Understanding Process Equipment for Operators and Engineers

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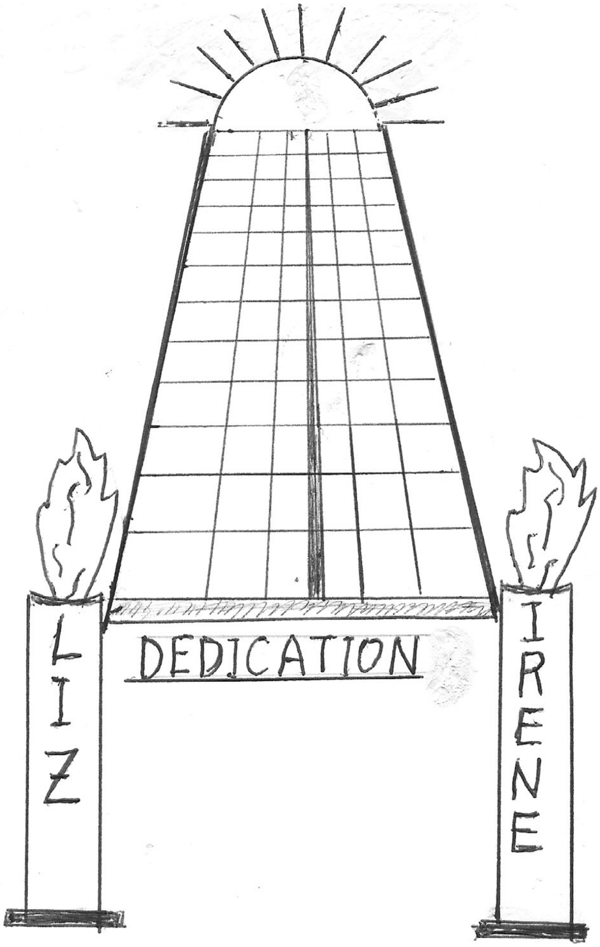

Author Note

I hope my book has helped you. If you have a plant problem, or a question about this text, email me at norm@lieberman-eng.com. I will be glad to help.

Norm Lieberman

Introduction: An Act of Creation

Our Universe exploded into existence 13,480,000,000 BC. The Big Bang! Quickly, a fraction of the energy coalesced in protons, neutrons, electrons, and photons. These elementary particles had the potential to form hydrogen, oxygen, carbon, and eventually, stars, planets, and life, in all its complexity and beauty.

The creator endowed the products of the Big Bang with the potential to form trees and frogs. Having done so, the creator allowed DNA to evolve into you and me without further interference.

As a design engineer, I have a similar role. I create process designs for refinery units. Having issued the PFD and data sheets, I leave my creations in the hands of operators and engineers. Rarely do I review or revise how they operate a unit I designed.

Technology evolves; product spec's change; equipment corrodes; and new catalysts are developed. And, my original design becomes progressively irrelevant. That is the nature of the Universe, and of our planet.

But sometimes, I have a desire to remind process operators and tech support engineers, as to the fundamental principles of my original design. That is why I have written this book. It contains no new ideas or modern technology. It is a voice from the past to remind the reader of basic process engineering concepts that form the foundation of units we operate.

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General Introduction

I have always wanted to become an important person. To go to important meetings with other important people. To become a team leader, or a manager, or a V.P., or even someday a Director. Mainly, I wanted an office, rather than my crummy cubicle.

So, in 1980, after 16 years, I left American Oil in Chicago. I was hired by the Good Hope Refinery in Louisiana as the Tech Service Manager. I had 18, mostly young, process engineers reporting to me. A few days after assuming my new position, one of the junior members of my staff knocked on my door.

"Excuse me Mr. Lieberman, do you have a second? I was wondering if you could explain to me how to calculate the pressure drop of water flowing through a hole?"

"Okay Dave," I said, "I'll get back to you after lunch."

But I did not get back to Dave after lunch, because I didn't know how to calculate the pressure drop (DP) of water flowing through a smooth hole. I didn't understand that the DP was mainly due to acceleration of water in the hole, plus additional loss due to turbulence and friction. And this was after working 16 years as a senior process engineer for American Oil. After a Master's Degree in Chemical Engineering from Purdue University. After a BS from Cooper Union in New York—one of the best engineering colleges in the United States.

As I sat there in my important paneled office, I started to mentally tabulate all the other process calculations that I didn't know how to do, without the aid of a computer simulation. And then I realized that the reason I didn't know how to do fundamental calculations was a consequence of my lack of understanding about how process equipment really functions or even how much of the everyday world around us works. For instance, I didn't know:

1. What causes draft in fired heaters?
2. What causes thermosyphon circulation in reboilers?
3. What factors affect distillation tray fractionation efficiency?
4. Why condensers lose heat transfer due to noncondensable accumulation?
5. Why condensate back-up in surface condensers slows turbines?
6. Why furnace tubes coke-up and fail?
7. What causes pump mechanical seals to fail?
8. What causes centrifugal compressors to surge and self-destruct?
9. How steam strippers work?
10. How thermodynamics apply to refinery equipment?
11. How vacuum ejectors work?

12. Causes of positive feedback loops in control systems.
13. What is “vapor-lock”?
14. How are flows and levels measured?

So, I made a decision. Either I could become an important tech service manager and attend important meetings with other important managers, or I could learn how process equipment actually works. And now, 38 years later, I’m still learning. Every time I go to a process unit to troubleshoot a problem, I learn something new. Something exciting. Something that makes me a better engineer.

The intent of these chapters is to share my knowledge about process equipment fundamentals. If you can’t explain how process equipment works to an operator, it means you don’t understand it yourself. If you cannot reduce an engineering principle to a calculation, then you do not understand the underlying principles.

Computer simulations and modeling are fine for examining complex alternative cases, but are not a substitute for a fundamental understanding of the underlying principles of chemical engineering. I hope that this book will lead you further along the path of knowledge and understanding that I am still traversing myself today. I will be very pleased to answer any questions you have about this text or those relating to your work.

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1

Basics of Process Operations and Engineering

The truth is you've wasted 4 years in University qualifying for an engineering degree or have wasted 2 years in an operator training class. That's not my fault. But now, as a process engineer in a refinery, or as an operator in a petrochemical complex, you need to understand how pumps, compressors, heat exchangers, distillation towers, air coolers, fired heaters, steam boilers, turbines, and vacuum ejectors actually work to do your job.

The reason you did not understand your professor about how process equipment works is that he didn't understand it himself. Everything I write is based on my own experience and field observations. For example:

1. Increasing excess combustion air in a fired heater will cool off the tubes in the firebox, but increase the heater absorbed process duty.
2. Nitric oxides (NO_x) can be reduced by increasing air flow to the secondary air register and reducing air flow to the primary air register.
3. Preheating combustion air to a heater will often reduce its heat absorption capacity.
4. The energy to spin a steam turbine does not come from the steam's pressure, but from the steam's heat content.
5. But, superheating steam to a steam turbine will cause the turbine to slow down.
6. A steam vacuum ejector is really a multistage gas compressor, but there are no moving mechanical components.
7. One way to stop a distillation tower from flooding is to increase the reboiler duty.
8. With a constant reboiler duty, lowering a distillation tower's top temperature with more reflux increases the vapor flow.
9. Ancient bubble cap trays have a better tray fractionation efficiency than modern trays, and also a lower pressure drop.
10. When starting a centrifugal pump, monitor the pump suction pressure, not its discharge pressure.
11. The feet of head produced by a centrifugal pump does not vary with the density of the liquid pumped.
12. Increasing the load on an ordinary alternating current motor-driven pump does not affect its speed.
13. Centrifugal, motor-driven American pumps, run 20% faster than European pumps. Why? Because they are American pumps.

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14. A butane reboiler will have more capacity, if its tubes are pitted and corroded, than a reboiler with brand new tubes.
15. A tiny amount of air or CO_2 in steam will reduce a steam reboiler, or a steam turbine surface condenser, capacity by 50%.
16. The latent heat of condensation of steam increases a lot as the steam pressure is reduced.
17. The auto-ignition temperature of gasoline is much higher than tar.
18. Hydrogen heats up as it expands, but all other gases cool off, as they expand.
19. Centrifugal compressors surge when the gas molecular weight is reduced.
20. Partly *closing* a cooling water outlet valve from an exchanger elevated 60 ft above grade can increase its water flow.
21. A rising level in a boiler can cause the boiler's indicated level to be reduced, and the boiler feed water level control valve to open, even though the water level is already excessive.
22. A gas flow meter will read a reduction in gas flow rate, even though the actual volume of gas flow has not changed, if the gas molecular weight is reduced.
23. A pressure above atmospheric can develop in the upper portion of a heater's firebox, even though a substantial vacuum exists in the lower portion.
24. Thermosyphon circulation rates may be reduced, as the exchanger heat input increases.
25. An effective way to improve a distillation tower fractionation efficiency is to reduce the tower's operating pressure.
26. The motor amps increase with molecular weight for a centrifugal compressor, but decrease with a higher molecular weight for a reciprocating compressor.
27. Fouled fin tubes on a forced draft air cooler which greatly reduces cooling air flow through the tube bundle does not increase the amp load on the fan's motor driver.
28. The outlet pressure of a heat exchanger can be higher than the inlet pressure, even though both pressure points are at the same elevation.
29. Closing a by-pass valve around a condenser will usually reduce the condenser's upstream pressure.

The preceding points are all somewhat counterintuitive. I'll explain the process basis for each of them in the following chapters. The application of these and other concepts I have discussed in subsequent chapters have led to the solution of many process problems and enhanced the capacity and efficiency of hundreds of process units.

I think most people are like me. I find it difficult to learn new stuff by reading, listening, or just watching. But, when I apply my own hands, and my own mind, to a particular problem, I remember and understand the solution for the rest of my life. And, as someone, who has been practicing solving process engineering problems in the field, without any hiatus, since 1964, that's a lot of learning opportunities.

We are fortunate to work in an industry where the nature of the equipment has barely changed in the past six decades. Becoming a proficient process operator, or process engineer, is still largely a matter of time and determination. And, I suspect, that will always be true as long as we try to alter the nature of hydrocarbons, minerals, water, and air to our own human purposes.

Things I Worry About

Troubleshooting Refinery Process Equipment is the most complex activity that mankind can undertake. I've been at it for 54 years, and I barely know what I'm doing. The main problem is that process equipment is normally not designed correctly in the first place. The original design errors are then compounded with:

- Corrosion
- Foulings
- Mis-operation
- Variable ambient condition
- Unexpected feed stocks
- Altered product spec's

In spite of my very long experience to the contrary, I often pray for help to solve process problems. However, instead of waiting for divine intervention, I've found it's better to understand some basic process principles such as:

- How wind affects draft?
- How reboiler duty affects thermosyphon circulation?
- How noncondensables cause vapor binding in steam condensers?
- What determines tray fractionation efficiency?
- The real function of steam in a steam stripper.
- How centrifugal pumps convert power to pressure?
- How steam turbines and steam engines are explained by thermodynamics?
- Effect of reflux on vapor flows in a distillation column.
- What causes centrifugal compressors to surge?
- How not to blow up a fired heater which has become flooded with fuel, and with fire and black smoke being emitted from its stack?

I also talk a lot about safety, and the last chapter in this text addresses that particular problem based on my many bad refinery experiences.

The Future Without Refineries

My world, the world dominated by vehicles burning hydrocarbons, will be gone in 20 or 30 years. Much of what I know about delayed cokers, visbreakers, fluid catalytic crackers, hydrocracking, and naphtha reforming will no longer be relevant to mankind's future in transportation.

I suppose even after gasoline and diesel will no longer be required, the human race will still need to process hydrocarbons for:

- Jet fuel
- Asphalt roads
- Lube oils
- Plastics
- Petrochemicals
- Carbon anodes
- Roofing shingles
- Baby oil
- Waxes

Still, the majority of petroleum refining process technology will be lost and forgotten—a relic of the past. However, I am quite sure that the human race will persist, as long as we require industrial technology as a species, to use:

- Centrifugal pumps
- Heat exchangers
- Gas compressors
- Vapor-liquid separators
- Electric motors
- Distillation columns
- Control valves
- Pipes, valves, and fittings

I imagine, if by chance, that I am reincarnated as a Process Engineer, in the year 3018, I will not feel out of place, or the skills I acquired in my current life cycle will be entirely outmoded.

If, on the other hand, I find myself in the netherworld, and am greeted by the Devil himself, I will also feel quite at home. Once Satan realizes my vast experience in Sulfur Plants and Fired Heaters, I am confident he will enroll me in Hell’s Process Engineering Department.

Safety Note

Hydrocarbon leaks in water cooling service can cause explosive mixtures to form below the distribution decks in cooling towers. Especially when cooling butane streams, this can, and has (Amoco Oil—Texas City), caused cooling towers to ignite. Check your cooling water outlets for combustible vapors perhaps once a month. The instrument that the operators use to issue vessel entry permits can be employed for this purpose. Vapors tend to accumulate beneath the distribution decks. Sludge buildup on these decks is another indication and result of hydrocarbon leaks into the cooling water return.

Note From Author: If you have questions relating to this text or your work in general, please email me at norm@lieberman-eng.com.

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Factors Affecting Tray Fractionation Efficiency

I was working on improving a refinery waste water stripper a few months back. The problem was about 1000 ppm of aromatics dissolved in the stripped water. Also, I needed to increase the unit capacity from 50 to 80 GPM.

The tower was equipped with 30 valve type trays (see Fig. 2.1). I've shown a sketch of the stripper in Fig. 2.2.

The experienced process design engineer should always begin any design retrofit project by running a plant test. In this case, I asked Jerry, the panel operator, to increase the feed rate from 50 to 60 GPM, and the reboiler duty from 4000 pounds per hour of steam to 4800 pounds per hour. That is, to increase both the water feed and the reboiler steam, by 20%.

Jerry objected to the 20% increase in reboiler duty. "Mr. Norm, that's an awful lot of extra steam! But, if that's what you want, I'll do it."

I was surprised by Jerry's caution. What would be more natural than to increase the reboiler duty in proportion to the feed rate?

Jerry's caution, I discovered, was based on the capacity of the boiler, in which the sour water stripper vapors were burned. This boiler was limited by combustion air supply from its forced draft air blower (see Fig. 2.2). A sudden surge in combustible hydrocarbons could overwhelm the boiler's combustion air supply.

Effect of Incremental Reboiler Steam

I had anticipated that increasing the water feed rate and the reboiler duty by 20% would result in an increase of 20% in the fuel flow to the boiler and a 20% increase in its combustion air rate. But, this was not the case. The fuel rate and combustion air rate both surged by about 40%. Twice what I had anticipated!

What had happened?

Entrainment and Tray Efficiency

The only possibility for the disproportionate stripping effectiveness of the reboiler steam was an improvement in contacting between the steam and the sour waste water. Since the "*STRIPPING FACTOR*" is constant:

$$S = \frac{VK}{L}, \quad \text{where } K = \mathrm{VP}_{\mathrm{H.C.}} \div P_{\mathrm{T}} \tag{2.1}$$

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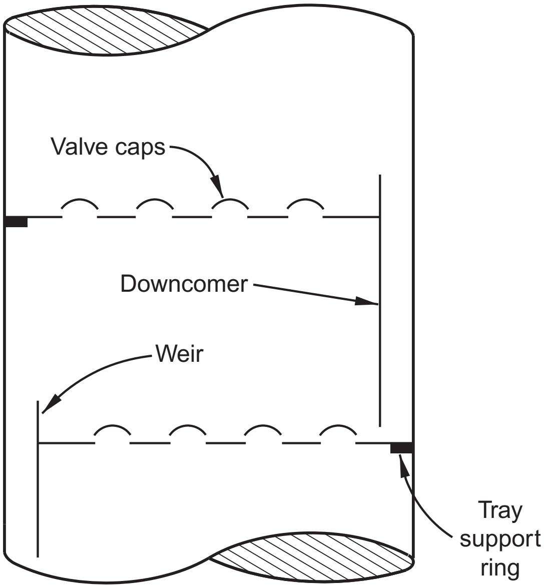

FIG. 2.1 Distillation tray.

- V = moles of steam
- L = moles of water
- $VP_{H.C.}$ = vapor pressure of the dissolved hydrocarbons
- P_T = tower pressure

The only possibility was that stripping efficiency must have been improved due to better vapor-liquid contact. That is, better tray fractionation efficiency.

Tray Efficiency Calculations

In university, I was taught several methods to calculate tray fractionation efficiency. Along with partial differential equations, particle quantum mechanics, and the history of the house of Hapsburg, I have forgotten them all. In ordinary process applications, tray fractionation efficiency is mainly a matter of:

- Tray deck levelness or sagging.
- Tray deck dumping, weeping, or leaking.
- Outlet weir levelness.
- Entrainment.
- Flow path length.
- Loss of downcomer seal.
- Tray deck hydraulic gradient.

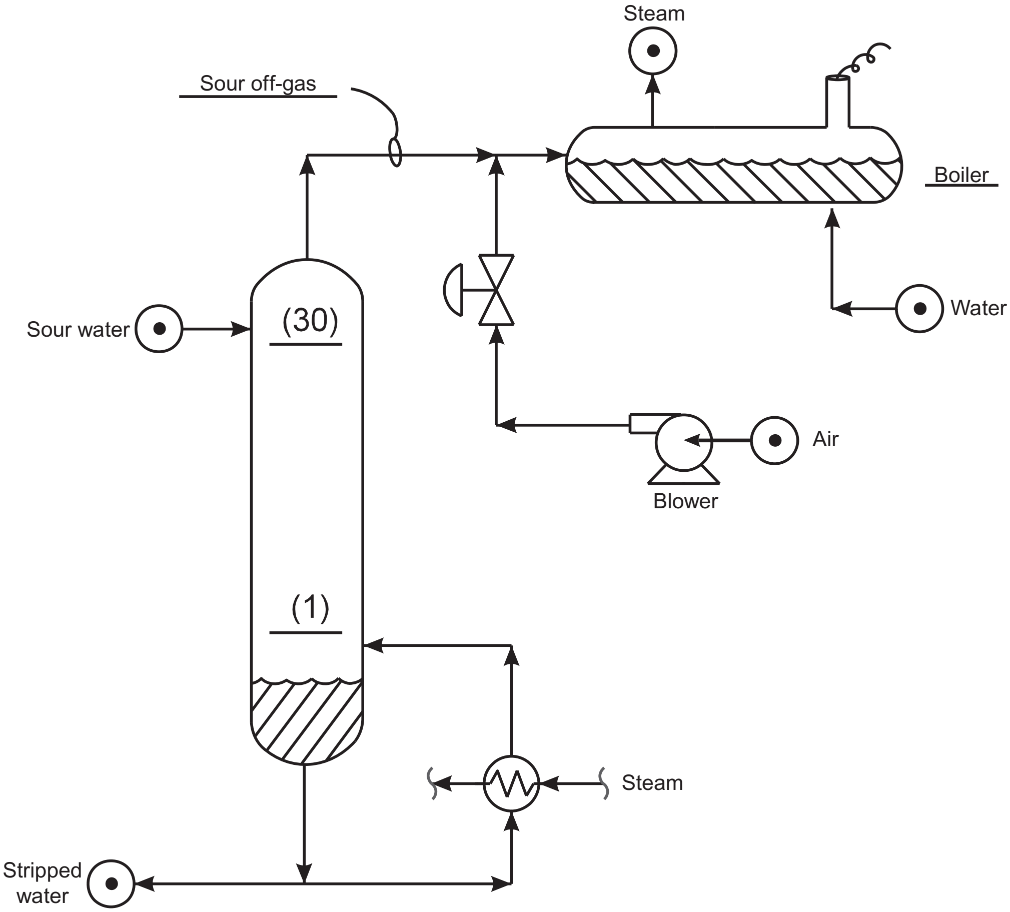

FIG. 2.2 Sour water stripper.

In my experience, it is the combination of the first two factors that account for much of poor fractionation efficiency in refineries and chemical plants. This problem of tray leakage, due to the tray deck being out of level or sagging, cannot be reasonably accounted for by calculation, when the out of levelness is not known. Even it was measured during a tower turnaround, it would change due to thermal expansion of a tray deck after the tower was restreamed.

Before the 1950s, when bubble cap trays were used (see Fig. 2.3), this was not a problem. Bubble cap trays don't leak even when moderately out of level. But now, we use valve trays (see Fig. 2.4) in most distillation towers. Valve trays have a movable cap that is pushed up by the force of the upflowing vapor. When that upward force is too little, the valve cap is supposed to fall closed to prevent the tray from leaking.

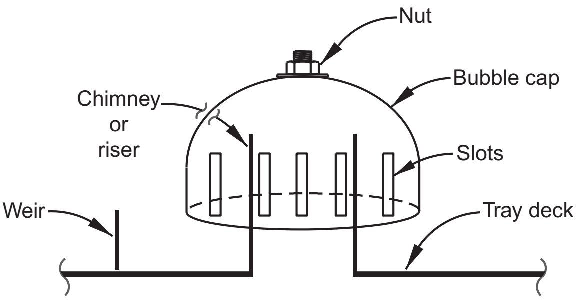

FIG. 2.3 Bubble cap. Note that top of riser is above top of weir to prevent tray deck leakage.

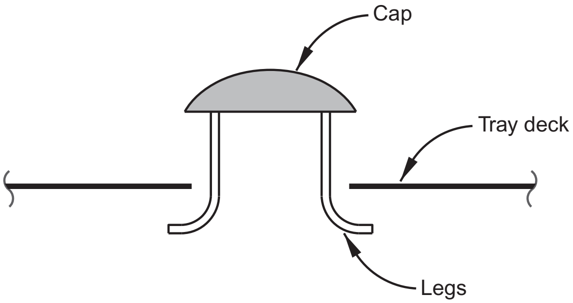

FIG. 2.4 Valve tray is subject to leakage.

The Problem With Valve Trays

I've shown a sketch of a valve tray in Fig. 2.4. The two main problems with this valve cap design are as follows:

- The cap sticks to the tray decks due to salts, corrosion products, and general dirt.
- The cap leaks almost as bad as a sieve tray. Perhaps, this would not be the case if the tray deck was dead level. From my experience in process operations, I have observed that as the vapor velocity (and tray delta P) varies with the vapor rate, so does tray fractionation efficiency.

Optimizing Fractionation Tray Efficiency

There are three paths to optimizing distillation tray efficiency:

- Process design
- Maintenance
- Operations

In this discussion, I am assuming either:

- Sieve trays
- Grid trays
- Valve trays (see Fig. 2.4)
- Tab trays

These are all perforated trays, which excludes:

- Bubble cap trays (see Fig. 2.3)
- Tunnel cap trays
- Shed trays
- Disc and donut type trays

Ninety-nine percent of modern distillation towers contain perforated trays, all of which are subject to tray deck:

- Dumping
- Leaking
- Weeping

The objective of maximizing tray efficiency is to operate the tray to minimize tray deck weeping, but without excessive vapor velocity, which promotes entrainment.

The Nature of Tray Efficiency

Maximizing tray fractionation efficiency is akin to an optimum running program:

- Running 0.1 miles per week will result in obesity.
- Running 8 miles every day will result in knee problems.
- The optimum is 4 miles a day, 5 days a week.

It's a balance between activity and rest.

Tray efficiency is also a balance between:

- Delta P dry
- Delta P hydraulic

Delta P dry is the pressure drop of the flowing vapor, as it accelerates through the tray's perforations. Typically, the hole area of a tray deck is 10%–14% of the area of the tray deck. A typical hole velocity ("V") is 10–25 ft per second.

The pressure drop (Delta P dry) of the vapor passing through the hole on the tray is as follows:

$$\mathrm{DP} = \frac{D_\mathrm{V}}{D_\mathrm{L}} \cdot K \cdot V^2 = \text{Inches of liquid} \tag{2.2}$$

where:

- D_V = density of vapor
- D_L = density of liquid

- K (sieve trays) = 0.3
- K (valve trays) = 0.9
- K (grid trays) = 0.6

The calculated "Delta P Dry" assumes the tray decks are clean. Certainly, in a refinery, this would be quite a poor assumption to make. Most tray decks are encrusted with:

- Iron sulfides
- Ammonium chlorides
- Amine salts
- Coke
- Polymers and gums
- Catalyst fines
- Silicates and carbonates

Or perhaps the trays have a lower than expected Delta P dry because of:

- Valve caps are blown off.
- Tray deck gaps due to mis-assembly of the tray panels.
- Tray decks have slipped away from tray support ring.
- Tray deck corrosion.

The calculated Delta P dry of a tray may not represent reality.

Delta "P" Hydraulic

The idea of tray efficiency is based on maintaining a balance between two forces:

- Delta P dry
- Delta P hydraulic

Excessive Delta P dry promotes entrainment. Liquid droplets from a lower tray are blown forcefully to the tray above. This blows the heavier stuff to the upper tray and reduces tray efficiency.

High Delta P hydraulic promotes tray deck weeping. Liquid droplets from an upper tray leak directly to the tray below. This allows the light components to trickle down onto the tray below and reduces tray efficiency. Delta P hydraulic is calculated by:

$$\text{DP Hydraulic} = \text{Weir Ht.} + \text{Crest Ht.}$$

This may not be exactly true. The problem is foam. Pour out a can of Coke into a glass. Initially, the glass will be full of brown foam. Wait a few minutes. You will now have half a glass of Coke. Why? Because the average density of foam is 50% of the nonaerated liquid. I will divide the weir height by two, to account for foam. This is an educated guess.

Added to the weight of liquid on the tray due to the weir is the "Crest Height." That's like the height of liquid overflowing the edge of a dam. The crest height depends on the rate of liquid overflowing the edge of the weir. That's the "Weir Loading." To calculate the weir loading, determine the flow of liquid in GPM (gallons per minute), overflowing the weir. Next, divide by the weir length (inches):

$$\text{Weir Loading} = \text{GPM}/(\text{Weir Length})$$

To calculate the crest height, in inches of “Clear Liquid”:

$$\text{Crest Height} = 0.4(\text{Weir Loading})^{0.67}$$

This is the “Francis Weir” Formula which is fine for dams, but does not actually apply to distillation tower trays. Why? Because the fluid overflowing the weir is not a clear liquid-like water but a frothy, foamy mixture of vapor and liquid.

I have seen videos of pilot plant distillation trays and have seen that the crest height appears more like several inches of froth. So, the crest height calculation is just a rough approximation of the depth of liquid on the tray, contributed by the weir loading.

To summarize, the weight of liquid on the tray, which I call “Delta P Hydraulic,” in inches of liquid is as follows:

$$\text{Wt.of Liquid} = \frac{\text{Weir Ht.}}{2} + 0.4(\text{Weir Loading})^{0.67}$$

Balancing Delta P Dry and Delta P Hydraulic

Valve caps on trays do not stop tray deck leakage. They serve no purpose and do not act as tiny check valves to retard liquid from leaking through the tray decks. It is the delta P dry, the flow of vapor through the holes on the tray decks, that retards tray deck liquid leakage.

If a tray was truly level, tray deck leakage would be of no great consequence. The drips of liquid would still come into good contact with the rising vapors. But trays in large diameter towers are not all that level. Therefore:

- The liquid will leak through the low points of the tray deck.
- The vapor will take the path of least resistance and rise through the high points of the tray deck.
- The result is vapor-liquid channeling and a loss of tray fractionation efficiency.

Increasing Delta P dry with more vapor flow, generated by more reboiler duty, will offset the tray deck levelness problem. But higher vapor velocities at some point will lift the foam level—or spray height—on a lower tray, to splash the foam onto the tray above. This will also degrade tray fractionation efficiency.

The problem with calculating the optimum balance between “Delta P Dry” and “Delta P Hydraulic” is that we do not know:

- The out of levelness of the tray deck.
- The fouling thickness on the tray deck.

- The out of levelness of the weirs.
- The foaminess of the liquid on the tray deck.
- The actual crest height of the foam above the weir.
- The out of levelness of the downcomers.
- The number of valve caps that are missing or stuck to the tray deck.
- The percent of liquid leaking around the tray support ring.

Optimizing Tray Efficiency—Process Design Perspective

From the process design perspective, optimizing the tray efficiency is pretty simple. Just calculate the required tray deck hole area based on:

- Delta P dry should be about equal to Delta P hydraulic.
- If Delta P dry is less than half of Delta P hydraulic, loss of tray efficiency, due to tray deck leaking, can be expected.
- If Delta P dry is double Delta P hydraulic, the liquid will likely be blown off the tray. Loss of tray efficiency due to jet flood (i.e., entrainment) can be expected.

In the 1950s, when we still used “Bubble Cap” trays (Fig. 2.3), tray deck leakage was not an issue. But with perforated trays (even valve trays), it’s a big problem in that trays in large diameter refinery fractionators are not all that level. And, with time, trays become progressively less level.

Optimizing Tray Efficiency—Maintenance

During a turnaround, we will want to improve two things: tray deck levelness and weir levelness. Tray levelness is best measured with a carpenter’s laser level. A reasonable maximum out of levelness of a tray, that will not affect tray efficiency too bad, is 50% of the calculated “Delta P Dry.” The more level the tray, the better the tray fractionation efficiency.

If Delta P dry is too small, compared to the out of levelness of the tray deck, then I would plug off a certain percentage of the valve caps to increase Delta P dry. This will reduce the tray capacity as limited by flooding. But, that’s a price we pay for enhanced tray fractionation efficiency.

Weirs that are out of level are also a problem. If the “Crest Height” is not high enough to overflow the high end of a weir, the liquid behind this portion of the weir will be stagnant. Areas of stagnant liquid have a zero tray fractionation efficiency. To overcome this weir out-of-levelness problem, we weld picket weirs onto the top edge of the existing weirs. This raises the crest height and promotes more even cross-flow of liquid on the tray deck (see Fig. 2.5).

Picket weirs also reduce tray capacity by increasing the depth of foam on the tray deck. But, that is a price we pay for enhanced tray fractionation efficiency.

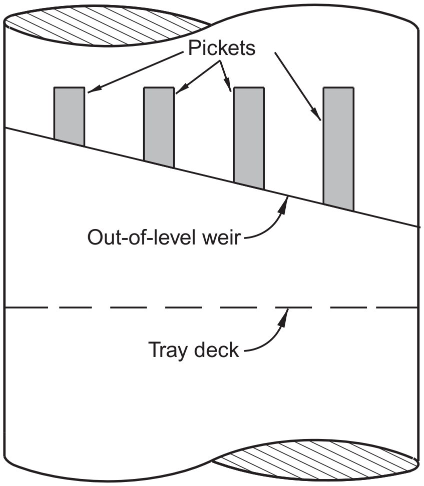

FIG. 2.5 Picketed weirs may improve tray function efficiency.

Optimizing Tray Efficiency—Operations

Returning to the test on the waste water stripper. I was monitoring the efficiency of the stripper's ability to reduce the light aromatic content of the stripped water, by checking the rate of hydrocarbon flow to the boiler at various feed rates. I increased the feed rate from 50 to 60 GPM and reboiler duty from 4000 to 4800 pounds per hour to match the 20% increase in feed.

To my surprise, the fuel gas rate to the boiler increased not by 20%, but by more than 40%. Why?

The ratio of vapor to liquid remained the same. Also, the volatility of the aromatics was unchanged. As a Process Engineer, I would say that the "Stripping Factor" remained the same.

Stripping efficiency is a function of two factors:

1. The stripping factor (see Eq. (2.1)).
2. The number of theoretical stripping trays or the tray efficiency.

The increase in stripping efficiency could only have been a consequence of enhanced tray efficiency. That is, I had increased the tray delta P, which:

- Diminished tray deck leakage.

But

- Without causing excessive entrainment.

The increased stripping steam rate had increased tray DP closer to the optimum. I call this optimum tray DP, the "Incipient Flood Point." Typically, it is a pressure drop, expressed in inches of liquid of about 18%–26% of the tray spacing. Below this DP, perforated trays will leak. Above this DP, trays will suffer lost fractionation efficiency due to entrainment or jet flood.

But, this 18%–26% tray DP target to optimize tray fractionation efficiency depends on the circumstances.

The optimum tray DP depends on the:

- Levelness of the trays.
- Levelness of the weirs.
- Fouling distribution between upper and lower tray decks.
- Downcomer becomes unsealed.
- Tower pressure control stability.
- Flooding being initiated toward the top or bottom of the tower.
- Valve caps being blown off.
- Foaming tendency of the components being distilled.
- Variability in the tower's feed composition.

Process Engineering in Action

What then is the correct way for an operator to adjust his tower to optimize tray efficiency? The answer is by field testing and observation. Based on lab samples, the operator will find, at a fixed ratio of "Reflux-to-Feed," which feed rate results is the best split between the distillate and bottoms products. The unit engineer should then measure the pressure drop across the trays. This is the optimum DP that corresponds to the maximum tray efficiency, which I call the "Incipient Flood Point."

To achieve this optimum tray efficiency, the operator has five handles on the problem.

1. Adjust tower pressure. A lower pressure increases tray DP.
2. Increase feed rate.
3. Increase reflux and reboiler duty. More reflux will require more reboiler duty.
4. Increase feed preheat at a constant reboiler duty.
5. Decrease feed preheat at a constant reflux rate.

The above assumes that the feed composition is reasonably constant.

I've referred to the earlier procedure as "Process Engineering in Action," because it combines field observations with our Chemical Engineering education.

Process Operation is complex. Distillation applications are the most complicated of all process applications. Tray efficiency can best be determined by field testing and observing the relationship between fractionation efficiency and tower differential pressure. And that relationship all depends on the individual circumstances.

The incident I described for the sour water stripper illustrates that the response of a distillation tower to increase reboiler and reflux rates can never be really anticipated, without operating experience on that particular tower. The effect of changes in vapor and liquid rates, on tray efficiency, is too unpredictable and variable. The only answer to this universal process problem is running a field performance test on the sour water stripper and observe the effect.

Bubble Cap Trays

I still use old style bubble cap trays on rare occasions in retrofits. More significantly, in refinery crude distillation units dating from the 1950s, bubble caps were always used. So, I have closed this chapter with a sketch of the archaic tray cap (Fig. 2.3), which cannot leak, and thus will inherently perform with better tray fractionation efficiency (but not capacity) than a perforated tray deck.

3

Function of Reflux in Distillation Towers

"Bobby why are you increasing the reflux on the de-prop tower?"

"Mornin' Mr. Lieberman. I'm increasing the reflux because there's too much C4 in my LPG."

"O.K. that's good Bobby. But how about the reboiler duty. The steam flow to the reboiler is on manual. You'll have to increase the steam yourself. Remember?"

"No Mr. Norm. I only have 2% propane in my butane bottoms product. It's right on spec. I don't need more reboiler steam. Check the 7:30 a.m. lab sample result. I'm fine with that."

"Bobby, we've discussed this before. You can't just crank-up the reflux rate without also increasing the heat to the reboiler. Remember? It was just last week on nights, we talked about this."

"Uh no… maybe you can explain that again."

"That's fine. Let me try to explain it a different way this time. Maybe I should illustrate the point with some examples. I'll explain it to you just the way I explained it to Jay Holcomb to help him with his crude unit and preflash tower. I used an example of an old still in Peru to help Jay. I'll tell you Jay's story and the story of the Peruvian still."

Jay Holcomb had been married twice, had four troublesome grown sons, and was the operations manager of a 30,000 BSD refinery, surrounded by the soy bean fields in southern Illinois.

"Norman," Jay said, "the project you suggested last year has been a disappointment. I diverted the crude flash drum vapors to the new air cooled condenser we installed in the October turnaround (see Fig. 3.1). Diverting the cool 350°F flash drum vapors out of the crude tower flash drum has, as planned, increased the flash zone temperature in the crude unit by 30°F. Also, we've unloaded the crude tower trays and have been able to increase the crude rate by 3000 BSD."

"Jay, that all sounds good. What's the trouble?"

"My trouble is that the light naphtha from the new flash drum vapor air cooler has a long tail. It has a 420°F end-point. That heavy tail is coking up my naphtha reformer catalyst. Also, my youngest son is getting divorced again."

"You should have had daughters. They're a lot less trouble."

"I hope your advice about my high naphtha end-point problem will be more useful. I need to get the ASTM D-86 Distillation end-point below 380°F max."

"Jay, I'll tell you a story that will illustrate the idea that I have in mind," I said.

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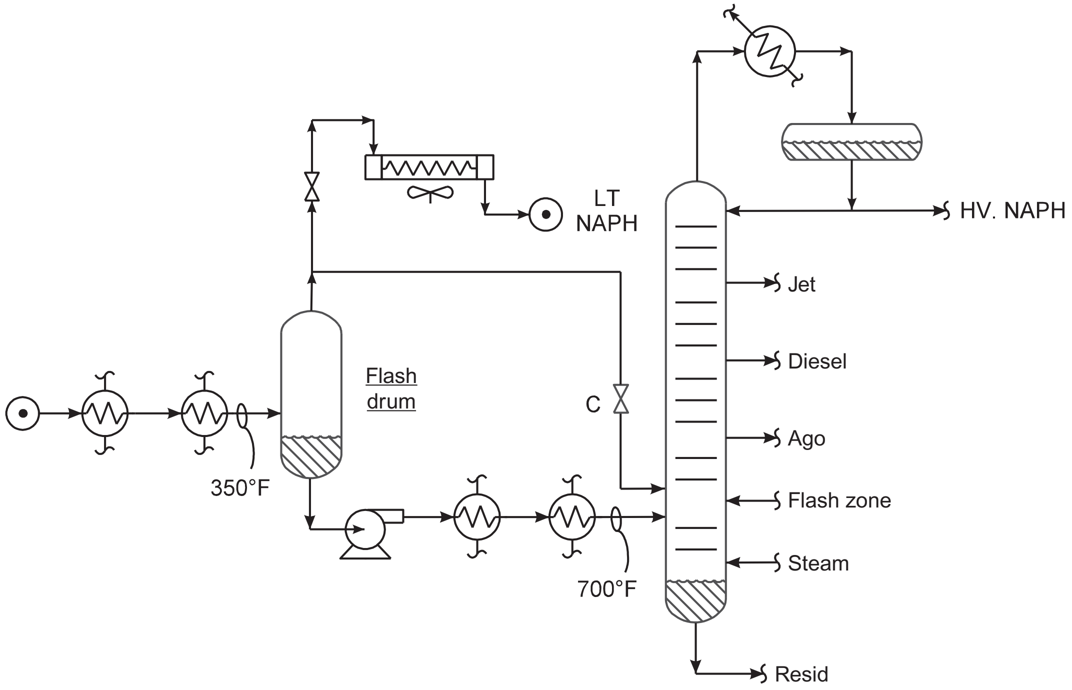

FIG. 3.1 Light naphtha has an excessive end-point.

I was visiting Peru a few years ago. On the road to Machu Piccu, a famous tourist destination, my driver suggested we stop for a tour at a distillery that made Pisco, the main ingredient of “Pisco Sour” the national drink of Peru.

I had drunk “Pisco Sour” at our hotel in Lima. I would normally choose Whiskey but thought I should try the local spirit. It would be interesting to see the distillery, particularly as this one was very old and in the remote countryside.

I was introduced to the lead operator, Juan Ramirez—a middle-aged man who appeared to be of Indian extraction, and very knowledgeable in regard to the distillery, which was indeed very ancient. Hundreds of what appeared to be clay jars or pots were piled in one corner of the site. Huge stacks of firewood in another area. Several old Ford trucks were lined up in front of the plant. And then there was the Still.

The Still was a huge copper pot, 8 ft wide and 5 ft high. It was set on a red brick structure above a brick-lined chamber into which firewood was being fed. A long coil of copper tubing condensed the vapors from the Still, which drained the condensed Pisco vapors, into a man-sized vertical copper pot. This second pot was elevated 16 ft above the top of the Still (Fig. 3.2).

Pisco is 25%–30% alcohol. Whiskey (100 proof) is 50% alcohol. At this distillery, on the road to Machu Piccu, they measured the alcohol content of the Pisco with a float.

