Technical Papers

32nd Annual Meeting

International Institute of Ammonia Refrigeration

March 14–17, 2010

2010 Industrial Refrigeration Conference & Exhibition
Manchester Grand Hyatt
San Diego, California

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ACKNOWLEDGEMENT

The success of the 32nd Annual Meeting of the International Institute of Ammonia Refrigeration is due to the quality of the technical papers in this volume and the labor of its authors. IIAR expresses its deep appreciation to the authors, reviewers and editors for their contributions to the ammonia refrigeration industry.

Board of Directors, International Institute of Ammonia Refrigeration

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Technical Paper #6

The Vertical Pipe Sizing Program (VPS 2010): When "Less is More"

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Abstract

In the world of ammonia refrigeration, it can sometimes seem that our intuitive sense of how things should work differs from the realities of the internal workings of the ammonia universe. One such example is when dealing with system performance and flow restrictions. As mechanical designer(s) we intuitively think bigger is better or that reduced flow restriction should allow improved performance. As some of our industry's experts have tried to help us understand, this is not always the case. When the real world verifies their counter-intuitive theories it can be a meaningful learning experience for everyone involved.

This was the case in a recent low temperature spiral freezer application with a flooded evaporator and surge drum accumulator. According to today's standard safety practices, the surge drum was mounted on the roof with a significant vertical lift required in the suction return line. This same vertical difference provided what proved to be an excessive down force on the liquid supply leg. The application of theoretical liquid return line sizing and liquid feed rate control per Mr. David Ross's spreadsheet program proved the truth that sometimes "Less really is More."

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Introduction

Nothing is more intriguing to the engineer or scientist as something which we believe should be quantifiable but defies our best efforts anyway. The fundamentals of system design can seem rudimentary but once in a while we find ourselves as designers dealing with phenomena that are difficult to explain.

The challenge of designing for two-phase flow in vertical liquid risers has been with us for a long time. Our inability to completely understand the complexities involved has been chipped away little by little over decades by numerous researchers but we are not there yet. This paper is one more step forward in producing successful results that our industry can reliably produce and easily use. It is a fact that the forces which dictate system behavior in this realm seem counter-intuitive to what we know about single-phase flow.

Our industry is participating in a research project on this very subject with the Danish Research Institute and this paper is not in any way intended to diminish or replace that effort. On the contrary, we whole-heartedly encourage it. The answer, as we understand it, is that the Danish Research Institute will produce mathematical model(s) that conclusively predict what is happening on the molecular level with the interactions involving liquid, gas, velocities, shear forces, gravity and buoyancy (forgive us if we left something out). Our intent is to simply offer a method that can define two-phase flow well enough to be useful until something better is learned.

A mechanical designer intuitively reasons that larger pipes allow greater flow and less pressure drop for a given flow rate. While this is typically true in single-phase flow and horizontal two-phase flow, if we try to apply this concept to a refrigeration evaporator circuit without understanding the impact of two-phase flow within a suction riser, our reasoning can lead us to the wrong conclusion. In other words, when faced with an installation that is not performing up to capacity, our first instinct may be to reduce pressure drop with larger piping or open up the hand expansion valves.

Within the last ten years, there has been significant progress from work by the members of the IIAR and ammonia community at large to deal with this topic. What had in years past proven to be a challenging problem has now become a manageable design task thanks to computers, and several creative and collaborative members of the IIAR.

Our industry has found through the application of mathematics, chemistry, physics, and engineering, as well as exhaustive empirical data, that contrary to many peoples' beliefs, the use of smaller risers and accurately calculated liquid flow makes for more effective two-phase riser installations. This appears to be a case where "less is more."

This paper provides the basis of the equations used to develop the VPS program that has been used to assist refrigeration engineers determine proper riser sizes and liquid feed when designing two-phase installations. Two well-documented case studies are provided that demonstrate the successful use of the program and have added to the improvement of the VPS program. These case studies are intended to show that even though the thought process and the equations developed can be confusing and counter-intuitive; they appear to be successful in producing better-working systems and provide further incentive for the continued research and documentation of two-phase refrigerant flow, especially in vertical risers.

One of the most comprehensive and detailed studies to date on the topic of two-phase flow was written by Mr. Bent Weincke in the IIAR Technical Paper #11 presented during the 22nd annual IIAR meeting.[1] In this paper, the author evaluated most, if not all of the information to date, comparing and contrasting a multitude of highly technical studies on the topic. Then, through a process of comparative analysis and review of the results of the various theoretical models, the author formulated a defensible approach to modeling the key components of two-phase flow.

While Mr. Weincke's Technical Paper 11 provided an incredible technical summation and insight into the topic, it required a high level of technical proficiency on the part of the designer to apply its findings.

Faced with the design challenge of a two-phase riser application, most designers will default to their experience or recommendations provided by manufacturers to size liquid feeds and risers. What the designer may overlook is that the manufacturers' design ratings and standards often come with caveats and limitations which exclude their application to the specific design conditions.

After dealing with these challenges for many years, in 2004, Mr. David Ross built upon Mr. Weinke's work by undertaking the challenge to develop a user-friendly tool to make this highly technical analysis accessible for everyone with a computer. The result was the Vertical Pipe Sizing (VPS) program which was presented at the 2005 IIAR conference.[2] The key to the VPS program was the use of the Kutateladze constant which at values between 3.05 and 3.2, appear to define the beginning of annular velocities. The formula takes into account the refrigerant properties and the system load to establish critical velocity limits for pipe sizing. The program then determines the pipe size that will sustain the proper velocity. The program does more than that, but this is its purpose in a nutshell.

Since the program's initial release it has been updated and refined, in no small part by input from some of IIAR's key contributing members. More importantly, the tool has been put to the ultimate test of multiple real applications where it has proven itself to be effective in providing successful solutions to tough design challenges.

This paper will present several case studies of systems where designers have applied this relatively accessible tool to design vertical two-phase risers capable of delivering stable, predictable and cost effective refrigeration solutions for both flooded and recirculated systems. It will also discuss further improvements and insights which have been incorporated into the VPS program since its first release in 2005.

The Problem, the Approach and the Correction

First, let's define the six riser flow patterns, both graphically and descriptively, per Mr. Bent Weinke's aforementioned Technical Paper #11: (figure #1)

a) Bubble Flow: There is a dispersion of bubbles in a continuum fluid.

b) Plug Flow: The concentration of bubbles becomes so high that bubble coalescence occurs, and progressively the bubble diameter approaches that of the pipe.

c) Churn Flow: Increased velocity causes a breakdown of the large plug-flow bubbles and leads to an unstable regime. In larger pipes, this causes an oscillatory motion of the liquid upwards and downwards.

d) Wispy-Annular: Liquid flows on the wall of the pipe as a film and the gas phase flows in the center. Some of the liquid phase is entrained in the gas core and as the liquid flow rate increases, droplet coalescence occurs leading to larger lumps or wisps of liquid in the gas core.

e) Annular: Liquid flows on the wall of the pipe as a film and the gas phase flows in the center. Usually some of the liquid phase is entrained as small droplets in the gas core.

f) Spray: The liquid film at the wall disintegrates and all liquid flows as small droplets dispersed in the gas.

