PRACTICAL PROCESS HYDRAULICS CONSIDERATIONS

by

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Introduction

A fluid moving through pipe or equipment is a well understood phenomena. There are some fluid flow areas that would benefit from improved understanding. Two-phase flow, flow through packed beds, and flow associated with phase changes are examples. But for the most part existing analytical methods predict behavior reasonably well. However, having a firmly rooted understand of what the fluid will do does not explain what affect this will have. Where hydraulic calculations meet process requirements can be thought of as Process Hydraulics. Application of the well defined fluid flow equations to achieve well defined process goals may follow a dimly lit path. Real-world illustrations highlight several ways you can get off-course.

Unusual Conditions

Often an equipment design basis does not account for abnormal conditions. Startup, shutdown, upset operation, feed changes, and maintenance requirements are cases where a compressor or pump may need to produce beyond its normal performance envelope.

Recycle and makeup hydrogen compressors for hydroprocessing reactor systems occasionally must operate within a varying range of hydrogen purity. Often an entirely different gas such as air or nitrogen must be accommodated for catalyst regeneration. When a compressor's hydrogen supply purity increases at a constant compression ratio the compressor discharge temperature increases due to a rise in the heat capacity ratio ($k = Cp/Cv$) of the gas being compressed. A reduction in hydrogen purity generally increases fluid horsepower requirements through an increase in molecular weight.

A Gulf-Coast refinery recently changed a hydrodesulfurizer hydrogen supply source from 70% purity reformer hydrogen, to imported 99% purity hydrogen. Due to the increase in heat capacity ratio, the reciprocating hydrogen makeup compressors experienced a significant rise in discharge temperature. The discharge temperature increased from 190 °F to 290 °F at the same pressure ratio. This resulted in immediate compressor valve reliability problems.

Like compressors, pumps can also be impacted by fluid property changes. Recently, a gas fractionator plant was starting up after a turnaround during which substantial debottlenecking construction had occurred. During the startup, the Product Ethane pumps, which deliver high purity Ethane to a pipeline, continually tripped off-line. Mechanically and electrically the pumps appeared in good shape. Investigation of the process revealed two problems: at startup the Ethane purity is low resulting in a heavier than normal hydrocarbon feeding the pumps, and unsuccessful startup preparation had left large amounts of Water in the unit. Among the problems caused was erratic Product Ethane pump performance.

Figure 1 depicts the Product Ethane pump performance curve. Operating points are shown for Ethane and Water. The pump operates at a relatively constant ΔP taking suction from a distillation column reflux drum and delivering Ethane to a pipeline. Equation 1 defines the relationship between pump pressure increase and the pump's differential Feet of Head which is plotted versus pump flow rate in the pump performance curve of Figure 1.

$$Necessary\ Pump\quad Head(ft) \;=\; \frac{2.31\quad x\quad \Delta P(psi)}{Fluid\quad Specific\quad Gravity} \qquad (1)$$

When the pumped fluid changes from Ethane to Water, the fluid Specific Gravity (S.G.) rises from 0.35 to 1.0. Equation 1 shows that to increase Water's pressure the same amount as Ethane's it takes only 35% as much Head. But, the pump must operate on its curve. Lowered pump head is produced by increased flow rates. This is illustrated by the operating points in Figure 1. The Water operating point is far out on the curve compared to Ethane operation.

Equation 2 describes the effect this has on the amount of energy the pump needs to do its job.

$$Hydraulic\quad Horsepower \;=\; \frac{Flow(gpm)\quad x\quad \Delta P(psi)}{1714} \qquad (2)$$

Hydraulic Horsepower is the energy imparted to the fluid by the pump to generate a pressure rise with a concomitant flow rate. When the S.G. of the pumped fluid increases substantially at constant ΔP, as in our case, the pump flow jumps according to its performance curve. Required Horsepower also jumps proportionally to the flow increase as shown by Equation 2. Because moving out to the end of the pump curve typically also moves away from the Best Efficiency Point, pump efficiency diminishes causing a magnified Horsepower demand on the device driving the pump - a motor in our case. These factors can combine to require more Horsepower than the motor driving the pump is rated for. The motor over-amps and trips. One way to work through these problems is to pinch down pump outlet valves. This raises the required pump head and lowers the flow rate thereby reducing the necessary pump driver Horsepower. Operation with heavier than normal hydrocarbon at startup probably can be economically accounted for in the pump design. For this pump, water operation would be costly to accommodate.