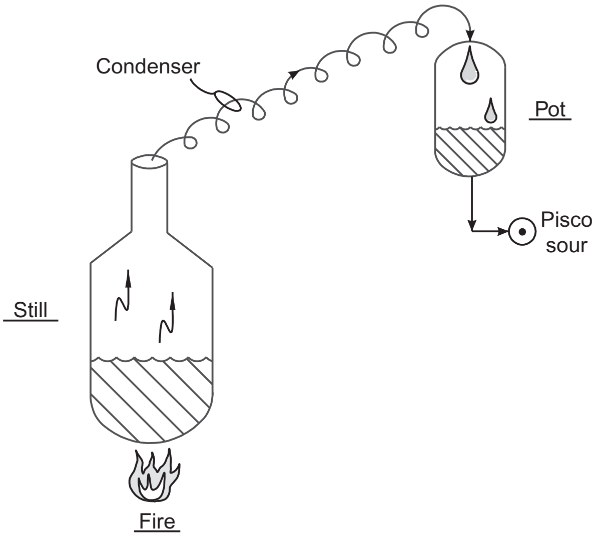

FIG. 3.2 Ancient still. Yield reduction for increased product purity.

If the Pisco was too weak, the line on the float would be above the liquid level in their test glass, indicating too high a water content. Meaning the Pisco density was too high.

"So Juan, what do you do to bring the Pisco up to 30% alcohol, if it's too weak?" I asked.

"So maybe, next batch, we don't cook it so hard. Use less wood. So, juice it cooks slower and colder. Mucho caliente makes Pisco too weak. Or maybe Señor, I put weak Pisco back into the Still, heat it up a second time. The second trip through our Still will make Pisco muy strong," Juan explained.

For me this is what Process Engineering is all about. It's that interface where I meet the plant operator. It's where we resolve our understanding of a problem using applied technology. Albert Einstein once said that which made America great is not freedom, or equality, or democracy. He said that America is the great land of applied technology.

Vapor Pressure Limitations

"Juan," I asked, "Why do you sometimes cook the juice too hard if it makes weak Pisco? If it vaporizes too much water with the alcohol?"

"Because Señor Norm, we make mucho dinero, by making more Pisco product."

This is the original problem of chemical and process engineering—the balance between product recovery and product purity. To keep his product on-spec, at a minimum of 30% alcohol, Juan had to reduce his product recovery by burning less firewood underneath the Still.

The problem is that alcohol has a large vapor pressure and water has a smaller vapor pressure. When the Still is fired harder, the vapor pressure of the water increases faster than the increase in vaporization rate of the alcohol from the fermented juice. So while the amount of juice vaporized increases, the amount of water vaporized increases faster, because the vapor pressure of the water is increasing at the hotter temperature in the Still.

Reducing the Still temperature reduces water vaporization, but also reduces the Pisco production. But, that's the price paid by Juan for making the Pisco stronger to meet his alcohol spec. It's a compromise between product yield and product purity.

I explained the concept of vapor pressure to Juan Ramirez, but perhaps I didn't offer a good explanation, as Juan not only didn't really grasp the concept, but began to show disinterest, and even possible signs of hostility!

So I said, "Maybe we can fix the problem Señor Juan, by using process engineering technology. Shall I explain?"

And Juan said, "No!"

So I explained anyway.

Two-Stage Distillation Process

"Señor Juan, we are going to create a two-stage distillation process using a partial condenser that will run continuously, process the Pisco all the time on-spec, and cost almost nothing to install. We can do it today."

When I said that we could do it today and that it would cost nothing, Juan smiled and said, "Bueno! Si! Si!"

- *Step One*—I'll cut the long coil of copper tubing in half.
- *Step Two*—The first half, I'll use to connect the Still to the vertical pot (Fig. 3.3)

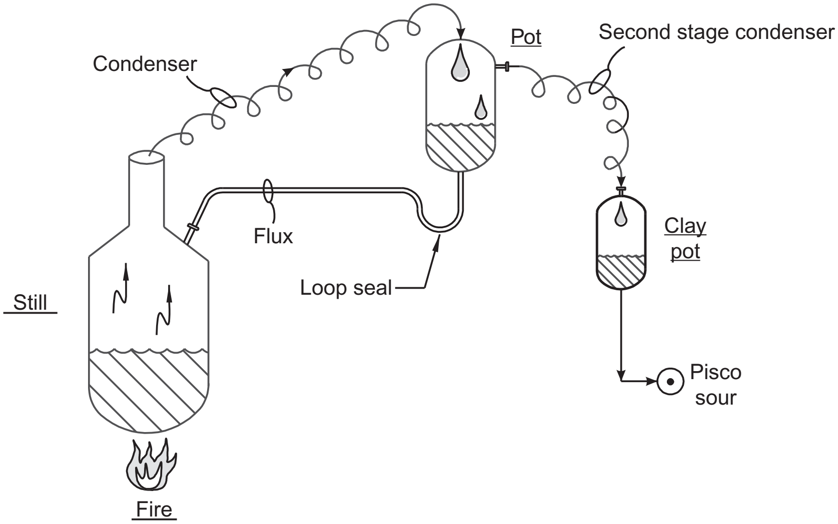

FIG. 3.3 Two-stage distillation apparatus increases purity without reduction in product yield.

- *Step Three*—The second half I'll leave connected to the top of the vertical pot and have it drain, by gravity, into one of those clay pots.
- *Step Four*—The liquid, which condenses (called "FLUX"), that accumulates in the vertical pot, I'll use a rubber hose with a 2 ft long loop, to continuously drain back into the Still (see Fig. 3.3).

"Señor Juan, the liquid that drains back into the Still will be re-vaporized in the Still, and is called flux or "Reflux." The purpose of the loop is to keep vapors from the still from backing-up into the clay pot's liquid drain line, which would stop the liquid draining through the hose. That is, the vapor would push the liquid back into the pot."

"But Señor Norm, suppose the Pisco is too weak. What should I do then?"

"Juan, just spray some water on the copper tubing between the Still and the vertical pot." That coil being like a first stage condenser, spraying water on the coil to remove more heat will make more flux, but less product.

"You see Juan, I'm just guessing about reducing the size of the existing first stage condenser (Fig. 3.2), when I said to cut the long coil of copper tubing by half. Maybe it should be a little bit longer to generate the right amount of reflux. But using an external source of cooling, the water spray, will allow you to compensate for lack of cooling by the air."

"But Señor Norm, suppose the Pisco becomes to strong! What should I do then?"

"Juan, just add more firewood to the Still. That will make more Pisco product too, by distilling over some extra water."

Origin of a Distillation Column

The technology of the partial condenser to produce a very strong alcoholic beverage originated in Arabia around AD 1000. The origin of the multistage, trayed distillation tower which we still use today, originated in Scotland in 18th century. It's called the "Patent Still." It had a reboiler, condenser, six bubble cap trays, and a feed preheater. I saw one being used to make apple brandy on a farm in the county of Somerset in England just a few years ago. It worked just fine.

The only things that have changed in distillation since the 18th century is that we now call "Flux," "Reflux," the Still now has a separate reboiler, and we use valve trays, instead of bubble cap trays.

My idea for updating the Pisco production technology was not original. I reproduced the small, two-stage, home-made distillation apparatus I had seen in a store front window of a liquor shop in Vilnius (Lithuania), a few years before my visit to Peru.

Removing the Naphtha's Tail

"Jay, we can do the same thing on your unit. Shut off three of the four fans on the air cooler. Then, we'll take the liquid, that was partially condensed in the air cooler outlet header box, and drain it back into the flash drum. Kind of like reflux. The remaining naphtha vapor, we

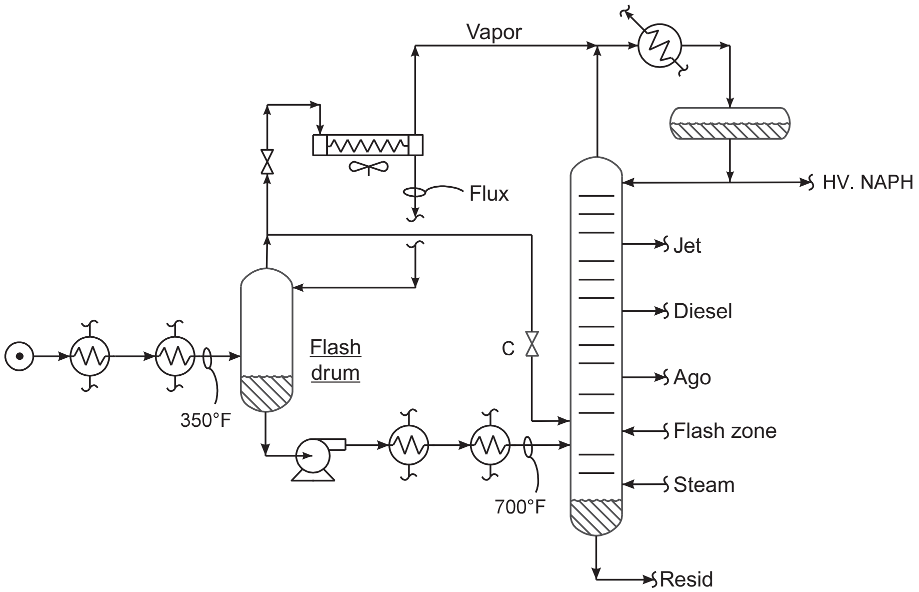

FIG. 3.4 The air cooler is acting as a theoretical separation stage.

can route to the overhead of the crude tower (Fig. 3.4) and comingle it with the crude tower overhead naphtha. The air cooler will be like the first stage condenser in Peru, which is also working like a theoretical, fractionation stage."

"But Norman. I'm going to wind-up with less naphtha from the preflash tower, by using the air cooler as a partial condenser. I don't know if I like that." Jay said.

"It's all a balance between product yield, and product purity. You can't have it both ways Jay. It's kind of like being married. You have to trade a part of your freedom for companionship."

"Ah yes Mr. Lieberman. I see it now. Wisdom comes with age." Said Jay, nodding his head in agreement.

Conclusion

"You certainly are quite the world traveler, Mr. Norm. Imagine you working all the way down in Peru," Bobby said, after listening to my story.

"Yes, Bobby. Just like Jay's story, the travelling has its good and not so good points. But the point of this story, is that the REFLUX COMES FROM THE REBOILER. If you increase the tower top reflux flow, to reduce the C4's in your LPG Bobby, the reflux drum is going to go empty, unless you refill it with the vapors generated from the steam reboiler. The reflux pump is going to lose suction and cavitate, and you'll tear up the pump seal again due to vibration."

"I sure am going to remember your story, and add steam to the reboiler next time I add reflux to the depropanizer. But Mr. Lieberman, wouldn't it be best to get the temperature control loop fixed on the reboiler steam, so it will add the steam automatically? It won't be that expensive to fix."

"Uh Bobby. We'll have to discuss that another time. I have to go to an important meeting at 9:30 this morning, I'd better get going."

4

How Distillation Trays Work

Distillation was invented before trays. It relied on flash drums (Fig. 4.1). The drums separated liquid from vapor, with no entrainment. Each drum was called, "A Theoretical Separation Stage." If the separation of vapor and liquid phases was perfect, the drum had a separation efficiency of 100%. The more drums in series, the greater the degree of separation that could be achieved between products. To avoid the mechanical complexity of many drums, the distillation column was invented.

A tray replaces a drum. If the tray really is working as well as a drum, it has 100% tray fractionation efficiency. If the tray is only working half as well as the drum, it is said to have a 50% tray efficiency. The usual cause of low tray efficiency is entrainment due to excessive vapor velocity.

As the distance between the liquid level on the tray below and the tray above gets smaller, it takes less vapor velocity to blow the liquid from the lower tray, up to the tray above.

For example, when separating propane from butane, the propane product has a butane specification of 2%. If the vapor rate flowing up through a tower increases, the droplets of liquid from a lower tray will be entrained and blown up to the tray above. As the liquid in the lower tray is rich in butane, the entrained droplets will increase the butane content of the upper tray. As the entrained droplets reach the top tray, the overhead propane product will be contaminated with butane, thus reducing fractionation efficiency.

A tray is intended to mix vapors and liquids together. The better the mixing, the more efficient the tray works. Tray efficiency is also a function of:

1. How well the droplets of entrained liquids are separated from the vapor's flow rising from the tray deck.
2. How well the bubbles of vapor are separated from the liquid flowing down from the tray deck through the downcomer.

Entrainment and Tray Efficiency

The main factors controlling entrainment are as follows:

1. Droplet size—the smaller the drop, the greater the rate of entrainment.
2. Viscosity of the continuous phase. High viscosity promotes entrainment.
3. Density difference between the vapor and liquid phases. Reduced density difference promotes entrainment.
4. Velocity—higher velocities promote entrainment as well.

These factors are summarized in "Stokes Law".

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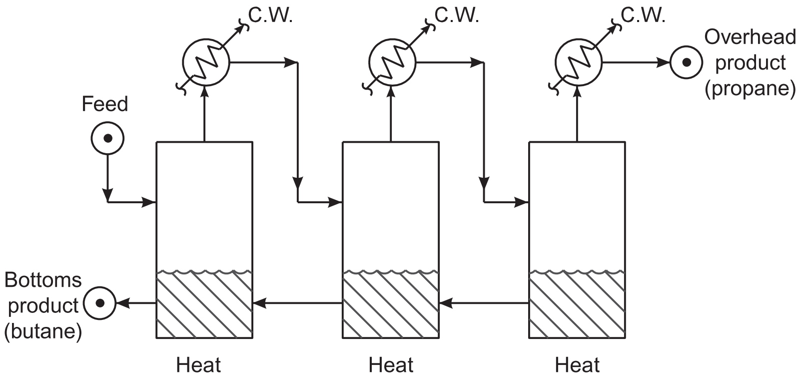

FIG. 4.1 Primitive distillation unit, three stage.

$$\text{Entrainment Proportional to } V \cdot (D_V \div D_L)^{1/2} \tag{4.1}$$

where:

- V = Velocity of vapor phase
- D_V = Density of vapor phase
- D_L = Density of liquid phase

The droplet size is usually unknown. Viscosity of a vapor phase is usually small and normally we neglect it. By small, I mean less than 0.3 cSt.

To minimize entrainment is to avoid high localized velocities of the vapor phase.

When the spray height from a lower tray reaches the tray above, tray fractionation efficiency is degraded, due to entrainment.

Effect of Foam

The fluid on a distillation tray deck is not actually liquid, but foam or froth. The vapor bubbling up through the holes on the tray deck mixes with the liquid from the tray above, to form foam. The foam overflows the weir (see Fig. 4.2) into the downcomer. But, if the downcomer is too small, then the foam fills up the downcomer, and backs up onto the tray above. The foam level on the tray then increases. As the froth, or foam level, increases on the tray, so does the height of the entrained liquid, or spray height.

Benefits of Reflux

More reflux improves tray fractionation efficiency, due to the increased vapor and liquid traffic on the tray deck. The problem is that the reflux comes from the vapor generated in the reboiler. If there is too much vapor flow through the tray decks, then the vapor velocity may become excessive, and this will cause excessive spray height or entrainment.

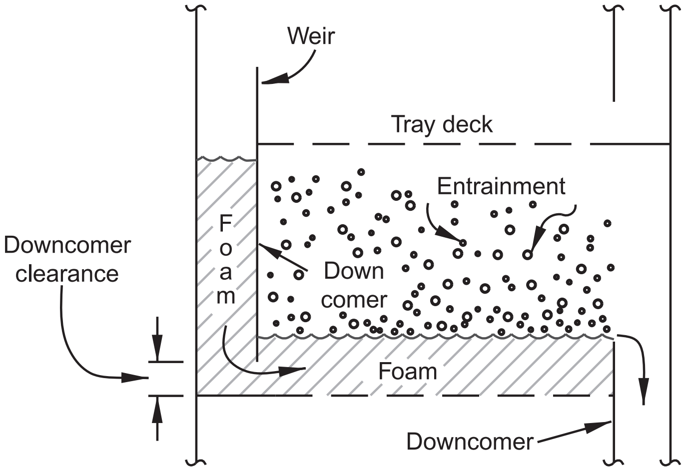

FIG. 4.2 Distillation tray. Entrainment reduces tray fractionation efficiency.

Excessive reflux and reboiler duty, will make fractionation worse, due to entrainment. One cannot increase the reflux rate without more reboiler duty. Reflux is condensed vapor generated from the reboiler that flow up through the trays, overhead condenser, and into the reflux drum.

Downcomer Back-Up

Downcomers often back-up onto the tray above and cause a high foam level on the tray. As a consequence, there may be entrainment of droplets of liquid to the tray above.

Normally, it's not the downcomer that has too small an area. Usually, the downcomer is too short. The height of foam in the downcomer is mainly a consequence of the pressure drop of the vapor flowing through the tray above the downcomer, as shown in Fig. 4.2.

As the vapor flows through this tray, it undergoes a pressure drop due to:

1. The restriction of the caps on the tray deck.
2. The weight of the liquid on the tray deck.

The pressure of the vapor below the tray pushes up the foam height in the downcomer. Once the downcomer becomes full of foam, the foam backs-up onto the tray above that downcomer. The extra weight of foam on the tray deck then increases pressure drop of vapor through a tray deck. This increases the force which is increasing the liquid level in the downcomer.

The problem feeds upon itself. Until the tower's trays, above the tray where the problem began, start to operate in a region called "*Fully Developed Flood*."

A tray downcomer can also fill with foam because it fails to drain fast enough. It is most often caused by a too small a downcomer clearance (see Fig. 4.2). Downcomer clearance is usually 2–3 in. Any clearance less than 1½ in. is likely to cause downcomer flooding and back-up on the tray above.

If downcomer clearance is greater than the tray's outlet weir height, the downcomer will also back-up. Vapor will blow into the bottom of the downcomer and displace the liquid flow. Fully developed flood will then result, due to this downcomer unsealing.

Keep in mind that the downcomer is full of frothy foam, and not the more dense liquid phase. Hence, it is rather easily displaced by the vapor flowing up through an unsealed tray downcomer.

Tray Efficiency

Certainly, tray fractionation efficiency is reduced due to entrainment, which is also called, "Jet Flood." However, most tray efficiency problems are caused by tray decks that are not level. The liquid on the tray deck may leak through the perforations on the trays, in those areas which are lowest. The vapor flow will be concentrated at the highest tray deck areas. This all results in vapor-liquid channeling, which is bad for tray vapor-liquid contacting efficiency.

The "Out of Levelness" of a tray that will cause a large loss in tray fractionation efficiency is related to the pressure drop of the vapor flowing through the tray deck holes or valve caps. If the low points of a tray (i.e., portions of the tray deck below the tray support ring) are less than the pressure drop of the vapor flowing through the tray deck perforations, then the liquid will leak through the tray deck and fail to come in good contact with the vapor flow.

To calculate the pressure drop of the vapor through the tray perforations:

$$\mathrm{DP} = K \cdot \frac{D_\mathrm{V}}{D_\mathrm{L}} \cdot (V)^2 = \text{Inches of liquid} \tag{4.2}$$

where:

- D_V, D_L = density of vapor, liquid
- V = hole velocity, feet per second
- $K = 0.3$ (sieve trays)
- $K = 0.9$ (valve cap trays)
- $K = 0.5$ (venturi-type valve trays)
- $K = 0.6$ (grid type trays)

Valve caps do not appreciably retard the tendency of liquid to leak through the tray deck panels. Bubble cap trays would retard the liquid leakage through the tray decks, but are not used in a modern distillation service. There is no standard "K" value for bubble cap tray decks.

Bubble Cap Tray Decks

Valve caps are supposed to act as check valves to retard liquid leakage through tray decks. This appears not to be the case. The valves do not seem to appreciably reduce tray deck leakage. What they certainly do in refinery dirty services is to stick to the tray decks. This restricts vapor flow and pushes up the foam level in the downcomer, which causes the tray to flood.

My practice is to use a grid-type tray. This is like a valve tray, but the cap is in a fixed position. It's more or less the same as a sieve tray, but with about 10%–15% more capacity than the sieve tray.

Bubble cap trays avoid potential tray deck leakage problems. They are, however, expensive, hard to clean, and have about 10%–20% less vapor handling capacity than a modern grid tray and slightly less capacity than a half-inch sieve tray deck.

5

De-entrainment of Vapors

It's raining and windy in England as I write. I went for my four-mile run anyway. The wind, which was blowing at about 15 miles per hour, blew the drizzle of rain drops into my face. The wind was entraining the rain drops, overcoming the force of gravity, or the settling velocity of the rain.

I do process calculations when I run. I decided to calculate the entrainment factor of the droplets of rain on my windy run. First, I calculated the density of air at 60°F and atmospheric pressure. At these conditions, one mole of air occupies 379 cubic ft. The molecular weight of air = 29 pounds. Therefore:

$$29 \div 379 = 0.076\,\text{lbs per cubic ft}$$

$$\text{The density of water at } 60^{\circ}\text{F} = 62.4\,\text{lbs per cubic ft}$$

According to George Stokes, the tendency of droplets of liquid to be entrained by a flowing vapor is proportional to:

$$(\text{Velocity of vapor}) \times (\text{Density of vapor}/\text{Density of liquid})^{½}$$

$$\text{The velocity of wind at 15 MPH is} = \frac{(15) \times (5280\,\text{ft})}{3600\,\text{SEC}} = 22\,\text{ft per second}$$

The tendency of droplets to be entrained by a flowing vapor is called the "*C*" factor. The greater the "*C*" factor, the harder it is for droplets of liquid to settle due to the force of gravity. In this case:

$$\text{“}C\text{”} = (22) \times (0.076/62.4)^{½} = 0.75\,\text{ft/s}$$

That's a big "*C*" factor. So, I wasn't surprised to have the rain blowing into my face.

Caution! If you are a resident of a planet with a different gravity than earth, guidelines in the next section will not apply. Also, if the droplets are a mist, created by reaching sonic velocity in process heater tubes, and then dropping to a subsonic velocity, the correlations discussed below are not applicable. This is a big problem in producing clean gas oil products in refinery vacuum towers, as an unstoppable entrainment of asphaltenes results.

Liquid-Liquid Separation

I started work as a process design engineer for American Oil in 1965. In the American Oil design manual I was issued, it noted, "Settling rate for water in hydrocarbon liquids is 25 ft per hour." So that day, after I went home, I filled a one-foot long bottle with kerosene. Then, with an eye dropper, timed how long it required for a droplet of water to fall to

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the bottom. About 30 s—which is about 120 ft per hour, not the 25 in my American Oil design manual.

I then mixed a few ounces of kerosene into my bottle and shook it violently. It required about 5 min for the two phases to separate completely.

I suppose it's rather like the tendency of wind to entrain the rain. If the drops of rain are large, they would fall vertically in a moderate wind. But, if the rain is falling as fine drizzle, a slight breeze will cause the tiny droplets to be blown in my face.

Last year, I collected a sample of water from the reflux drum of a refinery fractionator. The sample was clouded with a hydrocarbon haze of naphtha. It required about 2 h for the naphtha to float to the top of my graduated cylinder. However, when I added a few drops of acid to my sample, to reduce the pH from 8 to 6, the time for phase separation decreased from roughly 2 h to 30 min.

Once, I had a stream of 10% heavy naphtha and 90% water. When I pumped the mixed phase with a reciprocating pump, its effluent required 2 min to separate. When I pumped the stream with a centrifugal pump, it required 2 h to settle the resulting tight emulsion.

Liquid-liquid phase separation is a complex business, as we will see in this chapter.

Vapor-Liquid Separators

When I calculate the settling rate of droplets of liquid from an ordinary vapor phase, I will make three assumptions:

1. The droplets are not a finely divided mist.
2. The vapor phase has a very low viscosity (i.e., less than 1.0 c.s.).
3. The vapor phase density is very small compared to the liquid phase density.

On this basis, I use the following correlation:

$$C = V \cdot (D_V/D_L)^{0.5}$$

where:

- C = entrainment factor.
- V = vertical velocity of vapor phase, ft/s.
- D_V = density of vapor phase.
- D_L = density of liquid phase.

This relationship is universal in the process industry. However, the values for the "C" factors, shown below, reflect my own operating experience:

1. "C" less than 0.10 = Droplets will readily settle out by gravity.
2. "C" between 0.10 and 0.15 = If inlet velocity is low (20–30 ft per second) and there is a vertical height to vapor outlet of 4–6 ft, reasonably good settling of liquid.
3. "C" between 0.15 and 0.20 = Addition of a tangential baffle or vapor horn, in addition to items in #2, above is needed (Fig. 5.1).

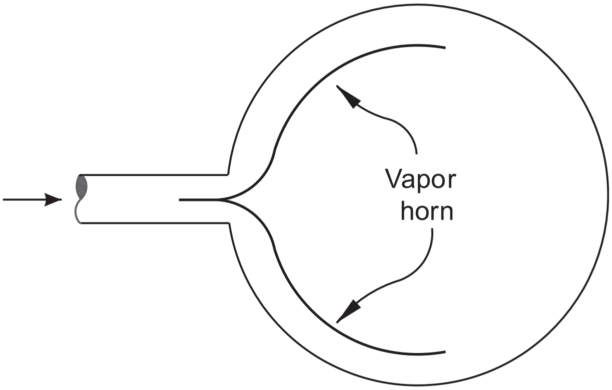

FIG. 5.1 Vapor horn promotes better phase separation.

4. "*C*" between 0.20 and 0.25 = Often, a 6-in. demister is recommended. This is in my experience a bad practice in most refinery applications. The problem is salts or iron sulfide corrosion products fouling the demister. The pressure drop across the demister increases until it tears away from the vessel wall. High localized vapor velocity results which promotes entrainment. It's best not to use a demister in any potential fouling or corrosive services, but to design for a "*C*" factor of less than 0.2 ft per second.

Vortex Tube Clusters

In the past few years, in many large vapor-liquid separators, I no longer worry about entrainment or "*C*" factors. I now use a "Vortex Tube Cluster" device. Droplets of liquid are separated from the vapor phase by centrifugal force—much like a cyclone separator or centrifuge. This device, which I use in Crude Oil preflash towers, has worked really well. It's purchased as a package that fits into existing vessels (E.G. Systems, Houston, Texas).

Wash Oil Sprays

One common method to suppress entrainment of liquids and/or particulates in a flowing vapor at high "*C*" factors is to use a spray chamber. The clean sprayed liquid will wash out entrained droplets of contaminants.

The spray nozzles cover a 120 degrees spray angle with a full cone of wash oil liquid. It's quite effective up to a "*C*" factor of 0.35 ft per second. The 120 degrees spray angle tends to collapse at a pressure drop of less than 6 or 7 psi. At a high pressure drop, the spray will turn into a mist which will be entrained. For a small spray nozzle (less than 1 in.), this happens at a nozzle pressure drop of more than 50 or 60 psi.

Vapor Distribution

For a small vertical vapor-liquid separator, 3 or 4 ft in diameter, I don't worry too much about distributing inlet vapor flow. But, I once designed a 26 ft diameter vessel for Chevron with a 30-in. inlet nozzle. In this case, I used a chimney tray to distribute the vapors across the diameter of the vessel with a small (i.e., 1 in. of water) pressure drop of the vapor flowing through the 6-in. diameter chimney.

An inlet vapor velocity of 20–30 ft per second can be ignored. On the other hand, an inlet vapor velocity of 100–120 ft per second requires some type of distribution device to dissipate the momentum of the vapor and prevent high localized vapor velocities which cause entrainment.

Settling Water Out of Hydrocarbon Liquid Phase

I spent a large portion of my time worrying about this problem. Mainly because refluxing water into a distillation tower causes:

- Corrosion
- Fouling
- Flooding due to foaming
- Off-spec products

Leaving water in the feed to hydrocarbon distillation towers will cause the same problems. I am worrying about a design for this particular problem today, when I am supposed to be on vacation with my wife.

I design drums settling water to be in a horizontal position as shown in Fig. 5.2. The hydrocarbon liquid hold-time is 3–5 min. Water settling velocity I use for a design is 30 ft per hour. Usually it can be twice that velocity and still achieve a reasonable water removal rate so that the hydrocarbon liquid phase will not have more than 2000 or 3000 ppm of water.

Coalescers

If the hydrocarbon viscosity is greater than 10 or 20 cSt, then water settling is impeded. Perhaps the water is emulsified in the hydrocarbons, because the phases have passed through a centrifugal pump together, or through a control valve with a big delta P (i.e., over 20 or 30 psi). Then, settling velocity may be really slow. Sometimes the water must be allowed to settle out in a tank for a day or two. Another common solution is the use of a mesh-type coalescer.

Usually, the mesh is packed into tubes the shape and size of fluorescent lights. These tend to plug rather rapidly and I dislike them in anything but clean services free of salts, particulates, and iron sulfide corrosion products.

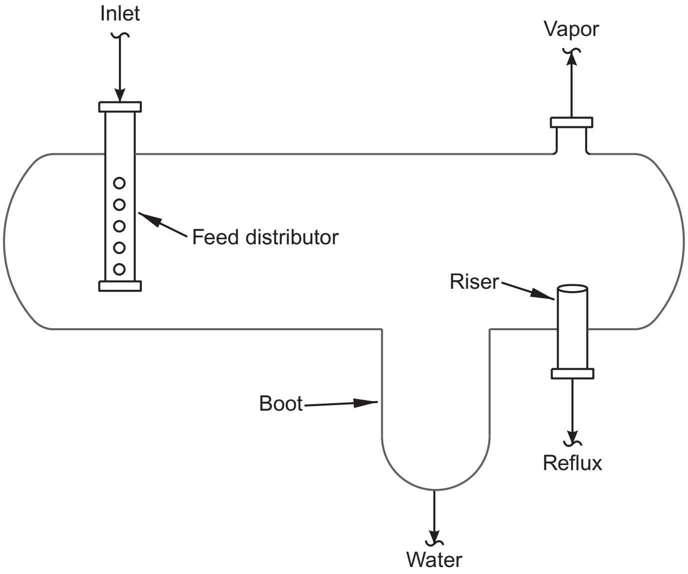

FIG. 5.2 Reflux drum.

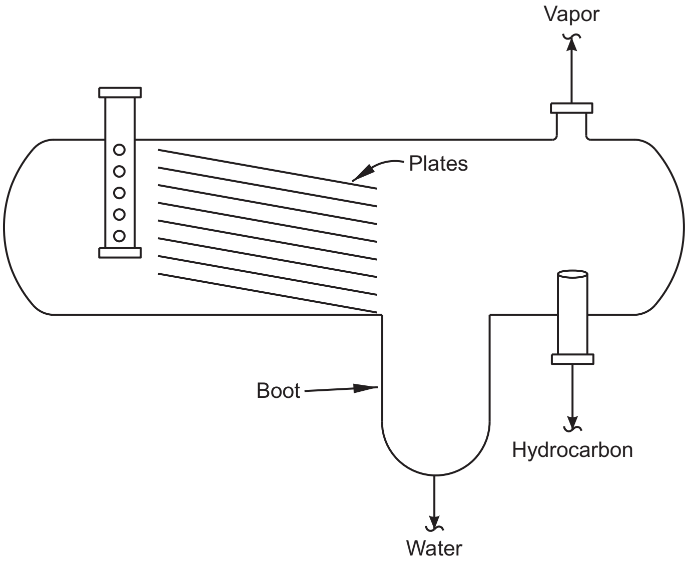

FIG. 5.3 Plates promote water coalescing.

A better choice is shown in Fig. 5.3. This is a plate coalescer. I copied my last design from the Internet. The plates are set at an angle of 10–15 degrees, and about 1 in. apart and span the diameter of the vessel. The plates work to shorten the distance that the droplets of water must settle (i.e., 1 in. instead of several feet). Once the water droplets reach the sloped metal plates, they coalesce and rapidly run down the sloping metal plates into the boot.

Foam

The formation of foam or froth is typically a consequence of particulates in a vapor-liquid mixture. Fig. 5.4 shows three split liquid levels in the gauge glasses. This is a definite indication of low density foam in the vessel. The liquid observed in the gauge glasses is not a measure of the level in the vessel. They are a measure of the density of the foam in the vessel, as compared to the density of the clear liquid in the gauge glasses.

Once the foam level rises to the inlet nozzle, then the liquid will be blown out of the vessel. One way to suppress foam is to use a silicon defoaming chemical. But, this often creates downstream problems with a reactor's catalyst. The better way is to remove the source of corrosion that is creating the particulates, which cause the foaming. Better to cure the underlying problem, than just treat a symptom.

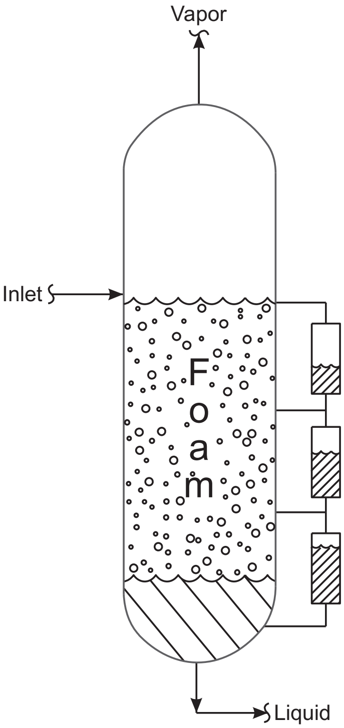

FIG. 5.4 Split liquid levels indicate foam.

Summary

It may seem to you that this chapter is a messy mixture of:

- Plant observations
- Child-like experiments
- Design rules of thumb
- Chemical Engineering principles

And that is the case. I am still fascinated by blowing soap bubbles because it is an experiment in surface tension. I love to watch a pot of boiling water boil over when the starch particulates are added from the pasta. I am always thrilled by swamp coolers that convert the sensible heat of air to the latent heat of evaporation of water. Refinery Process Engineering is all the same fun stuff but on a bigger, more dangerous scale.

6

Fractionator Wash Oil Grids

The annual NPRA Q&A session was being held in New Orleans. I was attending the evening gathering. I live in New Orleans, close by the convention center, and I had just met Pierre Tardo, one of my former Troubleshooting Seminar students from Saudi Arabia. Pierre had a question for me relating to the design of a wash oil grid. The function of the grid is to suppress asphaltene entrainment from the flash zone and to reduce contamination of the heavy gas oil product.

"Norm, what is the optimum depth of a vacuum tower wash oil grid?" Pierre asked. "Also, does the same criteria apply to a delayed coker fractionator?"

"Pierre," I replied, "the optimum depth of packing for a wash oil grid is zero. That applies also for:

- Vacuum towers
- Crude flash zones
- Coker fractionators
- Visbreaker fractionators"

"Zero!" Pierre responded in surprise. "But how could that be? Wash Oil grids are used in refinery vacuum towers and delayed coker fractionators all over the world. No, Norm. I can't agree."

"Well Pierre, I'd better explain. Let's get a drink and a plate of shrimp. I'll tell you two stories. One from the old Hevron plant in El Segundo, California. And the second story from the new Sosal Refinery in Saudi Arabia."

Delayed Coker Wash Oil Section

This 65,000 BSD coker fractionator had a long history of its wash oil grid coking (see Fig. 6.1). The result was black Heavy Coker Gas Oil Product. The Gas Oil would plug the feed filters at the Gas Oil Hydrotreater. Back-flushing of these filters became so frequent that it would force the coker to shutdown to renew the wash oil grid.

"Norm, it seems like they could've just used more wash oil to clean up the coker gas oil, rather than shutting down the coker? Shutting down a 65,000 BSD coking unit – that's like a million dollars a day lost!"

"Well Pierre, once the pressure drop, in inches of water per foot of packing, exceeds about two or three, the wash oil can't penetrate through the packing. It's just re-entrained by the vapor flow. With zero wash oil draining through the grid, the lower portion of the grid dries out. The grid cokes up. Then the grid pressure drop will increase exponentially."

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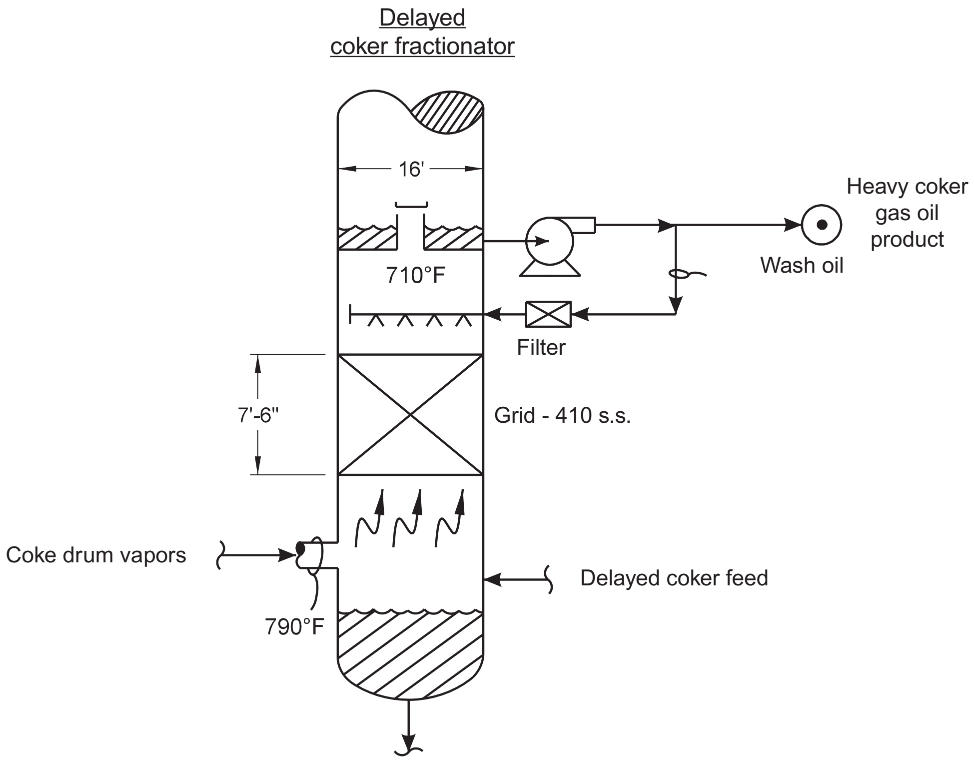

FIG. 6.1 Grid depth excessive. Optimum is zero.

“That sounds really bad. I guess the con carbon in the heavy coker gas oil also increases exponentially?”

“Exactly so, Pierre,” I agreed.

Excessive Coker Grid Height

The Hevron coker wash oil grid was:

- 7′-6″ high
- 410 s.s.—Heavy Duty Grid

This would produce two theoretical fractionation stages. The coke drum vapor enters the fractionator at 790°F (Fig. 6.1) and leaves the grid at 710–750°F.

The larger the wash oil rate, the bigger the vapor temperature decline. A taller grid also contributes to a bigger vapor temperature drop because the taller grid promotes:

- Improved vapor-liquid contacting
- Increased rates of the wash oil vaporization.

- The latent heat of vaporization of the wash oil comes from the sensible heat—or temperature—of the flash zone vapors.

For example, cooling off one barrel of vapor from 800°F to 700°F will evaporate about 0.60 barrels of wash oil. So, if you used 15,000 BSD of wash oil, and the total of the products was 20,000 BSD, 80% of the wash oil would evaporate inside the wash oil grid:

$$(20{,}000)\cdot(0.60)\div(15{,}000)=80\%$$

If you used only 13,000 BSD of wash oil, then 93% of the wash oil would evaporate inside the wash oil grid.

Wash Oil Distribution Problems

Pierre interrupted my story. “Okay Norm, I understand your point that a taller wash oil bed would promote greater wash oil evaporation. Also, an increased tendency to dry out the bed. But, this could be offset by simply using some extra flow of wash oil. After all, the incremental wash oil would not be lost. It would just be re-vaporized and recovered in the gas oil pumparound.”

“You’ve missed the point, Pierre. You’ve assumed that the wash oil is perfectly distributed. Because of problems with the wash oil spray distributor nozzles, if an average of 90% of the wash oil is vaporized, then in some portions of the grid, 80% will be vaporized, and in other portions, nearly 100% will be vaporized.”