Although we have defined six different flow regimes, they are in fact a single, changing continuum that progresses from one type of flow to another over a range of two-phase flow that begins at 0.0 fps and ends at "dry" gas defined as gas with liquid droplets at or below a Weber number of 6 [or diameters of 0.11in (2.8mm or less)].

Herein lies the problem to date: As these six definable two-phase flow patterns change from one type of flow regime to the next, it appears that we need different analytical tools for these changes and it may be that we need several different approaches, not just one. Our science has not yet yielded those tools although the refrigeration industry experts as well as academics around the world, are working on it.

VPS Evolution

After observing installations of Kutateladze (Ku) designed riser applications and receiving anecdotal success stories from contractors and consulting engineers, it was apparent that the Kutateladze constant was as good if not better than anything the refrigeration industry currently had for designing two-phase risers.

In addition to the anecdotal evidence, the formula presented at the 2005 conference had accurately predicted known points of empirical measurements, most notably those of Mr. Bill Richards in addition to those of Mr. David Ross.

In discussing the differences of threshold velocities with Mr. Todd Jekel of the Industrial Refrigeration Consortium in Madison Wisconsin, it was discovered that the formula had an error. After re-configuring the equation, the threshold velocities were lowered by approximately 10 fps at –40°F and +20°F but continued to have the same slope generated by the first VPS version.

The corrected Ku Formula as it is used in the program is:

$$\frac{\text{Ku}(3.2)\ {}^{*}g}{(\rho_g{}^{\wedge\,0.5})\ \sigma_g{}^{\wedge 2}(\rho_g - \rho_f)^{\wedge 0.25}} = U_g \text{ minimum riser velocity}$$

Given that loads often vary according to different stages of product and space cooling, this error was not noticed in operations and, according to one colleague with extensive experience with the software, "may have inadvertently accounted for water and oil accumulation effects." To this day, this colleague prefers the higher velocities of the first VPS version. In fact, the original higher velocities performed very well in real-life installations.

From this information and documented experience, it is reasonable to conclude that while the corrected design velocity has been lowered per the revised formula, there may in fact be little penalty for higher velocities up to some yet-to-be defined upper limit since the original flow velocities per the 2005 Technical Paper have been demonstrated to be within the annular range. More field-testing on higher velocities is thus warranted. The goal of the 2005 Technical Paper was to identify the beginning of annular flow. The upper limit of efficient annular and/or spray flow was not explored.

One of the points made in favor of the 2005 VPS program was that it correlated with the results of empirical data recorded by Bill Richards. Using the table from Richard's research as illustrated in Professor Stoecker's ***Industrial Refrigeration*** book, all twenty-eight (28) points in Richard's table have been checked against the VPS-2009b program.[3] This compares with the 2005 VPS program that was checked against six points.

The table now documents twenty-five (25) points that affirm the pipe sizing with the Richard's table. Of the three points left, the differences are all very close to the performance of each other and the table shows the velocity net differences in the shaded data. In all three of the cases, the VPS program called for 1.25" pipe which is not supported by the program, thus no psid information is available. (Figure 2)

The numbers in the first set by Bill Richards are his recommendations as to the ideal capacity for each of the riser sizes and recirculating rates. Those capacities were entered into the VPS program which generated the performances shown in the second set of numbers. Differences in selected pipe sizes are shaded and show the net performance differences between Bill Richards's pipe sizes and the VPS pipe sizes. In some of the larger capacities, there were two possible size selections but the table data reports only the pressure drop of the Richard's listed pipe sizes (5" vs 5," etc.).

Case Study #1:

This example provides a look at a 2009 project in which the application of the theoretical model was used to correct system deficiencies. This case is particularly interesting when we see the before, intermediate and after conditions because they demonstrate the benefits of both the riser sizing and liquid feed control capabilities of the program.

This application was a spiral freezer designed to freeze individual pieces of fried product. The freezer's performance objective was to freeze 5,500 pounds per hour of product from 65°F to a temperature of 10°F. The total load for product and freezer mechanisms was calculated and empirically verified to be 82 TR. The freezer evaporator design parameters were as follows: (figure #3)

Evaporator Rating: 100 TR with –40°F (saturated suction)

Evaporator Temperature Difference (TD): 13°F

Theoretical Leaving Air Temperature (LAT) –22 °F

Evaporator design: bottom fed, flooded

Evaporator Connections: two 6" suctions, two 4" liquid feeds

Elevation difference of wet suction outlet and surge drum: 18 ft.

The refrigeration system was a critically charged system with a flash cooler to receive liquid and cool it via the compressor economizer to 0°F. The liquid is then fed to a horizontal flooded accumulator rated at 125 TR.

In order to size the liquid feed and wet suction connections to the evaporator, the designer referenced standard flooded coil design recommendations per table IIAR 1–16.

The chart indicated that for the connected coil two 5" suction risers and two 4" liquid down legs were appropriate.

Upon starting up the spiral freezer and refrigeration system, it became apparent that the system had three operational problems. First and foremost, rather than meeting the 5,500 pound per hour production goal, the freezer could only handle 3,500 pounds per hour at a marginal 15°F to 16°F product temperature. Secondly, the liquid levels in the flooded accumulator surged wildly between 20% and 60%. Thirdly, the liquid level in the flash cooler was swinging so widely that ammonia was added and removed from the system several times to find the correct charge.

The compressors were maintaining a temperature of –40°F saturated suction in the engine room. A check of the pressure differences between the compressor suction and accumulator conditions indicated that the piping loss from engine room to the flooded accumulator was negligible.

Most interesting was the fact that the temperature probe in the center of the freezer indicated temperatures between –10° and –13° F. This indicated that at best, the air conditions in the freezer were at least 12°–14°F higher than expected.

After evaluating the conditions and data, a hypothesis was developed to explain the issues. First, the erratic operation of horizontal accumulator was attributed to intermittent slugs of vapor and liquid returning to the vessel in a slug or plugged flow fashion. This appeared to be propelling varying amounts of liquid up the riser and causing significant turbulence in the vessel.

Second, the inconsistent rate of refrigerant flow out of the evaporator was attributed to trapping liquid at the coil and risers during low load conditions. During defrost or under heavy production loads it was thought that this liquid was forced back to the flash cooler which was causing the flash cooler to swing under and over charged conditions.