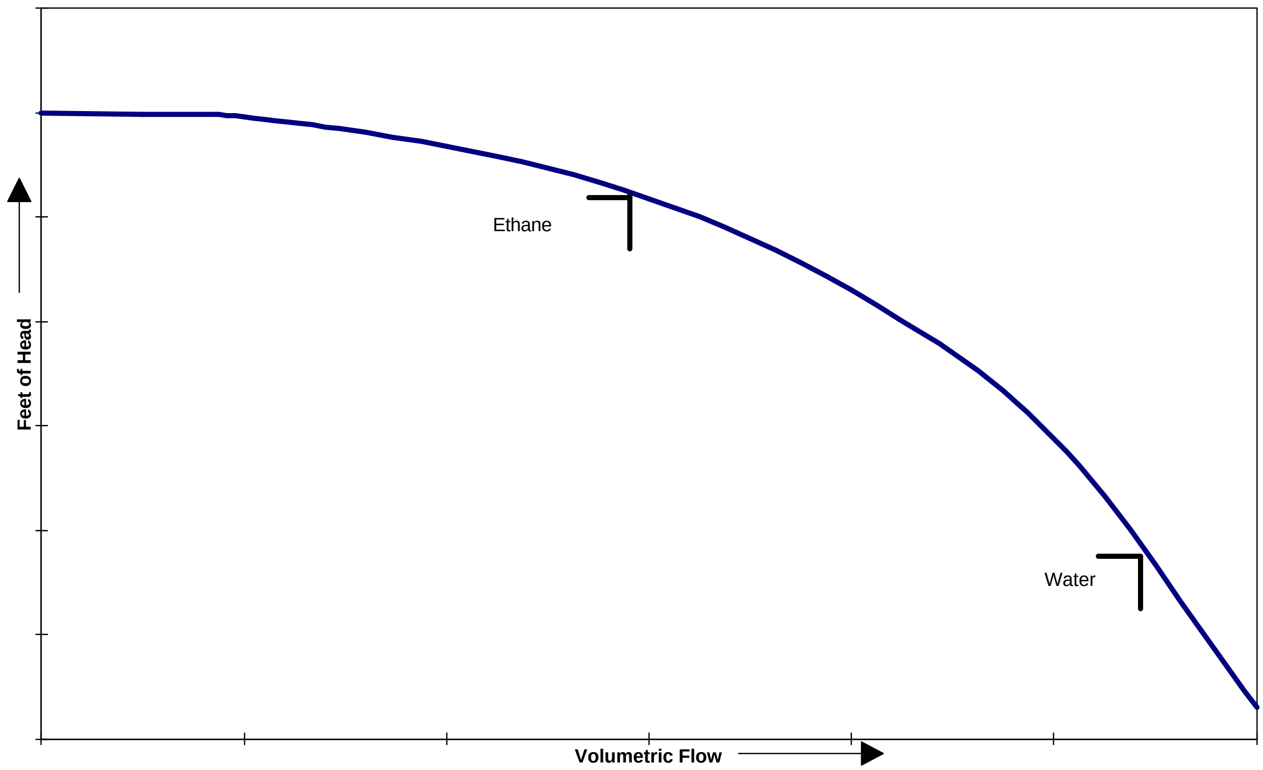

Figure 1. Product Ethane Pump Curve.

Another area Water can cause similar problems is in distillation column reflux operation. In systems where it is imicible and more dense than the other phase, water is often separated from hydrocarbon in a column's overhead accumulator by gravity difference. Water settles into a boot or behind a weir. Figure 2 provides an illustration. However, Water may be fed to the reflux pumps for several reasons. A process upset, accumulator fouling, or accumulator Water level control malfunction may diminish the accumulator Water separating efficiency. Under some circumstances, water may purposely be sent to the tower as reflux. This may be done to Water-wash the column to clean trays, or during a shutdown to clean the vessel.

Assuming column pressure and frictional losses are similar, reflux pumping of Water instead of hydrocarbon requires an increased pump pressure rise. This is due to the higher static head necessary to push Water to the top of the tower. In this case, the pump continues to produce the same Feet of Head at the same flow rate. The distance, or Feet of Head, to the column top has not changed. However, the pump ΔP rises in response to the Water's increased density. Equation 2 shows that the required Horsepower increases proportionally to the pump ΔP gain. The reflux pumps could demand more from their drivers than they are rated to deliver.