“What sort of problems with the wash oil nozzles have you experienced, which would cause a liquid distribution problem?”

“Pierre, I’ve got lots of experience in answering your question:

1. Nozzles plugged because of holes and/or mis-assembly of the wash oil filter baskets.
2. Corrosion products formed in the piping between the filters and the nozzles.
3. Coking, due to a loss of flow during a power outage, of the nozzles.
4. Too much wash oil hitting the vessel wall.
5. No overlap on the spray pattern on the top of the grid.
6. Spray nozzles’ internal defective.
7. Spray header piping flanges have parted.
8. Spray header distorted due to a pressure surge pushing up the wash oil grid.
9. Broken green glass from a beer bottle lodged in spray nozzles.”

“Hey Norm, hold up! Broken beer bottle glass – really?”

“I’ve seen it myself at the Coastal Plant in Corpus Christi.”

The point I explained to Pierre, that the more efficient the vapor-liquid contacting in the wash oil grid, the more concentrated the effect of top liquid maldistribution would be at the bottom of the grid.

“Best to avoid the problem Pierre, by not having a wash oil grid. Best just to rely on gravity settling.”

Selecting "C" Factor for Gravity Settling De-entrainment

If the vapor velocity is low enough, entrained droplets of liquid will settle out just due to the force of gravity. The tendency to entrain droplets of liquid is proportional to the "*C*" factor:

$$\text{“}C\text{”} = (\text{Vertical velocity of vapor, ft/s}) \times (\text{Density of vapor} \div \text{Density of liquid})^{1/2}$$

For gravity settling to be effective, my experience indicates a "*C*" factor of less than 0.15 ft/s is required. The "*C*" factor in the Hevron, El Segundo coker fractionator was 0.35 ft/s. Therefore, a wash oil grid was definitely required. But I thought, "the 7′-6″ grid height was excessive." So, during the 1995 turnaround, I convinced Hevron to reduce the height to 6′-0″, which helped extend run lengths. In 1999 the grid height was reduced to 4′-6″. And finally, in 2003, to 36″, which is close to optimum.

Then Hevron decided to expand the coker from 65,000 BSD to 85,000 BSD. I was given the contract to design the new fractionator, to replace the existing 16′-0″ ID tower. I could have designed a 19′ tower with a wash oil grid. But, I decided to design a 26′ ID tower without a wash oil grid and rely only on gravity settling.

Pierre was startled by the huge diameter of the new coker fractionator. "Norm, that's a real expensive tower."

"What's the difference? It's not my money. The main thing is that my client was happy. They never again had to shut-down because of wash oil grid coking."

"Norm, is this a true story?"

"You bet. Only the names have been changed to protect the guilty."

Vacuum Tower Wash Oil Grid

"Norm. How about the new vacuum tower in Saudi Arabia? That Sosol plant you mentioned ain't far from our plant."

"I'll tell you this story, which happened just a few months ago. The tower internal engineering contractor made an error in the design of the wash oil bed. They wanted to have a wash oil section that represented three theoretical contacting stages. So they designed a wash oil section with (see Fig. 6.2):

1. Four feet of structured packing with a small crimp size (i.e., ½ in.).
Plus
2. Eight feet of heavy duty grid.

"They had a 4″ support grid, but no hold-down grid."

"Did they use ½″ stainless steel thru-bolt rods to secure the bed to the support grid, like it says in your book 'Process Design for Reliable Operations'?"

"No!"

"What did they think was going to hold that packing in place if they got a pressure surge from wet stripping steam or a high liquid level?"

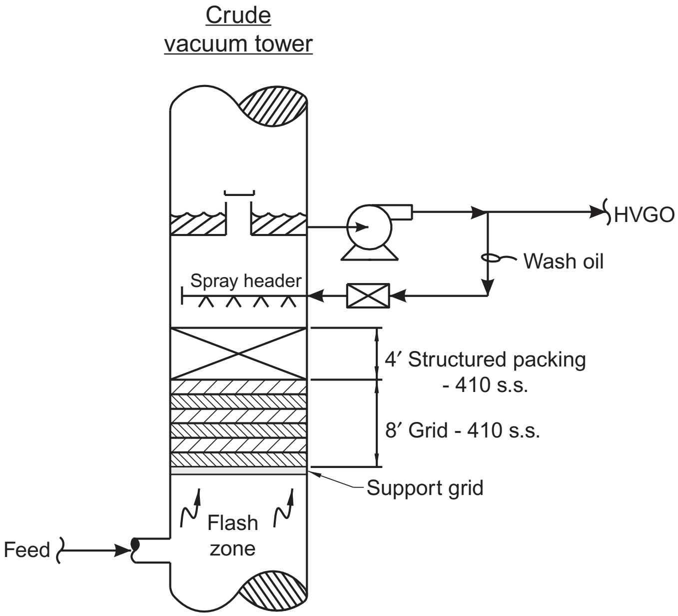

FIG. 6.2 Excessive wash oil bed depth and lack of hold-down grid are serious design errors.

"The weight of the packing itself," I answered.

"That don't sound too good. Norm, why did they think they needed three fractionation stages? Were they just trying to knock back entrained asphaltenes, like in the coker fractionator. To keep the Nickel and Vanadium out of the HVGO so as to protect the downstream catalyst in a hydrotreater reactor?"

"I asked the tower internal contractor why they had provided so many feet of packing. They said they knew it was wrong, but it was specified by the Engineering Design Contractor. That was what they told me," I told Pierre.

"The deep wash oil bed resulted in a very high percentage of wash oil evaporation. The bed dried out towards the bottom. Bed pressure drop increased from one mm of Hg, to ten mm of Hg, over a period of less than a year."

"So I guess they shut-down."

"Right, Pierre. They shut-down and replaced the bed in-kind. No changes."

"Norm. I can't believe that!"

"Yes, the same twelve feet of packing. Small crimp size too. They started back up, plugged the resid bottoms pump suction screen with coke. Ran the resid level above the feed inlet, and blew the new wash oil bed apart. Didn't have a hold-down grid. Wrecked the wash oil spray header. Asphaltenes and metals went sky high in the HVGO and…"

"Norm! Stop – you're making all this up. Things like this can't really happen in a major refinery. Is this the sort of nonsense you tell young engineers in your seminars?"

"Pierre, let me show you around the French Quarter."

7

Installation Error Causes Fractionator Flooding

Oscar Wyatt was the owner of the Coastal Corporation. I met Mr. Wyatt on several occasions, before he was incarcerated in a Federal Prison for purchasing crude from Iraq after the Gulf War. The government had prohibited such purchases, but that did not inhibit Oscar Wyatt from cutting a deal with Saddam Hussein, the President of Iraq.

Mr. Wyatt was a refinery guy, so we interacted at a technical level. “Lieberman! The new crude debutanizer is flooding! Ain’t that your design?” Mr. Wyatt was referring to the crude unit in Aruba.

“Mr. Wyatt, how do you know that the tower is flooding? It’s only operating at 70% of its design capacity,” I asked.

“Because, Lieberman, we got too much pentanes in the C_3-C_4’s, and too much butane in the naphtha. When the panel operator raises up the reflux to knock down them pentanes, we got even more pentanes in the butanes. That’s caused by entrainment, or jet flood.”

“You’re right, Mr. Wyatt. The fact that I saw the tower top temperature was increasing, rather than dropping, as the reflux rate was increased, pretty much proves we are suffering from vapor flood, or jet flood. Also, Sir, the tower top temperature was increasing at the higher top reflux rates. A sure sign of flooding and excessive entrainment.”

“Well, you screwed up the design somehow. I should’ve got some good ole boy from Texas to design my new debutanizer in Aruba. Not some damned Yankee engineer from New York.”

“Okay sir! I’ll go to Aruba and investigate.”

“Yeah Lieberman! Go find out what you done wrong. Ain’t no sense you hanging out in my office.”

Running a Tower Pressure Drop Survey

There are two ways to initiate a study of the causes of a flooding distillation tower:

1. Radiation scan to measure vapor-liquid densities vs elevation through a cross-section of the distillation tower. That is a “Tru-Tec” scan.
2. A single gauge tower pressure survey.

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The first option involves the services of an outside contractor, would take a couple of weeks, and involves a cost of $10,000. I can handle the second option myself in a few hours, without any extra cost. I'm careful to:

1. Make sure the pressure connection is not partially plugged.
2. Install the gauge in a straight-up position.
3. Tighten the gauge with a wrench, but never twist it tight by hand. That changes its calibration.
4. Check the pressure in a vapor space and not below a liquid level.
5. Make sure that the tower pressure is stable and not varying while the pressure measurements are being made.
6. Do not make measurements in a line flowing at an appreciable velocity (i.e., greater than 6 ft per second).

Observed Pressure Measurements

My plan was to measure three pressures:

- Above top tray (No. 40) on the tower top vent connection.
- At the feed point location, between trays (20) and (21).
- Below the bottom tray (#1).

I began at the tower bottom. I had two choices as to where to make this measurement (see Fig. 7.1):

1. At point "PT," which was used to measure the tower bottoms pressure.
2. At point "LG," the top tap of the bottoms' level glass.

Since both points were easily accessible, I decided to measure them both. Note from Fig. 7.2, that both points were at the same elevation. I anticipated both pressures at the "PT" point and the "LG" point would be the same. But from Fig. 7.1:

- The pressure at "PT" point was 160 psig.
- The pressure at the "LG" point was 165 psig.

Why the difference? My initial reaction was that I had done something careless to create the 5 psi discrepancy. So, I repeated my measurements several times. After an hour of repeated measurements, I was forced by the consistent data to conclude that the pressure at the "LG" point was actually higher by 5 psi than the "PT" point pressure.

But why?

There was only one possibility. The "IMPINGEMENT PLATE" was missing. The plate was shown on the vendor's vessel sketch. But, had it actually been installed in the tower?

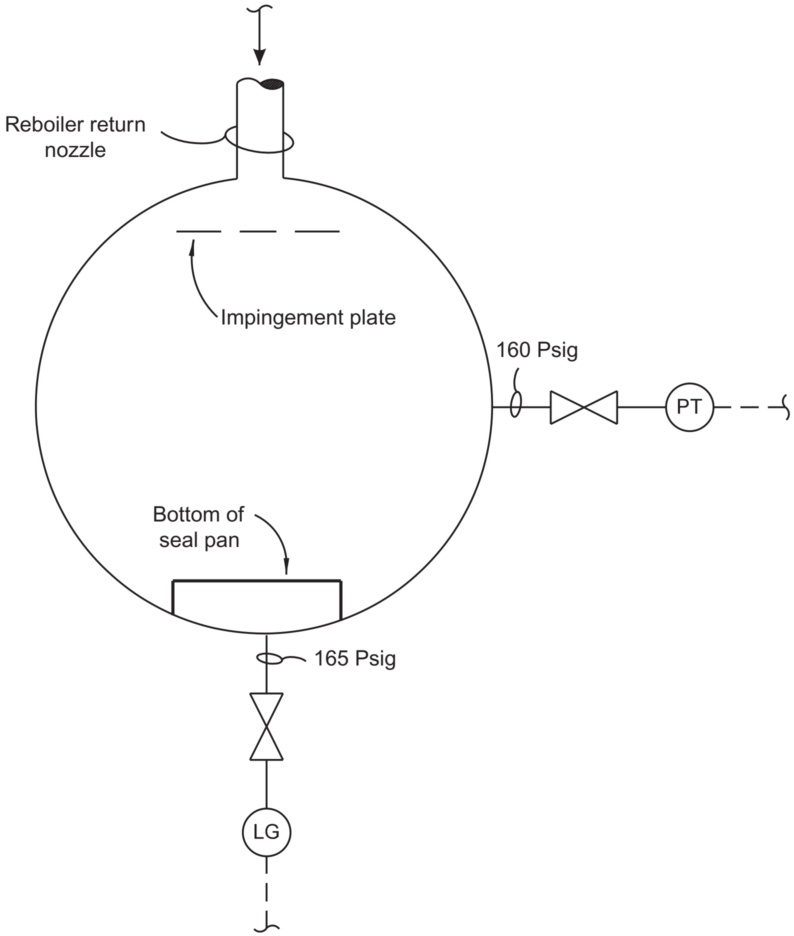

FIG. 7.1 Missing impingement plate.

Tower Internal Inspection

I reported my observations and conclusions to Nelson English, the Aruba refinery plant manager. I explained that the 6″ reboiler return nozzle was located 180 degrees away from the bottom's tray seal plan, slightly below the elevation of the seal pan. The cause of the extra 5 psi of pressure observed at the "LG" point was due to the conversion of momentum to pressure. That is, the high velocity of the vapor-liquid mixture in the 6″ reboiler return nozzle imparted to the opposite wall enough force to create 5 psi localized extra pressure just beneath the seal pan (see Fig. 7.2).

"Norm," Mr. English asked, "how does this affect the level in the seal pan and the liquid level in the bottom downcomer draining tray #1?"

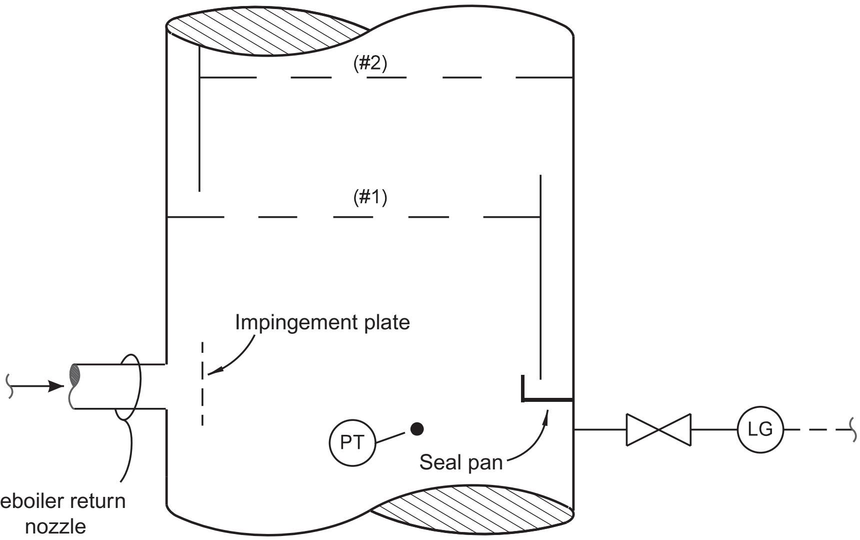

FIG. 7.2 Both PT & LG taps are located at the same elevation.

"Well Mr. English," I answered, "the s.g. of the naphtha is about 0.65. Therefore:

$$(5.0\,\text{psi}) \cdot (2.31) \div (0.65) = 17\,\text{ft}$$

This localized excess pressure in the region of the seal pan is sufficient to flood the downcomer from the bottom tray. And, as flooding progresses up the tower, all trays in the tower will also flood."

"Okay Norm, but why does increasing the reflux rate make the flooding worse?"

"Because," I explained, "more reflux requires more reboiler duty, which generates more vapor flow, which increases the flow of vapor in the 6″ reboiler return line, which increases momentum of the reboiler outlet nozzle, which then…"

"Okay Norm, I got it. Let's check with Oscar Wyatt and get his okay. If Mr. Wyatt approves, we'll shut the debutanizer down next Monday."

Conclusion

Monday morning, we bypassed the naphtha stabilizer, steamed the tower out, chemically cleaned it to remove pyrophoric iron sulfide deposits, and blinded it to make it safe for entry.

I was the first person to enter the tower. The reboiler return impingement plate was nowhere to be seen. I had it installed. Upon start-up, the tower was run at the design rate, meeting both naphtha RVP specs, and the max 2% pentanes in the overhead product.

The next week, Mr. Wyatt called me into his office. I guess to award me a gold watch, or a letter of commendation, or a silver trophy.

"Lieberman," he fumed, "this is entirely your fault. You've cost me a million bucks."

"But Mr. Wyatt, I…"

"Likely closer to three million! What with the shut-down costs."

"But Mr. Wyatt, I…"

"And all that damn off-spec LPG."

"But Mr. Wyatt," I interjected, "I'm the one who figured out the problem and defined the correct solution…the missing impingement plate."

"Oh yeah Lieberman. And what prevented you from inspecting the tower just after the internals were installed? Didn't I read in one of your books, *Process Design for Reliable Operations,* that part of every process design is the final inspection? To make sure all the internals are installed as per the designer's specifications? And didn't you write that the person most qualified to perform the final inspection was the engineer who designed the tower?"

Coastal was swallowed up by El Paso Natural Gas. Which was absorbed by Kinder-Morgan, a pipeline company. The last time I met Oscar, in Lake Charles, he had a broken arm and had become a bit senile but was still sharper than me.

I always tell this story to the young engineers who attend my Troubleshooting Seminar. The main point of the story being that I only was able to solve the problem by making the pressure drop survey myself, and being surprised at the two pressure points at the "LG" point and the "PT" point. I was surprised by this difference and my first reaction was to average the two readings. But, then I repeated my measurements and decided that I had done everything very carefully, and there had to be an explanation for the discrepancy between the two readings. If I had asked an operator to take the readings for me, then I would likely have just averaged both readings and lost my opportunity to fix the problem.

8

Oil-Water Separation

I was stuck at the rail crossing on River Road last month behind an endless string of UTX tank cars. Most refinery products are delivered by pipeline and truck, but some—diesel, kerosene, gasoline—are still shipped by rail car.

I know a lot about UTX, because I received a phone call from Paul Reicter last year. Paul was in charge of their tank car cleaning yard in Chicago.

"Mr. Lieberman, as you can see from the sketch I emailed you, we steam-out rail cars to remove diesel and other hydrocarbon residues. We then have several thousand BSD of an oil-water mixture that we must separate in a horizontal settling drum. Our problem is that we still have 3000–5000 ppm of residual hydrocarbons in the effluent water (see Fig. 8.1)."

"Paul, what's the vessel hold-time?"

"Four to six minutes, half full. We realize that's marginally too small. The water effluent flows into our aerated waste water treatment plant, which is also undersized. As a result, the water treatment plant discharge has an excessive BOD (biological oxygen demand) and violates the EPA water standards. Mr. Lieberman, we would like to retain your services as a consultant engineer to redesign our oil-water settler. Do you have the time to undertake this project," Paul asked.

"As the hookers in New Orleans say, 'If you got the money honey, I got the time.' $2,800 a day plus drafting charges."

pH Control

One method I have found effective in breaking oil-water emulsion is to control the pH at 6.5–7.0. Certainly a pH greater than 9–10 increases the separation time of the two phases. Typically, I have simply injected sulfuric acid controlled manually. But, I have also used an automated continuous pH control with good results.

A pH of less than 5½ is highly corrosive to a carbon steel vessel. Also, on nonstress relieved vessels, low pH will cause weld failure and leaks in a matter of days. You can draw your own conclusions, as I have:

- *Step 1*—Fill two, 1000 mL graduated cylinders with your oil-water sample.
- *Step 2*—Allow one sample to settle out by gravity, without pH adjustment, for 10–20 min.
- *Step 3*—Repeat Step 2 after acidifying to about 6½ pH.
- *Step 4*—As per above, find which pH maximizes the hydrocarbon layer in the cylinder in a given amount of time.

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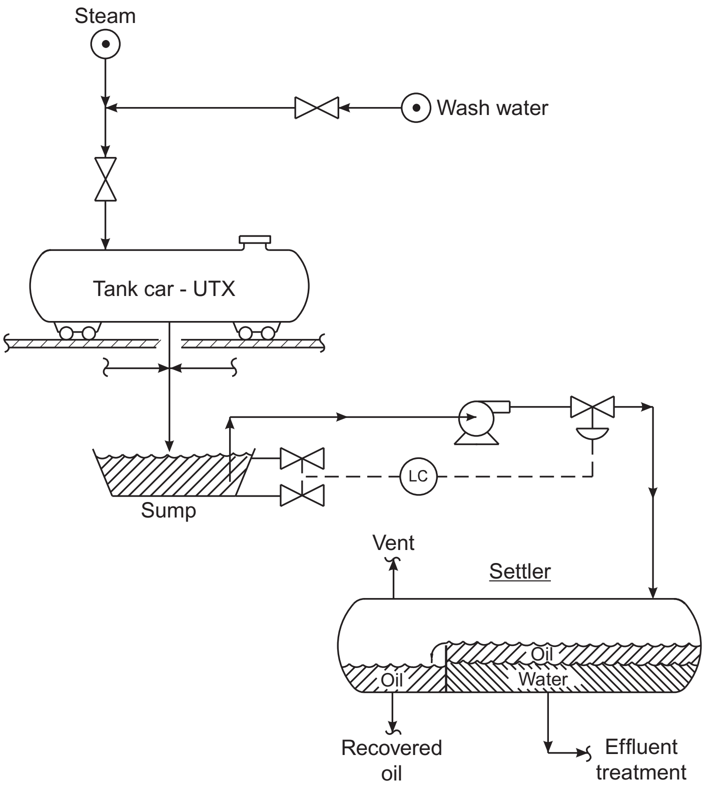

FIG. 8.1 Tank car steam condensate recovery system.

Control Valve Pressure Drop

When I worked for the American Oil Refinery in Indiana, we pumped an oil-water mixture out of a sump using an on-off reciprocating pump. That is, the pump only ran when the level rose above a certain point in the sump. There was no control valve on the pump's discharge line. A sample of the effluent in a quart bottle would separate cleanly into water and kerosene in 2 min.

One year, during a unit turnaround, the recip pump was replaced with a centrifugal pump, with also the addition of a sump level control valve on the discharge line (Fig. 8.1). A sample from the discharge of the pump required a half an hour to settle out. A sample from the level control valve required most of the day to settle. To minimize settling time due to the emulsification in the settling drum for the UTX unit, I suggested eliminating the control valve on the discharge of the sump pump shown in Fig. 8.1. We could then use a level switch that would turn the pump on at a given level and off when the level dropped below the set point. I would have suggested the use of a reciprocating pump, but did not think my client would appreciate this costly expenditure.

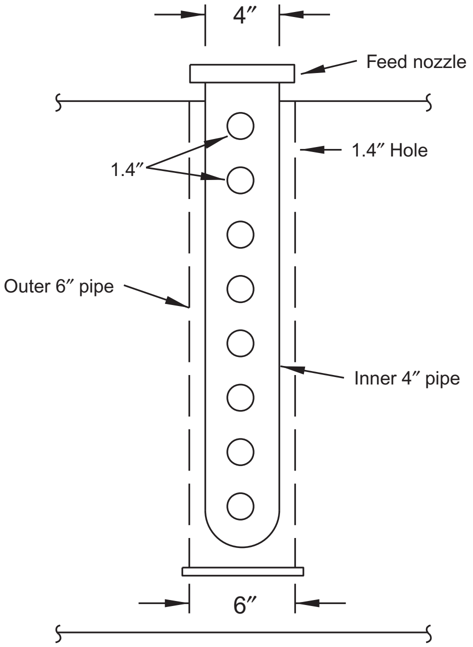

FIG. 8.2 Feed distributor for horizontal vessel.

Improved Feed Distributor

Fig. 8.2 shows conceptually the feed distributor that I use in this service:

- An internal vertical pipe, the same size as the feed nozzle, and the length of the diameter of the vessel, is added.
- Holes, about 4″–6″ apart are drilled in line with the vessel length. Total hole area equals approximately 50% more than the feed nozzle area.
- A pipe, one line size larger than the feed nozzle is placed around the above pipe with the same hole configuration. However, the holes in the larger outer pipe, are off-set by 90 degrees, to the holes in the smaller, inner pipe.

The above configuration effectively distributes the feed vertically across the entire settler diameter with a minimum of turbulence.

Coalescers

The most effective way to achieve an efficient separation between an emulsion of liquid hydrocarbons and or aqueous phase is to impose an electrical field across the mixture. This causes the polar water droplets to coagulate far more rapidly than just gravity settling.

For this tank car cleaning project, I rejected this option for two reasons:

1. An electrical precipitator is normally used when the hydrocarbon liquid phase is 90% to 99+% of the total mixture. In this case, the hydrocarbon phase was less than a few percent.
2. Retrofitting the existing settler into an electric precipitator would be a long, complicated, and costly project, which my client would not approve.

The most common sort of coalescer is to pass the oily water through a packed bed to accelerate phase separation. Typically, packing materials that I have seen used are as follows:

- Raschig rings
- Pall rings
- Fiber glass
- Stainless steel mesh
- Straw

I have found straw to be the best of the above five choices. However, all the packing materials have a fatal property in fouling service; they partially plug. Then, liquid channels with high localized velocity through the small areas that are still relatively open to flow. The high localized velocity retards gravity separation of the two phases.

Since the effluent (see Fig. 8.1) of the tank car wash water collection sump was full of black particulates, I decided to eliminate use of packing as a viable option.

Sloped Settling Baffles

According to Stokes Law, the rate of settling is a function of:

- Viscosity
- Particle size
- Density difference

The viscosity and the density difference were fixed by the fluid properties. By eliminating the sump pump discharge control valve, I had increased the average particle size of the oil. The other parameter left to me to alter was the settling distance.

Certainly, settling efficiency could be increased by a factor of 10, if the settling distance (and hence the settling time) could be reduced by a factor of 10.

Fig. 8.3 shows the common industry practice of the use of settling baffles. I have used this design on several occasions with good results, even in fouling services. A typical settling baffle configuration uses:

a. Baffles which span the diameter of the vessel.
b. Baffles sloped in the direction of flow by approximately 10% (i.e., 1.2 in. per foot).
c. Baffles spaced approximately 2 in. apart.

The relatively large space between the baffles (compared to the openings in a packed bed), resist fouling. Instead of the water phase having to drop several feet to separate itself from

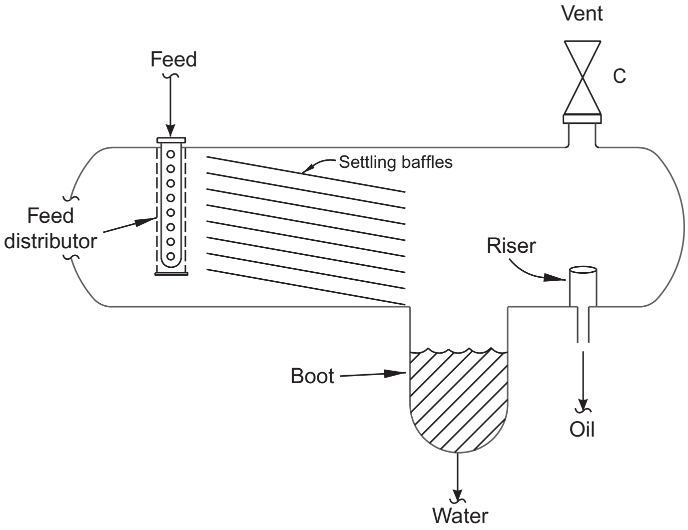

FIG. 8.3 Oil-water coalescer with sloped setting baffles.

the mixed-phased feed, it only has to drop a few inches. The water runs down the top of the baffle, and the oil slides down the underside of the baffle above.

Water flows into the boot shown in Fig. 8.3 and the oil overflows into the riser pipe (typically 6 in.).

This sort of coalescer is widely used in effluent treatment plants, and it is the design I selected to retrofit the UTX tank car cleaning facility yard in Chicago.

Particle Size and Settling Velocity

When I worked for American Oil in the 1970s, we used a velocity for droplets of water settling out of gasoline of 0.4 ft per minute. As a process design engineer, I noted the engineering contractors used a velocity for droplets of water settling out of propane of 1.0 ft per minute.

As a consultant, I have observed a maximum settling velocity for water in a propylene extractor tower of 1.3 ft per minute.

According to Stokes law, these rules of thumb are consistent:

- Specific gravity of gasoline = 0.75
- Specific gravity of propylene = 0.47

Therefore, the settling rate of water in gasoline is proportional to:

$$(1.00) - (0.75) = 0.25$$

For propylene:

$$(1.00) - (0.47) = 0.53$$

From the above, you would expect water to settle out twice as fast in LPG than in gasoline. But often this is not the case.

I have observed in a 1000 mL graduated cylinder that the rate at which gasoline and water separate can be closer to 0.4 ft per hour, rather than 0.4 ft per minute. It isn't that Stokes Law is inconsistent. It's that settling rate, according to Mr. Stokes, is proportional to the particle size distribution.

I have had a water-diesel mixture that settled quite readily in a few minutes in a 1000 psig receiver. If I allowed the mixture to pass through a control valve which reduced the 1000–80 psig, settling time would increase to a few hours.

To determine the liquid hold-time needed for adequate settling, we certainly need to consider the mixtures:

1. Continuous phase viscosity.
2. Density of the continuous phase, compared to the density of the dispersed phase.
3. History of the mixture (i.e., its particle size distributor).

But this third factor is always unknown. So, like everything else in process design, the engineer must ultimately rely on judgment and experience, when sizing water-oil settling vessels. The story I have just related about the UTX rail tank car cleaning project is a typical example. Eliminating the LRC valve shown in Fig. 8.1 was likely the critical improvement.

Summary

I gave a hand sketch of my ideas to Paul Reicter, the UTX supervisor of the rail tank car cleaning operation in Chicago, summarizing my ideas for:

1. pH control (i.e., 6½)
2. Eliminating the emulsification of oil and water through the centrifugal pump's discharge level control valve.
3. Retrofitting the settling drum (Fig. 8.1) with sloped settling baffles (Fig. 8.3).

"What do you think of my proposal, Paul? Shall I submit a firm cost estimate for my work? Three days at $2,800 per day, plus about $500 for drafting. I could start work on your project as soon as I receive the UTX P.O. number."

"Okay Norm. Thanks for your sketch of the settling baffles. Good idea! I've seen that before on a water treatment plant in Calumet City."

"Yeah Paul. If you had used a reciprocating pump in the first place, that would also have helped."

"I have a meeting I need to rush off to. I'll get back to you. Great speaking to you, Norm. Have a great day."

And, as I sat at the Rail Crossing on River Road in New Orleans, watching the UTX tank cars roll past, I recalled this conversation with Paul Reicter, who I never heard from again.

The lesson from this story is always get the P.O. Number first, before providing the technical solution to your client. It's a lesson I still have trouble learning after 35 years as a process design consultant.

Safety Note

Coalescers are often used to separate hydrocarbon liquids from sour water. The term "sour" refers to H_2S dissolved in the water. H_2S will react, in an aqueous phase, to form pyrophoric iron sulfide:

$$H_2S + Fe \rightarrow Fe\,(HS)_2 + H_2 \tag{8.1}$$

The term "Pyrophoric" refers to dry $Fe(HS)_2$ (a finely divided black solid) tendency to auto-ignite:

$$Fe(HS)_2 + O_2 \rightarrow FeO_2 + H_2O + SO_2 \tag{8.2}$$

The iron sulfide burns with a pale blue flame with minimal heat liberation. The danger is not heat, but evolved SO_2, which is fatal at a concentration above 1000 ppm.

I once issued a vessel entry permit to two pipe fitters to repair a settler that I had water washed, prior to permitting personnel access. The problem was iron sulfide is insoluble in water, but does not ignite until it is dry. A few hours after the pipe fitters' entry, the iron sulfide caught fire (at 75°F) and deadly SO_2 fumes were evolved.

I vividly recall the two men's hurried exit from the vessel—very angry, and looking for me, who they claimed had tried to kill them.

Iron sulfide needs to be removed from process vessels by chemical treatment in preparation for making them safe for entry. In Texas City, we used a "ZYME" solution. Hydrochloric Acid will remove all iron sulfide but is corrosive. Citric acid is perhaps a less aggressive solvent used for iron sulfide removal.

Black deposits in refinery process vessels are not typically coke but dangerous iron sulfide which auto-ignites into sulfur dioxide.

9

Hydraulics—Head; Pressure; Velocity

The two operators were standing next to the crude furnace charge pump having a heated dispute. They called me over to arbitrate.

“Look here Lieberman,” Carl yelled over the roar of a nearby steam leak, “we need to adjust the seal flush pressure to this pump. The seal oil’s heavy diesel. The suction pressure is 10 psig and its discharge pressure is 200 psig.”

“Mr. Lieberman,” Clara interrupted, “the seal flush pressure ought to be a little higher than the pressure inside the pump case, where the shaft goes into the pump through the mechanical seal. Enough pressure to keep the hot resid from blowing out of the pump case around the shaft, through the mechanical seal. If the seal blows, we’re gonna have a fire.”

“Yeah Clara. That’s why I said we need to adjust the seal flush pressure to like 210 psi. Ain’t that right Lieberman? Seal leaks are dangerous.”

“No! No! Carl, the pressure inside the pump case is actually a little bit below the pump’s suction pressure. Maybe eight or nine psi. Ain’t I right, Mr. Lieberman? Shouldn’t the seal flush pressure be set to like 15 or 20 psi if the suction is 10 psi?” Clara asked.

“Okay, let’s go into the control center, get us a coffee, and I’ll explain. You guys got some time?”

“Just explain things so that we can understand – no math or science stuff. None of your engineering B.S.!” Carl said.

Relationship of Pressure, Head, and Velocity

I had heard once that women are attracted to men driving clean cars. So, I decided to pressure wash my car, in the course of which, I learned two things. First, that women are attracted to men driving expensive, clean cars. The expensive aspect of the attraction being a lot more important than the clean part.

The second thing I learned was that water squirting out of the pressure washing hose is not high pressure water at all. The water is at atmospheric pressure. As I was washing my car in New Orleans, at sea level, the pressure of the water after leaving the hose nozzle was 14.7 psia.

My Bathroom Water Pressure

I have measured the water pressure in my house. It’s about 30 psig. It’s the same pressure in all of my seven bathrooms and in the entire neighborhood. The 30 psig pressure is a result of the height of the city water tower. The tower is 70 ft high (Fig. 9.1):

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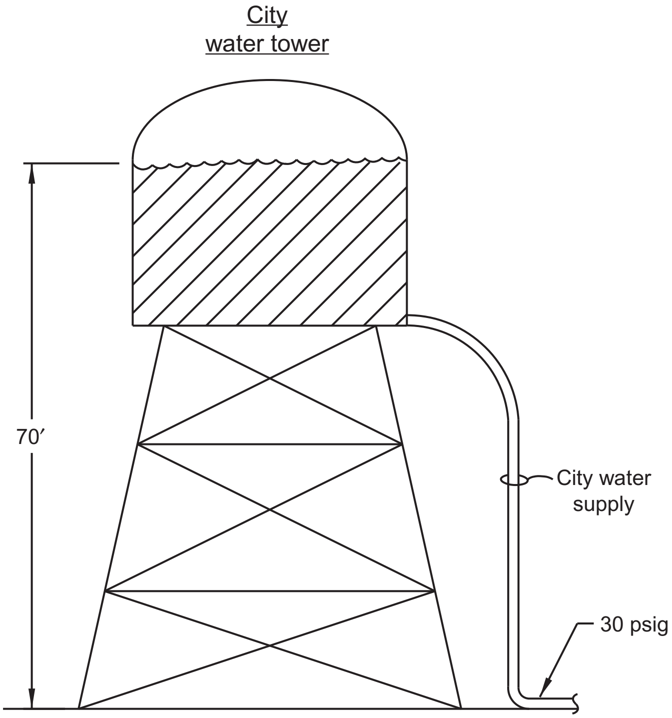

FIG. 9.1 Head pressure vs height.

$$70\,\text{ft} \div 2.3\,\text{ft/psi} = 30\,\text{psi}$$

Meaning, there are about 2.3 ft of cold water needed to create a head pressure of 1 psi.

If I squirt the water out of my garden hose through the hose's nozzle, the water pressure at the discharge of the nozzle is 0.0 psig (or 14.7 psia). What happened then to the 30 psig? It was converted into three other forms of energy:

- Velocity, or momentum, or kinetic energy.
- Heat due to friction.
- Potential energy (if I am pointing the nozzle upwards).

Converting Head Pressure Into Velocity

For water, or for any other liquid that is not particularly viscous, assuming that the nozzle has no friction, and that the water is squirting out of the nozzle parallel with the ground:

$$\text{Head of water (in ft)} = 2.1\,(\text{velocity})^2\,(\text{in ft per second})$$

For example, if the head of water in the water tower is 70 ft, the velocity of water flowing from the frictionless hose nozzle will be:

$$(70 \div 2.1)^{1/2} = (33)^{1/2} = 5.55\,\text{ft/s}$$

Unfortunately, my assumption that the nozzle is frictionless cannot be true. If half of the head of water (or the potential energy of the water) is lost due to the unavoidable friction losses in the nozzle, then the nozzle discharge velocity will be somewhat less:

$$(70 \cdot ½ \div 2.1)^{½} = (16.5)^{½} = 4.1\,\text{ft/s}$$

I really should not have said that any of the pressure was lost due to friction. Energy cannot just vanish. It’s that half of the potential energy is converted into heat. Meaning, the water coming out of the nozzle is slightly heated by friction.

In reality, somewhere between 20% and 60% of the head pressure would be converted to heat which is not enough to noticeably increase the temperature of the flowing water.

Converting Pressure to Feet of Head

It seemed to me, as a child, that if I squirted water from the garden hose nozzle straight up, and the city water pressure was equivalent to 70 ft, then I would be able to squirt the water up to the top side of a building 70 ft high. However, since I lived in a rented apartment in New York City, I did not have access to any garden hoses or even gardens.

But, when I purchased my own home, I did run this experiment. However, I was quite disappointed. At best, I could only squirt the water to the top of the brick chimney, about 30 ft high, of my two-story house. Apparently, the other 40 ft was lost due to friction in:

- The garden hose nozzle.
- The 100 ft of rubber hose.
- The faucet fittings.
- The copper tubing inside my home.
- Air resistance.
- My wife taking an unscheduled shower during my experimental observations.

Centrifugal Pumps

If water flows into the eye of the impeller (Fig. 9.2) of a centrifugal pump at 10 psig, the pressure of the water leaving the impeller is the same 10 psig! All the impeller has done is to accelerate the water. As a matter of fact, the pressure inside the entire centrifugal pump case is the same 10 psig. How then does the water pressure ever increase to the 200 psig pump discharge pressure?

Well, the pump discharge nozzle (see Fig. 9.2) increases from a 1 in. opening to a 4 in. opening. The velocity of the water would then drop by a factor of 16:

$$(4'' \div 1'')^2 = 16$$

The reduction in velocity is converted, not into pressure, but into feet of head. To convert from feet of head into pressure:

$$\text{Pressure} = \frac{(\text{Feet of Head})}{2.31} \cdot \text{S.G.} \tag{9.1}$$

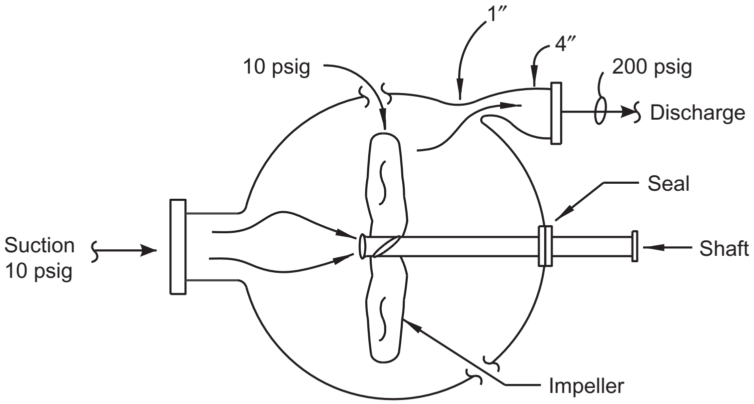

FIG. 9.2 Seal pressure should be a little higher than pump suction pressure.

"S.G." stands for the specific gravity of the liquid, corrected for the flowing temperature.

The S.G. for some of the fluids I work with are (at 60°F):

- Butane = 0.55
- Naphtha = 0.68
- Kerosene = 0.80
- Diesel = 0.85
- Asphalt = 1.00 (at 300°F)

When converting between head and velocity, we ignore the specific gravity. However, when converting from pressure to head and/or velocity, we need to take the specific gravity into account.