Lastly, there appeared to be a severe temperature penalty at the evaporator resulting from the stacked up liquid in the wet suction riser. The apparent temperature difference appeared to be approximately equal to or slightly less than the calculated static head penalty for the expected liquid vapor mixture. This temperature penalty appeared to account for the significantly higher than predicted air temperatures in the freezer and (most important to the customer) reduced production capacity.

As a side note, during attempts to optimize the freezer operation, the service technician was tasked with balancing the liquid flow from the accumulator to the evaporator. Consistent with the intuitive reasoning for managing single phase flow as noted in the introduction of the paper, the technician chose to fully open the liquid metering valve.

At this point in the project, the decision was made to apply the VPS program to diagnose the problem and identify possible solutions.

The vertical elevation difference between the wet suction outlet on the evaporator and the flooded accumulator was approximately 18 feet. Additionally as calculated by the program, the total equivalent length of pipe from the evaporator to the vessel was in excess of 100 feet. Given the design conditions, the program recommended installing two 3" wet suction risers and a single 4" liquid down leg.

The two existing 5" risers were replaced with two 3" risers as prescribed by the program. Since the two 4" liquid down legs had globe valves, one of the two was isolated and the recommendation was made to throttle the second globe valve.

The liquid header at the coil was 4" diameter pipe and the coil circuits had no internal restrictors. The liquid supply piping included 4" globe valves (one in each leg), one of which was closed and locked out permanently.

After this change, the system performance had improved significantly, but it was still not meeting performance expectations. The freezer production capacity had increased from 3500 pounds per hour to 4500 pounds per hour with product temperatures averaging 13°F. The average freezer temperature had dropped from –13°F to approximately –19°F. The violent turbulence in the flooded accumulator had ceased and the operating level had stabilized. The liquid levels in the flash cooler however were still fluctuating widely depending on production loading.

It should be noted that after the liquid risers had been reduced, the field technician had relied on his experience and intuition to balance the coil flow to what he perceived to be optimal flow. Again the globe valve on the single 4" liquid feed was left at 100% open.

After a great deal of discussion, the technician was convinced go against both his experience and intuition and he throttled the liquid feed valve to match the Cv on the liquid feed as prescribed by the VPS program. The spiral freezer temperature dropped an additional 2°F which resulted in a return air condition of –24°F. The production capacity increased to 5,000 pounds per hour with average product temperatures well below the 10°F target.

At this point, the customer's equipment was not able to produce the planned maximum production of 5,500 pounds per hour except for very brief periods of time.

Since the results at 5,000 per hour were well below the 10°F target, no further testing was done.

Again, the value of this example is that it illustrates that it was a combination of downsizing the liquid riser as well as throttling the liquid feed rate which delivered the optimum results.

The throttling was done at the available maximum continuous load of 5,000 pounds per hour. The flash cooler liquid level stabilized and the critical charge was adjusted and liquid level at the flash cooler became constant.

Case Study #2:

The second example is a side by side comparison of two –40°F crust freezers used for chilling of packaged meat loaves. In this case, the freezers had dual 30 TR evaporators designed to provide 60 TR at –40°F saturated suction. The evaporators in both freezers were fed with pumped liquid at a 4:1 liquid overfeed rate. The vertical rise from the evaporator suction to the wet suction header on the roof was approximately 26 feet. (Figures 4 and 5)

The first of the two crust freezers was installed with a 5" wet suction riser on each 30 TR evaporator, based on traditional sizing methods for horizontal suction piping at the insistence of the freezer manufacturer. The second freezer was installed with a 3" wet suction riser on each 30 TR evaporator per the VPS program. Both freezers had 1 ¼" liquid feed lines. On the first system, automatic fixed flow regulators were installed sized per the overfeed requirement but on the second system a ¾" hand expansion valve was added to the liquid feed system, in place of the automatic fixed flow regulators, sized per the VPS program. The ¾" hand expansion valves were opened 2.5 turns as also recommended by the program.

Temperature sensors were installed at the top of the wet suction riser for both freezers. An additional temperature sensor was installed at the suction connection to the new freezer riser which had been sized using the VPS program. Operational constraints prevented installation of a sensor at the suction of the 5" connected evaporator.

From the customer's perspective the most significant difference in the two freezers was that the freezer with the 3" riser started flawlessly and provided full production capacity. In contrast the freezer with the 5" riser was only capable of delivering 80% design capacity.

A comparison of the temperatures at the top of the two wet suction risers also indicated a performance difference. The temperature at the top of the 5" wet suction riser measured –24°F while at the top of the 3" wet suction the temperature a full 8 degrees colder at –33°F. Also noteworthy was the fact that there was not a measurable temperature difference on the 3" riser from the evaporator connection all the way up to the top of the riser indicating no significant loss in the 26' vertical riser.

Theory

On January 6, 2006 a method of calculating churn and semi-annular (wispy) flow was instigated. It was proposed that the theoretical model might behave something like a virtual pipe inside a pipe, with liquid forming the new, smaller "pipe" diameter and two-phase gas/liquid traveling up the center of the new "pipe." The 'virtual pipe-in-a-pipe' theory becomes applicable when riser gas velocities drop below calculated annular velocities due to a drop in evaporator heat load.

By prior definition, churn and semi-annular flow have a consistent but penalized gas flow and these two flow regimes define the limits to which this theory applies.

The key to estimating pressure drop within this combined flow regime requires a point at or very close to where the transition to annular flow behavior begins. With the calculated Kutateladze annular threshold velocities and VPS available, it became possible to calculate annular threshold velocities for any temperature and load in a given pipe size.

With a drop in load that results in a drop below annular velocity, liquid accumulates in the riser where it eventually reaches a new equilibrium and a new, fully entrained flow occurs in a smaller cross-sectional area of the riser. At this new annular Ku diameter, the riser is again working with an entrained flow in the center of the riser but at an increased pressure loss because of friction, sheer and other forces due to the accumulated liquid. The key to this calculation is the ability to estimate the virtual annular Ku diameter based on the temperature of the refrigerant and the mass flow reflecting the evaporator load.

This method does not delve into the second-by-second interactions of surface tension, sheer forces or effects from viscosity. The attempts to arrive at a usable formula that can predict fluid behaviors in these flow regimes are well-documented and to date have failed. Therein lays the purpose of the Danish Research Institute project, which will hopefully connect the rebellious dots.

At first, it seemed unlikely that the concept would have an analogous reality in real pipe but evidence that this was correct was found in an internet search by a colleague who discovered a paper entitled, *Vertical Two-Phase Flow, Part I: Flow Regimes* by Spedding, Woods, Raghunathan and Watterson (Department of Aeronautical Engineering, The Queen's University of Belfast).[4]

From their observations of churn flow progression they state, "With increasing gas rate, the Taylor bubbles broke through the enclosing liquid regions to form a gas passage in the pipe centre. The liquid near the wall was characterized by an oscillating instability in which liquid tended to first fall down against the net

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upward flow and then be carried upward again. Liquid bridging across the centre gas core was absent." This description, while similar to annular flow, has "oscillating instability" which is the result of insufficient gas flow (energy).