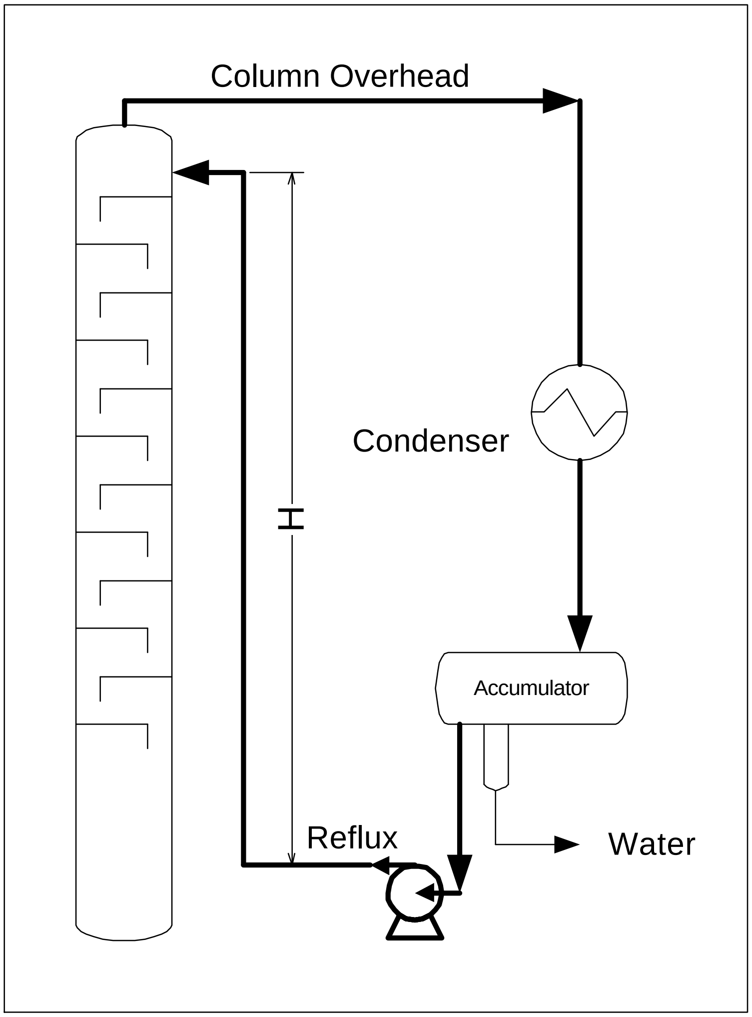

Figure 2 - Reflux Hydraulics.

Two Phase Fluids

Process streams at or near their bubble point can pose special hydraulic problems. The static head change from a piping rise may be a sufficient pressure reduction to result in vaporization of a bubble point fluid. Typically the amount of vapor formed is small as the cooling associated with vaporization retards the behavior. However, even a little vapor can sharply increase frictional pressure losses as the now two-phase fluid travels through a pipe. Higher frictional loss will raise the vapor fraction feeding additional frictional loss. Distillation columns utilizing heat pumps for combined condensing and reboiling are one place where the complications of bubble point fluids might arise.

Figure 3 illustrates a Heat Pump layout. A Heat Pump condenses a column's overhead vapor by using it to reboil the column from which it came. This is accomplished through compression of the overhead vapor. Where it is feasible, a Heat Pump requires the energy for compressor operation and a small amount of trim heating/cooling utility, where a conventional column needs full overhead cooling and bottoms heating utilities. There are many economic trade-offs between capital and utilities costs that affect the selection of a Heat Pump over a conventional design. But where it is practical, it can be very profitable.

Recently a Deisobutanizer (DIB) column operating with a Heat Pump began to have compressor trips during the summer. Compressor flow would begin to taper off, the low-flow kickback would open (the kickback had no cooler), and shortly thereafter the compressor would trip on high discharge temperature. An investigation revealed several contributing elements. One substantial factor was due to bubble point and equilibrium liquids.

The Figure 3 illustration captures the important points of the system under review. During the summer, upstream cooling constraints caused DIB operation changes which resulted in the liquid at Point A in Figure 3 approaching its bubble point. From Point A, the condensed DIB Overhead traveled a large vertical distance up into the piperack. Upon reaching Point B, the liquid had vaporized. The now two-phase pressure drop from Point B to the Reflux drum was higher than the winter's all liquid phase drop by 6 psi. Additionally, the material sent to the reflux drum was delivered at a higher temperature during the summer. Since the reflux drum contains a liquid in equilibrium with a vapor, a temperature rise results in a pressure rise. The Summer reflux drum pressure was 6 psi higher than the winter operation.

Together the two pressure demand increases amounted to a necessary Summer compressor ΔP 120% of the winter value. As the day heated up, the compressor loads built until their capacity was exceeded at which time the machine tripped. Upstream adjustments allowing subcooling of the condensed compressor outlet during the

Summer in addition to compressor maintenance to restore lost Horsepower reestablished column capacity.

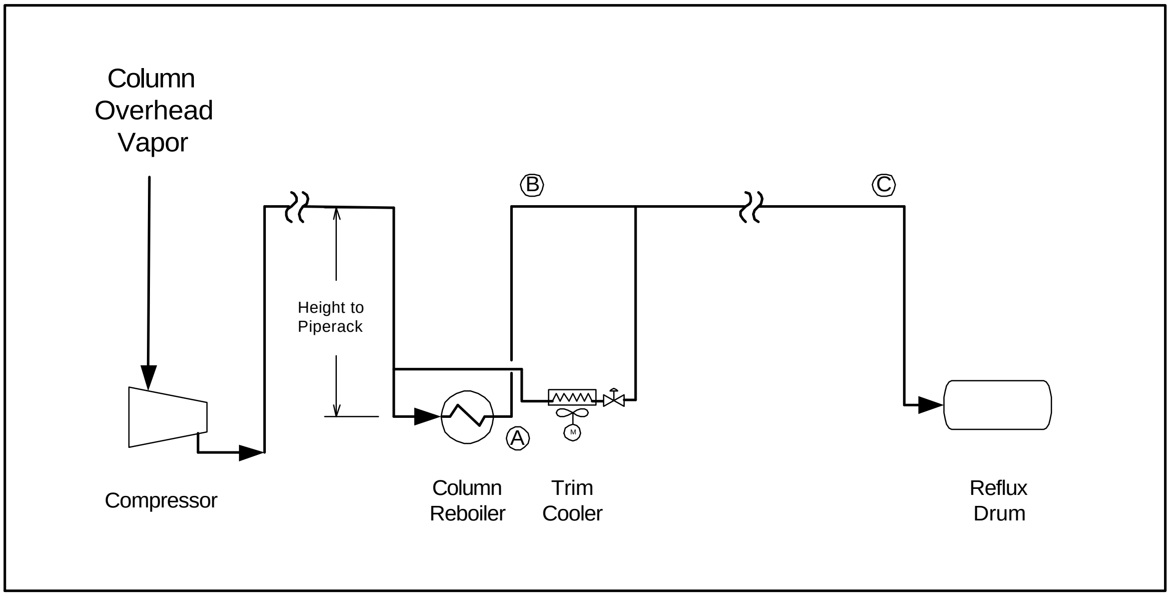

Figure 3 Heat pump equipment layout.