Orifice Coefficients

If I express head in "*inches*" rather than feet, orifice coefficients (OC) for different types of openings are as follows:
"OC"

- Theoretical (no friction) = 0.178
- Smooth outlet nozzle = 0.34
- Hole drilled top down = 0.28
- Orifice plate = 0.6
- Thin plate with punched hole = 0.9

We use this concept in measuring flows in the plant (see Fig. 9.3) by measuring the pressure drop across the orifice plate on the orifice flanges. Then we use the formula:

$$\mathrm{DH} = (\mathrm{OC}) \cdot (V)^2 \tag{9.2}$$

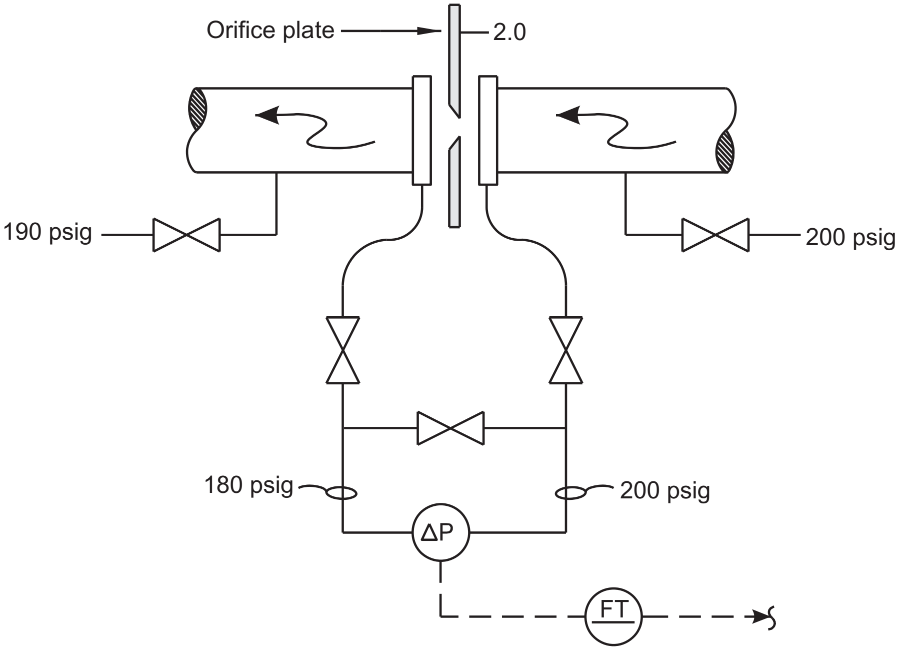

FIG. 9.3 Flow orifice with 50% pressure recovery.

For a refinery flow orifice plate, OC = 0.6:

$$\mathrm{DH} = 0.6 \cdot (V)^2 \tag{9.3}$$

to calculate "DH," measure the pressure drop across the orifice plate in psi. Then convert that "Delta P" to inches of liquid.

$$\mathrm{DH} = \frac{(\text{Delta P}) \cdot (27.7)}{(\text{S.G.})} \tag{9.4}$$

Don't forget to account for the flowing temperature. For hydrocarbons, S.G. is reduced by 5% for each 100°F hotter than 60°F.

Calculate V (ft/s) from Eq. (9.3). Look at the upstream side of the handle sticking out of the orifice plate flanges. The number shown, in inches, is the orifice plate hole size. Multiply V times the area of the hole (in square feet), to arrive at the flow in cubic feet per second. There are 5.56 cubic ft per barrel (42 gal).

Converting Velocity Back to Pressure

Clara had drained her mug of green tea. "So Mr. Lieberman, what happens to all that velocity as the flow comes out of the orifice plate? Is it all lost?"

"No. Some is indeed lost due to friction. But a lot of the velocity is converted back into pressure. That's called 'Pressure Recovery'. Maybe half could be recovered," I said.

“So,” Carl asked, “I guess this proves that Clara is right? The pressure at the seal faces is about the same as the pump suction pressure. All the impeller does is to increase the speed of the liquid escaping from the pump case into the discharge nozzle.”

“I’m still bothered by something,” Clara said. “On most of our pumps, Norm, the seal flush line comes right off the pump discharge. That looks to me as if the seal flush pressure is actually the pump discharge pressure?”

“No,” I explained. “There is a restriction orifice, with perhaps an eighth inch hole, that screws into the pump, and reduces the pump discharge pressure, for self-flushed pumps, down to the pump’s suction pressure, plus a few psi. Also, remember that what I’ve said does not apply to high viscosity operations, nor to positive displacement pumps. By high viscosity, I mean over 30 or 40 centistokes.”

“Okay, Mr. Lieberman. But let me tell you. Women are not attracted to men who drive expensive, clean cars. Women are attracted to men who give them expensive cars,” Carl concluded.

Pressure Balancing Holes

Just to ensure that seals are not overpressured, there are “Pressure Balancing Holes” drilled in the back of the impeller. This drains excessive seal chamber pressure, back into the pump suction. If holes plug with solids, seal damage will likely result. This is a common problem in plants that convert coal and coke to liquid hydrocarbons.

10

Common Hydraulic Problems

“Norman, the washing machine is banging again. Do something, you’re an engineer!” My wife was angry. The loud bang was not actually coming from our new washer. It was originating from the living room ceiling. It happened, without fail, during the rinse cycle. “That damned washer is broken. Call the repair man. I want it fixed today.”

“It’s water hammer,” I explained. “It’s caused by hydraulic surge in the cold water supply line in the attic. When the cold water supply solenoid valve suddenly shuts, a sudden increase in the flowing water pressure results in the hydraulic surge. I learned about this problem in fluid dynamics at University.”

“Just get it fixed. I’m going to the mall.”

Conversion of Velocity to Pressure

Hydraulic surge, which most people refer to as water hammer, is a consequence of the conversion of velocity of a flowing liquid to potential energy, when the flow slows or stops. By potential energy, I mean the increase in height of the flowing liquid as it slows down. This is not hard to understand. Just take a garden hose. The water escapes from the hose’s nozzle with a high velocity. If the nozzle velocity was 50 ft per second, and you pointed the hose straight up, the water might rise to a height of 30 ft. Meaning the velocity or the kinetic energy of the water (50 ft/s) had been converted to feet of head (or potential energy, 30 ft).

I can quantify this conversion of velocity to height:

$$\text{Hight (ft)} = 0.015\,(V)^2 \tag{10.1}$$

where V = velocity (ft/s).

This equation assumes that:

- There are no frictional losses.
- The fluid is incompressible (i.e., like water).
- There is no vaporization.

The conversion of velocity into potential energy, or elevation of a fluid, cannot occur if there is no open vertical space for the fluid to rise into. Then, the conversion of velocity is not to height or into potential energy, but into pressure. To convert from height into pressure:

$$\frac{\text{ft of height}}{2.31} \cdot (\text{SG}) = \text{PSI} \tag{10.2}$$

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where:

- SG = specific gravity. That is, the density of a fluid relative to water at 60°F.
- PSI = pounds force per square inch.

The pressure surge inside a pipe when the water flow suddenly stops is called water hammer and created a bang in my wife's washing machine water supply line.

Water Hammer Dampener

Having completed my analysis, I report to my wife and immediate supervisor. I showed her my proposed retrofit design (Fig. 10.1). She studied my sketch carefully and then said, "Norman I have three questions. One – how high will the riser pipe be? Two – how high will the water level in the riser pipe be? Almost full, or almost empty? Three – can I have your Master Card? I'm going back to the mall after lunch."

As a refinery process design engineer, I have learned the main rule of sound engineering design practice. Copy designs that you have seen working successfully in the field. On that basis, I installed a three-foot vertical riser halfway along the water supply line in the attic. I filled the riser halfway with water and then placed a screwed cap on top of the riser.

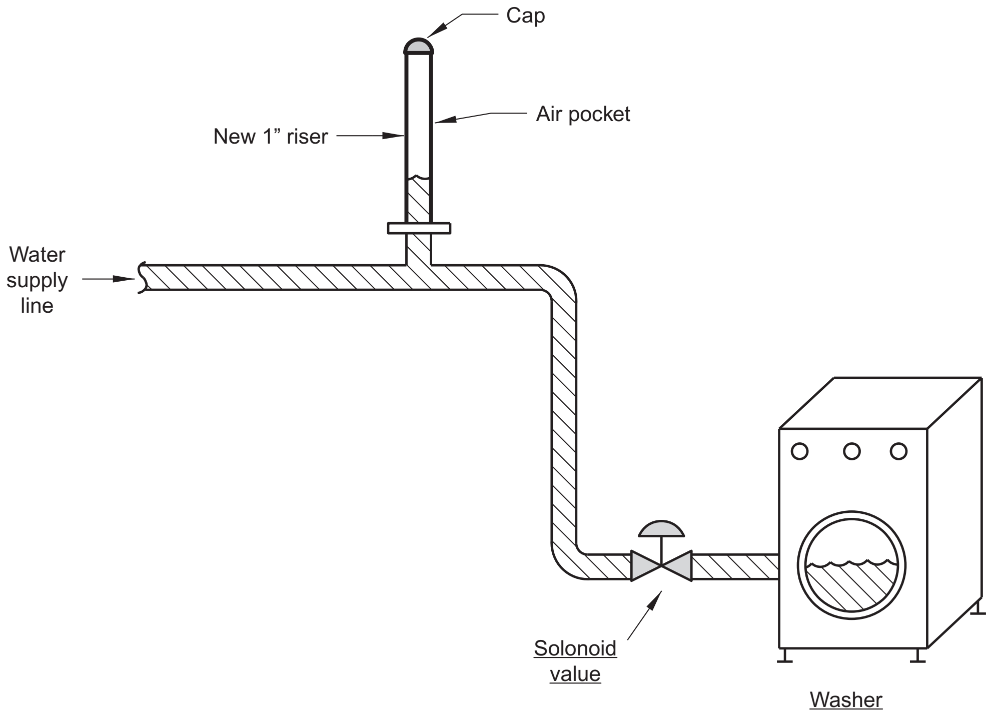

FIG. 10.1 Water hammer fixed with capped riser full of air.

Now, when the water flow to our washer suddenly closes during the rinse cycle, the diminished water velocity is still converted to increased pressure. But, most of this reduction in kinetic energy is used to compress the air inside the riser. The amount of energy needed to compress a gas to a greater pressure, being much greater than that needed to increase the pressure of a liquid.

The project, however, was not a success. My wife spent $580 at the mall for a toaster oven, which is now broken, and which I am supposed to restore to operational status.

A rather larger example of water hammer occurred about 10 years ago in Siberia. The Russians were starting up the world's sixth largest hydroelectric plant. During the opening ceremonies, when the visiting dignitaries were toasting the plant's successful start-up, a tree stump became caught up in the turbine blades in one of the huge turbine tunnels. The water flow stopped suddenly. In the ensuing hydraulic surge, 74 people were killed. I downloaded dozens of pictures. It looked like a small nuclear weapon had exploded. I suppose the bar screen, to keep out debris from the plant's intake, was not installed properly, or even omitted from the Russian's design?

Vapor Lock

Most readers will be familiar with the deadly explosion at the gasoline fractionator (i.e., raffinate splitter) at the BP Refinery in Texas City. What happened in brief:

1. The fractionator filled with liquid gasoline.
2. The fractionator's relief valve opened, as the tower was overpressured as a result of being overfilled.
3. The relief valve flowed into a blowdown stack vented to the atmosphere.
4. The liquid from the blowdown drum drained into a 6″ hydrocarbon condensate collection line that operated at 3 psig.

I've shown this facility (which I helped design in the 1970s) in Fig. 10.2. Note the "*LOOP SEAL*" on the drain line from the blowdown drum to the condensate collection line. The loop seal was needed to prevent the loss of hydrocarbon vapors from the refinery wide condensate collection system, continuously to the atmosphere.

During the design phase of this project in the 1970s, a senior engineer, Mr. Heinz, observed that the seal was subject to "*Vapor Lock*." That is, as the liquid flowed from the blowdown drum upwards through the loop seal, it would lose head pressure. The liquid could start to vaporize due to its loss of head pressure.

The evolved vapor could choke off the liquid flow. One volume of liquid, upon vaporization, expands to 200 volumes of vapor, and thus would retard the flow of liquid. That is, "Vapor Lock."

Mr. Heinz suggested that a vent be installed at the top of the loop to prevent this vapor lock. But, his idea was rejected on the basis that a small amount of vapor would be continuously lost to the blowdown stack to the atmosphere via the vent.

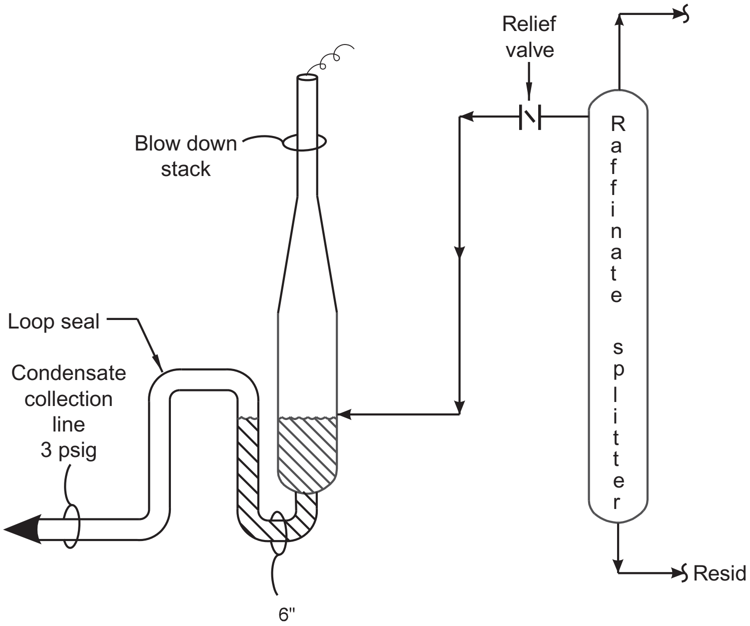

FIG. 10.2 Effect of loop seal.

So, it was not installed. And, as a result, BP lost $4,000,000,000 (United States) in the litigation that ensued from the failure of gasoline to drain out of the blowdown drum. The gasoline was ejected from the top of the blowdown stack, ignited, and exploded. Fifteen people were killed.

Vapor lock is most commonly encountered in steam condensate systems. High pressure condensate, at its boiling point, flows through a pipe and loses pressure due to friction. As the hot water pressure drops below its boiling point, it starts to vaporize to steam. The rapid volumetric expansion chokes off the flow.

To combat this problem at the Tenneco Refinery in New Orleans, I installed a series of flash pots, venting to the plant low pressure steam header, to remove the evolved steam. I recall having a flash pot every few hundred feet of piping.

Condensate Seal

This story is extracted from my vast experience in home air conditioner maintenance. My problem was that the 1-in. condensate drain line from the condensate collection pan underneath the evaporator coils did not consistently drain (Fig. 10.3). I tried pouring chlorine bleach down the line, which usually was effective, but this did not help. Then, I noticed a problem. The condensate drain line ran over the top of a 2-by-4 support beam in the attic. There was a small upwards slope of the PVC drain line, as it crossed the 2-by-4

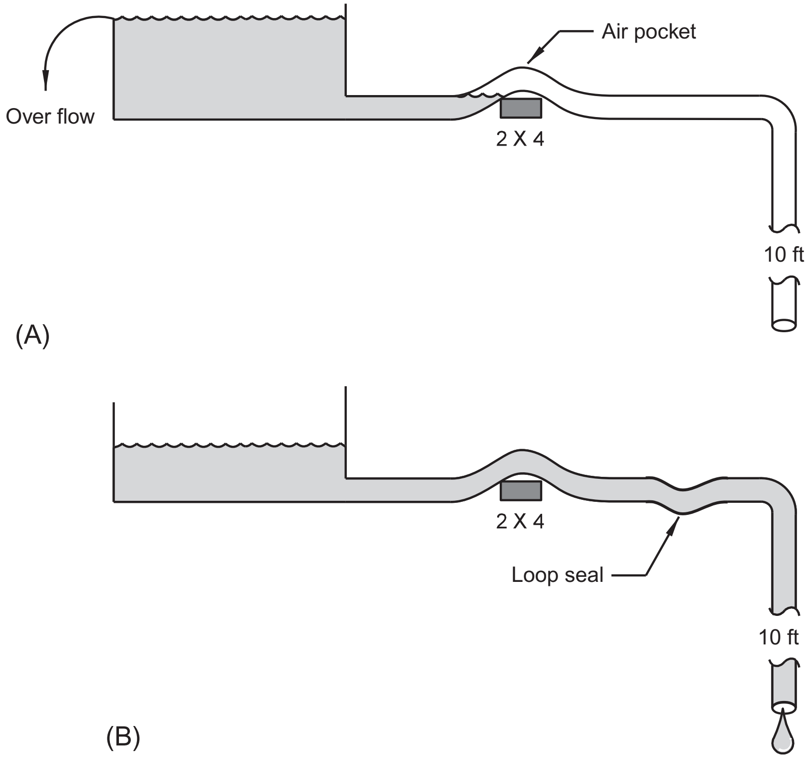

FIG. 10.3 (A) Air pocket stops condensate drainage. (B) Loop seal prevents formation of air pocket.

support beam. The height of the condensate collection pan was much higher than the top of the PVC pipe as it crossed over the 2-by-4 support beam. Apparently, there was sufficient liquid head driving force to overcome this small incremental elevation. But, what happens when the air conditioner shuts off? Then:

1. All the condensate will drain out of the 1″ PVC line.
2. The loop over the 2-by-4 support beam will fill with air that backs-up from the outlet of the drain.
3. Air will fill the loop where the PVC pipe crosses the 2-by-4 support beam.
4. This trapped air pocket will block drainage from the pan.
5. The pan will then overflow. Note, that the height of the loop was greater than the pipe ID.

After considering the problem, and rejecting the option of cutting out a section of my 2-by-4 attic support beam to straighten the 1″ drain, I devised the retrofit design shown in Fig. 10.3B. That is, a loop seal which prevented air from backing-up into the drain line upstream of the new loop.

The local hardware store sold the exact PVC loop seal piece I needed. I guess this must be a common solution to an equally common problem.

My installation has worked fine for over 20 years. I take as much satisfaction in this retrofit as I've done for any refinery revamp.

I have applied this same principle about the loss of a condensate seal to a refinery application on a side stream stripper at a refinery in Convent, Louisiana (Fig. 10.4A). In this case, kerosene flowed out of the main tower, by gravity, into the slightly higher pressure, but lower elevation side stripper. It worked fine, as long as the flow of kerosene to the stripper was consistent. However, if an occasional unit upset briefly interrupted the

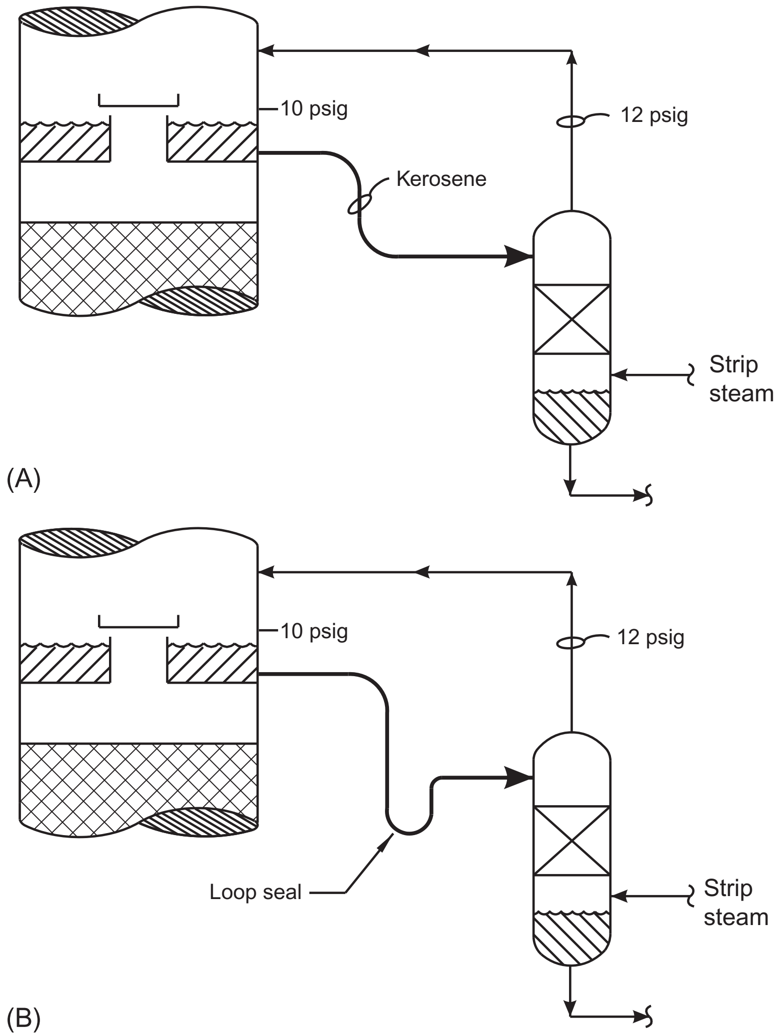

FIG. 10.4 (A) Kerosene stripper. (B) Loop seal added.

flow to the stripper, then the stripping steam would begin to flow up backwards the kerosene liquid feed line. With the feed line full of steam, there was no liquid head driving force to gravitate the kerosene into the stripper. The Texaco operators would then stop the flow of steam manually for a few moments to restore the feed flow to the stripper.

Recalling my home air conditioner project, I had a four-foot loop seal (see Fig. 10.4B) installed in the feed line. This preserved the liquid head driving force, and the stripper now functioned stably. This is a good example of applying a technical principle from everyday experience toward solving a common process plant problem. This is pretty much my normal method.

One of the conclusions to be drawn from these examples is that a hydraulic system that works well at steady-state conditions may not work during intermittent use. Another conclusion is that many such problems can only be resolved by direct field observations.

Steam Hammer

In May of 1980, I was starting up the 400 psi steam system on the Sulfur Recovery Unit in Texas City. It was a warm, humid evening. I had never done this before and was completely alone. As I opened the 10″ chain operated valve between the refinery 400# steam header, and the sulfur plant steam system, a horrendous banging sound began. Steam Hammer!

The cause of steam hammer is not related to water hammer. When steam enters a cold piping system, or cool condensate is flowing into the piping, the steam will begin to condense rapidly. The slugs of water created by this condensation will accelerate in those piping areas that are cooler. And being cooler, are at a lower pressure. When the high velocity slugs of water impact an elbow or tee junction, a loud reverberation results. There are three methods to handle steam hammer:

1. See Fig. 10.5. I learned this technique from a colleague in India.
2. Warm the piping slowly.
3. Ignore the clamor and hope the piping doesn't fail which was my procedure in Texas City in 1980, but is not recommended.

Summary

It may seem as if I have jumbled my problems at home with my problems at work, in this chapter. Very often, my inspiration for solving a refinery process problem is an

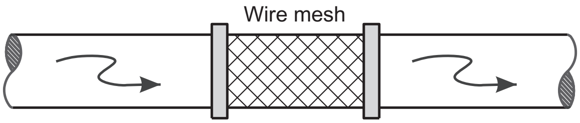

FIG. 10.5 Water hammer suppression device.

observation I have made when walking along the drainage canals in New Orleans, or looking at the fittings being sold in the local hardware store.

The concept of installing a vent on the top of a loop seal to break a vapor lock on a drain line, was something I learned on Bourbon Street, in the French Quarter.

At home, I have seven bathrooms, four air conditioners, and two washing machines, all of which require continuous field troubleshooting. It's almost a relief to tackle refinery process problems.

11

Applying Hydraulics to Process Problems

I became the tech service manager of the Good Hope Refinery in Norco, Louisiana, on December 2, 1980, after working for American Oil for 16 years. This was my first position as a supervisor, a job for which I discovered I was unqualified, for several reasons:

1. Lack of technical skills.
2. Lack of supervisory skills.
3. Lack of communication skills.

I have, after 54 years, only managed to master the first deficiency. Especially in my ability to overcome process hydraulic limitations, I have become quite proficient using a few simple correlations:

- Pressure drop through a hole:

$$\mathrm{DP} = K \cdot \frac{(\mathrm{DEN})}{28} \cdot V^2 \tag{11.1}$$

where:

- DP = pressure drop, psi
- DEN = density, lbs/ft^3
- V = velocity, ft/s

Typical orifice coefficients (K) are as follows:

- Theory (no friction or turbulence) = 0.178
- Smooth hole through a thick (i.e., ½ in.) plate = 0.2–0.3
- Hole through a thin (i.e., 0.08″, 16 gauge, 2 mm) = 0.4–0.6
- Smooth draw-off nozzle = 0.2–0.3
- A chimney with a hat = 0.9–1.2
- Flow orifice plate = 0.5–0.7

Similarly, the pressure drop through a pipe can be calculated by:

$$\mathrm{DP} = \frac{4}{\mathrm{ID}} \cdot \frac{(\mathrm{DEN})}{28} \cdot V^2 \cdot L \tag{11.2}$$

where:

- ID = Pipe inside diameter
- L = 100 equivalent feet

Understanding Process Equipment for Operators and Engineers. https://doi.org/10.1016/B978-0-12-816161-6.00011-4

If there are elbows, open gate valves, and other fittings on the pipe lines, I'll double the length of the pipe to calculate "*L*." For mixed-phase flow (i.e., vapor and liquid), with velocities above 15–25 ft per second, I use the average, mixed phase density. Below 15 ft per second, phase separation is likely.

All these methods are approximate, but then we don't know the actual roughness inside the pipe, so any more complex calculation is also only an approximation.

Pressure Recovery

The pressure in a feed pipe flowing into a vessel will usually be lower than the vessel pressure. I have measured this effect in a refinery in Aruba where the 18″ inlet nozzle entered a 30 ft ID vacuum tower. The pressure increased from 68 mm of Hg to 72 mm of Hg. I will use Eq. (11.1), with a $K = 0.1$, to calculate the conversion of velocity in pressure.

I have noticed for vapor coolers and condensers that the outlet may be at a higher pressure (discounting elevation effects) than the inlet. This is due to a large reduction in velocity, because of volumetric shrinkage of the flow. I observed this in Aruba on a hydrotreater reactor effluent exchanger, where an exchanger outlet pressure was 3 psig higher than the inlet, even though the outlet was 6 ft above inlet nozzle.

Vapor Lock in an Emulsion Line

I returned last month from Fort McMurray, Alberta, where I visited the Suncor Tar Sands plant. About 1% of the world's CO_2 emissions are a consequence of the tar sands operations. The extracted oil from tar sands is an emulsion of:

- Water—perhaps 30%
- Very heavy crude oil (12°API)
- Small amounts of air
- Diluent naphtha—several percent

Suncor had recently installed a new 16″ pipeline to transfer tar sands emulsion by 20 miles. The emulsion (Fig. 11.1) was to be pumped to a receiving tank. Calculated pressure drop was about 130 psi. This calculated frictional loss was verified by comparison to an existing pipeline in similar service. An extra 50 psi pump head was added as a safety factor. The 100 ft high hill shown in Fig. 11.1 was not considered a factor, as the 100 ft of head lost going uphill would be regained going downhill.

However, from the first day of operation, pump capacity limited the transfer rate to 70,000 BSD emulsion vs the design rate of 84,000 BSD. This was a very large economic penalty.

Having seen this problem before, I suggested the problem was "Vapor Lock." The emulsion was at its vaporization conditions in the feed tank at atmospheric pressure. When it reached the top of the 100 ft hill, it might be below atmospheric pressure. I asked Caroline, the project engineer, to check the high point vent valve at point "A" (Fig. 11.1).

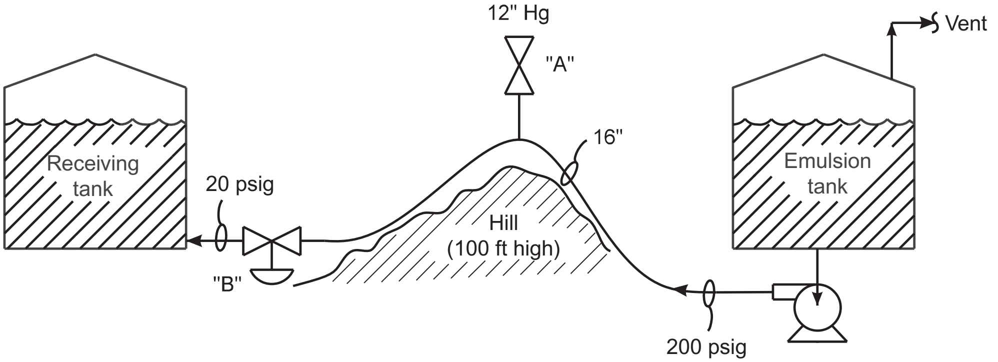

FIG. 11.1 Vapor lock.

She reported the pressure was 12″ Hg (i.e., minus 6 psig, or 7 psia—Fort McMurray being well above sea level).

"Caroline, you are experiencing vapor lock. Air and hydrocarbon vapors are choking off the flow at the top of the hill."

"But Mr. Lieberman, what should we do to get more flow?"

"Well, let's try closing valve 'B' a few turns to suppress the vapor lock."

Which we did, and flow came up by 3000 BSD. "Okay," Caroline said, "that helped a bit. What else can I do?"

"It's best," I suggested, "dear girl, to try to get smarter and think about the hydraulics of the system, and how it will react with vapor-liquid equilibrium, before laying 20 miles of pipe."

"Mr. Lieberman, I am not your 'Dear Girl'."

As I've noted already, my interpersonal skills have yet to be fully developed.

Cooling Water Piping Pressure Drop

I was contracted to increase the capacity of a refinery in Salt Lake City. The crude unit bottleneck was condenser capacity. The overhead condensers were fouling rapidly due to low water tube side velocity and high (i.e., 135°F) water outlet temperature. I measured the cooling water outlet pressure from the crude unit overhead condenser at 42 psig. Correcting for the elevation above grade (i.e., about 10 ft) of the condenser resulted in a pressure of 46 psig (Fig. 11.2).

The cooling water 12″ return line handled effluent from a dozen other coolers. It was 800 ft long. The total flow of water I calculated from the discharge pressure of the water supply pump located at the cooling tower and the pump's head versus Capacity curve.

The calculated return line pressure drop (Eq. 11.2) was 10 psi. The return line pressure at the base of the cooling tower I measured was 12 psig. The water had to flow to the top of the cooling tower 30 ft above my gauge. The observed DP of the return water line was then:

$$46 - 12 = 34\,\text{psi}$$

as compared to calculated pressure drop of 10 psi. Something was wrong?

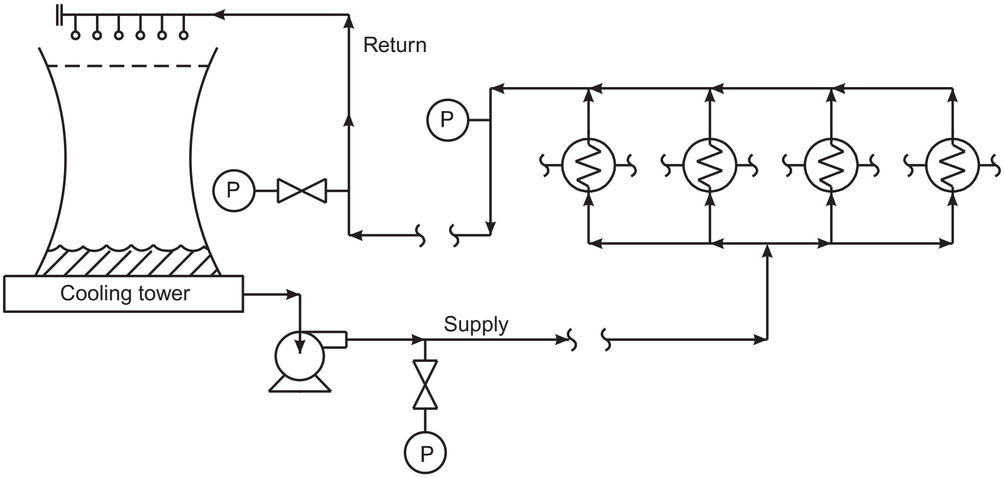

FIG. 11.2 LPG leak restricts water circulation rate.

So, I repeated my calculation and measurement for the water supply line with same flow and same distance. But the result corresponded to the 10 psi water supply line pressure drop I measured.

Water lines flowing at elevated temperatures are subject to scale formation unless a pH of about 6½ is maintained. I informed my client, Mr. Colson, that the observed 34 psi cooling water return line DP, as compared to the calculated 10 psi DP, indicated the line was restricted with hardness deposits and should be chemically cleaned.

"Mr. Lieberman," Mr. Colson, the plant manager, said, "we did that last month and it barely helped at all. Also, I do not appreciate your condescending tone when talking to me."

Rather disheartened, I went into the plant to investigate. I rechecked my pressure measurements. Everything was the same. But, when I removed the pressure gauge from the return line at the base of the cooling tower (Fig. 11.2), I noticed a bit of gas blowing out of the bleeder. So, I climbed to the top of the cooling tower to the water distribution piping. There I found hydrocarbon vapors blowing out of the distribution piping.

A sample of these vapors as indicated by gas chromatographic analysis showed that the crude unit debutanizer overhead condenser was leaking LPG into the cooling water return line. As the density of LPG vapor was 0.13 lb/ft^3, and the density of water was 500 times greater (62 lb/ft^3), a small LPG leak can greatly increase return water line pressure drop.

The leaking condenser was repaired. The flow of cooling increased by 30%, taking into account:

1. Cooling water pump curve.
2. Supply line pressure drop.
3. Exchangers' pressure drop.
4. Return line pressure drop.
5. Height of riser.

Note that items 1 & 5 somewhat offset item #4. But, 30% more water flow was a large improvement for the crude tower overhead condenser.

I expected Mr. Colson to reward me with a letter of thanks, or a certificate of merit, or a gold star, or something? But all I received was my consulting fee for 3 days, and a note suggesting that I try to improve my communication skills.

Fractionation Tray Efficiency

This is a story from the 1970s when I worked as an engineer for Amoco Oil. It happened at the long since vanished Amoco Oil refinery in El Dorado, Arkansas. Specifically, on an FCU debutanizer. The tower was equipped with sieve trays perforated with ½″ holes. The outlet weirs were 3″ high (Fig. 11.3). The tray decks were constructed of 16 gauge (i.e., about 0.08″ thick) steel. I inspected the tower a few months prior to the unit turnaround and reported that all trays to be intact but thin after 20 years of service.

The maintenance department decided to replace the trays with thicker, that is, ¼″ thick trays, during the turnaround. Other than the increase in thickness, the trays were a replacement in kind.

When the debutanizer was restarted, tray fractionation efficiency had dropped from 80% to 30%. I ran a computer simulation using data before and after the turnaround. Before the tray replacement, the 30-tray tower was developing 24 theoretical fractionation stages. Afterwards, just nine. What had happened?

Referring to Eq. (11.1), K for 16 gauge decks (0.08″) thick, is twice as big as thicker (0.25″) trays. Thicker trays reduce the pressure drop of the vapor flowing through the tray deck holes by one-half. As the weight of liquid on the trays, due to the weir height, was unchanged (see Fig. 11.2), the liquid largely drained through the tray decks, rather than mixing efficiently with the upflowing vapor on the trays. Especially as the trays were not very level, the liquid on the trays just drained through these low points on the tray decks. This reduced vapor-liquid contacting and thus fractionation efficiency.

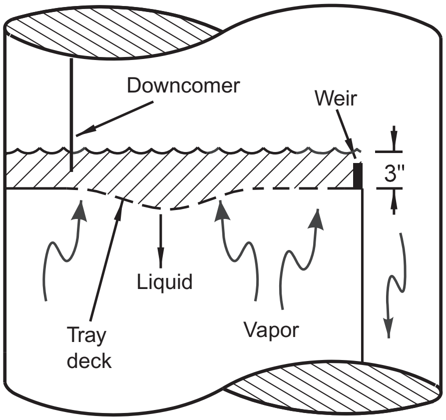

FIG. 11.3 Thicker tray decks will leak more than thinner tray decks.

The idea that a thicker hole has a lower DP than a hole drilled through a thinner steel plate, I find to be counterintuitive. But that's the way it is. I explained this to Larry Durland, the plant manager at El Dorado. It was a serious problem which was making it difficult for El Dorado to produce gasoline meeting required RVP (i.e., volatility) specs.

Mr. Durland stared at me blankly, "Lieberman," he said, "I have no idea what you are talking about. You are sadly lacking communication skills."

But that was a long, long time ago. I'm much better now.

Chimney Tray Pressure Drop

The orifice coefficient for a chimney tray, as noted at the start of this chapter, is 0.9–1.2. In a refinery vacuum tower in Puertollano, Spain, I measured a pressure drop of 4 mm of Hg across one chimney tray. This increased the flash zone pressure of the vacuum tower also by 4 mm of Hg, which then suppressed vaporization of the gas oil product out of the low value tar bottoms product.

The reason for the high chimney tray DP (and the reason for the large orifice coefficient) is shown in Fig. 11.4A. That being:

1. The vapor has to enter the restrictive chimney.
2. The vapor has to make a 90 degrees turn underneath the hat as it escapes from the chimney.
3. The vapor has to make a second 90 degrees turn as it flows between the hats.

All three of these paths are done as the vapor passes through sharp-edged orifices. A smooth, rounded orifice typically has half of the pressure drop of a sharp-edged orifice. To mitigate the effect of the chimney tray pressure drop, I had the chimneys and hats modified, as shown in Fig. 11.4B, with rounded openings.

The net effect was to reduce the flash zone pressure from 28 to 26 mm of Hg, which I thought was wonderful, but failed to impress Señor Perez, the plant manager. I think his expectations for a successful project included some sort of gift, or a nice dinner, or something that was beyond my technical expertise.

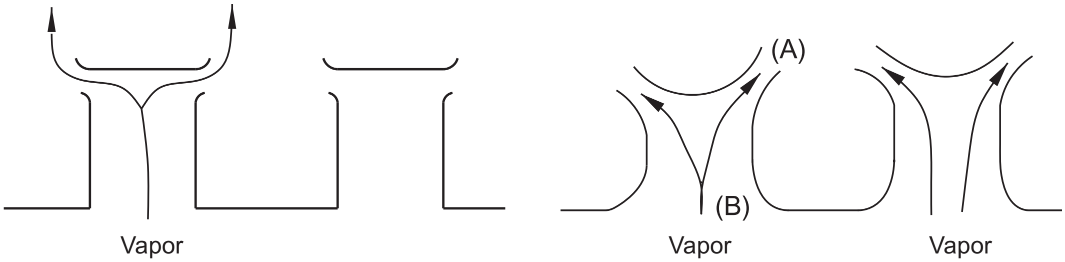

FIG. 11.4 Rounded edges reduce chimney tray pressure by half, for both (A, B) the entrance to each chimney and the entrance to each hat.

12

Troubleshooting Cooling Water Circuits

Plant cooling water systems are often regarded as a fixed entity that produces a circulating flow of water for process cooling, at an invariable pressure, and at a temperature that is only a function of ambient conditions. As time passes, the general reduction in performance of the cooling water circuit is “Little noted, nor long remembered.” But the reduction in cooling capacity due to reduced water flow and increased supply temperature, only too often, reduces plant capacity, product recovery, and operating efficiency.

Cooling Towers

Cooling water for most plants is supplied from a central cooling tower (see Fig. 12.1). Water is distributed or sprayed over wooden slats, or plastic packing, or other types of contact device. Air is drawn up through the slats or packing with a fan, or through a chimney. If a fan is used, it’s induced draft. If a chimney is used, it’s natural draft. The water is cooled by evaporation. For this to happen, the ambient air needs to be less than 100% humidity or below its water dew point temperature. To cool 10,000 lbs of water (or 20 GPM) from 100°F to 80°F, requires:

$$\frac{(10,000\,\mathrm{LB})\cdot\left(100^{\circ}\mathrm{F}-80^{\circ}\mathrm{F}\right)\cdot(1.0\,\mathrm{BTU}/^{\circ}\mathrm{F})}{1000\,\mathrm{BTU}/\mathrm{LB}}=200\text{ pounds of water evaporated}$$

Note that 1.0 BTU/°F/LB is the specific heat of water. The value of 1000 BTU/LB is the latent heat of vaporization of water per pound.