It has been proposed that the real question becomes, "when are the Taylor Bubbles so plentiful that they become one continuous bubble turning into annular flow?" At this point in time, the best answer we know of is that it occurs at a Ku number between 3.05 and 3.2 per the 2005 Technical submission. This may change when the Danish Research project completes its testing. But for now, based on experience from using the VPS program, we are very close to the final answer.

The next flow regime is Semi-Annular: "The central gas core through the liquid on the wall was more defined with no oscillatory up and down liquid movement. However intermittently, liquid bridges were formed across the gas core and were then broken. Semi-annular flow formed between the churn and annular regimes."

Next, Annular Flow: "The liquid continuum was mainly confined to the pipe walls with a clear gas core in the pipe centre. The liquid surface could exhibit roll waves or ripple waves as the gas rate increased. Liquid droplet entrainment in the gas core increased significantly with gas rate as annular plus droplet flow."

They then explain: "The transitions between each of the five main regimes did not occur suddenly but rather the flow pattern progressed from one to another through an intermediate regime that possessed mixed characteristics."

To scientists, this is a bit messy as the interactions of gas and liquid flowing up in a pipe are demonstrably complex. It is not known empirically whether the "liquid pipe" has a higher or lower wall friction than steel pipe. For all we know right now, the "liquid pipe" may at times exhibit both higher and lower friction loss compared to steel pipe. Based on Case Study #2, it is very likely close to steel as we do not see a significant difference beyond the pressure loss of a static column with mixed densities as will be explained later in this paper.

The observations presented in this portion of the technical paper are the best information we have found in published studies and hopefully of use in carrying this research further towards a definitive understanding.

Calculating Churn and Semi-Annular Flow Pressure Drop with the VPS Program

In January 2006, it was proposed to an industry colleague (with humor) that given a long enough observation period, even slug flow can be viewed as a relatively smooth event. For our purpose, slug flow can and should be absolutely avoided by designing the riser to the annular velocity. If you think there is going to be a large operating range for the application, you may want to go to a 2/3 – 1/3 dual riser and use the VPS program to design it.

While at this time, we cannot say with certainty where the velocity transition is from slug to churn flow, we can identify at what velocity "flooding" begins. Bent Weincke's 2000 Technical Paper says, "The Weber number is the most referenced dimensionless number used to characterize the behavior of droplets. The number that characterizes droplet breakage or scatter is often referred to as the critical Weber number. According to Grassmann (1982) and Prandtl et al. (1990), a free-falling droplet breaks or scatters at Weber numbers of approximately 12, the most common referenced value in the literature. However, depending on the application, critical Weber numbers in adiabatic flow can vary from 5 to 25. Therefore, it is essential to identify a critical Weber number originating from test series or experiments that relates to this problem statement."

"The experimental studies of gas-liquid interface behavior in vertical tubes by Wallis and Kuo (1976) give the most insight into the behavior of liquid entrainment. These authors investigated the critical conditions necessary for the gas-liquid interface to become unstable or break up. The maximum value of *Ku* at which instability occurred was estimated to be 1.87 ($We = 8.6$). For the value at which flooding in a large-bore tube begins, they recommend a *Ku* of 1.8 ($We = 7.94$). According to Prandtl et al. (1990), a droplet sustains its spherical rigid shape at $We < 6$ and remains stable at $We \leq 8$. At $We > 8$, strong deformation of the droplet occurs, potentially leading to droplet breakup and scatter. A We of 8 appears to be the most widely accepted value at which droplet instability occurs with subsequent entrainment. Because droplet stability was the criterion for deriving Equations 35 and 36, choosing a Weber number of 8 seems justifiable for Equation 36. This leads to a *Ku* of 1.8. Therefore, the vapor velocity at which all liquid is entrained in the vapor flow is defined as (37).

How low can you safely operate a riser is a very good question. Using the conditions of Case Study 2, which has a design load of 30 TR, the riser begins two-phase flow (bubble) at 12 TR Ku(1.87). At a Ku(3.05) velocity it is at 19.5 TR and at a Ku(3.2), it is at 20 TR. In terms of velocity, flow begins at 32.72 ft/sec, reaches Ku(3.05) at 52.99 ft/sec, Ku(3.2) at 55.69 ft/sec and 30 TR at 81.36 ft/sec.

The effective operating range for the 3" selection is very good for this example but you cannot count on that. It's all about where your design load velocity is in relation to the useful area left in the chosen pipe size. It can be very good, fine or not so good. Whereas VPS 2005 did a single-point analysis, VPS 2010 evaluates 70 points for one set of conditions. You will be able to view how good of a fit a particular pipe is and where these Ku numbers are in relation to each other as well as the design condition.

It is risky to say that you may be able to count on some useful level of performance at a particular percentage of the Ku(1.87) to Ku(3.2) range. Tex Hildebrand, the co-author of this paper cautions that at low temperatures, you lose performance very quickly at about 80% of full load. It is somewhat expected and correlates with experience that at higher temperatures the drop-off happens around 50% of full load velocity.

The following concept is based on the assumption that there is a two-phase gas flow at the center of the riser and that flow is surrounded by liquid. That, by definition, is a property of Churn and Semi-annular (wispy) flow regimes. Based on the Case Study #2 result, we propose that the following approach to calculating pressure drop in these two flow regimes can be used.

"VPS Program-Specific Information and Results Verification"

Before going any further, it is important (and comforting) to realize that 98% of the following process is performed automatically by the VPS program.

In the process of selecting the correct riser pipe for full capacity performance, a standard pipe diameter is recommended by the program but must be selected by the user. This feature was provided to allow the designer to see other diameters' results in the event that the program's choice was on the edge or the designer had other criteria to consider.

During the initial data entry the height of the riser is also entered. This information, along with riser diameter, provide for the calculation of the static head pressure at the bottom of the riser, albeit with a composite density as will be shown.

For this example, Case Study #2 will be used demonstrate the accuracy of the proposed methodology for calculating riser penalties associated with operating a system in the churn/semi-annular flow regimes.

Open the VPS program and select Single Evaporator and Recirculated at the top of the main page. Select Data Entry and enter the following data: 30TR, –40°F, 4:1 overfeed, 26' liquid height, 26' riser height, 0.5 psi evaporator loss and 10 psid liquid feed pressure. On the Equivalent Length side of the page, enter 30' for both the Supply and Return lengths, (2) SR 90's for the Supply, (2) LR 90's for the Return, (1) globe and (1) angle for the supply and (1) angle for the Return. *This data will already be entered in the program for those who request a copy of the VPS program.*

Click on Enter and you will go to the "Line Sizing Worksheet" where you will see the 5" riser (Case Study #2) selected in the SELECT Riser Line Size box and a 3" riser recommended below it. The projected temperature and pressure penalties of the 5" selection are calculated and listed at 10°F and 3.49 psid at the bottom of that section.