Condensing Fluids

Condensing fluids entail radical changes in fluid physical properties. Densities and viscosities can increase by an order of magnitude when fluids change from a vapor to a liquid. Liquid frictional flow losses may be an order of magnitude smaller than vapor pressure drops at equal mass flow rates. Static head effects go from negligible to substantial upon condensation. While the mathematical and analytical accounting of these changes is well defined, weaving their effects into the process can be difficult.

Recently, construction of a new propane refrigeration loop was completed. The system compresses Propane, condenses it, and then feeds it to heat exchangers where Propane vaporization cools the process. The Propane vapor from the exchangers is then returned to the compressor through a suction knockout to complete the loop. Plot space constraints impacted the project. In order to reduce the construction footprint, the Propane condensers were stacked vertically and piped as shown in Figure 4. However, the Process Hydraulics of the design resulted in the effective negation of both of the bottom two exchangers. At least part of the # 2 exchanger was also impaired. The static head inherent in the liquid outlet piping caused the bottom two exchangers to run liquid full. Only the pressure drop resulting from the # 1 exchanger accommodating approximately three times its design rate permitted part of the tubes in the #2 exchanger to be exposed and function. The condensing capacity of the loop was cut by more than 50%. At sufficiently large pressure drops through exchangers 1 and 2, exchangers 3 and 4 might become operable. This could be accomplished by inserting orifices in the exchanger 1 and 2 piping. However, this solution consumes expensive compressor Horsepower. Rearrangement of the exchangers to establish all their centerlines at a common elevation is necessary.

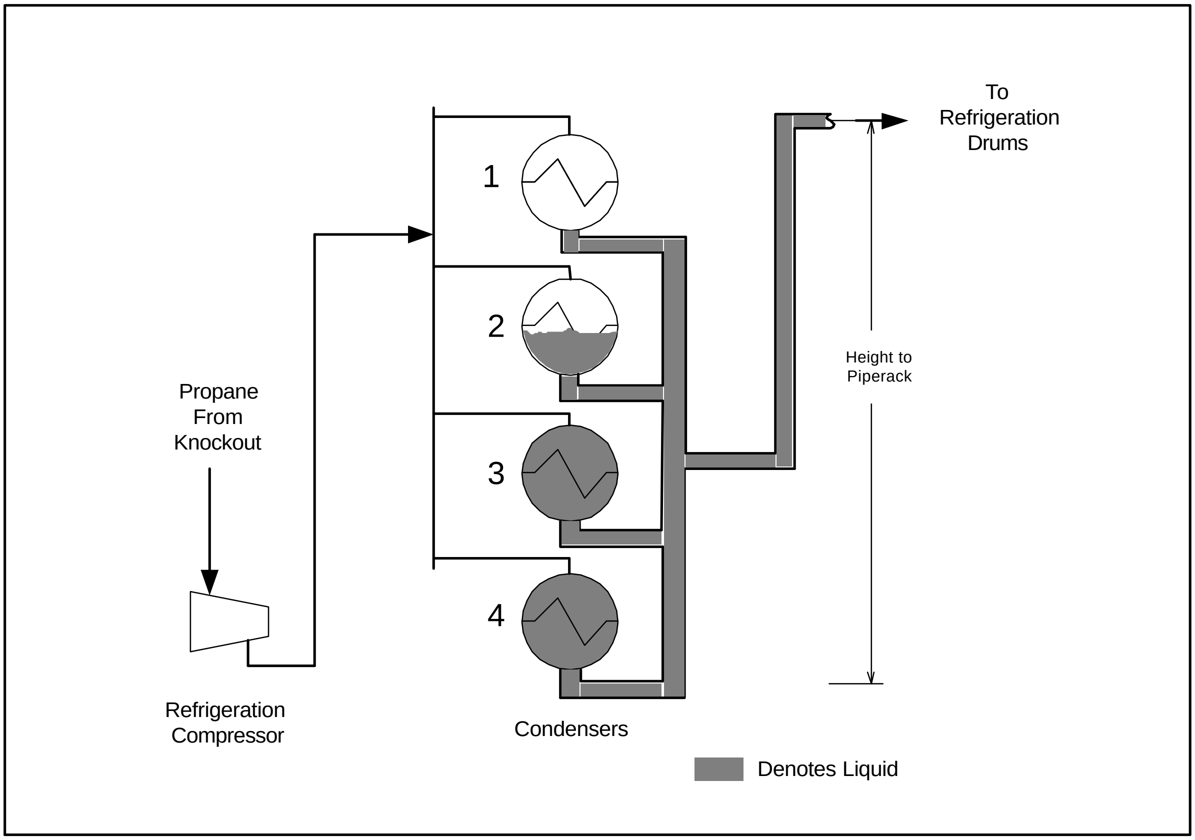

Figure 4   Propane condenser layout.