For the water to evaporate into air, air must be below its “Wet Bulb Temperature.”

Wet Bulb Temperature

Wrap a wet cloth around the end of a thermometer. Swing it around for a few moments. If you’re in my home town of New Orleans and the relative humidity is 100%, nothing will happen. If you’re in Laredo, Texas, in the middle of the summer, and the air is bone dry, the thermometer reading may drop 15°F.

The best a cooling tower can do is to reduce the cooling water return temperature to its wet bulb temperature. For example:

- If the cooling water return temperature from your cooling tower is 80°F, and the wet bulb temperature is 75°F, I call that a 5°F approach to wet bulb, which is really good performance of the cooling tower.

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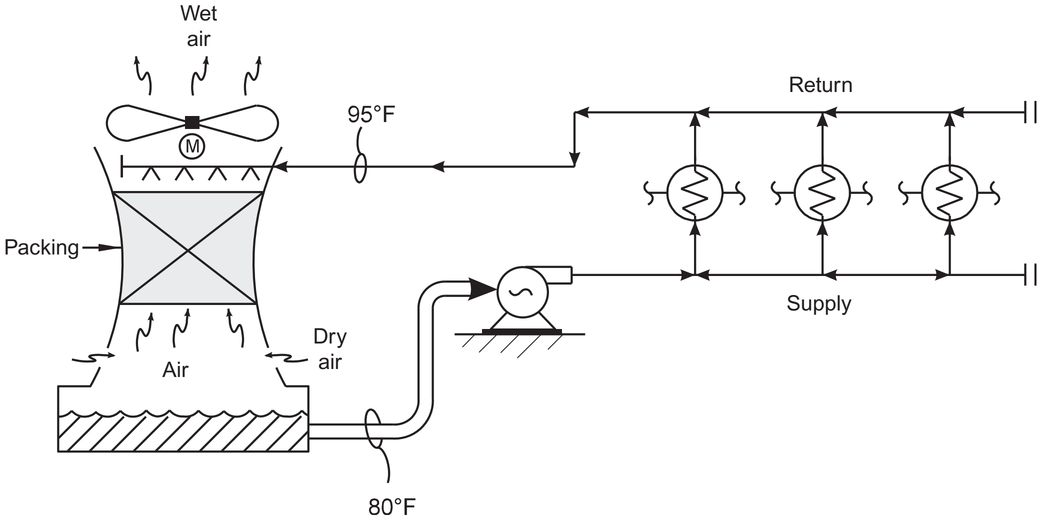

FIG. 12.1 Induced draft cooling tower.

- If, on the other hand, the water return temperature to your unit from the cooling tower is 80°F, and the wet bulb temperature is 65°F, that's a 15°F approach to the wet bulb temperature, which indicates poor cooling tower performance.

  What causes a large (i.e., bad) approach to the wet bulb temperature?
- Reduced air flow
- Poor water distribution
- Tower packing or slat deterioration

Water Pressure

Both the supply and return pressure of cooling water appear to the plant operator to be constant. The supply pressure being the cooling tower pump discharge pressure (typically 60 psig). The return pressure being that required to lift the water to the top of the cooling tower (typically, 15 psig or 35 ft). The difference between the 60 psig and the 15 psig (i.e., 45 psi) is not the pressure drop available to circulate the cooling water through the unit's coolers. We have to also take into account the pressure loss in the supply and return water circulation lines.

Time, Temperature, and pH

"Mr. Lieberman, we require a computer model, 'Pinch Point', analysis of our naphtha reformer. Are you interested in providing a quotation for this project, not to exceed $100,000? Our lack of cooling capacity on the reactor effluent is overloading our recycle gas centrifugal compressor motor, due to excessive molecular weight of the recycled hydrogen," Katherine asked.

"Katie, if you've got the money, I've got the time," I answered.

Katherine was the Technical Service Manager of the Great Northern Refinery in Casper, Wyoming.

"Can we then base the contract on a not to exceed $100,000? If so, assume your final report is due by November 15th. Can we agree on this contractual basis, Mr. Lieberman?"

"Well yes, Katie. But our computer correlations are proprietary, and while we will tabulate results, we cannot share our correlation with Great Northern. Is that acceptable?" I asked.

"Certainly, Norman. I quite agree. Only results need to be reported on the spread sheets."

Pressure Drop Survey

Fig. 12.2 shows the results of the field pressure drop survey of the plant's cooling water circuit:

- Pump suction = 6″ Hg vacuum
- Pump discharge = 67 psig
- Cooling water pressure at exchanger inlets = 58 psig (at grade)
- Cooling water pressure at exchanger outlets = 48 psig (10 ft above grade)
- Cooling water pressure at base of cooling tower, on cooling water return line = 12 psig

The 6″ of Hg pump suction pressure is equivalent to a vacuum of minus 3 psig. That is, each 2 in. of mercury vacuum equals 1 psi of negative pressure. Thus the pump was developing:

$$3 + 67 = 70 \text{ psi of differential pressure}$$

The line loss from the pump discharge to the exchanger inlets on the reformer unit was:

$$67 \text{ psig} - 58 \text{ psig} = 9 \text{ psi}$$

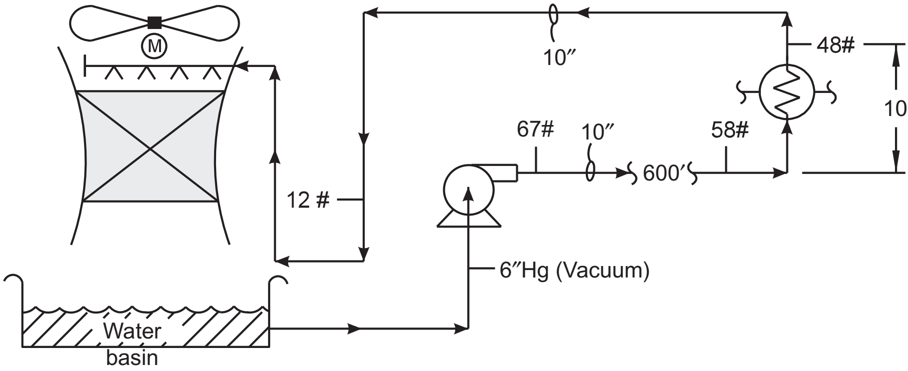

FIG. 12.2 Pressure drop survey.

The pressure drop through the process coolers was:

$$58\,\text{psig} - 48\,\text{psig} = 10\,\text{psi}$$

Correcting for the 10 ft of elevation (2½ ft of water = 1 psi), the pressure loss due to friction in the coolers was:

$$10\,\text{psi} - (10\,\text{ft}/2.5\,\text{ft}) = 6\,\text{psi}$$

And, the pressure drop through the return header was:

$$48\,\text{psig} - 12\,\text{psig} = 36\,\text{psi}$$

Frictional Loss in Water Piping

For my birthday, my wife Liz bought me a Doppler Flow Meter, to measure velocity in cooling water lines. At the cooling water pump discharge, I measured a velocity of 8 ft per second. Line size was 10″. Using a relationship for pressure drops in clean piping:

$$\text{DP} = \frac{4}{\text{ID}} \cdot \frac{(\text{DEN})}{62} \cdot \frac{(V)^2}{28}$$

where:

- DP = pressure drop in 100 equivalent feet, psi.
- DEN = density in lbs/ft^3.
- V = velocity, feet per second.
- ID = pipe inner diameter, inches.

I calculated for 600 ft of the main straight piping that the pressure drop for the supply line should be 7 psi, as compared to the 9 psi observed. As the cooling water return line ran parallel to the supply line, and the flow and ID were identical, I expected its pressure drop for the return water line would also be about nine, rather than the 36 psi observed in Fig. 12.2! Something was very wrong?

Hardness Deposits

“Mr. Norm, when we opened our exchangers to clean deposits from our water coolers, I found our supply line had an inch or so of hard deposits. Looks kinda grayish-white, I reckon,” Travis, the outside operator, noted.

Well, since pressure drop varies with:

$$\text{DP varies (ID)}^5$$

where:

- DP = pressure drop, psi
- ID = pipe diameter, inches

For a 10″ pipe with an average of an inch of deposits (i.e., 8″ of pipe ID, average), DP would increase by:

$$[10'' \div 8'']^5 = 2.0$$

That is, the deposits would cause the pressure drop in the clean water supply line to double, from about 9 psi to about 18 psi. But how about the remaining 18 psi pressure drop in the warm, cooling water return line? Also, what had caused the deposits observed by Travis?

Water pH Control

I checked the circulating water pH. It was 6½.

"Mr. Norm. We got a new plant manager. The old manager tried to save on cooling tower chemicals. So we used to run 7½ to 8 pH. Mr. Johnson, our new manager, he told us to follow our Chemical Treatment Guidelines from the water treatment vendor, which says hold 6½ pH, by addition of sulfuric acid. Seems like our exchanger don't foul-up so fast since then," said Travis.

Of course my equation for frictional loss in piping assumes normal pipe roughness and not that of rougher hardness deposits, or scale. So, that was also part of the high cooling water return pressure drop problem. The 6½ pH target has to be adhered to carefully. Using extra acid to produce a lower pH (i.e., less than 6½) will greatly accelerate corrosion rates in the carbon steel cooling water pipes as I found out the hard way in 1974, at my Texas City alkylation unit.

Cooling water lines can be chemically cleaned with a, "Chelating Solution." But, this is an expensive, off-line procedure.

A Localized Restriction

Fig. 12.3 depicts the rest of this problem. I did a second, more detailed pressure survey. The 10″ line, just before it returned to the cooling tower, dipped underneath a gravel road. On either side of the road, I could measure the water pressure. The observed DP was (26 – 12) = 14 psi, compared to my calculated DP of 1 psi! The next day, I had the road excavated, and found a 10″ gate valve, 90% closed, buried underneath the road, which Travis and I could not move. (Later that year, we had a 6″ "hot tap" bypass installed, which increased water flow by 15%.)

Why or when this valve was installed, no one knew. And why it was mostly closed was also a mystery. However, there was a third problem. Having calculated water circulation rates from the observed water velocity at the pump discharge and the 10″ line, I realized the pump was running well below the manufacturer's flow vs head performance curve.

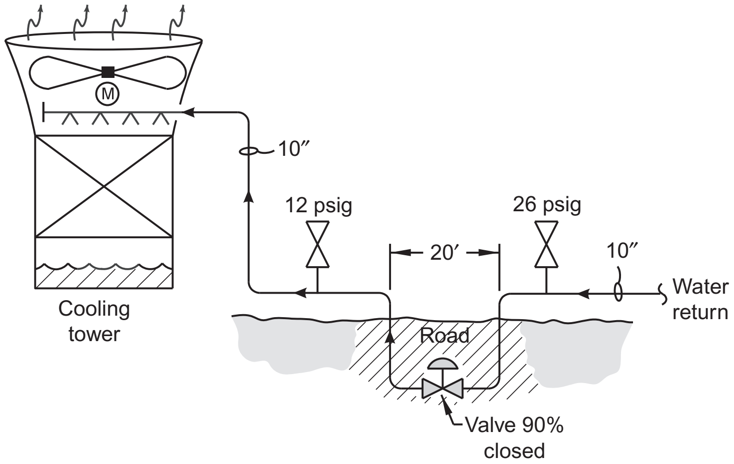

FIG. 12.3 A buried valve partially closed restricted cooling water circulation.

Air Leaks in Suction

“Katie,” I observed, “I want to show you something.”

“What’s that Norman? Have you decided to share your ‘Pinch Point’ computer simulation technology with us? I’m interested.”

“No Katie. That’s too proprietary. What I want to show you is what happens when I shut down the spare cooling water pump for a minute.”

I left the “A&B” pumps running and pushed the stop button on the “C” pump. A few small jets of water blew out of the pump’s suction flange and from several small holes in the suction line to the “C” pump.

“What’s the point Norman? It’s only water. And the leaks are small. I can’t see any safety problems,” Katherine said.

“Well Katie. The pump suction is under a slight vacuum because the pumps are elevated above grade (see Figs. 12.1 and 12.2) by about six feet. Air is drawn into these leaks, into the pump suction and through the pump’s impeller. The air bubbles reduce the flow of water, and the pump’s discharge pressure and flow,” I said.

“Miss. Katie,” Travis interjected, “I believe Lieberman’s right. I’ve noticed that when it rains hard, that our cooling water pumps’ discharge pressure increases a few pounds. I reckon that’s ‘cause that rain replaces the air leakage with water. Let me show you, Miss. Katie.”

When Travis sprayed a torrent of fire water across the suction lines of “A&B” pumps, their discharge pressure increased by 3 psi.

Approach to Wet Bulb

“Mr. Lieberman, did you ever check our approach to the cooling tower’s water return temperature to wet bulb? The tower is designed at 20,000 GPM for an 8°F approach.”

"I have. You're running at 10,000 GPM for a 13°F approach to wet bulb," I answered.

"Any suggestions to reduce this approach?"

"Water distribution may be bad. Perhaps, the old wooden slats in your cooling tower have deteriorated. Check that fan belts are not slipping. Increase the fan blade angle to 22°. Also, check that the blade tip-to-shroud clearance meets the design specification. If motor amps allow, run the fan faster, by putting a smaller wheel on the fan's shaft. This will help increase air flow, but motor amperage will increase a lot."

"Katie, I also have three additional ideas which you may wish to consider:

1. "Try to minimize the pressure drop through the spray nozzle, or cooling water distribution valves, at the top of your cooling water cells. Likely, each psi you can eliminate, will increase water flow by about 1%.

   "On a plant in Kansas, Katie, I found my client was holding a back-pressure on their entire cooling water system, upstream of the return water distribution header. There was a 6″ vent on the distribution header, which would periodically blow water out, and rain down on the unit control center. I opened all the distribution valves, blocked-in the 6″ vent, and opened the now obviated back-pressure valve. Water flow increased by about 10%. The lesson—simple solutions come about from field observations.
2. "A cooling water return temperature above 135°F is going to promote fouling with carbonate deposits of your water coolers. However, this depends on the quality of the water. For some systems, even 125°F return temperature is excessive.

   The company that monitors the chemical treatment of your cooling water system should provide a maximum permitted water cooler outlet temperature for each exchanger to avoid runaway water-side fouling and plugging of the exchanger tubes. Be careful to follow their advice.
3. "Occasionally, cooling towers need to be shocked with large doses of chlorine to kill off bacteria that promote fouling. Spend an hour each month with the rep from the company that treats your cooling tower. Just don't assume he knows what he is doing. Occasionally, these guys are quite incompetent, and are just out to sell the refinery chemicals."

Safety Note

Hydrocarbon leaks in water cooling service can cause explosive mixtures to form below the distribution decks in cooling towers. Especially when cooling butane streams, this can, and has (Amoco Oil—Texas City), caused cooling towers to ignite. Check your cooling water outlets for combustible vapors perhaps once a month. The instrument that the operators use to issue vessel entry permits can be used for this purpose. Vapors tend to accumulate beneath the distribution decks. Sludge buildup on these decks is another indication of hydrocarbon leaks into the cooling water return.

13

Thermosyphon Circulation and Draft

When I was young, I lived on the fifth floor of an old apartment house in Brooklyn. Our apartment had steam heat (Fig. 13.1). I grew up in a room with my sister until she married. It was my room, and I miss it still.

I was not a normal child. To this day, I don't have any friends, which didn't bother me. I've never been that concerned with people. I was more interested in the radiator in my bedroom. In the building's basement was a coal-fired steam boiler. Lumps of coal were shoveled into the fire box by the building Super.

The low pressure steam rose up 50 ft to the radiator in my bedroom, where it condensed to water. The water was denser than the steam. So, the water sunk back down to the boiler. The steam, being less dense than the water, flowed up back to the radiator.

Sometimes, the steam condensate return line would clog with hardness deposits. I could see some of these deposits accumulating around the stem of the steam valve (Fig. 13.1). Mom would then bang on the radiator with a black hammer to knock the hardness deposits off the condensate return line. Mom said she was "bringing up the heat." But, I guess she was restoring condensate drainage. Or, as I would now say, mom was restoring thermosyphon circulation of the steam and water, which is driven by density difference.

That much I figured out by myself. But, what I could not grasp was the function of the small, inverted metal cone, on the top of the radiator. What was that for? I studied this question for 38 years. Through high school, University, and at Amoco Oil, where I worked as a process design engineer. I never asked my parents, or my sister, or Professor Albright, or Bill Duval, the lead process engineer at Amoco in Chicago. And then 1 day, during the long strike in 1980, it happened. A moment of insight!

I was working as a scab at the Amoco Refinery in Texas City during the strike by the OCAW refinery union. I worked 12-h shifts as the outside operator at the Sulfur Recovery Unit.

"Hey Norm," old Joe Patrocelli called, "open the non-condensable two-inch vent on the amine regenerator (Fig. 13.2) steam reboiler. It looks like we've gassed-up the shell. Seems like the steam won't condense. Leave that two-inch vent valve open a couple of turns for the rest of the shift."

Yes, that's it, I thought. The little cone on top of my childhood radiator was a high point vent to purge out gases, that with time, would become trapped in my radiator. If the bedroom radiator filled up with gases, then it would make it harder for the steam to flow into the radiator. The radiator would "gas-up" according to Old Joe. The only thing was that the vent Joe told me to open was not the high point vent

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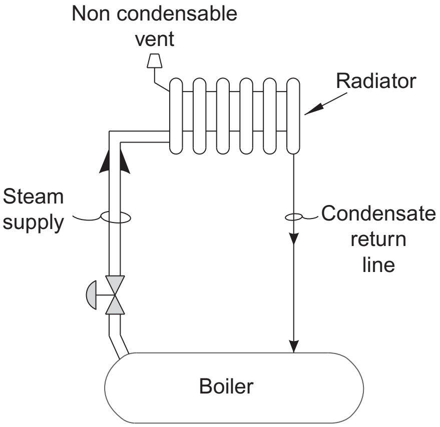

FIG. 13.1 Circulation through steam radiator in New York.

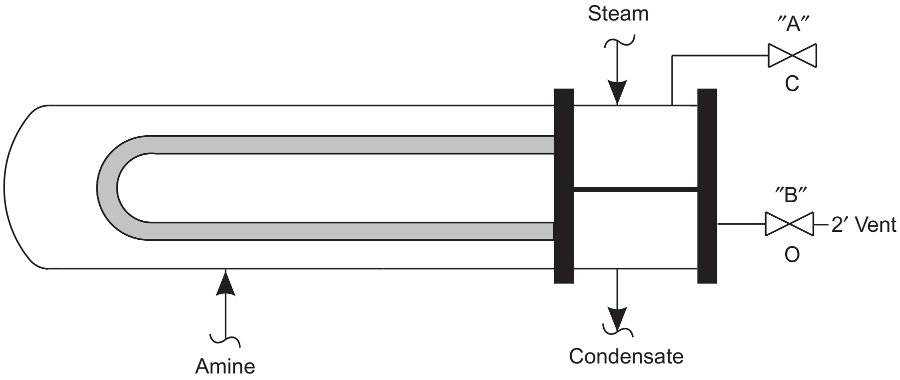

FIG. 13.2 Noncondensable vent on a reboiler.

(valve "A"—Fig. 13.2) on top of the reboiler's channel head. The 2-in. vent was on the side of the channel head (valve "B")?

I always suspected that my professors at Purdue University and my dad and Chief Engineer, Bill Duval, and even my big sister Arline didn't really know all that much. But Joe Patrocelli, a former engine room officer on a US Navy Aircraft carrier during the Battle of Leyte Gulf in 1945, was different. Old Joe was the local representative for the Creator of the Universe.

"Look Mr. Engineer," Joe explained, "gas in the steam to the reboiler blows right through the top half of the channel head and the tube bundle with the steam. It's only when the steam totally condenses in the bottom pass of the tubes, that the gas accumulates and can be vented out. If you open valve 'A,' all you gonna vent is steam! You got that Norm?"

So, I vented valve “A” anyway and steam blew out, and it didn’t help increase the reboiler duty. So, then I tried valve “B” and gas blew out and the reboiler duty doubled. And that’s how I came to know for certain that Lieutenant Joe Patricelli was a representative of a higher authority.

But what was this gas? I suppose the gas in my radiator in Brooklyn would have just been dissolved air from the steam boiler feed water. But, in Texas City, the boiler feed water was first steam stripped to remove all traces of air in the Boiler Plant’s deaerator. So, then I inflated a plastic sample bag with the vent gas and sent it to the lab. The gas analysis was mainly CO_2! But from where? So I asked Old Joe.

“Well Norm,” Joe explained, “you got them carbonates in the boiler feed water! They break down in hot water to CO_2. Not only will they gas-up the reboiler but they’ll cause bad corrosion. You know Norm. The CO_2 will concentrate itself up to form carbonic acid (H_2CO_3) which is real corrosive to carbon steel tubes! Like they learned us in the Navy, best just to vent off those gases where they accumulate. I guess Norm, in 1945, you were just still a baby.”

“Yeah, Joe, but I was already wondering even then about this steam condenser venting problem. I guess I needed the 35 years to understand it all.”

Draft, Thermosyphon Circulation, and the Air Lift Pump

It is not true that University is a complete waste of time. It’s a place to meet girls. Or, so I’ve been told. In my engineering school, Cooper Union in New York, there were no girls. But my student exemption kept me out of Vietnam. So, my University experience did serve a purpose.

One of the subjects we studied, which I did not understand, was draft. I don’t wish to imply that there were subjects that I did understand. It’s just that I remember, most distinctly, not understanding furnace draft. Draft is kind of like the boiler and radiator heating system in my apartment house in South Brooklyn. It works by density difference. The density difference in my apartment house was between:

$$\text{Low pressure steam} = 0.04\ \text{lbs/ft}^3$$

vs

$$\text{Steam condensate} = 60\ \text{lbs/ft}^3$$

You could multiply this density difference by the height of my fifth floor bedroom, above the boiler in the basement, to get the circulation driving force.

For a fired heater, the draft driving force causes combustion air to circulate through the fire box, burners, and up the stack. The density difference that creates draft is between the hot flue gas in the stack and the colder atmospheric air. To calculate draft, I always make a few simplifying assumptions:

- Forget about the difference between the molecular weight of air and the flue gas. It’s more or less the same.

- Forget about frictional losses of the flue gas flowing through the stack.
- Forget about the shape of the stack, creating high exit velocities and inducing extra draft.
- Forget about the effect of the wind blowing across the top of the stack.

First, let's calculate the density of air at standard conditions (60°F and 14.7 psia). Remember that there are 379 ft^3 per pound mole. Then:

$$29/379 = 0.0765\,\text{lbs/ft}^3$$

The 29 is the molecular weight of air.

The stack temperature is 600°F. Don't forget we've got to convert this to Degrees Rankin (°R):

$$459 + 600 = 1059^\circ\text{R}$$

Density varies inversely with the absolute temperature (°R). The 60°F, the temperature at standard conditions is then:

$$459 + 60 = 519^\circ\text{R}$$

The density of the 600°F flue gas is as follows:

$$(0.0765) \cdot \frac{519^\circ\text{R}}{1059^\circ\text{R}} = 0.0375\ \text{lbs/ft}^3$$

Draft is due to density difference. So, let's subtract the density of the flue gas from the density of the air in the atmosphere:

$$\text{Draft driving force} = 0.0765 - 0.0375 = 0.0390\,\text{lbs/ft}^3$$

But, draft is also due to the height of the stack, which is 100 ft.

$$100\,\text{ft} \times 0.0390\,\text{lbs/ft}^3 = 3.90\,\text{lbs/ft}^2$$

There are 144 sq. in. in 1 sq. foot. So:

$$3.90\,\text{lb/ft}^2 \div 144\,\text{in}^2/\text{ft}^2 = 0.0271\,\text{lb/in}^2$$

There are 27.7 in. of water in 1 psi (lb/in.^{2}). Therefore:

$$\text{Draft equals } 0.0271\,\text{psi} \times 27.7\,\text{in.}H_2O\,\text{per psi, or } 0.75\,\text{in.of water}$$

I have been told that I am the last person on earth who knows how to calculate stack draft, without the aid of a computer. Not true. There are dozens of engineers, who can execute this calculation. Unfortunately, they have exceeded their "Sell by Date" and are due, like your author, to expire in the near future.

Imagine a column (100 ft high) of cold, dense air, pushing against a column of hotter, less dense, flue gas. That's draft.

The Air Lift Pump

When I was 12 years old, my dad bought me a fish aquarium (Fig. 13.3). The fish died, and I was left with a vacated aquarium which was okay, as I didn't like fish, but did like to play with the aquarium. The two features I could play with were:

1. The water level.
2. The air flow rate to the riser.

The thing that these changes affected was the rate of water circulation between the elevated filter and the aquarium tank. As air flow increased, the water circulation rate increased up to a point and then it began to decrease. Finally, the water flow completely stopped.

How can I account for this and calculate the water circulation rate?

The first step in designing, operating, or troubleshooting process equipment is to understand how things work. In this case, the underlying process principle is the same as the furnace draft. That is, density difference. The density of the water in the aquarium minus the density of the air-water mixture in the riser line flowing into the filter.

Let's assume that I've adjusted the air flow to the riser line, so that the percent of air and water appears to be 50/50 in the riser line. The density of cool water is 62 lbs/ft^3.

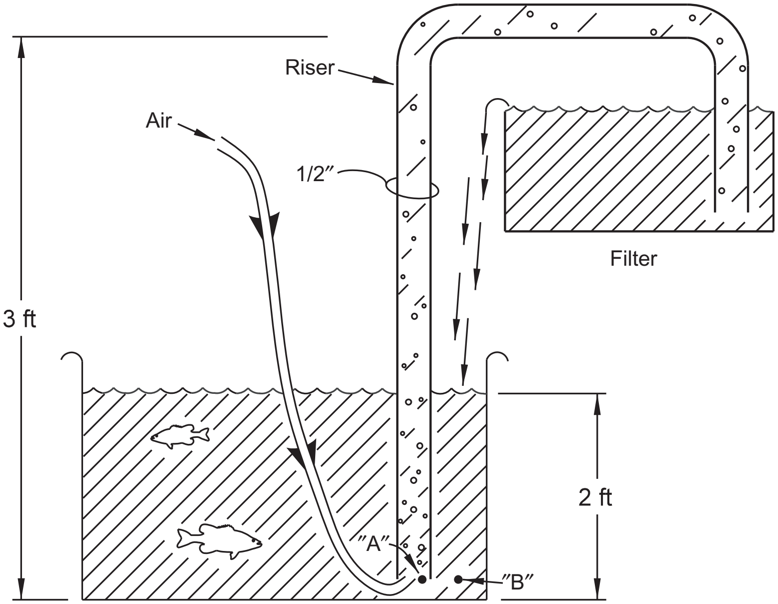

FIG. 13.3 Air lift pump.

The density of air is small, 0.07 lbs/ft^3, so I'll neglect it. The density of the emulsion (air-water mixture) in the riser line is then:

$$62\ \text{lbs/ft}^3 \times 50\% = 31\ \text{lbs/ft}^3$$

The height of the riser tube is 3 ft. The height of the water in the aquarium is 2 ft. Therefore the pressure at pt. "A" at the base of the riser tube is (Fig. 13.3):

$$31\ \text{lb/ft}^3 \times 3\ \text{ft} = 93\ \text{lbs/ft}^2$$

The pressure at pt. "B" at the bottom of the aquarium is:

$$62\ \text{lb/ft}^3 \times 2\ \text{ft} = 124\ \text{lbs/ft}^2$$

Therefore the air lift pump circulation driving force is:

$$124\ \text{lb/ft}^2 - 93\ \text{lb/ft}^2 = 31\ \text{lbs/ft}^2$$

Or, since there are 144 sq. in. in 1 foot:

$$\text{Driving force} = 31\ \text{lbs/ft}^2 \div 144\ (\text{in.})^2/\text{ft}^2 = 0.22\ \text{lb/(in.)}^2 = 0.22\ \text{psi}$$

How Fast Will Water Circulate?

What I like about process engineering is:

1. Calculating some variable—like flow.
2. Measuring that variable.
3. Comparing the measured variable to my calculated variable.

But how can I calculate flow?

Well, the driving force for the water circulation, caused by the water flow, has to equal the 0.22 psi that I just calculated.

How can I calculate the frictional loss through the system? I'll assume that the only resistance to flow is the riser itself. And then I can use the formula:

$$\text{DP} = \frac{4}{\text{ID}} \times \frac{\text{DENSITY}}{62} \times \frac{(V)^2}{28} \times \frac{1}{100}$$

- DP = pressure drop, psi.
- ID = riser pipe inner diameter, inches.
- Density of mixed phase flow, lbs per ft^3.
- V = velocity in riser tube, feet per second.
- L = length of the tube, feet.

The "ID" of the riser or emulsion tube in my aquarium was 0.5 in. "L" was 3 ft. DP was 0.22 psi. Density was 31 lbs/ft^3.

Therefore:

$$V^2 = 0.22 \times \frac{(0.5)}{4} \times \frac{62}{31} \times 28 \times \frac{100}{3}$$

Therefore:

$$V = 7.16\,\text{ft/s}$$

And then, to arrive at the water flow from the filter:

$$\text{Pound of water} = \text{Area} \times \text{Velocity} \times \text{Density}$$

$$\begin{aligned}\text{Pound of water flow per second} &= \frac{0.5\text{ in.}}{144} \times 7.16\text{ ft/s} \times \ 31\,\text{lbs/ft}^3 = 0.77\,\text{lbs/s}\\ &= 0.77 \times 60 = 46.2\,\text{lbs per minute}\\ &= 46.2 \div 8.34\,\text{lbs/gal of water} = \underline{5.55\,\text{GPM}}\end{aligned}$$

I then diverted the flow of water from the filter into a bottle to measure the rate of water flow directly. And that is what Process Engineering is all about—comparing a measured variable to a calculated variable.

The equation used above to calculate pressure drop in the riser line is the standard for frictional loss in piping. However, the coefficient used of four is derived from my experience in making field measurements in refineries and petrochemical plants. I use this coefficient for vapors, liquids, and mixed phase flow. The "L" term is really for 100 equivalent feet of pipe. That is, you should add in equivalent feet for elbows, gate valves, and such. I rarely bother and will just double the actual length of a process line to allow for various fittings, except when it's a long, straight run of piping.

Why then did the flow of water in my aquarium drop as I continued to increase the air flow? Because, as the velocity in the riser tube went up, so did the frictional losses. And when the frictional losses exceeded the air lift pump driving force, water circulation through the aquarium filter slowed and finally stopped.

Furnace Draft Losses

In the same sense, the draft in a fired heater is also consumed by friction. The frictional losses in the furnace are as follows:

1. The brick lining of the stack. Typically very small, if the stack velocity is less than 50 ft per second.
2. The delta P through the convective tube bank. Typically, 0.1–0.2 in. of water, except if the convective tubes foul with ash. Then the pressure drop of the flue gas may increase to 0.5 plus in. of water.
3. The delta P through the burners, which is adjustable, but typically 0.2 in. of water when wide open.
4. The delta P through the stack damper, which is also adjustable and is typically negligible when wide open.

I had assumed earlier in this chapter that I was neglecting the effect of wind blowing across the top of the stack when calculating the furnace draft. But the effect of wind is large. I have observed that gusts of wind will suddenly increase the heater draft by 0.1–0.2 in. of water or by perhaps 20%. This often has a dramatic effect on the flow of combustion air through the burners.

Thermosyphon Circulation

Most distillation towers are equipped with reboilers. I've shown such a reboiler in Fig. 13.4. As there is no pump, what causes the liquid to circulate through the reboiler?—the same force that causes the circulation in my aquarium. That is, the air lift pump principle due to density difference. The density difference is caused by the partial vaporization of the reboiler feed or the distillation tower bottom's product.

Partially vaporizing the feed to the reboiler reduces the density of the reboiler effluent. The mixed phase vapor-liquid emulsion flows up the riser pipe. This emulsion is pushed up and back into the tower by the head of liquid in the bottom part of the tower. The emulsion is less dense than the liquid above the reboiler. To calculate the reboiler's thermosyphon circulation driving force, I would:

- *Step 1*—Calculate the density of the emulsion based on its percent vaporized.
- *Step 2*—Multiply the emulsion density by the height of the riser.
- *Step 3*—Calculate the density of the liquid in the tower bottoms.
- *Step 4*—Multiply this liquid density, by the height of the liquid above the reboiler inlet nozzle.
- *Step 5*—Subtract the pressure calculated in Step 2, from the pressure calculated in Step 4. This is the pressure drop available to drive the "Thermosyphon Circulation."

Some of this differential pressure is consumed in overcoming:

- Nozzle entrance and exit losses.
- Frictional line losses.

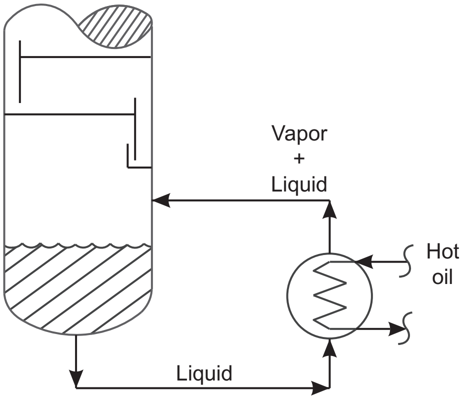

FIG. 13.4 Circulating thermosyphon reboiler.

However, most of Thermosyphon Circulation differential pressure driving force is used to overcome the reboiler bundle shell side pressure drop. When the heat input to the reboiler is increased, two things happen:

- The thermosyphon driving force increases.
- The reboiler pressure drop also increases.

And at some point, the frictional loss through the reboiler will cause the thermosyphon circulation rate to go down, rather than up. When circulation is largely lost, the plant operators refer to this as “Vapor Lock.”

Going Home

It seems as if most of what I know, and much of what I think about process engineering, is built on my childhood observations. As a small boy I was fascinated by the wind rotating my toy plastic pin wheel, by a helium filled balloon, by rainwater flowing into a sewer drain, by boiling water and more than anything else, I was fascinated by fire.

My mother noticed that there was something different about me. “Norman,” my mom would say, “why can’t you be like your sister? She’s so friendly. Why don’t you make some friends?”

“Mom,” I would answer, “can I have that broken thermometer? I need to collect its mercury.”

14

Latent Heat Transfer Concepts

In this chapter, I have summarized with a few stories and references to everyday experiences the interrelated concepts of:

- Latent heat transfer
- Sensible heat transfer
- Reboiling
- Condensing
- Vapor binding
- Flux limitations
- Nucleate boiling
- Effect of noncondensables
- Specific vs latent heats
- Heat transfer rates

To understand how reboilers and condensers function, the above subjects have to be considered together. It's a complex, but necessary process.

My high school Chemistry teacher, Mr. Rosenblat, did not like me. My problem originated in Chemistry Lab.

"Mr. Rosenblat, I don't see any reason to use boiling stones."

"Lieberman. In this experiment, the addition of boiling stones to the flask is standard experimental procedure."

"But Mr. Rosenblat, my mother boils water at home without any problems – and without boiling stones."

"You're not at home, Lieberman. This is 9th grade Chemistry Lab. Here, we use boiling stones. Also, work with your assigned partner, not Linda."

Forty years passed. One morning my wife, Liz, called me from our bedroom, "Norman, make me a cup of tea. Use my teapot – not the microwave. Could you brew my tea in the new Starbucks cup? I'll be a few minutes."

So, I filled the new Starbucks mug with tap water, set the microwave on high for 6 min, and waited for the water to come to a boil—which did not happen. This puzzled me.

Next, I placed the mug of stagnant, hot water, on the kitchen counter and dropped in Liz's tea bag. The water exploded with tremendous violence. Liz came into the kitchen, looked at her empty new cup, and asked, "So where's my tea?"

I suppose I should have anticipated this teacup problem, because the previous winter, I had a similar incident on a crude distillation unit. I had drawn a sample of light naphtha from the reflux drum. The drum was operating at 20 psig, 60°F. I had filled a glass bottle

Understanding Process Equipment for Operators and Engineers. https://doi.org/10.1016/B978-0-12-816161-6.00014-X

with naphtha and taken it to my office. I left the bottle on my desk for an hour and it warmed to 75°F room temperature. But, no loss due to evaporation was evident.

I carefully unscrewed the cap without generation of vapor. So, I dropped a piece of chalk from my blackboard into the bottle, and the naphtha sample erupted from the bottle's top. Now, every time I draw a naphtha sample in the plant, I drop a small pebble into my bottle. Mr. Rosenblat would be proud of me.

Nucleate Boiling Sites

My mother never had any problems boiling water, even without boiling stones. Her pots were old and scratched. Once I knew what to look for, I could see that boiling was initiated at these scratches. Liz's warning not to make "microwave tea" was correct, but only because the tea cup was new and unblemished. I had boiled water in our microwave before without difficulty in our old cups.

What does this mean in a refinery application? Let me apply this "boiling stone" or "nucleate boiling" limitation to a situation where I have had firsthand experience. That is, boiling normal butane on the shell side, using 30 psig steam on the tube side of a horizontal propane-butane splitter, in a Houston refinery.

The tube bundle was old. The tubes themselves were carbon steel. Ten percent of the tubes had been plugged due to leaks. A new alloy tube bundle was installed during the turnaround. When the reboiler was returned to service, the maximum reboiler duty that could be obtained was a small fraction of its previous capacity, even with the larger number of tubes.

The lesson is that for exchangers in sensible heat transfer service (meaning heating material up to a higher temperature), smooth tubes are best, especially because they stay cleaner. But for latent heat transfer, especially when boiling a pure component like normal butane, a rough surface that promotes nucleate boiling is best.

To overcome this unexpected limit on-stream in the Houston refinery, the plant switched from 30 psig steam on the tube side to 100 psig steam. This doubled the temperature difference (Delta T) between the condensing, higher pressure steam, and the boiling normal butane. And since:

$$Q = U \cdot A \cdot (\text{Delta T})$$

where:

- Q = reboiler duty
- U = heat transfer coefficient
- Delta T = temperature difference between the boiling butane minus the condensing steam
- A = exchanger area

The engineering expectation was that the reboiler duty would double, but in fact, the reboiler duty declined further.

This is a true story. As I, your author, must always be the hero of my stories, I heroically suggested the solution: to reuse the old, carbon steel tube bundle with its rough, pitted surface.

When I related this story at my Troubleshooting Seminar for the Exxon Chemical Plant in Baytown, operators in attendance laughed in an uproarious fashion, "Hey Norm! We had that exact same story on our propane-propylene splitter. Even the part about trying higher pressure steam."

Heat Flux Limited Situation

It could be that I'm getting nucleate boiling limitation confused with a "Flux Limited Situation." Or, it could be that both are a manifestation of the same problem.

One day, when I was 16, I carried out an experiment. I took my mother's new stainless steel frying pan and got it really hot on the stove. Then, I carefully poured a glob of water onto the hot surface. Just like a glob of mercury, the water rolled gently and gracefully around the pan and slowly evaporated.

The glob of water was not really touching the hot surface. It was floating on a thin layer of steam. It was kind of a tiny layer of insulation that retarded heat transfer between the pan and the glob of water.

When I shut off the gas to the burner, the pan slowly cooled, and nothing happened for a few minutes. But suddenly, the glob of water collapsed, spread across the pan, and flashed to steam in a few seconds. Apparently, when the pan surface gets cooler, the layer of steam formed that supports the glob of water gets too thin. I don't really know why it happens this way. Try it yourself and draw your own conclusions.

Mom always considered these experiments to be a manifestation of my antisocial personality. But, when I started making big bucks troubleshooting refinery problems based on these experiments, she stopped complaining.

