high pressure, high temperature liquid. But when fed into an evaporator, a subcooled stream of liquid is not a good thing because it causes brining. As I stated previously, the resulting negative impact to rated capacity is greatest with top-fed evaporators.

Figure 4 is a partial view of a pressure enthalpy diagram zeroing in on saturated and subcooled liquid between 25.6 psia (1.77 bar) and 13.9 psia (0.96 bar). To the right of the saturated liquid curve, percent vapor mass increases. To the left of this curve, liquid becomes increasingly subcooled (colder). Liquid lying directly on this curve we ol' fart engineers call "on the bubble". Said another way, liquid at saturation cannot hold any more heat without beginning to change state (into a vapor).

Looking at this figure, we can visualize two paths for subcooling a stream of liquid:

- Z to A: we take heat away from a liquid stream while holding its pressure constant, or
- C to A: we leave $T_{ref}$ constant while raising the fluid pressure (example: in a pump)

Z to A would take place inside a heat exchanger; a shell and tube intercooler is one such example of this process. C to A is regarded as isothermal (constant temperature) because liquid is incompressible. When warm product (usually boxed) initially enters a blast cell, liquid will also be entering the evaporator very close to a saturated condition because the suction riser is handling a high percent vapor mass ratio and doing so at a high upward velocity, >2750 fpm. But after an hour of chilling, refrigeration load has steadily decreased, therefore liquid stacking begins and backpressures begin rising. After many long hours, $T_{ref,supply}$ didn't change but its pressure did, rising by 11.7 psi. At this new pressure but same temperature, our liquid stream is now chilled by 23 ºF below its boiling temperature as seen in Figure 4.

The issue becomes one of liquid, subcooled to a temperature less than the temperature at saturation. Bottom-fed evaporators can normally handle a degree or two deviation from saturation, particularly at high initial temperature differences, example 10 ºF (5.6 ºC). This is a common metric used for selecting air-to-refrigerant evaporators in U.S. industrial markets. However, top-fed evaporators are not so forgiving of recirculated return line pressure drop. I know this first-hand, having made this design error many years ago.

### *3.5 Recirculated Top-Feed vs Bottom Feed – So, Which Is Better?*

Both methods have their own set of advantages and disadvantages. Top-feed is normally employed for high temperature evaporators (no hot gas defrost sequence) with a 10 ºF TD or greater. US evaporator manufacturers usually recommend a higher overfeed ratio for top-feed, normally ~4:1 in lieu of the 3:1 ratio for bottom-feed. My recommendation for top-feed would also include an oil drain connection because dragging oil uphill in a riser containing a mixed-phase of ammonia at low temperature, using vapor velocity is problematic and likely fruitless. Remember that the oils typically used for ammonia are very viscous at low temperature and don't flow easily.

If you'll look carefully at a top-fed evaporator undergoing a brining condition, it will manifest itself by the presence of little white stripes of frost running along the bottoms of tubing thus affected (Figure 1, right-hand side). I've estimated the heat transfer film effectiveness of a subcooled liquid stream during laminar flow to be roughly 10% of a boiling film and that may even be optimistic. Frost occurring from a subcooled stream does not extend out away from tubing and onto finned surfaces until the ammonia stream reaches its saturation temperature. Then when one factors in the small internal tubing surface area ($A_{frost}$, Figure1) in actual contact with liquid, it becomes apparent that tubes thus affected fall far short of their design capacity when liquid isn't boiling inside. Another factor to recall are again the fluid Reynolds Numbers. Even at overfeed ratio = 3:1, liquid flowing in a tube is still in a laminar zone ($N_{Re}$<2000); therefore heat transfer is a small fraction of that associated with a boiling film and $N_{Re}$>3000. Now you see why evaporator manufacturers state that gross refrigeration capacities (fan heat excluded) shown on their submittal drawings are based upon saturated liquid entering each refrigerant circuit.