A similar situation also arose during the debottlenecking of a Debutanizer column. The existing overhead condensing capacity was inadequate for the revamped design. Additional exchanger area was added ahead of the existing flooded condenser as illustrated in Figure 5. However, two sets of exchangers were added at very different elevations. Under normal conditions, the pressure drop through the higher exchanger was not greater than the liquid head the lower exchanger had to overcome to rise into the piperack where it joined the outlet of the upper exchanger. Therefore, there was zero flow through the lower exchanger. In order to make this system work, a block valve on the outlet of the upper exchanger is kept pinched down to superimpose a pressure drop equal to the 30 foot liquid differential between the exchangers. In this way flow through the lower exchanger is achieved.

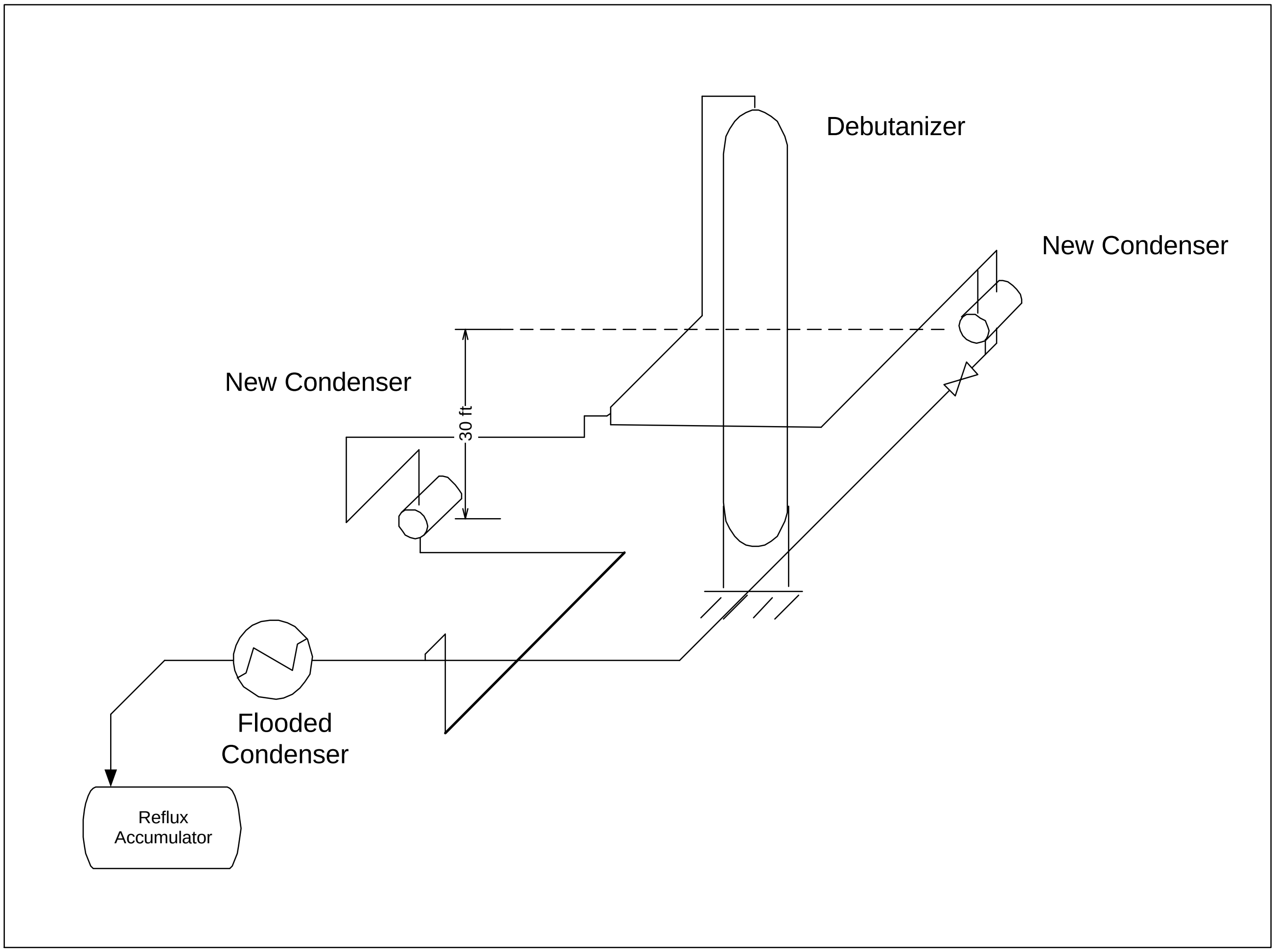

Figure 5 Overhead condenser layout.

HOT VAPOR BYPASS

The hot vapor bypass configuration is a common method employed for fractionation column pressure control. Figure 6 illustrates a typical system. Hot vapor bypasses are used where the column overhead is totally condensed. The system's name comes from its design in which part of the column overhead is bypassed around the overhead condensers.