In the engineering literature [1], the flux-limited situation I have described ranges from 12,000 to 20,000 btu/h/ft^2.

Sometimes on new heat exchangers or boilers, I do indeed encounter this limitation. But, rarely on older units, when the heat transfer surface on the vaporizing side has become roughened with age.

Sintered Metal Tube Coating

On an isopentane vs normal pentane splitter that I designed for Amoco in Texas City, I used a rough, sintered metal tube coating, on the tube exterior (steam was on the tube side), that nicely overcomes this flux-limited situation. This was marketed, I recall, by the departed Union Carbide Corporation. A victim of the Bhopal, India tragedy.

An alternate to sintered metal coating would be to lightly sandblast the tubes. On new installations, the flux limitation, which is related to the nucleate boiling limitation, should not be ignored in sizing the surface area of reboilers, when the shell side fluid is a pure component like propane or butane.

Effect of Circulation Rate on Maximum Reboiler Flux

At the Amoco Refinery in Texas City, we found that contrary to our expectations, a higher reboiler circulation rate reduced the reboiler duty. I do not know the optimum rate of circulation, but this observation is supported in the literature [1]. Note, that in this case, the fluid was a pure xylene component, circulating through the shell side of a horizontal reboiler. Amoco R&D calculated the optimum restriction orifice size, to maximize reboiler duty, and it worked great.

Of course, if fouling was a factor, this would have been a poor design feature. Also, heat transfer was 99% latent heat because of the low DP and the xylene being pure.

Vapor Binding

I have often been surprised how a small leak of propane or butane in a steam reboiler greatly diminishes the reboiler duty. When I worked in Texas City, a minor contamination of the 30 psig steam supply to my butane splitter reboiler seemed to cut the reboiler duty in half. I know from experience that CO_2 contamination of steam that originates from carbonates in the boiler feed water (BFW), even in concentrations of only a few ppm, has a large detrimental impact on the steam condensation rate in steam heaters, unless accumulated CO_2 is periodically vented.

In steam turbine vacuum exhaust surface condensers, small air leaks, drawn into the condensers, significantly reduce heat transfer efficiency of the surface condensers, even though the ratio of air to steam entering the condenser is tiny.

I recall quite vividly (see Fig. 14.1) on my Alky Unit in Texas City, how opening a ½″ vent on the channel head of a 60″ ID reboiler, increased the steam reboiler duty by enough to increase the tower reflux rate by 30 plus %.

Whether the noncondensable vapor is air, nitrogen, CO_2, or light hydrocarbons, the effect is the same. The name in the literature for this nasty problem is "*Vapor Binding*."

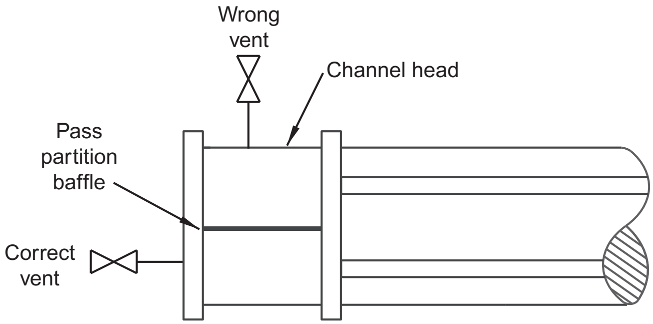

FIG. 14.1 Vent from beneath pass partition baffle.

In a refinery, the most common problem that causes vapor binding is tube leaks of ethane, propane, or butane from the process shell side into the steam tube side.

The way I visualize vapor binding based on reading the literature references [2] is as follows:

1. Ten thousand molecules of steam and 10 molecules of propane flow into a tube together.
2. The steam condenses to water and runs out of the tube.
3. The propane vapor molecules remain inside the tube, because the carrier (i.e., velocity) steam has disappeared.
4. The propane molecules accumulate inside the tube and disproportionately reduce the partial pressure of the steam.
5. The reduction in the steam partial pressure suppresses its condensation temperature.
6. The "Delta T" temperature driving forces is reduced.
7. The reboiler duty falls.

Effect of Vapor Venting

As I do not have X-ray vision, I am just guessing as to the actual mechanism of vapor binding inside a tube for a reboiler, or outside a tube for a steam turbine surface condenser. But what is not a guess is the dramatic effect of venting condensable hydrocarbon vapors from the channel head of my alky giant DIB (De-isobutanizer) tower—the world's largest (see Fig. 14.1—note vent is beneath pass partition baffle, and not the high point vent on the top of the channel head).

In this case the reboiler steam supply was contaminated with propylene from a leaking upstream heat exchanger. This was in 1975. I had great difficulty in understanding how venting off a very few pounds of noncondensables could influence the condensation rate of 40,000 pounds per hour of 30 psig steam. My hourly operators, lacking a University education, were not the least surprised.

"Yeah Mr. Norm," they explained, "adding gas or air to the steam makes the steam weak. You can't expect that reboiler to work hard, with weak steam."

"Nucleate Boiling Limitations," "Flux Limited Situations," and "Vapor Binding" are problems affecting latent heat transfer of pure components that are not always accounted for in troubleshooting and equipment designs. From an engineering perspective, the effects of these problems are counterintuitive. However, from a broader perspective that includes everyday life experiences, these effects are consistent with many common observations [3].

Relationship of Sensible to Latent Heat of Steam

"Mr. Norm," Gerry asked, "could you explain something? I gave my new girlfriend, Linda, a gold pendant for Valentine's Day, and she got really mad. What'd I do wrong?"

"It was probably 12 carat gold, and lacked a diamond. Next question."

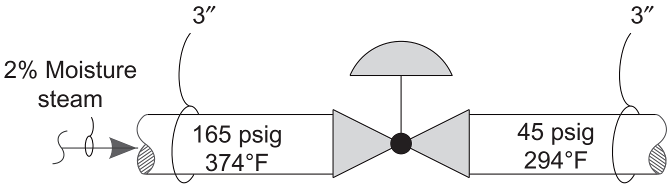

FIG. 14.2 Adiabatic expansion of wet steam.

“Okay. So, I was looking at the flow control valve on the steam supply to the refinery de-butanizer reboiler (see Fig. 14.2). The temperature of the steam is dropping by 80°F across the valve. Why?”

“There are three probable reasons Gerry:

- Speed
- Latent heat
- Evaporation”

“I was thinking also about ambient heat losses, Norm. Ain’t that part of the cooling too?”

“Well, yes. But, let’s discuss that problem last.”

Conversion of Heat Into Kinetic Energy

“Gerry, to accelerate a fluid like steam requires energy. Like when you want to accelerate your truck from 20 MPH to 60 MPH. That takes energy from the engine.”

“Uh! I forgot to tell you Norm that the steam line on both sides of the control valve was the same 3″ size.”

“Okay. But how about the steam pressure?” I asked.

“Well! That dropped across the valve from 165 psig to 45 psig.”

“Okay, Gerry. That means the pressure dropped from 180 psia to 60 psia. Volume of steam, which is inversely proportional to the pressure:

$$\text{Volume proportional}\,(1/\text{pressure})$$

would have tripled. And,” I explained, “the velocity of the steam would have also tripled.”

“Sure. We converted pressure to velocity. Just like across an orifice plate on a flow meter.”

“What you’ve said, Gerry, is entirely correct for a liquid, which is an incompressible fluid. However, for a vapor, like steam, which is compressible, most of the energy to accelerate the vapor comes not from the pressure, but from the heat of the steam itself.”

“I’ve heard that before. That’s what Thermodynamics is all about. But Norm? Still, the reduction in the flowing pressure of the steam must free up quite a lot of energy to help the steam accelerate through the control valve. You know. Converting pressure to velocity?” Gerry asked.

"Gerry, what you say is certainly true for both liquids, like water, and for vapors, like steam. The reduction in pressure of water can be used to make the water flow faster. You can see how powerful that effect is by visiting Niagara Falls. There, the head pressure of water is spinning turbines to generate electricity.

"The difference is, that water is an incompressible fluid. Steam is a compressible fluid."

Effect of Molecular Attraction

"Look pal. If you really like Linda, there are two approaches:

- A single perfect red rose.

  Or
- An $8000 gold watch, studded with diamonds."

"Norm, what does molecular attraction have to do with Linda? I'm not following your ideas."

"Okay, when a liquid loses pressure, its volume does not increase by any significant amount. But, of course, when a vapor loses pressure, its volume expands in direct proportion to its reduction in absolute pressure. That's part of the Ideal Gas Law."

"When steam molecules come closer and closer together, they get hotter. I mean anytime a gas like ethane, air, butane, or steam, is compressed, the gas heats up.

"But when molecules of gas get further apart, the gas gets colder. The further the molecules are from each other, the greater the distance between each molecule of steam.

"So, when a gas like steam or nitrogen or propane expands through a control valve, and the molecules of vapor move further away from each other, the gas is going to get colder. That's called a 'Joule-Thompson Expansion'."

Hydrogen Expansion

"Very interesting. That explains why the temperature of the steam cooled off by 80°F, as it flowed through the reboiler steam control valve. Did you say that's true for all gases?"

"Actually, it's not true, Gerry, for hydrogen. You remember in the hydrogen production plant in Aruba, that I showed you that hydrogen, when it expands from 1,000 psig, down to 860 psig, it heated by 100°F. Helium does the same thing. They act backwards. Hydrogen is said to have a 'Negative Joule-Thompson Expansion Coefficient'."

"Yeah. I see the analogy between Linda and the Valentine's gift. An inverse reaction. Are you gonna explain the Negative Joule-Thompson Expansion Coefficient for hydrogen, Norm?"

"No pal. I don't understand it myself."

Latent Heat Effects

"So Norm! When steam molecules join together to form liquid, water, heat is liberated. Is that all correct?"

"Exactly so Gerry. That's called the 'Latent Heat of Condensation'."

"So then Norm, it follows that if the steam molecules start out really far apart, they must liberate a greater amount of that latent heat of condensation."

"Exactly so."

"Then it also follows," said Gerry, "that the lower pressure steam should have a larger Latent Heat of Condensation, than does higher pressure steam. I'm saying this Norm, 'cause the lower the pressure of the steam, the further apart are the molecules of water. And then, when they come together, and coalesce into liquid water, the lower pressure steam must release more latent heat. Is this all correct?"

"Dead-on Gerry! The latent heat of condensation of 165 psig (11 BAR) steam is around 880 BTU per pound. The latent heat of condensation of atmospheric pressure steam is around 990 BTU per pound."

"Yeah. I got it! So when steam expands, some of its temperature, or sensible heat, is converted to its increased latent heat content! Oh boy! I can't wait to tell Linda about this. She'll think I'm real smart!"

Evaporation

"I don't think that's a real good idea, Gerry. But listen. Likely, a lot of the 80°F temperature drop of the expanding steam was simply due to evaporation of moisture. Let's say the 165 psig steam had 2% moisture, or liquid water. The specific heat of steam is about 0.56 btu/lb/°F. Then, the 2% of water will evaporate to steam, and cool the rest of the steam off by:

$$880\,\text{btu/lb} \cdot 2\% \div 0.56\,\text{btu/lb/}^{\circ}\text{F} = 30^{\circ}\text{F}$$

Ambient Heat Losses

"The ambient heat loss you suspect, Gerry, is likely a real small factor. You can estimate it, by using the following equation:

$$Q = U \cdot A \cdot \text{Delta T}$$

where:

- Q = ambient heat loss, btu/h
- A = uninsulated exposed metal surface, ft^2
- U = heattransfer coefficient (btu/h/ft^2/°F)
- Delta T = temperature difference, °F, between the hot metal surface and the ambient air."

"Right Norm! But what is a typical value for 'U' in this equation?"

"For moderate windy conditions, of 10–15 MPH, use:

Surface Temperature, °F	***U*, btu/h/ft^2/°F**
150	One
300	Two
450	Three
600	Four

As you are only looking, in this case, at the control valve itself being uninsulated, and the adjacent pipe has 4″ of insulation, 'A' will be small and the ambient heat losses, will likely be negligible."

Effect of Surfactants on Foaming

While performing the experiments mentioned in this chapter, you should verify a common light hydrocarbon distillation problem, which I call "Foam Induced Flooding." Fill a pot halfway with water and bring it to roiling boil. Now, drop a handful of pasta into the pot. It will boil over. Your problem is the starch particles have reduced the surface tension of the water and caused it to foam up. In a refinery, the same problem occurs due to particulates of iron sulfide corrosion products. The reboiler return liquid foams up, creates a high froth bottoms level covering the reboiler return nozzle, and the entire tower floods.

Especially in light hydrocarbon distillation towers, this is an extremely common problem resulting in reduced fractionation efficiency due to tray deck flooding.

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[1] D.G. Kern, Process Heat Transfer, McGraw-Hill, New York, 1950, pp. 459–461.

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15

Steam Flow Through Reboilers

"Mr. Lieberman, sir. May I ask you a question? I've only worked in the plant for a few months. I'm totally lost."

I was teaching a Refinery Troubleshooting Seminar at the Etton Refinery in the United Kingdom. Tad, who looked like he was 16, but was likely 23, was not my brightest student.

"What's the question, Tad? We only have a couple minutes before the class starts."

"Well, I got this packed stripper that's reboiled with 3 bar steam. We used to be able to strip 120 tons per hour, but lately all I can strip is 90 or 100 tons an hour. I'm not getting enough heat out of the steam reboiler."

"What's the reboiler look like?"

"Well, sir, I have a sketch of it right here (see Fig. 15.1)."

"Tad. Could you not call me 'sir'? That makes it sound like I'm an old person. Call me Norm!"

"Okay, sir. I guess what really bothers me, is that none of the senior engineers is willing or able or has time to help me. This is a big company, but I feel like I'm all on my own with my problems."

"Look. It's time to start the class. Let's go out into the plant at 5:00 P.M. and look the problem over in the field. What have I taught all of you, Tad, about solving refinery process problems?"

"Well, sir…er, Norm. It's like picking up girls at a bar. You have to get closer to the problem."

Lost Reboiler Duty

Most refinery reboilers, at least in the United States, are horizontal shell and tube exchangers. The process fluid is circulated through the shell side either by thermosyphon circulation, or less commonly, with a pump.

The steam is on the tube side. For low-pressure steam, as in this case, the steam flow is controlled by varying the rate of condensate flow from the condensate pot (Fig. 15.1). As the condensate outlet control valve closes, the steam condensate (water) backs-up and fills some of the tubes. Tubes full of water do not condense any steam.

In a steam reboiler, 90%+ of the heat is derived from the latent heat of condensation of the steam and less than 10% is from sensible heat of the hot water.

Since the Etton Refinery was a subsidiary of a US company, the reboiler was horizontal. A European refinery would have used a vertical reboiler, with the steam of the shell side, and the circulating process fluid on the tube side.

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Condensate Back-Up

The most common problem with steam reboilers using low pressure steam is condensate back-up. Something is restricting the flow of steam condensate drainage. One problem could be the condensate drain valve is too small. That's uncommon. More likely, is the pervasive problem of "Vapor Lock."

When hot steam condensate drains out of the channel head (see Fig. 15.1) of the reboiler, it is close to its boiling point. As the hot water flows through the condensate drain line, it can lose pressure due to:

1. Frictional losses in the drain line.
2. Acceleration losses, as the condensate flows through a smaller fitting or pipe.
3. Increases in line elevation.

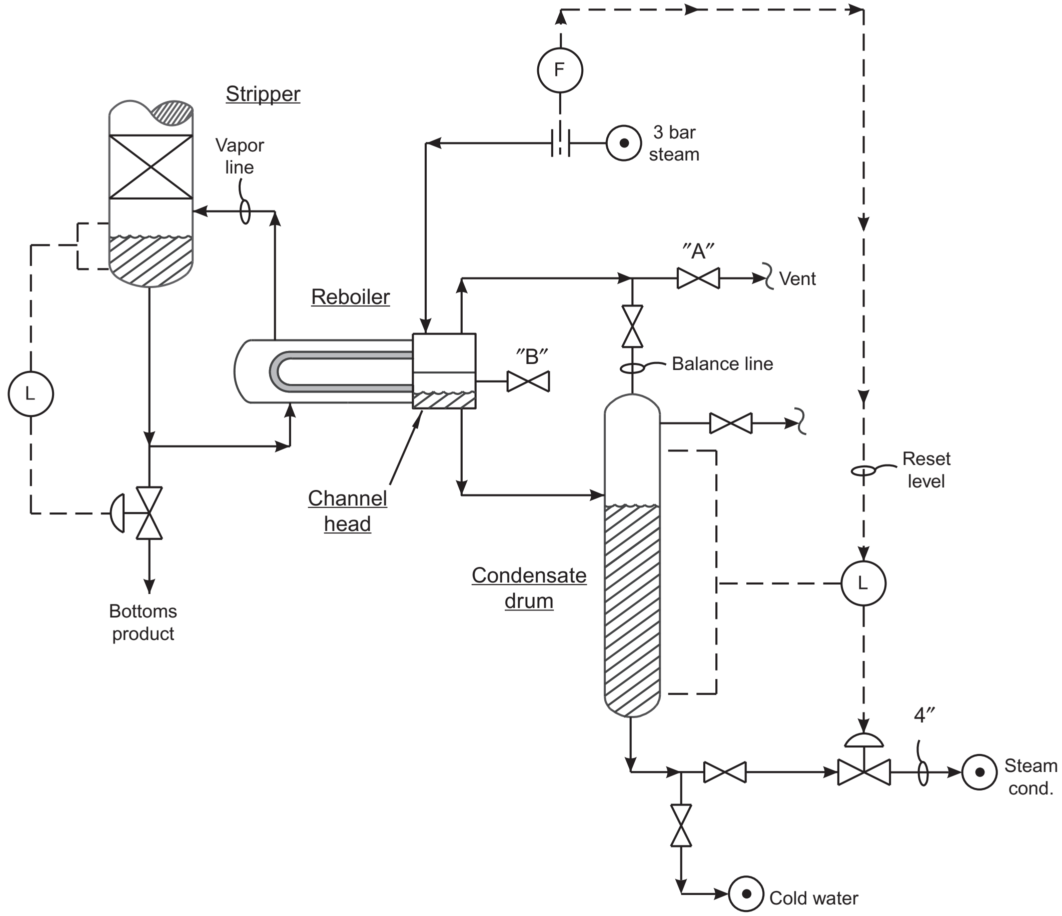

FIG. 15.1 Thermosyphon reboiler.

The hot condensate will have a tendency to partly vaporize. The evolved vapor may increase the volume flow by a factor of a hundred. This huge volume of evolved steam chokes off the flow and backs-up the condensate in the channel head. Result is "VAPOR LOCK" and the loss of reboiler duty.

Blowing Condensate Seal

Every operator on the planet knows how to handle the problem of condensate back-up. They divert the condensate to the deck which creates two problems:

1. The valuable condensate, which is supposed to be returned to the deaerator, or boiler plant, is lost to the sewer.
2. If the drain valve is opened a little too much, steam escapes with the condensate. This is called "Blowing the Condensate Seal," which is bad.

Bad, because the steam is wasted. But far more important, the heat transfer efficiency ("*U*" value) is severely degraded. Whereas high velocities aid sensible heat transfer, high vapor velocities may retard latent heat transfer.

Blowing the condensate seal can instantly reduce reboiler duty by 20%–50%. This is based on my experience and those of field operators.

Noncondensables in Steam

A very small amount of noncondensable vapor contaminating the steam can reduce reboiler duty by half. The noncondensables are due to:

- CO_2—Residual carbonates left in the boiler feed water from the demineralization plant. The carbonates then break down into CO_2 in the boiler.
- N_2—In Texas City, I once found nitrogen in a low pressure steam supply to my alky DIB tower reboiler. Someone had inadvertently cross-connected nitrogen into the steam at some other unit.
- *Light Hydrocarbons*—A small tube leak may cause noncondensable ethane, propane, or butane to enter the steam tubes.

The most common problem is CO_2, formed from carbonate dissociation.
The noncondensables will accumulate in the tubes and retard the rate of steam condensation if they are not vented off frequently. At one plant in Australia, the accumulation of CO_2 was reducing the reboiler duty in a debutanizer by 20%–30%, over a period of several days, before it was vented off from the channel head.

The problem is described in the literature [1] as "Vapor Binding." I have always been amazed how a little bit of noncondensable venting has such a dramatic effect on steam condensation rates—provided the venting is from beneath the channel head pass partition, and *not* on the channel head top vent [2].

Lack of Thermosyphon Circulation

The reboiler shown in Fig. 15.1 is called a "Circulating Thermosyphon Reboiler." The liquid circulates through the shell side of the reboiler due to density difference between the cold inlet side and the hot outlet side.

The cold inlet side is denser because it is all liquid.

The hot outlet side is less dense because it is a vapor-liquid mixture.

The density difference, multiplied by the height of the liquid in the vessel, as compared to the reboiler elevation:

$$\frac{(\text{Liquid ht.}) - (\text{Reboiler ht.}) \cdot (\text{SG})}{2.31} = \text{DP} \tag{15.1}$$

where:

- Ht. is in feet
- SG = specific gravity of the liquid (i.e., 1.00 for water)
- DP = psi

The DP is the thermosyphon circulation driving force. Note that I've neglected the SG of the outlet fluid as its volume is largely low density vapor.

If the thermosyphon circulation rates get too low, reboiler heat transfer duty will suffer. Operators also call this vapor lock.

If the vessel level rises above the reboiler return nozzle, this will also impede the rate of thermosyphon circulation.

The Big Fix

Tad and I did not begin our field investigation until 8:00 p.m. It required 3 hours for my safety orientation. "Tad," I explained, "let's first open the non-condensable vent. The CO_2 will never accumulate above the channel head pass partition baffle, but only beneath the lower channel head partition."

"Yes, Norm, I can see that. The upper part of the channel head will be continuously purged with steam. CO_2 could only accumulate after the steam is condensed. I'll crack open valve 'B' (see Fig. 15.1), and continuously purge the channel head."

The flow of reboiler steam, as indicated on the local flow indicator, climbed by 5%–10%.

"Looks like we did some good, Tad," I observed.

"Yeah! That venting helped. But not by 50%. What's the next step?"

"We need to determine whether we're suffering from condensate back-up or have blown the condensate seal."

"How can we do that?"

"Real easy," I explained. "You monitor the reboiler vapor line skin outlet temperature with your infrared temperature gun. I'll close the steam condensate outlet gate valve part way. Either the reboiler shell outlet temperature will go up or down."

"If it goes down, that means we're losing reboiler duty due to condensate back-up in the tubes. If the reboiler outlet temperature goes up, that means we have been blowing the condensate seal."

"Hey Norm. The reboiler outlet temperature is dropping. We must be hurting from condensate back-up, and not blowing the water seal."

I saw that Tad was checking the water level in the condensate drum, which was well below the bottom of the channel head (see Fig. 15.1). "But Norm, now I'm really confused! How can you say we're losing reboiler duty due to condensate back-up, when the level in the damn pot is a foot lower than the bottom of the channel head?"

"I'll explain; it's complicated. It's due to a common design error:

1. Note that the balance line between the condensate drum and the channel head is connected above the channel head pass partition baffle, at the top vent (valve 'A').
2. Let's assume the pressure drop of the steam through the tubes is 0.1 BAR or one meter of water.
3. Therefore, the pressure in the bottom half of the channel head will be 0.1 BAR less than in the condensate drum.
4. Therefore, the water level in the channel head is going to be about 3 ft, or one meter, higher than the water level in the condensate drum.
5. Which means that almost half the tubes will be full of water."

Correcting Design Error

"They should have connected the balance line below the channel head pass partition baffle," Tad observed. "At valve 'B.' Then the pressure in the condensate drum, and the lower half of the channel head, would have been the same. I guess even big companies can make mistakes."

"Okay Tad. Close off the valve connecting the top of the channel head to the drum. Then, I'll open the atmospheric vent on the drum just a little to reduce the condensate drum pressure by a bit."

"Look Norm, look! The reboiler steam flow just jumped up. And the reboiler vapor outlet temperature is also rising fast! This is great! You got any other ideas we could try to get more heat out of the reboiler?"

Vapor Lock

"I'm not sure, but we may still be suffering from some condensate back-up due to vapor-lock in the 4″ condensate drain line. As the condensate loses pressure in the line, it will tend to partly vaporize back to steam. Even a few percent vaporization can restrict condensate flow and cause the condensate to back-up in the reboiler's channel head."

"So that's vapor-lock. But what can we do about it? Install a 6″ line? Buy a pump?"

"No, Tad. We want to fix the problem tonight. We'll inject some cold water into the 4″ condensate line. This will suppress the tendency of the hot condensate to flash. Let's just remember to call 'Utilities' and have them divert the condensate to the boiler feed water treatment plant."

Increasing Thermosyphon Circulation

"Norm. That cold water injection into the 4″ condensate drum outlet has also increased reboiler heat. But, it's getting kind of late. Should we call it a night?"

"It's only 11:30! Let's try one more thing."

"Okay, but then let's go to the gate."

"Tad, increasing the rate of flow through the reboiler shell side will often, but not always, improve heat transfer. If a lot of the reboiler duty is in the form of sensible heat transfer, higher circulation rates will help. But, if it's all latent heat transfer, higher circulation rates may not help at all."

"But how can we tell for sure if the higher circulation rate is doing us any good?"

"If the tower bottoms temperature increases, we're moving in the right direction," I answered.

The chief operator came over to us, "You guys want to order lunch? They deliver at about 1:00 A.M."

"No," Tad said, "Mr. Henry, we'll be leaving before then."

"Henry, raise up the bottoms level in the tower from 50 to 80 percent. Tad, increasing the tower level will increase the liquid head driving force that causes the flow through the reboiler. That is, it will increase the Thermosyphon Circulation driving force (see Equation #1 above)."

As I waited for plant security to return my passport at the gate, Tad said, "Mr. Lieberman, I forgot to tell you that my grandfather, Harry Jenkins, took your Troubleshooting Seminar back in 1983. Sir, I suppose that you don't remember my granddad? Anyway, he said hello."

"No, Tad. I don't remember your grandfather. But I do remember that I've asked you not to call me Sir."

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[1] C.L. Henderson, J.M. Marchello, Film condensation in the presence of a non-condensable gas, J. Heat Transf. 91 (1969) 447–450.

[2] R.E. Putman, Steam Surface Condensers, ASME Press, 2001.

16

Heat Exchange in Condensers and Reboilers

High velocities and smooth tubes are often an aid in promoting sensible heat transfer in heat exchangers. However, when considering latent heat transfer services such as condensers and especially reboilers, high velocities are often an impediment to heat exchange. This is true for both shell side and tube side flow. Operators observe this when the steam side of a reboiler "blows its condensate seal." Reboilers are often limited by "flux rates" if the tubes are too smooth, as a consequence of retubing an existing reboiler bundle.

However, the main subject of this chapter is an even more common problem that operators experience. That is what I call "Vapor Binding."

Vapor Binding

Ordinarily, the steam sides of reboilers and surface condensers are clean. The steam side fouling factor of 0.001 is normally a sufficient allowance for the steam side fouling with silicates and carbonates. However, there is another type of temporary fouling that greatly inhibits the rate of steam condensation. This is called vapor binding and is a consequence of noncondensables in the steam. There are four origins of noncondensables in steam:

- Tube leak in reboilers, such as those that serve deethanizers, depropanizers, and debutanizers.
- Carbon dioxide due to the thermal decomposition of the carbonates that are not effectively removed in the preparation of boiler feed water (BFW).
- Nitrogen inadvertently introduced into the plant steam supply.
- Air leaks into steam turbines that exhaust to a vacuum due to defective turbine shaft seals and/or surface condenser external leaks.

Fig. 16.1 illustrates this common problem of vapor binding. The steam contacts the cold metal surface of the tubes and drips off. The air, CO_2, or noncondensable light hydrocarbon molecules do not condense. As vapor velocities in a steam condenser are inherently low—at least toward the latter portion of the exchanger—the noncondensable molecules remain in close proximity to the cold exterior of the tube. Thus, they prevent efficient contact between the tube and the condensing steam. Or, in more conventional terms, the steam side heat transfer film resistance increases.

To the observer, it appears as if the exchanger is fouled. And so it is, by vapor binding.

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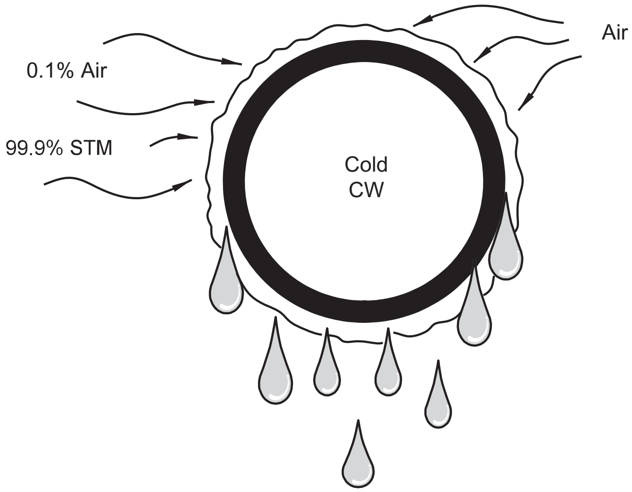

FIG. 16.1 Vapor binding reduces rate of steam condensation.

Depropanizer Reboiler Tube Leak

I was once in charge of the operation of a 23,000 b/d alkylation unit C_3–C_4 splitter. The 30 psig steam reboiler had developed a tube leak. The rate of heat transfer, as measured by the reduction in the 30 psig steam flow, had declined by approximately 50%. However, the operators were easily able to restore the heat transfer rate by venting the channel head. Venting (see Fig. 16.2) from point A at the high point of the channel head was ineffective at restoring the rate of reboiling. Only when the vent at point B was opened, did the rate of steam condensation in the reboiler increase. Point B was located on the channel head, just below the lower pass partition baffle. Venting at point A only serves to vent fresh steam. Venting at point B, after the steam has condensed, purges the noncondensable butane

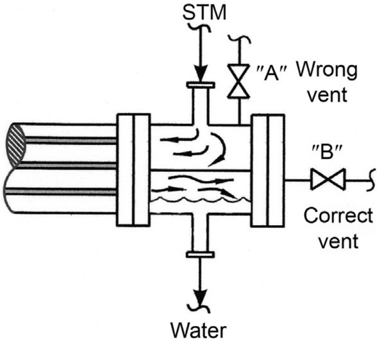

FIG. 16.2 Minimize vapor binding by venting at "B", not "A".

vapors out of the tubes and helps to strip off the noncondensable vapors binding to the outside of the tubes.

Venting From a Vertical Thermosyphon Reboiler

How can a vertical reboiler, with steam on the shell side of the reboiler, which is the typical configuration, be vented for noncondensables? The answer is to periodically blow the condensate to the sewer, through valve C (Fig. 16.3). Experience has shown that this is a far from satisfactory answer. But there does not appear to be any other alternative.

Carbon Dioxide Accumulation

Steam is always contaminated with CO_2. In earlier days, when BFW was prepared by hot lime softening, CO_2 accumulation was a major problem. But, even with modern ion exchange demineralization of BFW, there will be some carbonate contamination of BFW, and hence some CO_2 in the steam. In one refinery, heat transfer coefficients on a debutanizer would slip down over a period of a week, until the CO_2 was blown out of the channel head. While the reboiler was a horizontal exchanger, with the steam on the tube side, the refiner had neglected to connect a valve to the nozzle at point B (Fig. 16.2) and was forced to periodically blow the channel head clear of CO_2, by bypassing the condensate steam trap.

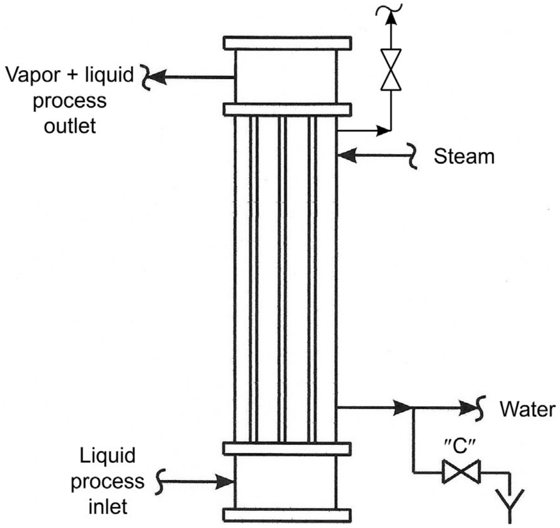

FIG. 16.3 Venting from a vertical thermosyphon reboiler cannot be done efficiently.

In addition to vapor binding and the consequent loss of heat transfer rates, accumulation of CO_2 promotes the formation of carbonic acid (H_2CO_3), which is quite corrosive to carbon steel tubes, even at a moderate pH of 6 [1].

Air Leaks in Surface Condensers

A common problem observed in vacuum towers is excessive back pressure at the ejector discharge due to low heat transfer coefficients in the downstream surface condensers. The problems with surface condensers are as follows:

- Shell side condensate back-up.
- Tube side water fouling.
- Vapor binding on the shell side.

While most of the vapor binding in refinery vacuum towers is due to cracked gas rather than air leaks, the literature describing this problem is based on air leaks in steam turbine exhaust vacuum surface condensers [2]. Whether I am dealing with hydrogen, methane, ethane, and H_2S, or air, the problem and the solution are much the same.

Surface Condenser Air Baffle

A typical design "*U*" value for a refinery vacuum tower surface condenser may be 180 btu/h/°F/ft^2. In practice, even for clean surface condensers, observed *U* values are 40–80. Much of the problem appears to be due to leakage of steam around the leaf seals or seal strips (Fig. 16.4). This allows the condenser inlet steam to bypass most of the tubes and overload the downstream ejector [3]. The best way to retard this leakage is as follows:

- Seal strips should extend ¼″–½″ beyond the ID of the shell, before the tube bundle is reinserted into the shell, so as to crush the strips against the shell ID.
- Seal strip material should be 316 stainless steel of a special flexible grade.
- Three pairs minimum should be used.

In addition, the design of the tube support baffles should allow the vapor trapped just below the air baffle shown in Fig. 16.4 to flow without hindrance to the ejector. Retrofitting an existing condenser shell, as shown in Fig. 16.5, can ensure that this objective will be accomplished.

Cooling Water Inlet Location

In one plant, I improved the performance of the vacuum tower precondensers by reversing the cooling water flow. The general rule for condensers is to have the cold water inlet on the bottom of the channel head and the warm water outlet on the top of the channel head.

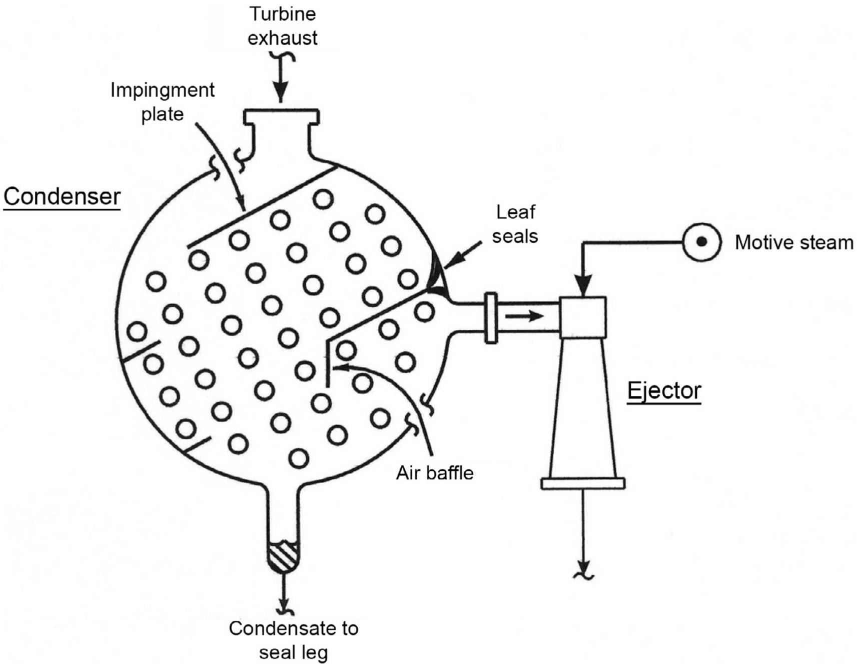

FIG. 16.4 Vapor binding increases surface condenser operating pressure and turbine horsepower.

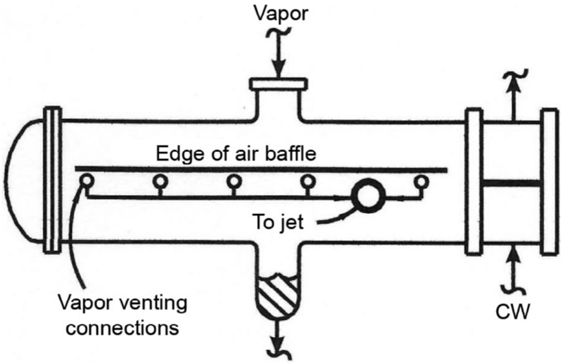

FIG. 16.5 Vapor venting connections underneath top edge of air baffle reduce vapor binding.

For vacuum system condensers, the correct practice is to have the cool water inlet as close as possible to the vapor outlet nozzle that is flowing to the downstream ejector. This will minimize the vapor outlet temperature to the ejector. The lower the vapor outlet temperature, the less the water vapor content of the ejector inlet vapor. This unloads the ejector, which may then develop a greater vacuum.

There should be no concern about high liquid outlet temperature. This temperature does not control the vapor outlet flow. Indeed, a depressed liquid drain temperature is a sign of condensate back-up. Submerging any condenser tubes in condensate is a common way to degrade the condenser's performance. Typical causes of condensate back-up are as follows:

- Leaking seal legs.
- Sludge in the seal drum.
- High seal drum liquid level.
- Excessive seal drum pressure.
- Wax in the seal legs.

If the liquid level rises to cover the lower edge of the air baffle (Fig. 16.4), there will be a sharp break in the vacuum. The cause, in one case, was the lack of sufficient capacity in the seal drum's condensate pump. A high seal drum liquid level backed up the seal legs into the condenser.

Effect of Smooth Reboiler Tubes

On one occasion, I had a horizontal circulating thermosyphon reboiler, with steam on the tube side, retubed. The service was a normal butane-isobutane splitter. The old, carbon steel reboiler bundle had a large percentage of its tubes plugged due to leaks. Also, the exterior surfaces of the tubes were rough and pitted due to extensive corrosion. When the new bundle was commissioned, it demonstrated a capacity of less than half of the older, discarded bundle.

The problem was a loss of nucleate boiling sites. The exterior pitting of the tubes promoted rapid normal butane vaporization. The smooth tubes of the newly retubed bundle resulted in a "flux limited" situation [4]. With smooth tubes, this occurs at about 12,000 btu/h/ft^2.

To overcome this flux limited situation, apparently caused by a loss in the tube surface roughness, the steam supply to the higher pressure 100 psig refinery steam header was lined up, and the flow of 30 psig steam shut off. This made the loss of reboiler capacity worse. Finally, the new reboiler tube bundle was extracted from its shell and aggressively sand blasted to restore surface roughness. Upon its return to service, the reboiler performed adequately.

An alternative and perhaps more conventional way to avoid this problem is to use a sintered metal coating on the exterior of the tubes, which provides a uniform, rough reboiler surface. This is called a "high flux" tube coating.

De-superheating Vapors

The discharge flow from a propylene compressor was superheated due to the heat of compression. While 80%–90% of the downstream compressor condenser's capacity may be latent heat removal, and only 10%–20% sensible heat removal, a much larger proportion than 10%–20% of the condenser surface area will be devoted to de-superheat, rather than to condensation. This is a consequence of the relatively low heat transfer coefficient for sensible heat removal, as compared to latent heat removal.

Spraying a slip-stream of liquid propylene product into the compressor discharge, while reducing the temperature difference driving force, will actually improve overall condenser capacity by eliminating the need to de-superheat the compressor discharge, by direct heat exchange with the condenser tubes.