Now enter 3" in the SELECT Riser Line Size box. The penalty is 1.65°F and 0.54 psid. Subtracting the 1.65°F penalty from the 10°F penalty = 8.35°F, which represents the theoretical net difference between the two risers.

We are focused on the temperature penalty because that is what was measured in Case Study #2. To conclude, the Contributor of Case Study #2 measured "a full 8°F colder" between the 5" and 3" risers as measured at the top of the riser. That is close enough for our purposes in refrigeration and we encourage the IIAR membership to include instrumentation in their projects to increase our knowledge and skill in providing the most efficient systems for our clients.

The simplest way to describe the "Virtual Pipe in a Pipe" theory is this: imagine a riser pipe of 4" diameter. Now place a 3" pipe inside it. This 3" pipe is made of liquid ammonia and is formed by the steel on one side and a relatively steady center core of two-phase gas/liquid in the center. The gas/liquid will form in the center because that location has the lowest resistance to flow.

Without knowing anything of the intricate and complex interactions between the gas and liquid or effects from gravity and buoyancy, we do have a pretty good picture of the volume and density of both the liquid and two-phase components in the pipe. By adding the weights of the liquid and two-phase components together and dividing by the cross-sectional area of the pipe, we get a psid 'snapshot' of a column with composite densities. Simplistic as this method is, it is apparently a reasonable approximation.

One might ask why this wasn't considered before. The answer is because there wasn't a starting point to work from. It required a fairly accurate estimate as to where annular velocities begin and end. Once that was found, it became possible to calculate either the velocity of entrained flow given a specific pipe size or the size of a pipe that will have entrained flow given specific load conditions. Lastly, it required validation from third-party sources that observers have witnessed center cores of two-phase flow in a lab.

Compelling Reasons for Utilizing the VPS Methods

Avoidable losses, which can and will occur with less than adequately designed mixed flow risers, can be prevented by applying the **VPS methods**.

To address more directly the overall nuances and effects of inadequate design, let's break down the applications to two distinct areas of concern which affect both component selection as well as operating capacities and efficiencies:

1) Excessive energy consumption will be experienced due to the need to operate at a lower saturated suction temperature at the evaporator to overcome the static head penalty. It should be noted that this only provides a partial solution. The external static head penalty can be reduced by lowering the Saturated Suction Temperature; however, the effective heat transfer within the evaporator may not be totally recoverable due to internal friction, alterations

of the scrubbing effect of localized velocities, flow distribution patterns, oil films coalescing at the lower temperature, etc.

2) Increased equipment capacity will be required to meet the operational conditions for both the booster compressors as well as the high stage compressors of the system. Operating at a lower saturated suction temperature increases the volumetric flow rate (CFM/TR) dramatically at the booster compressor, without improving capacity. This means larger capacity compressors as well as increased horsepower (compression ratios are significantly increased).

Let's look at the implications of:

A. New system designs

It behooves all system designers, application engineers, etc., to become thoroughly familiar with the application of, and utilize the VPS METHODS, on any system, whether flooded or overfed, where the two-phase flow riser(s) to the flooding drum or to the control bank incur a vertical lift of more than just a few inches (for low temperature applications) to a mere foot or two on high temperature applications.

By careful analyses, the system design can be optimized for best performance, lowest first cost of equipment (because it is properly sized), and at lowest power cost.

B. Previously installed systems that require remediation should be modified to obtain best performance.

If the evaporator(s) in question is (are) part of a system which has additional available capacity, it is possible to simply bite the bullet, lower the saturated suction temperature to overcome the loss, and continue ahead. However, the cost to eliminate the problems by applying the VPS METHODS compared to the cost of continued inefficient operation should be carefully considered.

For instance, considering a 30 ton evaporator designed to operate at –40°F Saturated Suction Temperature that suffers a 10°F lift penalty and is serviced by a typical two-stage system with a 25°F Intercooler/high stage suction operating at 95°F Saturated Condensing Temperature, would result in the following:

(Based on preliminary calculations):

Booster Compressor would have to be increased in Suction CFM by 34%.

Booster Compressor horsepower would then be increased by 28%.

Booster Compressor compression ratio may be increased by 35%.

High Stage Compressor would have to be increased in Suction CFM by 11%.

High Stage Compressor horsepower would then be increased by 3%.

The overall energy of the system is increased by 14%.

Correctly applied, VPS METHODS will yield a lean, mean, and green system!

From this analysis we should perhaps consider looking at temperature penalties to qualify good design, and not pressure as the first criterion. Our business is making things cold; not keeping minor pressure fluctuations to a minimum.

What we learn from this has the potential to improve system efficiencies and reduce first-costs to Owners, not to mention easier startups for contractors and manufacturers.

This paper follows the same path as the 2005 paper. This is not intended to be the end of what has been a very long discussion in our industry. Rather it is another page in our book that will move us along to more refined answers in the future. And we shouldn't avoid using tools like this; we still don't know how magnets work.

There is no great genius without a mixture of madness.

—Aristotle

General Recommendations

Most of the following recommendations were included in the 2005 Technical Paper but they are worth including again. These are a few lessons learned that might increase your success.

Installation

- Install the vertical piping with as few 90° angles and tees as possible. These can disrupt annular flow. Adding horizontal length penalizes you any way you look at it.
- If you must have automatic control valves in either line, use low-pressure drop valves such as gas-powered checks or sealed motor valves. Stay away from valves that require pressure drop to operate.
- Install the liquid line HXV at the evaporator inlet elevation if possible. Stray no further than you have to from that elevation as the HXV should "see" all of the available static liquid pressure in order to be set correctly.
- Use top quality thermometers, pressure transducers and gauges. Slightly inaccurate information is worthless information on these types of systems.
- Install a method to drain oil (and water!) at your evaporators' lowest piping location. An alternative to that is hot gas applied to the suction line and a condensate drain line with a defrost regulator (and check valve) tied into the liquid feed at the same location you would drain oil. Flooded evaporators naturally make great water stills.

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- Insulate and vapor barrier all piping with great care and attention to quality. Experience has indicated that any water formation next to the piping will tend to stay there and in most cases leads to rapid loss of wall thickness.

Design

Perform the product or space load calculations yourself to make sure you know what the true peak, normal, and minimal load is going to be. Do not allow unknown safety factors from other people to creep into your design.