Hot vapor bypasses effect pressure control through adjusting the overhead condenser exchanger surface area. By raising and lowering the liquid level within the condenser, the effective heat transfer area is increased and decreased through uncovering and covering exchanger tubes. An increase in the surface area lowers the column pressure. A decrease in the surface area raises the column pressure. Because covering tubes is required to adjust heat transfer, air-coolers are not preferred in hot vapor bypass service. The height to width ratio of an air-cooler is small compared to a shell and tube exchanger. A change in liquid level of one inch could result in an air-cooler's entire tube row being covered. A shell and tube exchanger has many rows - covering one of them is an incremental change. However, an air-cooler may have only 10 or fewer rows - covering one of them is a large step-wise change in heat exchange. One method used to mitigate this effect in air-coolers is to slope several bottom tube rows, or all rows, toward the outlet so that the effective height to width ratio is increased.

The liquid level in the exchanger is adjusted through the hot vapor bypass flow. Equation 3 details the bypass pressure balance. Within reason, it is not possible to oversize hot vapor bypass piping. However, it is possible, and a common problem, for bypass piping to be undersized. Increased hot vapor bypass flow raises the pressure in the accumulator (P_C). This pressure increase must be balanced by an increase in the H_1 liquid head as illustrated in Figure 6 (the accumulator level controller maintains a constant H_2 liquid head). Because the pressure at Point B (P_B) is greater than that at Point C (P_C), distance H_1 must be shorter than H_2. This leads directly to the conclusion that the condenser exchangers must physically be below the accumulator. Otherwise, the condenser pressure P_C cannot force liquid up into the exchanger against the greater pressure P_B. Also because P_C is less than P_B, the liquid exiting the overhead condensers must be subcooled or it will vaporize in the accumulator. The overhead condensers in a hot vapor bypass system must be oversized (typically by approximately 25%) to ensure subcooling. Using the hot vapor bypass overhead pressure control scheme there is no direct accumulator temperature control. In practice, the accumulator temperature is approximately 10 °F subcooled thereby delivering a consistent reflux temperature at constant overhead composition and pressure. In summary, P_B and H_2 are constant, while P_C and H_1 vary in direct proportion.

$$P_B + H_1 \times (\text{Liquid SG})/2.31 - \text{Friction loss BC} = P_C + H_2 \times (\text{Liquid SG})/2.31 \qquad (3)$$

$P_B > P_C$
Friction Loss BC = Small

An accumulator drum in hot vapor bypass service often has special features. The material bypassing the condenser is hot and part of it must remain a vapor upon entering the reflux drum. That means the vapor must be in equilibrium with the liquid in the drum. How is this accomplished if the liquid in the accumulator is subcooled? Figure 6 depicts the exchanger liquid and bypassed vapor entering the overhead drum separately; vapor in the top, liquid in the bottom. Although the physical layout of the piping entering the vessel may be different, the internal distribution of liquid and vapor should be kept stratified. Vapor entering the vessel top interacts with the upper layer of the liquid level. This layer is kept quiescent through design of the liquid and vapor inlets so that the calm layer is at a higher temperature than the bulk of the subcooled liquid. This allows a stable vapor zone to coexist with a large body of subcooled liquid and maintains a steady liquid density. In addition to special liquid and vapor inlet internals, hot vapor bypass accumulators may have internal baffles to help maintain a still upper liquid layer. Typically horizontal vessels are used in hot vapor bypass service to deliver an optimal liquid/vapor interface area.

Some hot vapor bypass designs mix the bypassed vapor and the subcooled liquid before entering the accumulator. This increases the required hot vapor bypass flow and may result in the accumulator vapor space being filled with a lighter component as the heavier material is condensed and absorbed by the liquid. This mixing before the drum results in two-phase flow and its attendant instability. If the vapor and liquid are mixed upstream of the accumulator, static head hydraulics will be directly impacted.

The hot vapor bypass configuration could permit non-condensable components (inerts) to accumulate either in the exchanger or the overhead drum. Vent lines are typically installed to allow system purging.

If the hot vapor system hydraulics are configured properly, process control is straight forward and is shown in Figure 1. The drum level control typically adjusts reflux flow as most systems using a hot vapor bypass have a large reflux to distillate ratio. P_C does not directly reflect the column pressure and should not be used as the controlled variable in the pressure control scheme. Note that because a steady drum level supports functional column pressure control, advanced control methods such as gapped or dead-band drum level control are typically not appropriate in hot vapor bypass systems.