This is a variation of the practice of injecting steam condensate into a superheated steam supply to a reboiler to increase the reboiler duty. This was tried by varying the steam condensate injection rate into 400 psig, 650°F, superheated steam and noting the effect on a vertical, thermosyphon reboiler's capacity. The benefit was small, but noticeable. It is probably not worthwhile for a new installation, but a useful method to expand the capacity of an existing steam reboiler that is using superheated steam or an existing condenser, on the discharge of a light hydrocarbon compressor.

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17

Crude Unit—Heat Exchanger Fouling

"Just look at that bundle! Half the tubes are plugged. And the plugged tubes are all bent. Look at that channel head pass partition baffle! It's also bent. Pushed out towards the channel head outlet nozzle. And that damn partition baffle has pulled itself away from the channel head tube sheet gasket (Fig. 17.1)."

Mike Greenly, the first line supervisor on No. 12 Pipe Still seemed to think that somehow I was to blame for the distorted bundle. It was true I had calculated the operating "*U*" (heat transfer coefficient) was only:

- Eight to nine btu's/h/ft^2/°F

vs the design coefficient of 45.

Also, I had recommended that the exchanger be taken off-line for cleaning. Mike had a hard time isolating the exchanger, and this had made him even more angry. "Mike, why are you mad at me? This is an unavoidable consequence of tube side fouling. That's not my fault – anymore than it's your fault."

"Look Lieberman! You told us to take E-307 off-line for cleaning. It was your idea. No mention about them tubes and bent pass partition baffle. I'm gonna have to get the entire bundle re-tubed. Also, I'm going to need a new channel head pass partition baffle. How you suspect I'm gonna cool the diesel pumparound without E-307? Man! And we're sure to lose twenty degrees of crude preheat."

"Would you like me to explain, Mike, how this happens? And, what we can do to avoid a similar failure with a re-tubed bundle?"

"You engineers all the time talking too much. Go ahead, explain. You babble on and I'll get me a coffee."

Which Side Should Crude Be Placed on

Ninety percent of refiners place the crude on the wrong side on a shell and tube heat exchanger. That is, on the tube side upstream of the desalter. High viscosity fluid like crude should normally be placed on the shell side. Laminar flow can't develop on the shell side due to the turbulence which I call vortex shedding, as the liquid flows perpendicularly across the tubes between the tube support baffles.

But, putting the dirty crude on the shell side makes it difficult to clean the shell side unless:

- Tube spacing = 1½″
- Tube size = 1″
- Tube pitch = rotated square

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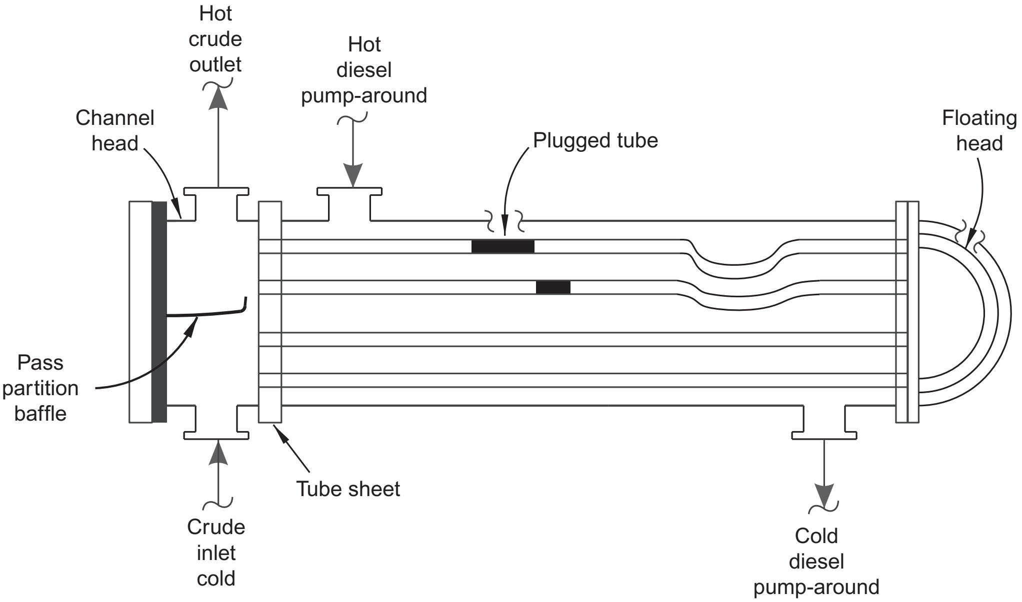

FIG. 17.1 Plugged tubes cause failure of channel head pass partition baffle.

This is the design practice I learned from the now idle Standard Oil Refinery in Aruba. It creates ½″ cleaning channels between rows of tubes. Unfortunately, most refineries place the crude flow on the tube side, regardless of the crude oil's viscosity. High viscosity materials are best placed on the shell side, to avoid loss of heat transfer due to laminar flow, which may readily develop on the tube side, due to a low Reynolds number (Re):

$$\mathrm{Re} = \frac{(\text{Density}) \cdot (\text{Velocity}) \cdot (\text{Tube diameter})}{\text{Viscosity}}$$

The only way to realistically offset high viscosity is a high velocity. But any velocity much above 12–14 ft per second will cause erosion.

Why Tubes Foul

Tubes foul in crude oil service due to four factors:

- Low velocity
- Dirt
- Heat
- Surface roughness

Low velocity is a big problem. Any velocity less than 3 ft/s is going to promote fouling. Any velocity above 12–14 ft/s is erosive. The problem is that once a slow tube starts to foul it will:

- Have less flow.
- It will get hotter because of the less flow.

- Its velocity will diminish because of the lower flow.
- The lower flow will lead to more fouling.
- More fouling will cause less flow.
- Less flow will cause the tube to get hotter.

It's like a positive feedback loop has been created that generates fouling. At the same time, and for the same reason, the tubes that have started out running a bit faster will see more and more flow as the slower tubes foul. The faster tubes will run cooler as the crude from the slower, hotter tubes is diverted through the swifter, cooler tubes.

This uneven fouling rate of individual tubes will continue until:

- Some percentage of the tubes are totally plugged and heat up to the shell side temperature.
- Some percentage of the tubes are running free, and clear and cooler. And fast enough to prevent further fouling.

The tubes that are plugged and hot experience thermal expansion. However, the tubes that are clear and colder grow less because they are cooler. The colder tubes restrict the horizontal expansion of the tube bundle (see Fig. 17.1). The Floating Head expansion is constrained by the colder, shorter tubes.

But the hotter tubes must grow. No force in the universe can resist their expansion. So, since they cannot grow horizontally, they do the only thing possible. The hot tubes bend and twist. And that's what we see when we pull a crude preheat exchanger bundle—the effect at differential rates of thermal expansion between hot and cold tubes.

Effect on Heat Transfer

The distorted tubes block and restrict areas on the shell side. Shell side flow stagnates in these blocked areas. This leads to shell side fouling and flow distortion. This is apparent when the bundle is pulled.

If half of the tubes are plugged, the tube side pressure will increase from around 10 to 40 psi. The increased pressure drop will cause the pressure difference across the Channel Head Pass Partition Baffle (see Fig. 17.1) to also increase by a factor of four. Eventually, the pass partition baffle will bend and pull away from the channel head tube sheet. And now, some of the crude flow will entirely bypass the tube bundle. The lower crude flow velocity, due to this bypassing, will make the problem worse, by promoting higher rates of fouling as the tubes run hotter and slower.

Surface Roughness

One way to mitigate the fouling tendency of tubes running at a low velocity is to use smoother tubes. That is, use alloy tubes. In crude service, stainless steel would not be advisable due to chloride stress corrosion cracking. After all, crude oil is typically 0.5% sea water, upstream of the desalter. But, we could use chrome tubes that would resist corrosion and stay smoother longer, and hence cleaner.

When using alloy tubes instead of carbon steel, make the entire bundle, including:

- Tubes
- Support baffles
- Tube sheets
- Impingement plate
- Seal strips

from the same material, to avoid galvanic corrosion between dissimilar metals exposed to an aqueous phase (i.e., the salt water in the crude, upstream of the desalter).

For ordinary carbon steel tubes, a reasonable design minimum tube side velocity, depending on the type of crude, might be:

- Louisiana light—5 ft per second
- Heavy venezuelan—10 ft per second

These are guesses based on my limited experience. Certainly, anything less than 3 ft per second is looking for trouble, excessive rates of tube side fouling, and distorted tubes.

Increasing the Number of Tube Side Passes

Mike Greenly stared at me with open hostility, "Lieberman, you should have designed these crude bundles right off with higher tube velocities!"

"Mike, I didn't design this exchanger train. The crude unit was an American Oil Central Engineering Design. Don't blame me."

"Yeah Lieberman, it was another of your engineering screw ups! What we gonna do now, Mr. Engineer?"

"Well Mike. We can change the exchanger pass configuration. We'll go from two to four, tube side passes. We will have two, pass partition baffles, in the channel head. Plus, a single new baffle in the floating head (see Fig. 17.2). This will double the tube side velocity from three feet to six feet per second."

"Okay Norm. But I'll have to have both tube sheets re-machined. But that's okay, if it will prevent the tubes from fouling. But how about the DP? Won't the tube side pressure drop also double?"

"No Mike, it will go up by a factor of eight. From five to 40 psi."

"Eight! Why's that Norm?"

"Because Mike:

1. Velocity will double.
2. Pressure drop varies with the linear velocity, squared.
3. The flow path length will also double, because the crude has to make four, instead of two passes through the tube bundle. That is 80 ft vs 40 ft."

"Yeah I can see that. But I don't know if we got enough extra head on the crude charge pump for all that extra pressure drop?"

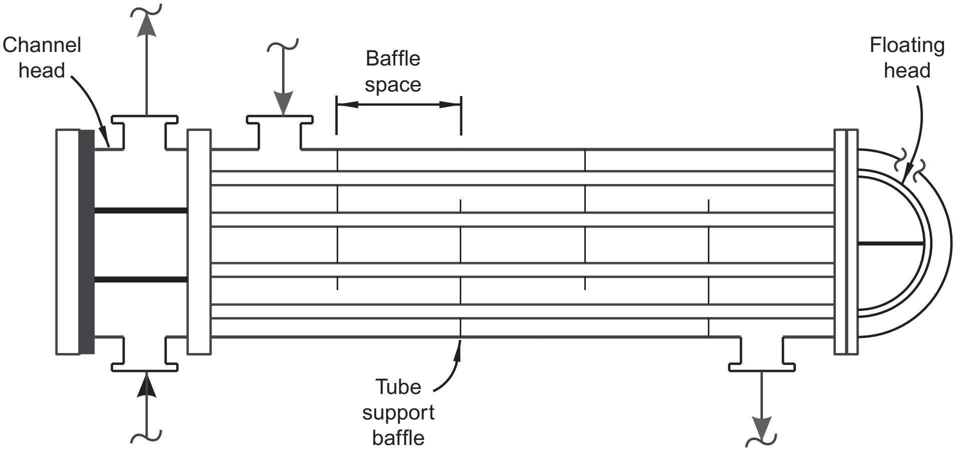

FIG. 17.2 Four pass exchanger.

"Actually Mike, it's not all that bad. The higher tube side velocity will suppress fouling in the tubes. A lot of our pressure drop in the crude exchanger train is due to fouling. Especially if we make the tubes out of 410 chrome steel, which will stay smoother than carbon steel tubes. The pressure drop will be higher than before. But, after a year of operations, maybe the DP won't be all that bad compared to your current operation."

"Look Norm, I guess that you're pretty smart. Can't you just figure out a way to have clean crude tubes, and a low crude Delta P?"

"No Mike. It's kind of like your plan to keep your girlfriend and stay married to Marge. That didn't work out either."

"Yeah Norm! I get your point. You can't have it both ways."

On-Line Spalling

"But look here Lieberman. You got any other ideas as to how I could partly clean a crude preheat exchanger without taking it off-line completely?"

"What do you have in mind Mike?" I asked.

"Like, inject some kind of magic chemical into the crude that would clean an exchanger, without pulling the bundle. Without all this isolating, blinding, and draining and hydro-blasting? Norm, I'm not asking about the shell side. The diesel pumparound and other products are clean. Only the crude side – the tubes are fouled."

"Okay Mike. But only on exchangers upstream of the desalter."

"Yeah! These are the worst ones."

"You might want to make some notes:

- *Step One*—Open up the by-pass gate valve on the crude side (see Fig. 17.3).
- *Step Two*—Block-in the crude side inlet valve.
- *Step Three*—Wait 15 min, and then open up the crude inlet valve and close the by-pass valve.

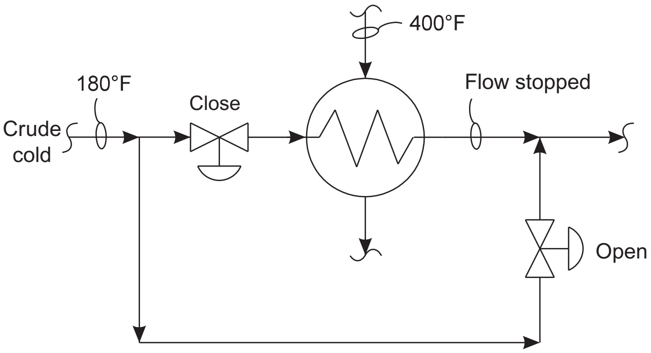

FIG. 17.3 On-line spalling of crude preheat exchanger.

During the 15 min, the tubes will heat up 100–200°F closer to the shell side temperature. The deposits of dirt inside the tubes will spall-off. The sudden restoration of crude flow will flush the deposits out of the tubes, and into the desalter."

"How do you know that's going to work, Lieberman?"

"Because, Mr. Greenly, I've tried it on No. 11 Pipe Still last month. Heat transfer rates doubled, and pressure drop was cut by half."

"Sounds good Norm. It's like spalling coke out of a fired heater tube. Same idea. Can't hurt to try. But why didn't you tell me this before? You damn engineers. Don't know how to communicate with us regular folks. You all go around with your heads in the clouds."

Shell Side Cross-Flow Velocity

Note that I have said that the higher viscosity fluid ought to be on the exchanger shell side, to avoid laminar flow and poor heat transfer on the tube side.

However, laminar flow can occur on the shell side as well, if the cross-flow velocity is too low. Cross-flow velocity means the velocity between the tubes at the edge of the tube support baffles (see Fig. 17.2).

If this velocity is above 3 ft per second, then the tubes induce "Vortex Shedding." This means as the liquid flows perpendicularly across the tubes, it creates twirls and turbulence which destroy the heat transfer film resistance boundary layer. So, I will select the "Baffle Spacing" as shown in Fig. 17.2 that will create a cross-flow velocity of 3–5 ft per second. Sometimes this requires a tiny baffle spacing of only 4 in. and a huge number of baffles per shell. Once, I had an exchanger with 50 tube support baffles in a single exchanger shell!

Cutting the baffle spacing by half doubles the number of baffles and increases the shell side pressure drop by a factor of eight (not four). Both the shell velocities have doubled and so has the flow path length, and pressure drop varies with velocity squared.

I don't leave it to the exchanger vendor to specify the baffle spacing. I do this myself. Also, I specify the number of tube side passes, the tube spacing, the type of tube pitch (triangular or rotated square), and the size of the tubes.

18

Guide to Revamping Process Units

Basically, there are two distinct approaches to retrofitting process units to increase efficiency and/or capacity:

1. Computer modeling.
2. Extrapolation from plant current operating conditions.

In the following examples, I have recalled three actual stories that illustrate this fundamental problem which process engineers must confront.

I have found after many years of revamping process units that the best method of creating a design that results in achieving the refinery's requirements is to rely on prototypes and to personally conduct a unit performance test.

Computer modeling is currently the generally accepted method when retrofitting distillation towers, heat exchangers, flash drums, piping systems, and rotating equipment. In the 1960s, this was not the case. The question explored in this chapter is, has the computer modeling resulted in improved process design or has it been a distraction from the fundamental problems.

The older methods largely relied on extrapolation from prototypes, field data, and plant performance tests.

Preflash Tower Case History

As an example of the problems that I have typically encountered, refer to Fig. 18.1. This is a crude tower preflash drum. The crude unit was limited by heat removal capacity. Ambient conditions at the time of my observations were about 30°F. The operating conditions I observed were as follows:

- 26 psig at the top of the tower
- 438°F vapor outlet
- 162,000 lb/h vapor flow
- 390,800 lb/h liquid flow

The vapor flow of 162,600 lb/h I calculated from an orifice-type flow meter was based on a lab analysis of the off-gas collected in a sample cooler. The liquid flow was measured at two points:

1. At the pump discharge orifice place.
2. The individual, dual pass flow meters at the crude preheat furnace inlet.

Both of the observed liquid flows were in substantial agreement.

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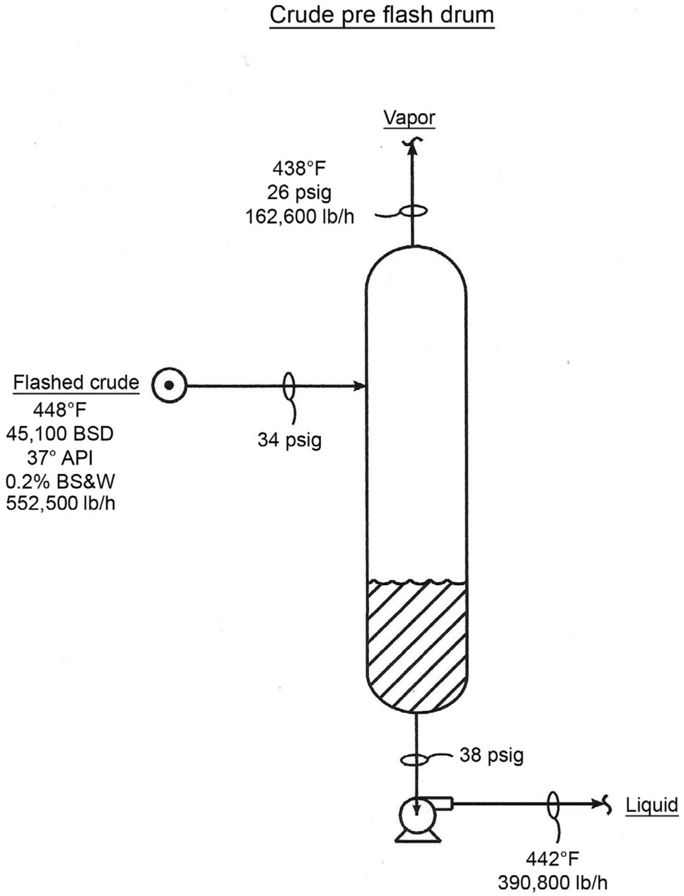

FIG. 18.1 Observed crude flash drum operating conditions.

Simulation Comparison

A computer simulation of the 37°API crude flash at:

- 438°F
- 26 psig
- 45,100 BSD

Indicated the flows of vapor and liquid to be:

- 184,600 lb/h (vapor)
- 367,900 lb/h (liquid)

That is, the calculated vapor flow, at the observed operating temperature and pressure, was ***13.6%*** larger than that observed in the field during normal plant observations.

The object of this particular crude unit retrofit design was to increase vaporization in the crude preflash drum, with the objective of reducing pressure drop in the downstream crude unit fired heater, by improved heat exchange upstream of the preflash drum. I was then faced with the choice to use as the design base case:

- The 162,600 lb/h vaporized as observed in the field.
- Or, the 184,600 lb/h vaporized as calculated from the computer simulation model.

The objective of the project was to increase the vaporization in the crude preflash drum to about 200,000 lbs/h. Thus, the decision as to which base case vapor flow to use, the observed in the field or that calculated from the computer simulation, would have a very large effect on the design strategy I employed in selecting changes to the upstream crude preheat exchanger train.

Design Basis

To be consistent with the observed operation, I selected the observed crude vaporization rate as a basis for further design calculations. The difficulty then arose, as to how to use the existing crude unit simulation model for design calculations, as the starting point for the simulation was not consistent with the percent vaporization assumed as the basis for the unit retrofit?

This example typifies the problem discussed in this chapter. That is, the inherent inconsistency between observed versus simulation model predictions for:

- Crude unit product draw-off temperatures
- ASTM D-86 5%–95%, gaps and overlaps
- Pressure drops
- Exchanger enthalpy changes

The problem I describe, I have also encountered, not only for crude towers, but for:

- Alkylation units
- Coker fractionators
- Visbreaker fractionators
- Vacuum lube oil towers
- FCU fractionators

One conclusion that I can draw for certain is that to ignore the inconsistency between observed data and simulation results will result in a design that does not reflect reality.

Distillation Tower Modeling

The designer modeling a distillation tower must input two parameters:

- The number of theoretical trays.
- The Equation of State.

These parameters are needed to determine the vapor-liquid traffic in the distillation tower to achieve a given split between the components in the feed. The vapor-liquid traffic is a function of the reflux rate, the pumparound duty, and the reboiler duty.

The number of theoretical stages is used to determine the number of actual fractionation trays, or the height of packing, and hence the height of the distillation tower, based on the tray efficiency, or for packing HETP (height of packing equivalent to a theoretical separation plate or stage). The lower the tray efficiency, or the greater the HETP, the taller the distillation tower.

Selection of Tray Efficiency

I have seen tray efficiency vary in distillation towers between 10% and 100%. In refinery distillation towers, as opposed to pilot plant testing equipment, tray efficiency is mainly a function of the mechanical condition of the trays. In particular:

- Tray deck levelness.
- Outlet weir levelness.

The larger the diameter of the tower, the more critical these level parameters become in their effect on tray fractionation efficiency. Longer weirs, with low weir loadings (1–2 GPM per inch), are more adversely affected in regards to tray efficiency by out-of-level weirs compared to shorter weirs, with very high weir loadings (12–14 GPM per inch). But how does the designer take the weir levelness into account when determining tray efficiency? The designer could assume a reasonable out of levelness for the weir, but that is equivalent to assuming the answer.

The same problem occurs for determining the effect of tray deck levelness on tray efficiency. For perforated tray decks, such as valve, sieve, or grid trays, the trays are subject to leakage. Localized leakage may occur when the pressure drop of the vapor flowing through the tray openings (delta P dry) falls well below the weight of liquid on the tray (delta P wet). Localized tray deck leakage will promote vapor-liquid channeling, poor vapor-liquid contact, and impaired tray efficiency.

The larger the flow path length between the inlet downcomer and the outlet weir, the more the out of levelness of a tray deck will reduce tray fractionation efficiency. The designer can assume a reasonable degree of tray levelness and then assume a reasonable tray efficiency, based upon the assumed tray levelness. But, assuming a degree of tray deck levelness, the tray efficiency is also being assumed, and hence the number of trays, and thus the height of the fractionator.

The use of a computer model gives the designer the false impression that the number of trays in a distillation tower is being calculated with a degree of precision that does not correspond to reality. The designer would do just as well, or even better, by estimating the number of trays for a particular application based on an existing prototype in a similar service and of similar size, rather than expending time in computer modeling. This engineering approach to distillation tower design assumes that there are existing prototypes to base new designs upon. In my work, there are many prototypes for distillation service.

Equation of State

The first decision that the engineer makes in establishing a distillation tower simulation is selection of the Equation of State from perhaps 25 choices. The Equation of State is used to calculate the Relative Volatility of each component. The Relative Volatility is then used to calculate:

- The reflux rate
- The reboiler duty
- The condenser duty
- Vapor-liquid flows
- The number of theoretical separation stages or trays
- The tower diameter

based on required degree of fractionation specified.

But, how does one know which Equation of State to select?

I have long experience on this subject. At one plant, a new propylene-propane splitter was being designed to replace an existing tower that could not make the spec's at the new, higher design feed rates. This fact was supplemented by a computer model of the splitter, which showed that the tower would be limited by operating at over 105% of jet flood, at the required design feed rates and reflux ratio.

I found that the historic limit to the tower was not flooding, but reboiler duty. The reboiler was limited by the steam condensate drainage rate. Diverting the condensate to the sewer for demonstration allowed the new design feed rates to be achieved. The reflux rate had previously been limited by the reboiler duty. The reboiler duty had, in turn, been limited by the steam flow to the reboiler. The steam flow to the reboiler had been limited by steam condensate back-up. The ultimate constraint was that the relatively short 3″ steam condensate drain line had not been sized to allow part of the steam condensate to flash to steam. It was just designed for hot water. A common piping error.

What was wrong with the computer model of the splitter? Nothing. Depending on the Equation of State selected, the simulation could be used to predict that the existing splitter tower was too small or that it was adequately sized for the new design rates. Selection of low tray fractionation efficiency values reduced the theoretical capacity of the splitter even further.

Incidentally, the new tower was constructed, in spite of my observations based on field test work, because the computer modeling had *implied* that the existing tower was too small.

Pressure Drop in Process Lines

The following discussion assumes mixed-phase, low viscosity, fluid flows. By low viscosity, I assume less than 10–15 cSt. Most process streams are below this viscosity. Mixed phase vapor-liquid flows will be maintained above 20–25 ft per second.

To calculate a pressure drop using a computer correlation in refinery process piping systems requires inputting the piping friction factor. The friction factor is derived from the roughness factor. The roughness factor is assumed and is not measured. Thus, the calculated piping losses, using pressure drops calculated by computer, are based on a guess and are not actual precisely calculated values.

As an example of the problems introduced by the overreliance on computer-derived piping pressure drops, I was recently involved in a project in a Kansas refinery to debottleneck a crude distillation unit. The capacity limit was the fractionator relief valve shown in Fig. 18.2. As the crude rate increased, the pressure drop in the overhead vapor line also increased until the pressure at the top of the fractionator approached within 5 psi of the relief valve set-point pressure.

The observed delta P between the top of the fractionator and the inlet to the downstream air coolers, as shown in Fig. 18.2, was 16 psi based on the difference between the two (PT_1 minus PT_2) pressure transmitter outputs recorded on the control room consul.

The computer model of the overhead piping system calculated a Delta P for the 500 ft of 14 in. diameter pipe of 14.5 psi, which was in a reasonable agreement with the observed pressure loss of 16 psi (see Fig. 18.2).

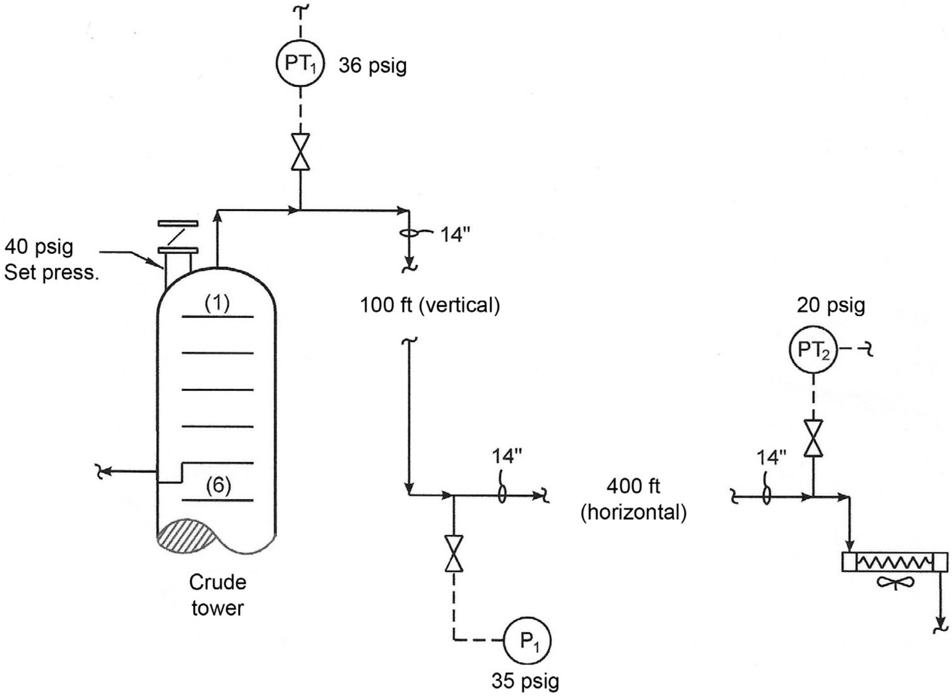

FIG. 18.2 Observed crude tower overhead piping pressure losses.

Using the following simplified equation:

$$\mathrm{DP} = 4/\mathrm{ID} \cdot (\mathrm{DEN})/62.3 \cdot (V)^2/27.7 \tag{18.1}$$

where:

- V = velocity, ft/s
- ID = pipe inner diameter, inches
- DP = line loss per 100 equivalent feet, psi
- DEN = mixed phase density, in pounds per cubic foot

I calculated a pressure loss for the 500 ft of 14 in. pipe, of 6 psi. I assumed that the piping was clean. My calculated pressure drop (6 psi) was well below the observed (16 psi) pressure drop. It was my theory that the assumption that the 14 in. piping was clean was wrong. The line had been in service for many years and had never been cleaned as there were no clean-out flanges at the elbows. Also, crude tower overhead vapor lines foul due to:

- Accumulations of iron sulfide corrosion products.
- Amine chloride salt sublimation.
- Degraded olefinic light slop components in crude charge.
- Poorly dispersed filming amine.

My client was faced with a decision. My recommendation was to clean out the vapor line during the forthcoming crude unit turnaround. This would require the costly addition of four, 14 in. clean-out flanges. The alternate recommendation, based on the computer-generated pressure drops, was to replace the existing 14 in. line with a new 18 in. line. Note that pressure drop varies with pipe ID to the fifth power.

To discriminate between my calculated 6 psi DP and the computer calculated DP, I made a field pressure measurement at P_1 (see Fig. 18.2) of 35 psig. The field observed pressure loss in the 100 ft of vertical piping was 1 psi. I then concluded that the 15 psi pressure drop in the remaining 400 ft of horizontal piping was three times greater than our calculated DP of 5 psi, due to fouling. Solid deposition would preferentially accumulate in the long horizontal run of 14 in. pipe and not in the 100 ft of vertical piping.

What was wrong with the 14.5 psi pressure drop calculated using the computer piping model? Nothing. Depending upon the specified surface roughness factor selected for the calculation of the friction factor, the model could predict a DP of 6 psi or 14.5 psi.

As the cost of replacing the existing 14 in. line with a new 18 in. line was prohibitive, the refiner selected the more moderate cost option of the four new 14 in. line clean-out flanges. The horizontal run of line was cleared of many buckets of a black sludge and then hydro-blasted clean. On start-up, the overall line pressure drop fell to approximately 5 psig, and the relief valve no longer limited the crude charge rate, which was increased by about 15%.

Rigorous Determination of Friction Factors Inside Process Piping

The correct way to calculate pressure losses in piping was developed in 1857 by M.H. Darcy, “Recherches Experimentales relatives au movement de leau dans les tuyaux,” Mallet-Bachelier, Paris, France. Or, a more accessible reference is, “Handbook of Hydraulics,” King & Brater, fifth ed., McGraw Hill, 1963.

Basically, the pressure drop between two points is measured in the field, in a particular service, and then extrapolated to the new service. This is the method that I employed by use of Eq. (18.1) above.

Crude Preheat Exchanger Network

A typical crude preheat exchanger train will utilize heat from:

- Crude tower overhead vapors
- Desalter brine
- Top pumparound
- Heavy naphtha product
- Kerosene product
- LVGO pumparound
- Kerosene pumparound
- Diesel product
- Diesel pumparound
- Vacuum tower overflash product
- Atmospheric gas oil product
- HVGO product
- HVGO pumparound
- Vacuum Resid

Integration of these sources of heat to preheat crude is a complex matter, especially when trying to represent an existing heat exchanger train. There are a number of proprietary computer models available to implement a solution to this computer problem. Greatly adding to the complexity of the task is that the “*U*” values (heat transfer coefficient) vary with:

- Fluid properties
- Fluid velocity
- Fouling factor

While the stream fluid properties and velocities may be calculated or inputted with little source of error, the fouling factors are another matter. They must be assumed by the process engineer. Unfortunately, in crude preheat, heat transfer service, it is my experience that the controlling resistance to heat transfer is not film resistance, but fouling resistance on the crude (i.e., tube) side of the crude preheat exchangers.

What then is a reasonable crude side fouling resistance? In °F/ft^2/h/btu, I would use:

- 0.002–0.100

Depending upon the circumstances.

If the design engineer specifies crude side fouling factors in the range of 0.002–0.004, a crude preheat temperate as high as 520°F may be calculated. If fouling factors of 0.050–0.10 are assumed, a crude preheat temperature as low as 360°F may be calculated.

While the computer-generated crude preheat exchanger model can rapidly calculate the crude preheat temperature, the result largely depends on the assumed fouling factors. The impression is the crude temperature to the heater has been calculated with precision. The reality of the situation, is that this temperature is mainly a function of an arbitrary guess as to the tube side fouling factors.

It is not uncommon, in my experience, for "*U*" values of 10–15 btu/h/ft^2/°F to be observed. Often, this is a result of the refiner charging the crude unit, out of the same tank used for crude receipts, and for recycling refinery slops to this same tank.

The problem with the heat exchanger computer simulations is that they cannot be used to predict the real crude preheat temperature with any degree of accuracy. These programs are more of a distraction than an aid. They divert the process engineer's focus from the main problem. That is, the cause and the prevention of crude tube side fouling.

The key to high crude preheat temperatures is to:

- Never charge the crude unit from the tank in which the crude is received.
- Always add some of the desalting chemicals to the crude charge tank.
- Do not run mixers on the crude charge tank.
- Do not have the crude draw-off nozzle on the bottom of the crude charge tank. Employ floating suctions.
- Remove sludge from the crude tanks. Do not attempt to flush tank sludge through the unit heat exchangers.
- Keep the tube side velocities in preheat exchanger train, above 6 ft per second.

Vacuum Tower Bottoms Composition

A large portion of my work is retrofitting refinery vacuum towers to minimize vacuum residue and to maximize HVGO recovery by reduced flash zone pressure. I have found that computer modeling of the vacuum tower does not consistently reflect plant operations. The problem is in the flash zone vapor-liquid flash calculations. For the computer model to execute the equilibrium flash calculations, the number of moles of liquid must first be calculated. The moles of liquid depend on the molecular weight of the liquid. However, the liquid phase, to a great extent, consists of asphaltenes, which are not liquid, but complex aromatic rings. These asphaltenes, which are solids, are dispersed in the liquid, oils, and resins.

The method I use to achieve a vacuum residue yield that corresponds to reality is to manipulate the composition of the vacuum tower feed until the expected result, based on a prototype vacuum tower operation, is duplicated in the computer simulation. Without reliance on such a prototype operation, I would not have confidence to produce a design based on a simulation.

To a lesser extent, this same limitation, due to the asphaltene content in crude, applies to:

- Crude atmospheric towers.
- FCU fractionators (problem is slurry composition and not asphaltenes).
- Coker fractionators.

Performance Test Data

Survey using handheld infrared temperature indicator.
Pressure readings using a single pressure gauge.
Pressure drop across distillation trays using a single gauge.
Position of control valves.
Exchangers partly bypassed.
Pump and compressor motor amps.
Speed of turbine-driven compressors and pumps.
Pressure drops across flow meters.
Cooling water supply and return pressure and temperatures.
Ambient conditions (including barometric pressure if a vacuum system is included in survey).
Gas chromatic of fuel gas and process streams.
Sample for ASTM D-86 distillation.
Heater flue gas excess O_2 and consumption.

19

Problems in Computer Modeling of Process Units

I was the first person in history to design a complex distillation tower using a digital computer. Not the first distillation tower designed by computer. That was common in the early 1960s, using a program marketed by IBM. My 1966 design of the "B" Delayed Coker Fractionator in Texas City was the first including:

- Pumparounds
- Product side draw-offs
- Side stream steam strippers

I wrote the program that employed "Fortran Four," using "Stream-Add," and "Equip SEP" statements. Conversion was done by a "Do-Loop" using 50 cycles. Run time was 30 min and the cost was $300 per run.

I assumed that each equilibrium separation stage was equal to a tray with 100% tray efficiency. The "Relative Volatility" of all components, I based on the pure component's vapor pressures. That is, I ignored nonideality factors. What I should have done is used the appropriate "Equation of State" to determine the correct relative volatilities of individual components. But that degree of accuracy was beyond me in 1966.

"B" coker was commissioned in 1968 and performed quite well. You can still see it in operation from State Highway FM 1764 in Texas City, just south in the now Marathon Galveston Refinery.

Computer Simulation of Fractionators

When a process engineer designs a distillation tower using a computer simulation provided by ASPEN, he or she must specify (Fig. 19.1):

- Equation of State selected from perhaps 30–60 choices.
- Tray efficiency.

The incremental cost to run the program is negligible, and run time is a few seconds.

I have observed tray fractionation efficiency between 5% and 100%, depending mainly on the:

- Tray levelness—Most important
- Weir levelness
- Percent of jet flood

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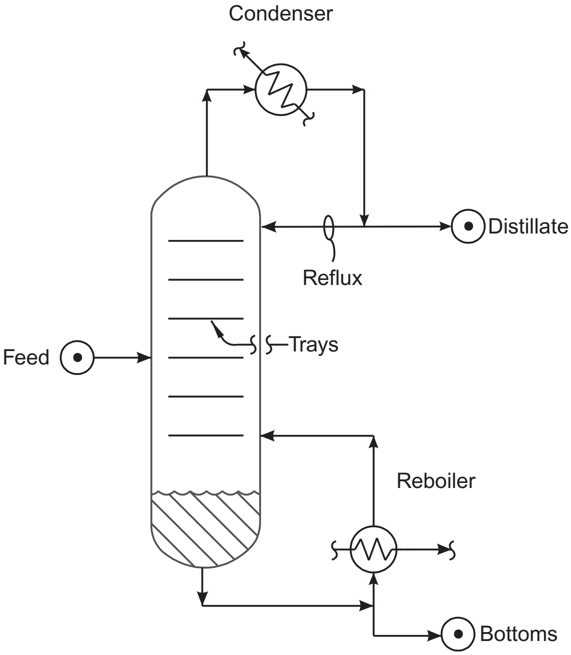

FIG. 19.1 Distillation column.

- Hole or valve cap open area
- Tray flow path length—Least important

Tray efficiency is therefore just a guess. But, how does one know what Equation of State to select for the computer model? The equation of state is used to determine the relative volatility of components. This is then used to calculate:

- Reflux rate
- Reboiler duty
- Condenser duty
- Tower diameter
- Number of stages
- Product rates
- Product parities

As the selection of the equation of state is normally just a guess as is the tray efficiency, the computer model output is also just a guess which is often wrong.

Oil-Water Settling Velocity

The rate at which hydrocarbons and water separate is used by computer models to size settlers. Typically, a settling rate of:

1. 0.5 ft per minute is used for settling water from diesel oil.
2. 0.7 ft per minute is used for settling water from light naphtha.
3. 1.0 ft per minute is used for settling water from LPG (i.e., propane).

is assumed for horizontal gravity settlers, for low viscosity (i.e., less than 10 est) fluids for the computer model.

However, in reality, the sizing of such settlers is mainly a function of other factors:

- Has the hydrocarbon-water mixture been pumped with a centrifugal pump?
- Has the mixture passed through a control valve with an appreciable pressure drop?
- Has the mixture undergone a large reduction in velocity?

While we do not often pump fluids with reciprocating pumps, mixed phases pumped with recip's settle ten times faster than fluids pumped with a centrifugal pump. I made this measurement myself in the American Oil Refinery in Indiana.

Trying to calculate a settler size with a downloaded computer method, without taking into account what has caused the dispersion of the phases, is an exercise in futility. The largest common variable is the pressure drop through the upstream control valves. Increasing the pressure drop through the control valve from 20 to 200 psi will increase settling times by a factor of 10 to a 100.