- Base the riser design on the actual load and then add safety considerations into the design. Most of that safety will be put into the make-up liquid and liquid feed leg sizing anyway because your suction risers can stand higher velocities with little additional penalty, particularly when a relatively drier gas (defined as a higher gas-to-liquid ratio) is produced and the starting velocity point is at the flow-reversal threshold. In many cases our industry practices add "safety factors" which provide more capacity than our base calculations. With two-phase risers, adding more capacity may result in underperforming systems due to excess liquid in the riser. A clear case of when less is more.
- Velocity is the most important piping design criterion followed by pressure and temperature drop. Too much speed in the liquid feed line reduces your capacity safety factor and too little speed in the wet suction riser will allow flow to reverse. In the case of the riser, the program converts the pressure drop into a temperature penalty because it directly relates to the evaporator's operating point. The pressure/temperature relationship is non-linear; one-size-fits-all pressure drop rules don't work. Temperature penalties for high-temperature evaporators will be around 1°F/1 psi pressure drop, whereas low temperature evaporators may experience a 2°F/1 psi [1.6°C/decibar] loss.
- Obtain or specify accurate information on the evaporator design. If you have a long vertical distance or if there are a lot of horizontal detours to get to the coil, determine if you can work with the evaporator manufacturer to design to a lower overfeed rate. If you have less liquid to lift, you will have a broader capacity range

that the riser will work well over. If the evaporator is to be recirculated feed, ensure that the orifice selection is carfully engineered.

- When you design for a specific overfeed ratio, tell the manufacturer what it is when you place the order. Alternatively, you can ask the coil manufacturer to select the minimum overfeed ratio to ensure a fully wetted tube surface, but nothing more.

Nomenclature

A	area of pipe (ft^2)
C_v	flow coefficient
H_s, H_r	height of fluid columns (ft)
hfg	latent enthalpy change (Btu/lb)
L	equivalent length of pipes, fittings, valves & entrance losses (ft)
OF	total mass circulated per mass evaporated, expressed as ratio (OF:1)
Q	heat transfer rate of heat exchanger (Btu/hr)
X	vapor fraction of two-phase flow (dimensionless
ρ	density (lb/ft^3)
ΔP	pressure differential (lb/in^2)
σ	surface tension, (dyne/cm) & (ft-lb)
g	gravity constant, (32.174 ft/sec)
Ku	Kutateladze constant, dimensionless
u	velocity, (ft/sec)

Supply and return d diameter of piping (ft) f friction factor (dimensionless) can be derived without iteration by using the Churchill equation in a spreadsheet (Welch, 1999)

References

Bent Weinke. Technical Paper #11, "Two-Phase Flow Behavior in Pipes, Valves and Fittings," in *Technical Papers*, 22nd Annual Meeting, International Institute of Ammonia Refrigeration, March 19-22, 2000.

David Ross. Technical Paper #7, "Modern Evaporator Piping: Two-Phase Riser & Gravity-Feed System Design" in *Technical Papers*, 27th Annual Meeting, International Institute of Ammonia Refrigeration, March 13-16, 2005.

Wilbert Stoecker. *Industrial Refrigeration Handbook*, McGraw-Hill, 1998.

Spedding, Woods, Raghunathan and Watterson. *"Vertical Two-Phase Flow, Part I: Flow Regimes,"* Department of Aeronautical Engineering, The Queen's University of Belfast.

Bent Weinke. Technical Paper #3, "Sizing and Design of Gravity Liquid Separators in Industrial Refrigeration" in *Technical Papers*, 24th Annual Meeting, International Institute of Ammonia Refrigeration, 2002.

Figure 1: The problem and part of the answer

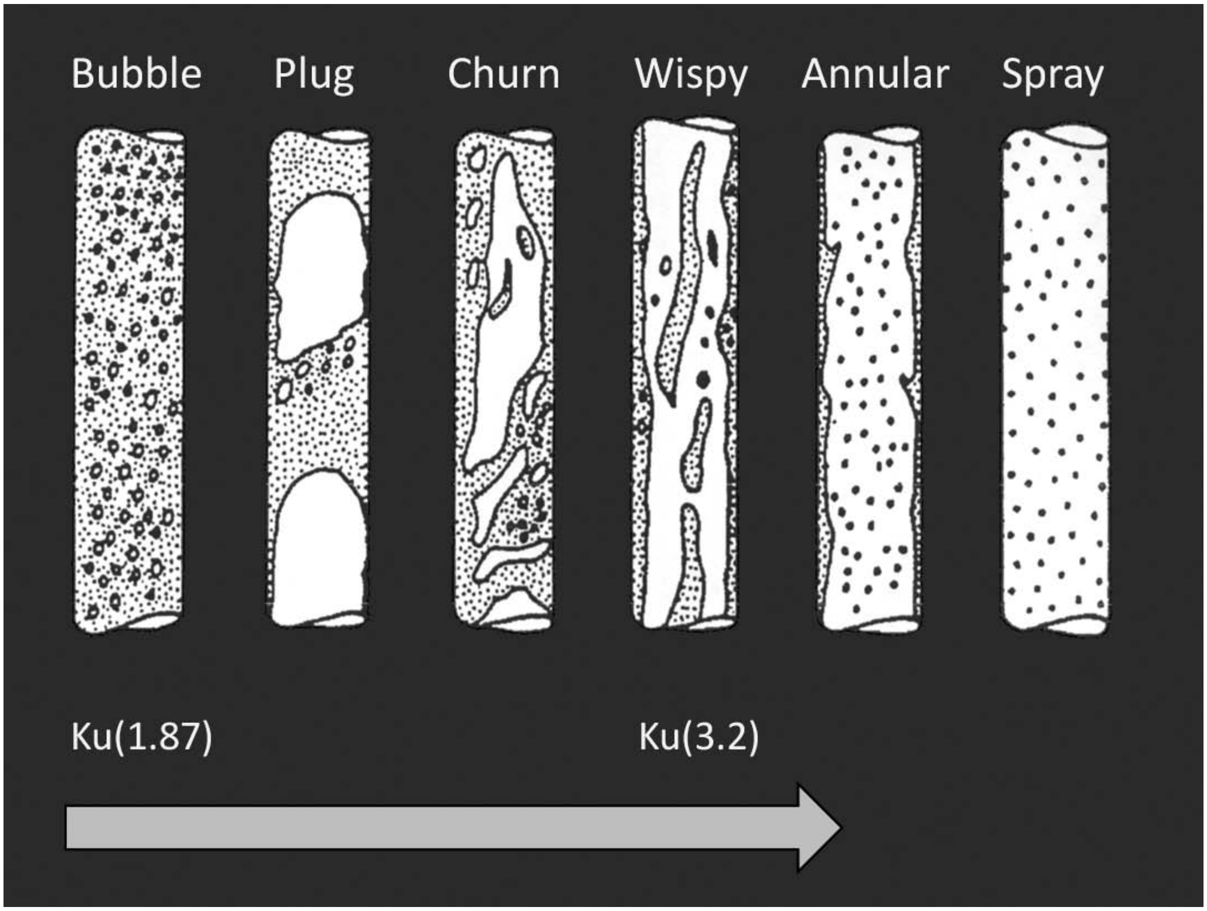

Figure 2: Bill Richards

-40F/C evaporator   Bill Richards / Stoecker "Industrial Refrigeration" (Book 1)