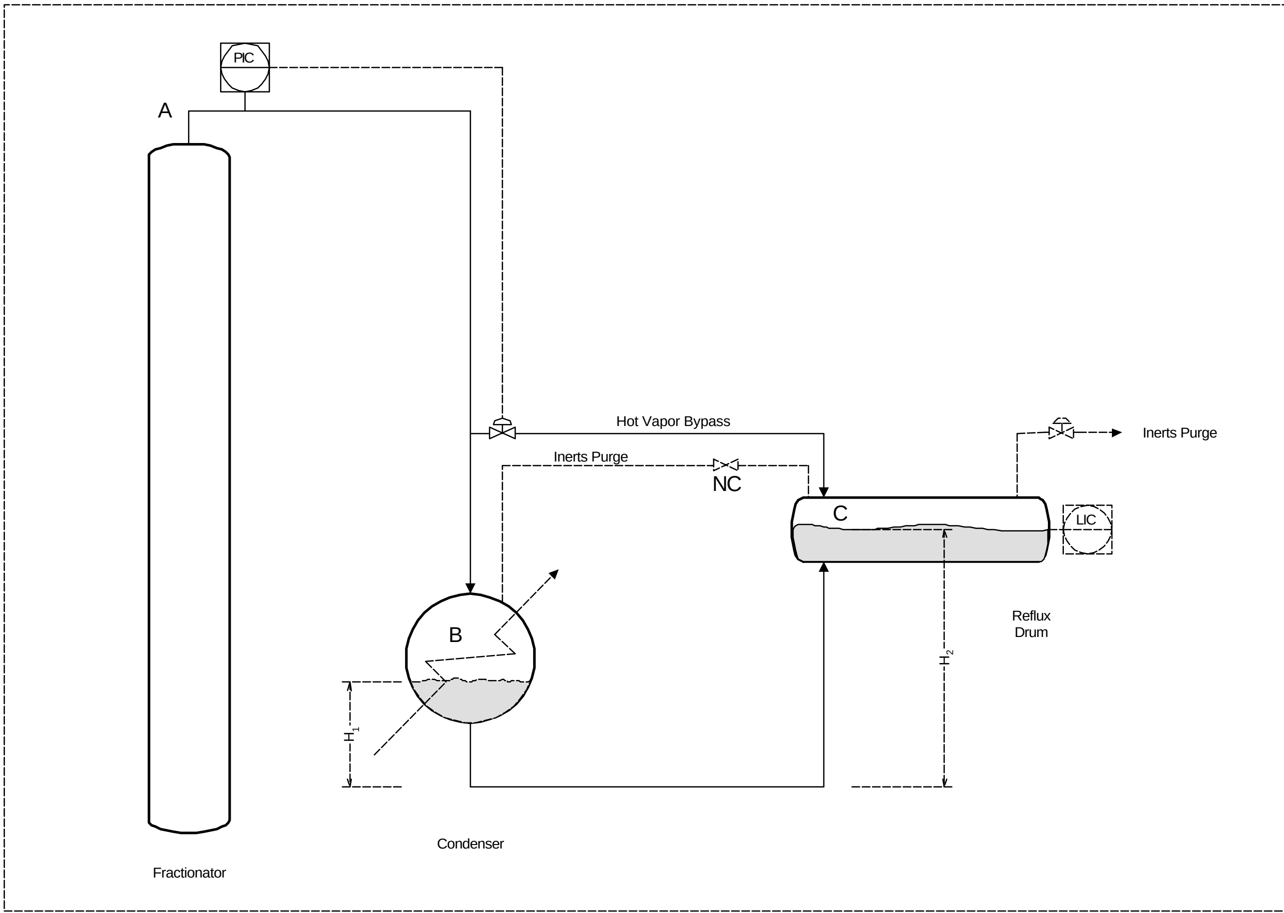

Figure 6 - Hot Vapor Bypass Process Hydraulics

Figure 7 depicts a recent Depropanizer Hot Vapor Bypass arrangement. Several of the pitfalls discussed above are present in this design:

- The exchanger elevation is above the reflux drum.
- The Hot Vapor Bypass vapor joins with the exchanger outlet before entering the reflux drum.
- The pressure control takes its signal from the reflux drum.
- An air-cooler, in this case without sloped tubes, is used.

On startup this system did not control the column pressure. Currently the column is operated with the Hot Vapor Bypass blocked, the exchanger outlet partially closed, and the column overhead pressure controller adjusting the air-cooler louvers. The exchanger outlet must be pinched to reduce the exchanger surface area by backing liquid up into the tubes. Because the exchanger surface area was oversized, as for Hot Vapor Bypass service, with one fan in operation and the louvers closed excessive cooling is delivered.