### *2.1 Computer Software Limitation*

I'd like to offer a few words regarding computer software used by many manufacturers for determining their heat transfer capacities. Evaporator performance software, whether it applies to the residential, commercial or industrial refrigeration market sectors, assumes the following regarding the state point of liquid fed to an evaporator:

$$T_{ref,supply} > T_{sat,evap} \qquad (9)$$

When considering a mechanically pumped liquid overfed evaporator, one would logically (but erroneously) assume these two temperatures to be equal, therefore $T_{ref,supply}$ would be at saturation. However, the software I'm familiar with cannot accept a zero value for quality ('x' – a dimensionless value between 0 and 1) because a divide by zero computational fatal error would occur. To compensate, the user must either input a value slightly greater than $T_{sat,evap}$, say 0.001 degrees, or the software does this internally by default. Either way, the user "assumes" liquid *is* at saturation (on the 'bubble') when in fact this is theoretically not true when examining the delivery of pumped liquid to an evaporator as adiabatic. Oftentimes liquid line insulation systems approach this value, especially when the liquid line set is relatively short. If the ambient temperature is low (winter operation, Northern climates), this may be entirely true.

## 4 Conclusions

As long as recommended diameters for recirculated return piping are followed (IIAR, IRC), and the piping contains no places where liquid can become trapped and back pressure regulators are avoided, issues associated with subcooled liquid and brining normally do not arise. But when long risers are involved, my recommendation is to think about using gravity recirculated evaporators instead.

Solutions (there are several) to liquid stacking in long risers can become complex. Much depends upon how the manufacturer circuited the evaporator and whether it is top-fed or bottom-fed. But suffice it to say, getting liquid out of suction risers is **highly** desirable. One such way would be to install a pumped receiver set at the base of the riser and pump the liquid back up a new liquid return riser. Couple this with a dual step liquid feed arrangement previously discussed. Others solutions exist but they are a wee bit more complex and will not be discussed here.

Using a gravity recirculated evaporator avoids the riser issue altogether because liquid is not returned up the riser, only vapor and a vapor static head of ammonia is negligible.

## References

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EVAPCOND, a computer-aided evaporator analysis tool, NIST, National Institute of Standards and Technology

## Symbols& Abbreviations, List of

| | |
|---|---|
| A | area, $ft^2$ |
| $V$ | velocity, feet per minute (fpm) |
| TD | temperature difference |
| T | temperature, ºF |
| m | mass, lb (dot over signifies rate) |
| Re | Reynolds Number |
| $Q$ | Heat energy, Btu (dot over signifies rate) |
| $U$ | overall heat transfer coefficient, Btu/$ft^2$-ºF |
| $J_v$ | superficial (without the presence of liquid) vapor velocity, feet per minute (fpm) |
| g | gravitational constant, 32.2 ft-$sec^2$ |
| $\rho_l$ | density of liquid, $lb_m/ft^3$ |
| $\rho_v$ | density of vapor, $lb_m/ft^3$ |
| σ | surface tension, $lb_f/ft$ |
| psi | $lb_f/in^2$ |