	Pipe Size						
recirc rate	1.5"	2.0"	2.5"	3"	4"	5"	6"
2.5	5.3	9.9	15.2	25.6	52.4	92.2	146.0
3.15	5.0	9.3	14.3	25.0	49.3	86.7	137.0
4	4.8	8.9	13.8	24.0	47.4	83.4	132.0
5	4.6	8.6	13.3	23.1	45.6	80.1	127.0

-40F/C evaporator   D. Ross VPS-2010

	Pipe Size						
recirc rate	1.5"	2.0"	2.5"	3"	4"	5"	6"
2.5	0.9 psid	0.18 psid	0.17 psid	0.17 psid	0.18 psid	0.19 psid	0.20 psid
3.15	0.29 psid	0.20 psid	0.19 psid	0.20 psid	0.20 psid	0.21 psid	0.22 psid
4	0.8 f/sec	2.1 f/sec	0.23 psid	0.23 psid	0.24 psid	0.24 psid	0.25 psid
5	3.1 f/sec	1.22 psid	0.26 psid	0.27 psid	0.27 psid	0.28 psid	0.29 psid

Results in psid represent pressure drop penalty and agreement with Richard's results.

Shaded data indicates VPS 1-1/4" selection which is not supported for riser duty.

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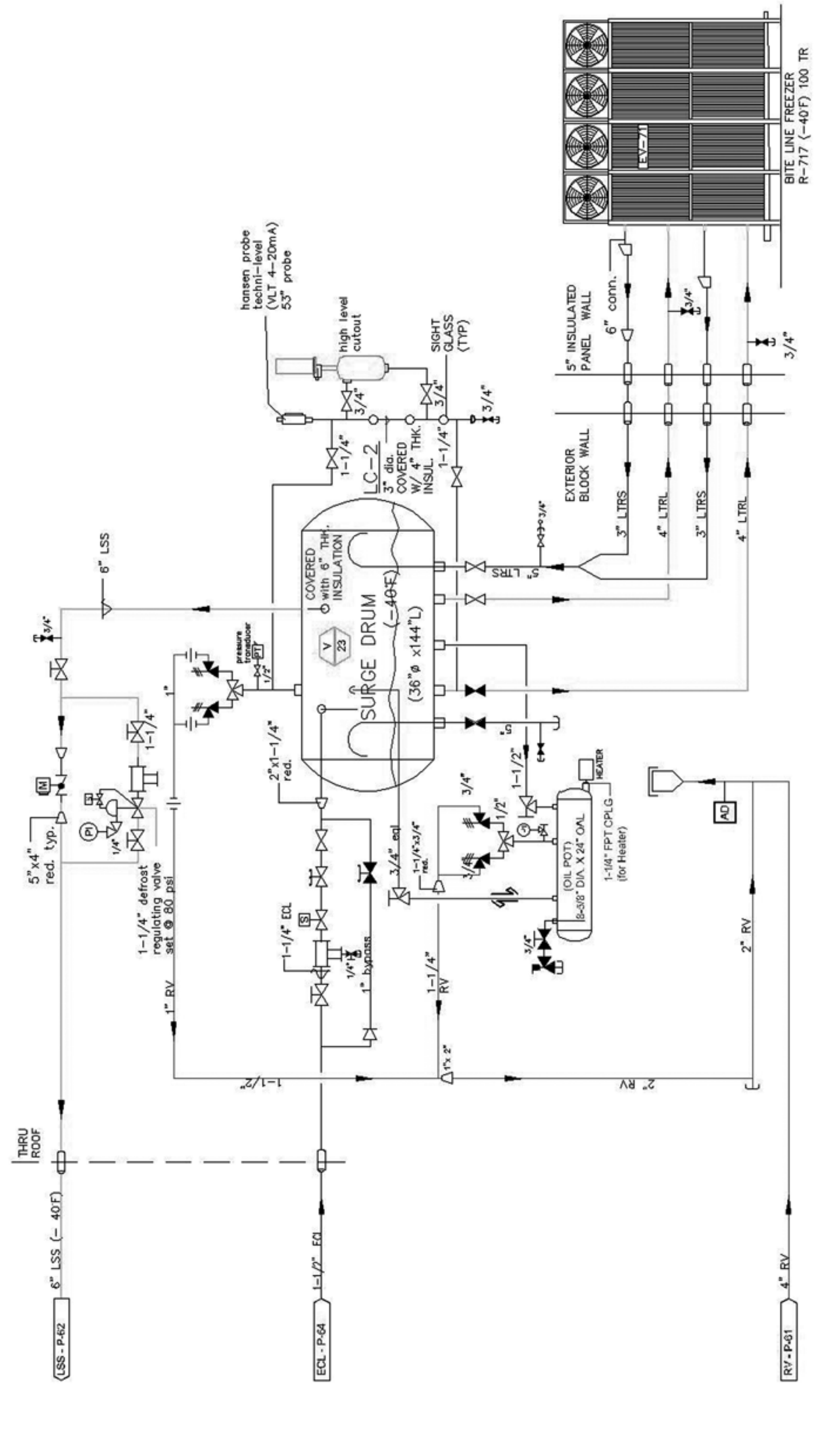
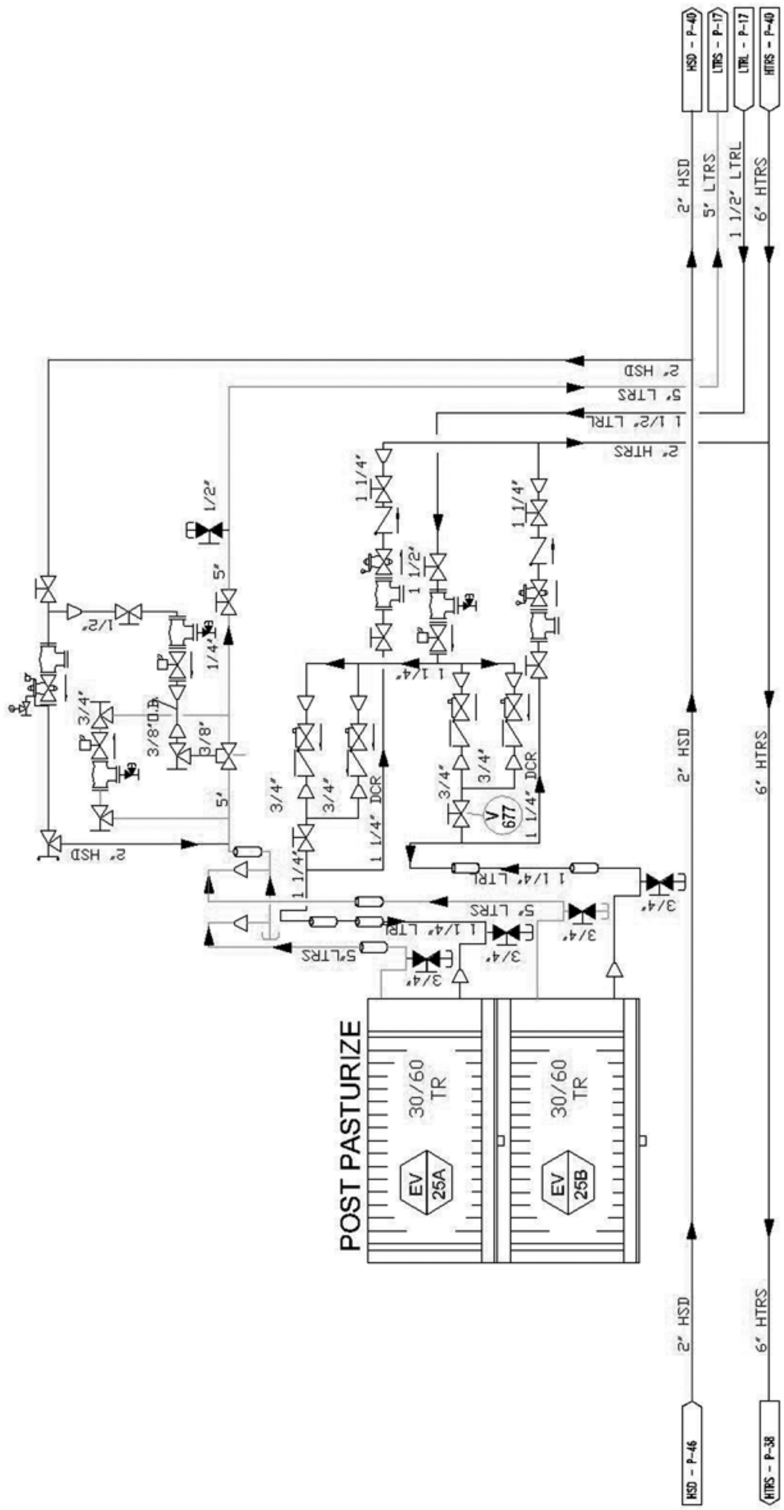