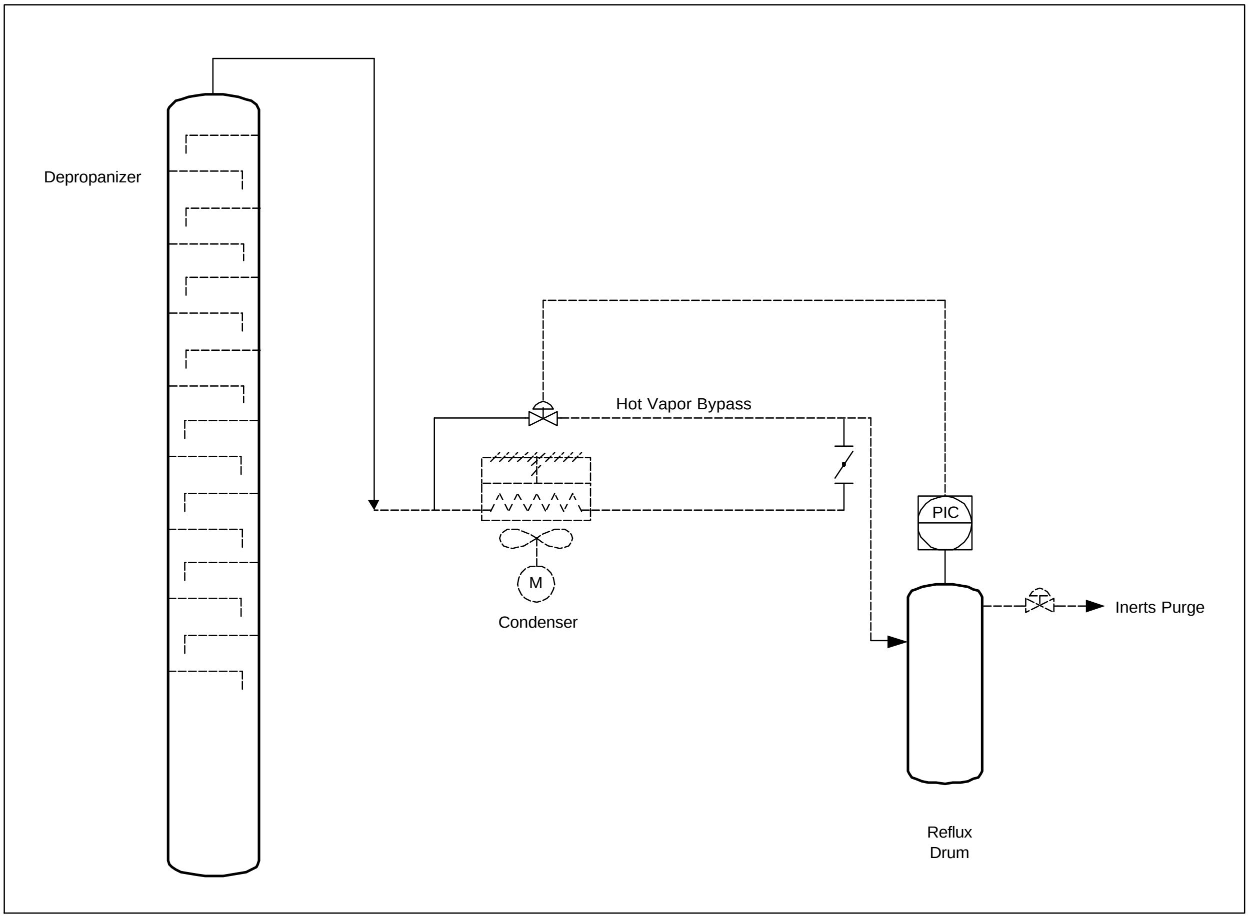

Figure 7 - Depropanizer Hot Vapor Bypass

Piping and Plot Plan

Moving from the paper of process design to the reality of construction can take many twists and turns. Some of these detours can take the form of unexpected piping lengths that impact Process Hydraulics. In a recent NGL fractionator revamp, several Propane refrigeration system problems resulted from plot plan and piping design decisions.

The propane system under review is a closed refrigeration loop consisting of gas turbine driven compressors, water cooled condensers, and process exchanger/evaporators. At the initial phases of detailed design, the pipe hydraulics were calculated from process design information, pipe lengths were estimated from the plot plan, and a normal number of fittings (ells, tees, etc.) were considered in calculating equivalent lengths. However, as detailed design progressed, the equivalent length of the discharge piping increased dramatically due to equipment location alterations, piping appurtenance requirements, and adjacent equipment and piping obstructions. Despite the changes, piping hydraulics were not reviewed to confirm the original assumptions of hydraulic sizing.

This resulted in a “built-in” 15 psi pressure drop between the compressor discharge and the condensers for the unaltered piping diameter originally selected. The pressure drop absorbs an unplanned load of 500 BHP from the compressor gas turbine drivers for $60M per year of additional variable cost. Along with this constraint in horsepower is a more costly impact on process throughput due to a refrigeration system capacity reduction of 700 tons. It is difficult for the process design to foresee all construction constraints. Likewise it is hard for detailed design and construction to accommodate all process design goals. There are real incentives for design and construction groups to have absolutely no communication: It is easier to build something without regard to function, and it is easier to design something if you don't worry about whether it can be built. The primary incentive for thorough communication and review between the competing groups is higher profitability through improved safety, better operation, and lowered cost.

Other problems can crop up when being too conservative during pipe diameter selection. One situation occurred with sizing of lean amine piping at an NGL fractionation facility. The amine system consists of three contactors operating in parallel being fed from a common regenerated lean amine supply. The piping to two of the three contactors is greatly oversized for their normal flow rates, and the resulting low velocity causes settling of suspended solids along the pipe bottom. The material builds up over time until constriction created velocities great enough to re-entrain the solids are generated. This sloughing off results in sudden fouling and foaming of the two amine contactors. The section of piping in question is 10” nominal diameter. With a volumetric flow rate of 200 gpm, the process velocity is a low 0.8 ft/sec. Low velocities in this service are desirable as they retard erosion/corrosion. However, bigger is not always better.

Summary

Many of the perils associated with Process Hydraulics should be addressed by thorough Piping and Instrumentation Diagram (P&ID) definition and by equipment specifications that include operation outside of the norm. Critical equipment layout parameters such as relative elevations should be detailed on P&ID's. Because piping design and construction usually take place apart from process design, the P&ID serves as the formal template for informing downstream design disciplines. Equipment specs that consider upsets, shutdowns, and maintenance activities are more likely to deliver machinery well mated to its task.

Author Biographies

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Don Schneider is President of Stratus Engineering, Inc., Houston, Texas. Previously he worked as a senior engineer for Stone & Webster Engineering, and as an operating and project engineer for Shell Oil Co. He holds a B.S. from the University of Missouri-Rolla, and an M.S. from Texas A&M University, both in chemical engineering. Don is a registered professional engineer in Texas.

Author's Previous Publications

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