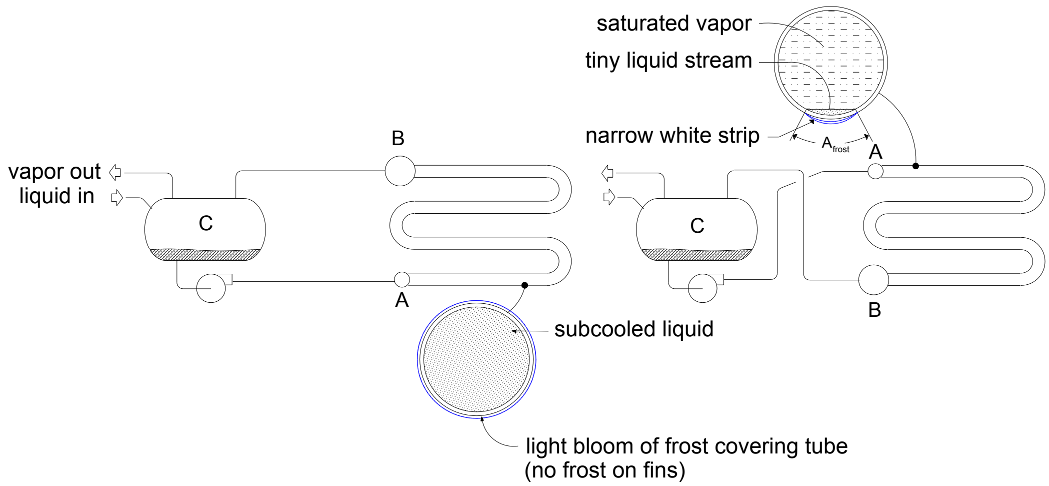


**Figure 1** The pressure difference between C and A, including button orifices, is met by the pump.