Figure 3: 103-P66-IIAR2-Model

Figure 4: 125-P43-IIAR3-Model



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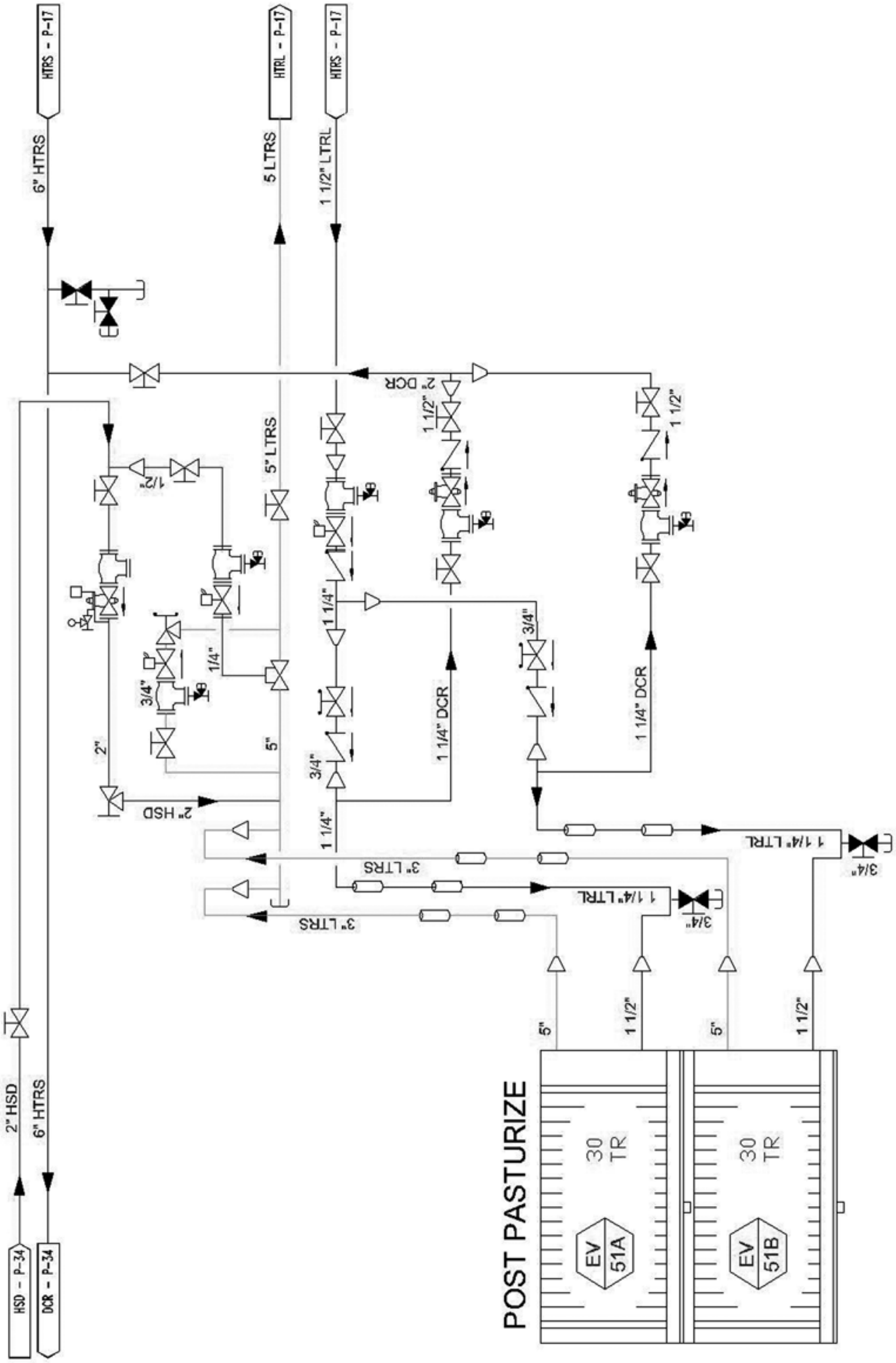

Figure 5: 125-P35-IIAR3-Model

Notes:

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