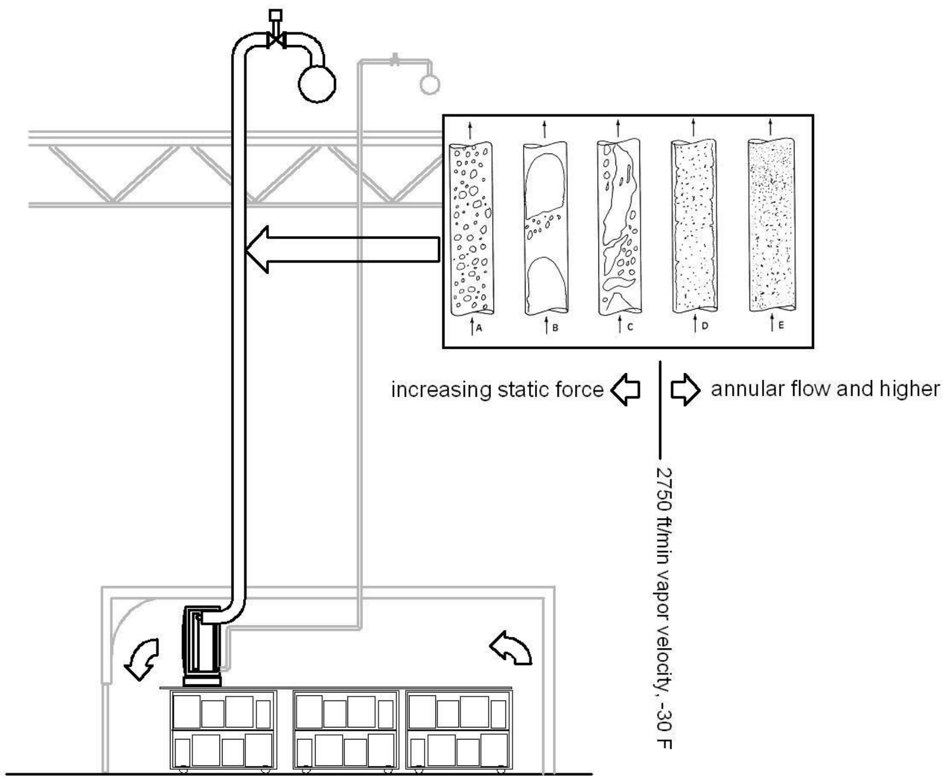


**Figure 2**

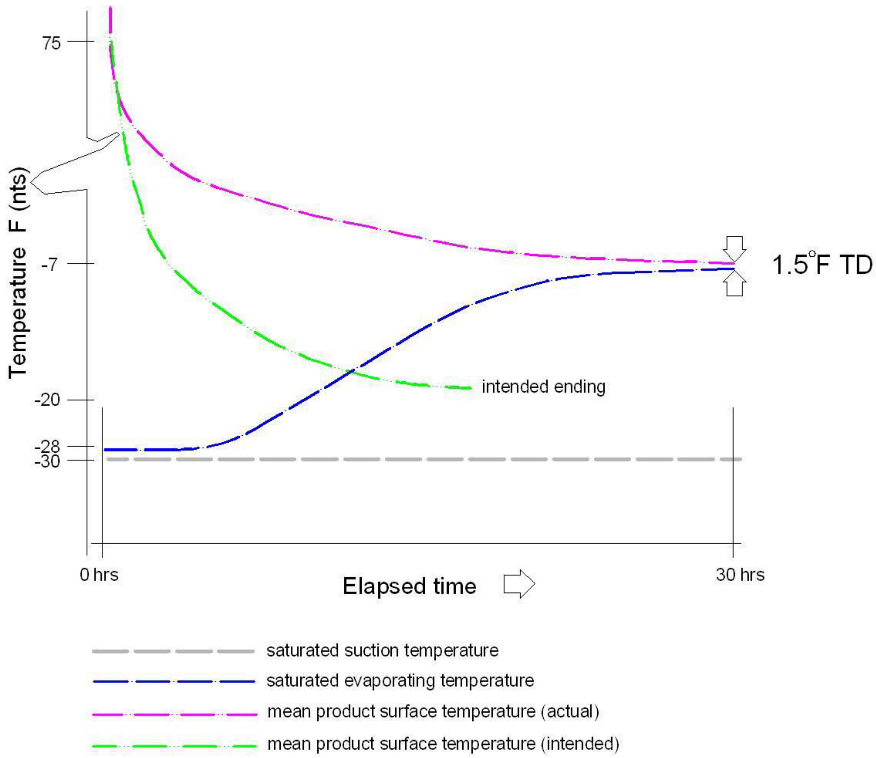


**Figure 3**

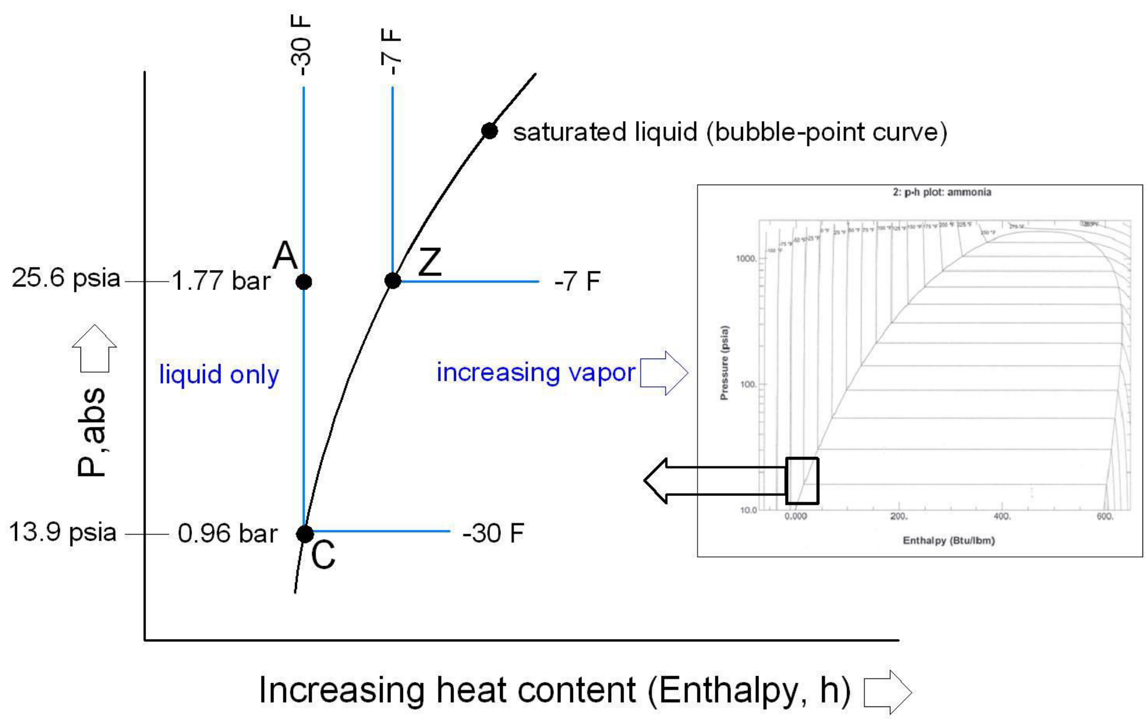


**Figure 4**