Heat Exchanger Efficiency

Calculating the performance of a complex heat exchanger network is typically done with a "Hextran" computer program. The program itself calculates the heat transfer coefficient ("*U*") based on fluid (Fig. 19.2):

- Viscosity
- Velocity
- Thermal conductivity
- Density

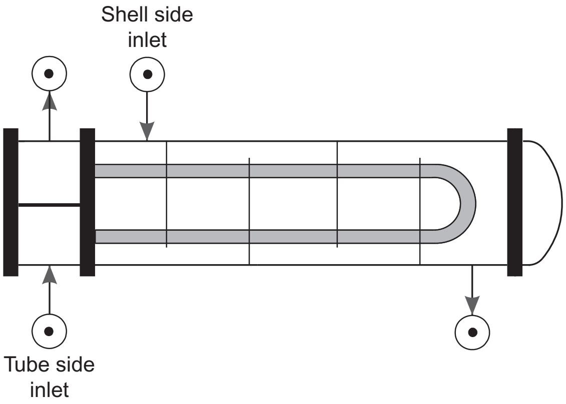

FIG. 19.2 Heat exchanger.

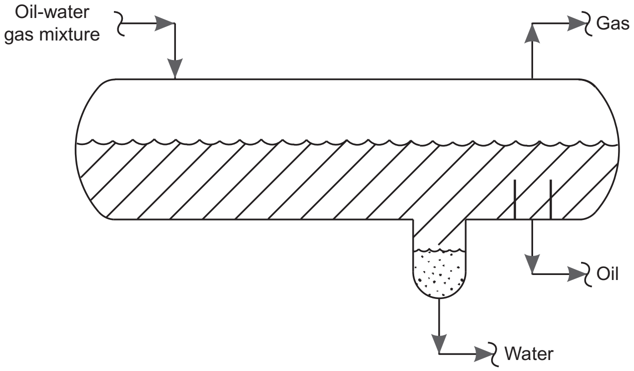

FIG. 19.3 Settler.

as well as the geometry of the heat exchanger. That is:

- Tube size
- Tube pitch
- Baffle spacing
- Baffle cut
- Pass configuration
- Seal strips—Location and number
- Internal clearances
- Tube metallurgy

Unfortunately, the user has to provide the tube and shell side fouling factors, which range from 0.10 to 0.001 btu/h/ft^2/°F, for each side.

For most heat exchangers I have worked with in refineries, at least after a year or two of service since being cleaned, heat transfer coefficients are predominantly a function of fouling. Rates of fouling are largely unknown and unknowable unless they are measured in the field.

My conclusion is that unless a heat exchanger network analysis is based on field measurement, its predicted performance in regards to the:

- Pressure drop
- Heat transfer rate

cannot be calculated with a computer model. The main problem is that rates of fouling are largely a function of:

- Velocity
- Tube surface roughness
- Nature and concentration of the fouling deposits.

While the heat exchanger network computer simulation generates a great deal of information, it is all ultimately based on a guess as to the fouling factors, which are extremely variable, especially as fluid velocity changes.

Pressure Drop in Pipes

To calculate the pressure drops in pipes, we input to the piping hydraulic program:

- Line length
- Fittings and valves
- Pipe ID
- Flow rate
- Fluid viscosity
- Fluid density
- Friction factor or roughness factor

The friction factor is calculated from the roughness factor. Typical roughness factors were derived in 19th century England using cast iron pipes flowing water. However:

- Have they been updated since then?
- Do they apply to your system?
- Will they change with time and the pipe's environment?

Who knows? Not me!
Having guessed at the friction factor, the resulting calculated pressure loss is also only a guess.

Fired Heater Limitations

Most natural draft fired heaters are usually limited by combustion air and/or draft. Both draft and air are functions of (Fig. 19.4):

1. Stack height
2. Flue gas temperature
3. Fire box height and temperature
4. Pressure drop of flue gas in the convective section
5. Wind
6. Ambient temperature
7. Tramp air leaks into the convective section

Wind, as I have observed, changes draft by 0.1–0.2 in. of H_2O which can be 20%–40% of the total available draft. The other big unknown variable is cold tramp air leaks, which suppresses the draft developed by the stack, by cooling the flue gas. Also, the third major unpredictable variable is fouling on the fins in the convective tube bank. Therefore, available computer models of fired heater capacity cannot be a true reflection of reality.

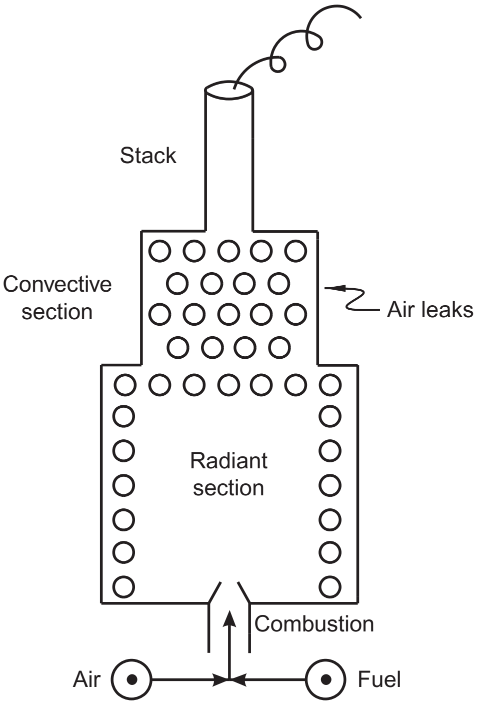

FIG. 19.4 Natural draft fired heater.

Effective Design Methods

At the start of this discussion, I described my design of "B" coker in Texas City in 1966, using a computer model. How did I address the above design concerns?

Adjacent to "B" coker was the old (and now demolished) "A" coker. I spent a week gathering plant data covering the operation of "A" coker and adjusted my coker computer model so that I could simulate the performance of "A" coker. I then made small changes to the "A" coker model for my design of the new "B" coker.

I have continued to use this method of process design for the last half century. Computer modeling is fine. But, we must base the model on a firm foundation. That being a plant performance test where the resulting parameters are used to adjust the models:

- Tray fractionation efficiency
- Relative volatility of components
- Piping pressure drops
- Heat transfer coefficients
- Fired heat capacity
- Oil-water settler sizing

As an old Harry Belafonte song teaches, “A House Not Built on a Firm Foundation, will not Last.” A process design for dual, new 150,000 BSD crude units, built for a new refinery in Saudi Arabia, based on a computer model in an Engineering Contractor’s office in Europe, has proved a disappointment to their clients. The contractor failed to base their design on an existing crude unit’s operation. I always tell the young engineers working with me on design projects, “copy whenever possible – calculate only as a last resort.” The trick is to know which designs to copy.

Safety Note

Dangers of Nitrogen

In 1982 I had a great idea to improve the poor reliability of our “Instrument Air System” at the Good Hope Refinery in Louisiana. I had the nitrogen supply line connected to instrument air. The control valve that would allow nitrogen to flow into the instrument air header would open when the instrument air pressure fell below 30 psig.

At that time, our control room instrumentation was powered by instrument air. Each instrument used 5–10 SCFM of air.

One afternoon, while I was observing the refinery operation in the control room, a strange alarm sounded, followed by a flashing red light. Everyone looked around puzzled. It was the “Low Oxygen” alarm which was set at 18% O_2. My N_2-to-Instrument air pressure back-up system had been triggered by low air pressure.

The effect of nitrogen dilution of air is not shortness of breath or any physical discomfort. We killed three workers in Texas City when they entered a reactor that had been purged with N_2. The effect of excess nitrogen and low oxygen is death.

I have advocated since that incident at Texas City that N_2 should have a few % of CO_2 added when it is used for vessel purging. The CO_2 will cause shortness of breath and alert personnel of the lack of oxygen.

20

Does Computer Modeling Reflect Reality?—Diesel Freeze Point

This is a true story. Word for word. Number for number. All I've done is to change the names except for Kathy. I believe this is the most important section in this book.

The reader will be disappointed that at the end of this short chapter that I did not resolve the huge discrepancy between the lab data and the computer model. Politicians, lawyers, and bankers can function in the theoretical world. Process Operators and Engineers are constrained by reality.

Freeze point is often the limiting factor in the production of diesel from certain crudes, depending also on the location in which the diesel is marketed, and the time of year. In a US refinery, the primary objective was to maximize diesel production, which was limited by freeze point. An engineering consulting firm from Houston was retained to study the possibilities of enhancing diesel production from crude.

A number of cases were developed using an Aspen computer model simulation of the crude distillation train. Seventeen cases were developed, which showed that diesel flow could be increased between 3000 and 11,000 BSD. Capital investment for the various cases ranged from four million to twelve million dollars depending on the incremental recovery of diesel from the heavier products.

The Technical Manager, Ed, retained me to review the 17 cases and to recommend the most cost-effective option. In my preliminary discussions with Ed, I drew the conclusion that he did not understand the various cases, and had retained me to assist him in their evaluation.

The volume of material that the consulting company had generated in the past year was immense. Reams of simulation results and dozens of preliminary process flow sheets with modifications shown in red. Each case was accompanied by the calculated incremental diesel production and a capital investment cost estimate.

I concluded that the most economically attractive case was the option that resulted in the 11,000 BSD of the maximum incremental diesel production.

Whether or not any of the 17 cases was practical to actually implement, I could not determine. Much of the proposed process modifications involved the repurposing of existing fractionators in a manner that seemed very complex, and possibly mechanically improbable.

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The Lab Analysis

At this point, Kathy, the young crude unit engineer, decided to procure unit samples for laboratory analysis. She obtained eight samples. Two sets from each parallel crude unit for:

- Atmospheric gas oil
- Light vacuum gas oil
- Heavy vacuum gas oil
- Vacuum Residue

Each sample was distilled in the lab using an ASTM, D-1160 distillation apparatus to recover its diesel component, as limited by the freeze point. D-1160 being a comparable test to TBP (True Boiling Point) Distillation.

Vacuum tower seal drum condensate was being recycled back to the crude unit desalter. So, Kathy quite rightly, ignored this stream in her analysis of the potential maximum incremental diesel production, as limited by freeze point. She emailed me the percent of the recoverable diesel from each of the six streams listed before. My calculations showed that based on her lab data and plant flows, 800 BSD of recoverable diesel existed; *not* 11,000 BSD!

When I reported this to Ed, the Project Technical Manager, he was not pleased. After all, Ed had spent $500,000 during the past year with the Houston consulting engineering company on computer modeling, case studies, cost estimates, report preparations, and meetings.

“Mr. Lieberman, please devote another day to reconcile the 800 BSD with the 11,000 BSD for incremental diesel production. I had anticipated that the lab analysis program that I suggested to Kathy last month, would support the simulation case studies, and not refute them.”

I reviewed the simulation results and method used by the consultants. Their calculations and assumptions seemed consistent with engineering principles. I finally concluded that their theoretical studies and modeling results did support their projection of an incremental 11,000 BSD of diesel product.

Kathy had based her analysis on 180,000 BSD of crude, rather than the refinery maximum of 210,000 BSD. So I increased her 800 BSD of incremental diesel production to a 1000 BSD.

And then Ed arranged a conference call to discuss my conclusions, which were as follows:

1. Both methods seemed correct.
2. I could not resolve the 10,000 BSD of diesel production discrepancy.
3. Kathy’s lab method represented reality.
4. The computer simulation model represented theory.
5. As the refinery was marketing real diesel with a freeze point specification limit, rather than theoretical diesel, the lab results should be used as a guide to justify any actual or real capital investment.

"But Norm, we can't justify any of the 17 alternate investment cases on a 1000 BSD of diesel production. I've spent $500,000 on developing these crude unit simulations, cost estimates, and case studies."

"Yes Ed. Refinery problems are often like that. They rarely correspond to theoretical computer modeling, unless the models are based on actual plant data, including product composition analysis," I answered.

This is a true story. Anyone who wants to determine the actual identities of the participants will have to phone me. Use your land line. I'm not going to correspond by email, get hacked by WikiLeaks, and wind up on the CNN News Channel.

Post Script

As I noted, this is a true story. Brent, a more experienced plant engineer, redid Kathy's analysis. He concluded that some of the high end-point diesel production could be blended to mitigate the freeze point limitation, which allowed potential maximum production, to increase to 1800 BSD vs the 1000 BSD I had calculated. We are planning to modify steam stripping trays to achieve this goal next year.

Freeze, Smoke, Cloud Point Limitations

Kerosene, Jet Fuel, Heating Oil, and Diesel Fuel product rates are sometimes limited by freeze, smoke, or cloud points, rather than by 95%, or final boiling points. These limitations are highly crude dependent.

Typically, a 200,000 BSD refinery might yield 40,000–50,000 BSD of virgin kerosene and diesel. By greatly improving fractionation efficiency between kerosene, diesel, and the heavier products, incremental distillate yields of perhaps 2000–3000 BSD, at most, might be anticipated.

Typically, efficient side-stream steam stripping of light gas oil components is the most cost-effective method to achieve this goal.

The above observations are based on plant experience, rather than a computer model.

Haze Point Limitation

Jet fuel or kerosene has to meet a haze point specification. The haze point is not related to the cloud point limitation. Haze is a function of the dissolved water content in jet fuel. As the jet cools, water becomes less soluble, precipitates, and the jet fuel takes on a hazy appearance. Haze is controlled by drying. Typically, with beds of rock salt that gradually are dissolved by the haze.

Cloud point is related to heavy waxy components which may precipitate out of solution as the jet fuel is cooled. It is not related to the moisture content of the distillate product. Cloud point is controlled by reducing the heavy, waxy tail on the product by improved fractionation efficiency. It is highly crude dependent.

21

Effect of Combustion Air Preheat on a Fired Heater

Colonel Noraguard was a B-17 bomber pilot during World War II. That was long before I met him at the American Oil Refinery (Amoco) in 1971.

"Lieberman," the Colonel said, "we need to increase crude rate by 10%."

"Sorry sir," I explained, "we're limited by the fuel gas rate to 400,000 SCF/HR. Above that firing rate, the radiant section tubes get too hot. We can run more crude at 400,000 fuel gas, but then the heater outlet temperature drops, and diesel recovery suffers. That would not be on balance, economically, a good idea."

"Look here, Lieberman. I didn't single-handedly defeat the Nazis, to be limited by this little problem. Suppose we preheat the combustion air. For every 360°F increase in the combustion air temperature, fuel gas consumption is reduced by 10% (Fig. 21.1)."

"Colonel, that is true."

"And then we could reduce the fuel gas rate to 360,000 SCF/HR at our current 100,000 BSD of crude charge."

"Quite true sir."

"And then we could go back to the 400,000 SCF/HR limit, and run 10% more crude, without exceeding the firing limit you just noted."

"But Colonel, we would…"

"After all, Lieberman, our stack temperature is 860°F. Preheating the combustion air would drop it to about 540°F. I guess we might lose a lot of draft due to the cooler stack flue gas temperature. Would we need a taller stack or an F.D. blower? What do you think?"

"No sir! We have lots of extra draft. We keep the stack damper 60% closed."

"Do we have enough fuel gas pressure to sustain a 10% higher firing rate? Do we need to drill out the burner tips? Are we limited by excessive burner tip pressure?"

"No sir, the current burner tip pressure is only 9 psi. They're rated for 34 psi. We have lots of burner capacity, and lots of fuel gas. Also, the burner air registers are only 2/3 open. So air, draft, and fuel are not the limiting factors," I explained.

Colonel Noraguard had been the lead bomber pilot for the thousand B-17 bomber raid over Cologne in February 1945 that killed 10,000 people. He came to work every day at the Sugar Creek Refinery with that same sense of determination and unshakeable purpose.

"Sir, could I just explain the problem in detail? It has to do with what I call 'A Radiant Tube Heat Flux Limit.' It's the radiant heat density of the lower row of roof tubes (i.e., the shock tubes – see Fig. 21.1). Could we just look at my sketch for a moment? I could explain…"

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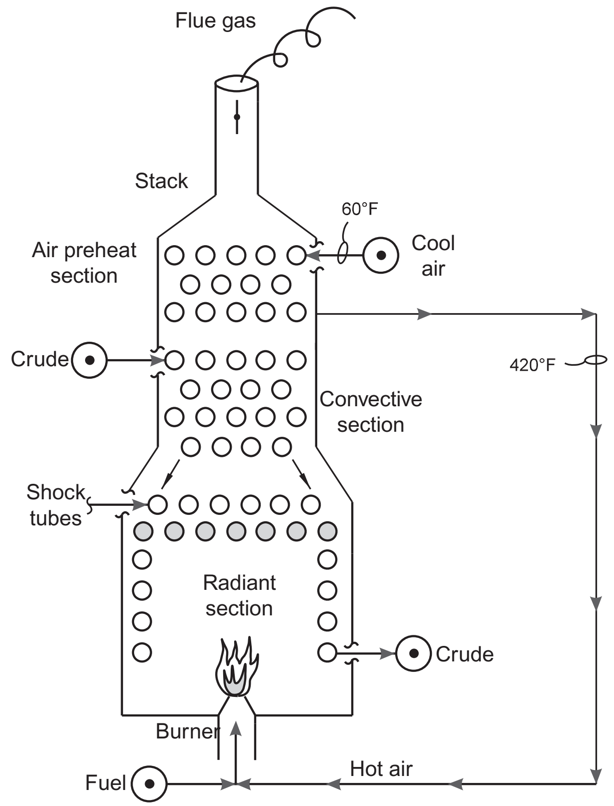

FIG. 21.1 Air preheat increases rate of coke formation inside tubes.

“No! I’m a busy man. I’ve got this entire Amoco refinery to pilot – err, to manage! I will get together with the project group today and initiate the installation of the combustion air preheater ASAP. With a crack-spread of $15 to $20/barrel, that’s $7,000,000 a year we will net. I reckon that air preheat project won’t cost but one or two million dollars. Pretty attractive investment, don’t you think so Lieberman?”

Effect on Flame Temperature

What I had tried, and failed to explain to Colonel Noraguard, was that:

- *Step 1*—Preheating the combustion air by 360°F by heat exchange with stack flue gas, would increase the burner flame temperature by the same 360°F. The flame temperature might then increase from roughly 2200°F to perhaps 2500°F.
- *Step 2*—Radiant heat transfer rate varies with:

$$(T_{\mathrm{R}})^4 - (T_{\mathrm{A}})^4 = \text{Radiant Heat Transfer Rate} \tag{21.1}$$

where:

- ➢ T_R = Radiant source temperature, °R (i.e., Rankine)
- ➢ T_A = Absorption surface temperature, °R (°R = °F + 459)

- *Step 3*—The radiant tubes, including the shock tubes in the roof, would get much hotter.
- *Step 4*—The total heat absorption in the radiant section would increase.
- *Step 5*—The total heat absorption in the convective section would diminish because the flue gas rate had dropped by 10%.
- *Step 6*—The stack flue gas rate would decline because we would be firing 10% less fuel due to the air preheater.

It's certainly true that air preheaters save fuel. But, they also increase the radiant section firebox temperature which increases the heat flux (btu/h/ft^2) of the radiant section tubes.

As these tubes run hotter, they will coke up faster. I have seen a heater run length drop from 2 years to 8 months in the Texaco Refinery, El Dorado, Kansas, due to the addition of a combustion air preheater.

Effect of Improved Air-Fuel Mixing

Improving air-to-fuel mixing efficiency in a burner may permit excess O_2 to be reduced from 6% to 2%. This will have a huge positive impact on energy efficiency. But, it will also increase the burner's flame temperature a lot which will raise the firebox temperature. And also reduce the heat recovery in the convective section which will cause the radiant section tubes to turn from cherry red to reddish-orange. And then accelerate the formation of coke inside these tubes. Indeed, one of the outcomes of optimizing the combustion air rate to a fired heater is a dramatic *decrease* in the heater's run length between tube de-coking intervals (i.e., spalling or pigging).

Excess Air

The reader can conclude from the above that the detrimental effects of a combustion air preheater can be partially offset by increasing the excess air. I had suggested this previously to Colonel Noraguard. But, he had once attended an Energy Conservation Seminar in Houston, where he had learned that excess oxygen should be rigorously controlled between 2% and 4% in the flue gas so as to save energy. I could not convince him that if his objective was to maximize the crude rate, the 2%–4% target should be ignored.

"Colonel," I explained, "it's kind of like in the war. You can be the lead bomber pilot destroying cities, or you can be a humanitarian. You can't have it both ways."

Effect of Air Preheater

Was the crude heater retrofilled with the air preheater?

Yes!

Did the crude unit run 10% more crude charge afterwards?

Yes.

For just over a year. But, then the resulting damage to the shock tubes in the radiant section required that the sagging and distorted 12% chrome radiant section tubes be replaced, at a cost of one million dollars.

However, Dr. Horner, the Amoco Refining Vice President, refused to sign the appropriation for the new heater tubes that I handed to him.

"No," he said to his assistant, the Planning Manager for Amoco, "we can shut Sugar Creek Refinery down, and run the incremental crude at our Texas City and Whiting, Indiana, plants. They're all connected by the Plantation and Capline pipeline systems. Sugar Creek is just an old, outdated plant anyway."

And all the operators, maintenance, and clerical guys and gals at Sugar Creek were terminated. And Colonel Noraguard, Lead Bomber Pilot, was promoted to be the plant manager at the giant Amoco Whiting, Indiana Refinery, on the scenic shores of Lake Michigan.

He retired in 1980. I met him and his wife by chance a week later, at Chicago O'Hare Airport. A really nice elderly couple. On their way to their retirement home in Casper, Wyoming.

Radiant Tube Wall Failure

When the radiant heat flux density in crude oil service exceeds 14,000–16,000 btu/h/ft^2, coke will tend to stick to the interior wall of the tubes. Especially, if the mass flow rate is less than 100–120 lbs/ft^2/s.

The coke inside the tubes prevents the crude from cooling off the tube wall. The tube skin temperature will increase. For heater tube material of 12% chrome, once the tube wall heats to 1350–1400°F, the tube will be subject to "High Temperature Creep" which will result in the tube diameter increasing at the hot spot.

The bulging tube will progressively lose wall thickness, until the wall fails at the hot spot. Such failures, if not addressed properly by the unit operators, often result in explosions, which have caused fatalities.

Overheated tubes are best identified using an optical pyrometer. Cherry red being reasonably cool, and yellow being dangerously hot. The common use of a tube-skin TI is less reliable and at best only monitors a few locations on the tubes.

22

Fired Heater—Safe Shutdown Due to a Tube Failure

Kathrine was laughing. The tiny waiting room echoed with her hysterical laughter. Gary Adams lay sedated and unconscious behind the glass partition. Third degree burns covered his face, arms, and chest where his polyester uniform shirt had melted onto his skin.

Gary and Katie were my friends and coworkers at the Good Hope Refinery in Southern Louisiana. Gary was my assistant on the FCU-Visbreaker complex. His wife, Katherine, was at the edge of hysteria, and her laughter was unnerving. “Thank God,” I thought, “Gary was wearing cotton jeans, and not polyester pants.” This being in the days before “Nomex” coveralls at our refinery (i.e., 1982).

Heater Tube Failed

“Lieberman, get your skinny ass down to the Visbreaker. We just blew-out a tube on the Visbreaker Vacuum Heater,” Gary screamed over the radio, the previous afternoon.

“What’s happened Gary?”

“We got black smoke and fire blowing out the damn heater stack. We got smoke blowing out the sight ports. I got the guys putting fire water on the box. I tell you Norm, I can urinate with more pressure than we got in that fire water.”

“Sounds bad Gary!”

“Yeah! I told plant management before not to use fire water for process cooling. In an emergency, we don’t got enough pressure!”

“Okay! I’ll be right over. Don’t forget the pass steam. Don’t want to coke-up the tubes.”

Upper Explosive Limit

I stood on the Mississippi River Levee, 200 ft back from the heater. Black smoke blew out of every sight port and oily, black flames raced from the stack. Tommy Jones was blocking in the fuel gas, and Boon Babin was lining up the pass steam to the coils to blowout the residue before it coked up.

Gary himself was closing the 4″ heater feed valves on all four passes and making sure the control valve bypasses were shut tight.

It looked bad. But in reality, the heater was in a safe position. At least for the moment. That’s because the heater firebox was in a “Fuel Rich Situation” above its “Upper Explosive Limit.” It was too fuel rich to explode.

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I knew this for certain. I could see the unburnt hydrocarbon vapors from the failed tube igniting in the flue gas that was racing out of the heater's stack. Of course, the concentration of hydrocarbons in the firebox would diminish with time as Gary blocked in the feed and Tommy closed off the fuel. And then, as the fuel-to-air ratio diminished, the heater box would drop into the explosive region. And then—the box would explode. I had seen this happen before at American Oil 15 years back, in Texas City.

Snuffing Steam

Gary should have introduced Box, or Snuffing Steam, into the firebox at this time. This would have retarded the air flow into the firebox and allowed the box to cool off in an air-deficient environment. That is, the steam would have slightly raised the box pressure above atmospheric pressure and temporarily precluded the air flow into the box due to draft.

But I knew for certain that Gary would not do this. Because, a few months back, I had the four, 6 in. snuffing steam lines blinded off from the visbreaker heater.

But why?

Because, the snuffing steam lines were full of steam condensate (water). Thus, their use would have blown water into a 1600°F brick box, and created a huge pressure surge, which would then have blown burning visbreaker tar all over the plant (see Fig. 22.1).

I had suggested to Gary Adams that in the event of a tube failure, he instruct the operators to close off the air registers underneath the heater and shut the stack damper. This would prevent an explosive mixture of air and fuel from developing inside the firebox.

"Okay Mr. Lieberman. In the event of a busted tube, I'll call you to close off the damper and secondary air registers. I'm not gonna ask any of my guys to risk their lives. Us Cajuns are too smart for that. That's a good job for a New York Yankee like you."

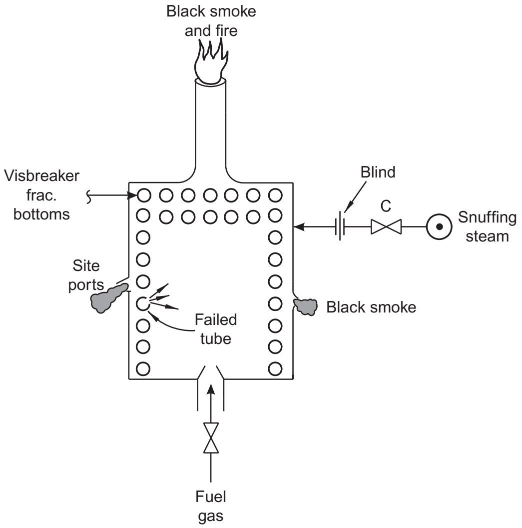

FIG. 22.1 Stuffing steam could not be used when tube failed. It had been blinded due to water in steam.

Thus the call from Gary!

As Gary Adams completed closing off the visbreaker feed valves, a slight explosion shook the unit. Gary and 20 operators and maintenance guys were scattered across the deck with their polyester shirts on fire. The heater itself, and the rest of the unit, had not sustained very much damage.

Only later did I learn about fire-retardant Nomex coveralls. Afterwards, six pairs were provided to all operators and maintenance personnel. Don't wear polyester clothing underneath your Nomex Coveralls. "Better safe than sorry."

Abandoning Box Steam Installation

The above story is true. It happened in 1982, exactly as I've explained. What is equally true is that the world's largest operator of refineries and chemical plants has also disconnected snuffing steam from process fired heaters for the same reason I've described before.

This first came to my attention about 10 years ago, in the course of instructing my "Process Equipment Troubleshooting Seminar" for this operator at their largest Gulf Coast Refinery. The reason for this corporate wide decision was the same as I made in 1980 at the Good Hope Refinery. That is, water contamination of the Snuffing Steam supply lines. The result has been periodic and repeated explosions in their fired heaters, due to tube failures, while shutting down the failed heater.

Corrective Action

Recently, another major operator of refineries in North America shared with me their response to this serious, chronic problem. I have shown their common sense design for a Snuffing Steam Installation in Fig. 22.2. Key features are as follows:

1. Snuffing Steam valve located at grade.
2. Snuff Steam valve located 100–200 ft back from the firebox.
3. Lines steam traced and insulated.
4. Carbon Rupture Disks rated to rupture at 5 psig at the firebox snuffing steam inlet nozzles.
5. A ½ in. drain hole at all low points.
6. "Car Seal" open valves at the firebox inlet nozzles.
7. Upstream of the main snuffing steam supply valve, a condensate drain line equipped with a steam trap.
8. Downstream of the main snuffing steam supply valve, a drain valve, car sealed open.

Note that activation of this system only requires an operator to open one valve (Valve "A"), at grade. Leakage through this valve will be drained by the ½ in. weep holes (Fig. 22.2).

Typically, the snuffing steam valves I have seen are 6″ or 8″ carbon steel valves 20 ft up in the pipe rack, with chain operators. When I have tried to pull open such valves in an emergency, the valve was stuck due to it not being moved or greased in 20 years. All I accomplished was having the chain break off and hit me in the head.

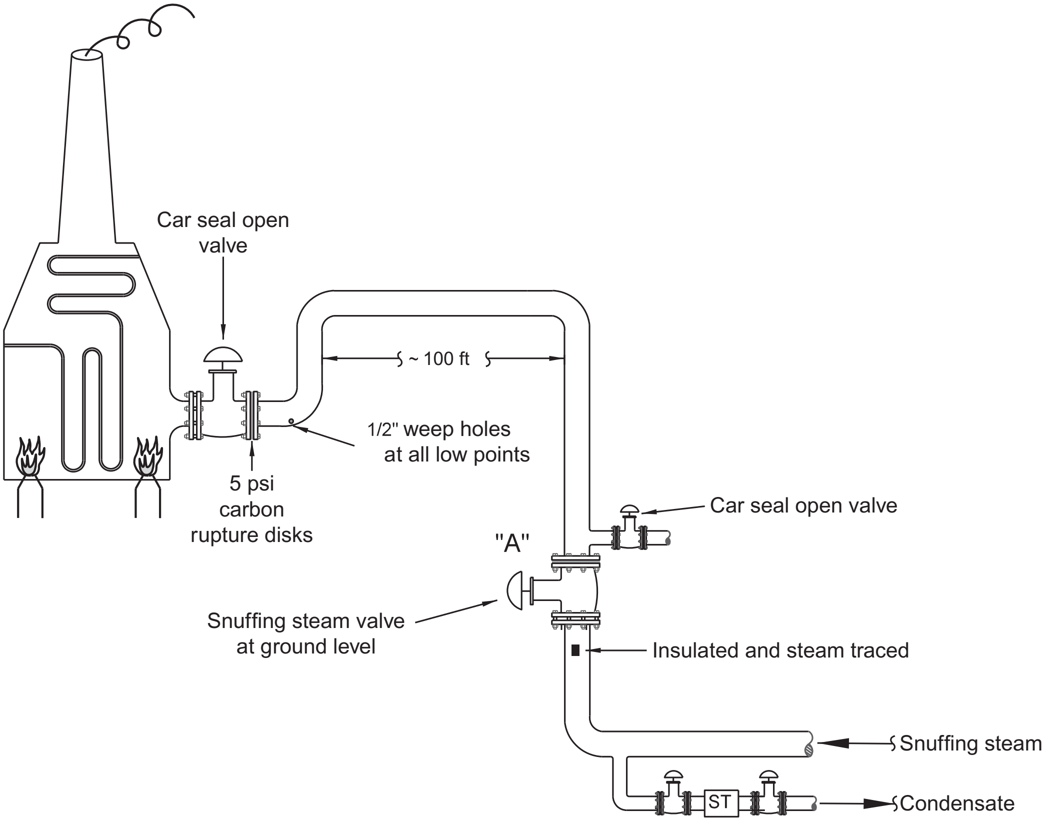

FIG. 22.2 Heater snuffing steam correct design.

A final note. Operators who are not trained will not know what to do in an emergency. Operator training consists of me walking out to valve "A" (Fig. 22.2) with Bobby Brickhead and saying, "Look! When you got a fire in the heater, turn this valve counter-clockwise. Got it, Bobby?"

"Oh! No, Mr. Norm. Which way does a clock turn again? I forget!"

Going Home

Gary Adams went home 2 weeks later. A few days more and he was back at work. His beard had been burnt off and his hands were covered with red scars. A month later, there was a giant sewer fire at the Diesel Hydrotreater. Gary, a decorated Vietnam War Veteran Infantryman, was in the forefront of the firefighting crew. Quite fearless.

I, who had served my country fearlessly during the conflict with a Student Exemption in New York, observed the conflagration from a safe distance on River Road, "Discretion being the better part of Valor."

23

Vacuum Towers

On a lube oil vacuum tower, the tower top pressure was swinging from 25 to 50 mm Hg. The pressure would run steadily at 25 mm Hg for 15 min and then surge upwards for a few minutes, before falling back down. This erratic pressure is bad for any vacuum tower, but especially serious for lube oil base stock production.

The system consisted of the following components connected in series (see Fig. 23.1):

- Precondenser
- Primary jet
- Primary condenser
- Secondary jet
- Secondary condenser
- Third Stage jet
- After-condenser

Each jet was spared, but only one jet was working in each position.
Using a portable digital vacuum gauge, I conducted a pressure survey and obtained the following data:

- Precondenser inlet = 28 mm Hg
- Primary jet inlet = 16 mm Hg
- Primary jet discharge = 58 mm Hg
- Secondary jet discharge = 76 mm Hg
- After-condenser inlet = 0.6 psig (i.e., about 780 mm Hg)

From this data, I drew the following conclusions:

1. The compression ratio of the primary jet is about 3½ (i.e., 58 ÷ 16 mm Hg).
2. The compression ratio of the secondary jet is about 1.3. (Neglecting the delta P across the primary condenser.)
3. The compression ratio of the final jet is about ten. (Neglecting the pressure drop across the secondary condenser.)

I now made the following additional observations:

1. A small increase in the pressure at the after-condenser inlet, caused the third stage jet to begin to surge (i.e., it started to make a soft, moaning or hunting sound). The inlet pressure to this jet increased by perhaps 25 mm of Hg (i.e., by 1 in. of Hg vacuum).
2. The precondenser inlet pressure rose rapidly (i.e., the tower top pressure) by about the same 25 mm Hg.

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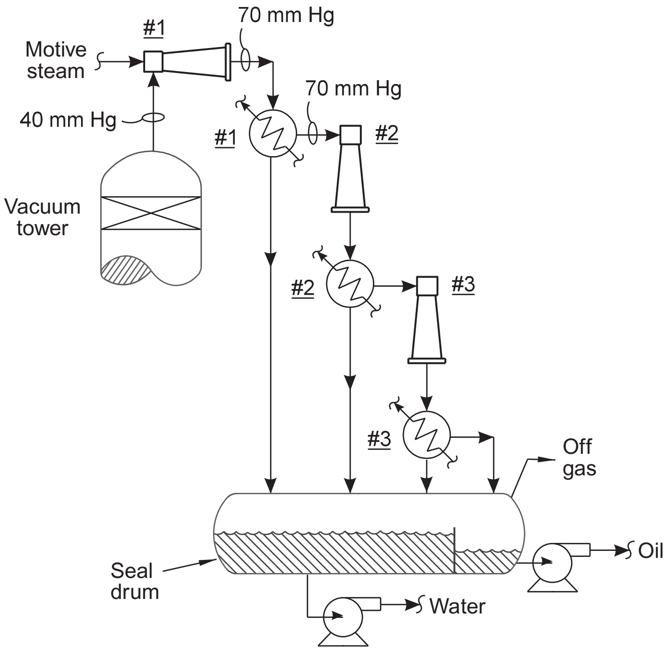

FIG. 23.1 Increased cooling water to #1 condenser temperature causes vacuum to break by exceeding the critical discharge pressure of the first stage ejector.

3. After a few minutes, the pressure at the after-condenser slipped back down and the normal operating vacuum was restored.

I then noted that the refinery flare gas recovery system pressure oscillated by 0.5–1.0 psi, and that this oscillation could not be mitigated. As the ejector seal drum was floating on the flare pressure, and the final condenser vented to the seal drum, changes in pressure at the discharge of the third stage ejector could not be avoided.

How Jets Work

The ejector motive steam was about 350°F. The motive steam as it passes into the ejector cools to 100°F. The lost 250°F has been converted into kinetic energy (see Fig. 23.2). This is an “Isentropic Expansion.” It’s the speed or kinetic energy of the motive steam that compresses the vacuum tower cracked, noncondensable gases. As the steam slows down in the body of the ejector (i.e., the diffusor), its kinetic energy is converted to pressure. However, a lot of the kinetic energy is also converted back into heat or temperature. It’s the heat of compression.

Using my infrared gun, I noted that the temperature increase along the body of each ejector stage was:

- Jet #1 = about 80°F
- Jet #2 = about 10°F
- Jet #3 = about 130°F

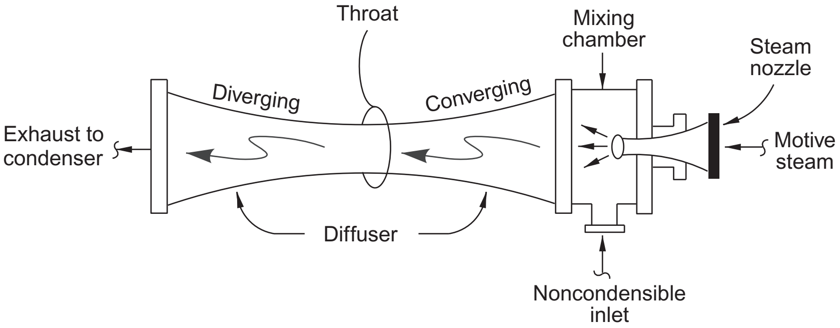

FIG. 23.2 Components of a converging-diverging steam jet.

Clearly jet #2, the second stage ejector, was producing very little compression work. This was consistent with my previous pressure measurements.

Nonlinear Performance of Ejectors

A steam jet has two modes of operation. One mode I call the sonic boost. The second mode of operation I call the velocity boost. This indicates the stage of compression that occurs in the diverging portion of the ejector's diffusor, as the vapors slow. The ejector vendors call the sonic boost mode of operation, "critical flow." They call the second mode of operation, when the jet is only doing half its required work, the "throttling" mode.

I've explained this in detail in my book, "Troubleshooting Vacuum Systems," Wiley Books, 2013. As the discharge pressure from an ejector rises past a point, it will suddenly lose a large portion of its compression ratio. When this happens, we say the vacuum or jet has "broken." Hence, a relatively small increase in an ejector discharge pressure may sometimes lead to a disproportionate increase in the ejector suction pressure. Apparently, the small back-pressure increases from the refinery flare system were backing-up and creating a disproportionate increase in the vacuum tower pressure by causing the secondary ejector to break or lose its sonic boost (i.e., it was forced out of "critical flow").

Fixing the Problem

To correct the problem, I lined up the spare secondary ejector to run in parallel with the operating secondary ejector. This did no good. So, I next blocked in the motive steam to the secondary ejector that had been running for years. This also had no positive effect. Only when I remembered to close the suction valve to the ejector that had been previously in operation, did the discharge pressure from the second stage condenser increase from 76 mm of Hg to about 140 mm of Hg. Also, the temperature increase along the body of the second stage jet gradually increased from about 10°F to roughly 90°F. One lesson to learn from this exercise is that you cannot run a weak jet in parallel with a stronger jet. Noncondensables from the discharge of the stronger ejector will recycle back through the weaker jet.

The result of this change was that small increases in the pressure in the seal drum, no longer effected the discharge pressure from the primary ejector, and thus did not cause the primary ejector to periodically lose its sonic boost. Thus this ejector's performance no longer caused the vacuum tower pressure to periodically double. However, much larger increases in back-pressure still had the same effect.

Limitation of Water Vapor Pressure

The improved performance of the secondary jet effectively stabilized the vacuum tower pressure, but did not improve the vacuum. The minimum vacuum tower pressure was a function of the condensing pressure in the precondenser. The condensing pressure in the precondenser was limited by the vapor pressure of the water, at the precondenser vapor (not liquid) outlet temperature. This vapor temperature was a function of the cooling water outlet temperature from the precondenser. While the precondenser pressure could certainly go well above the vapor pressure of water at the outlet temperature, it could not fall below this value. Many vacuum towers are limited, not by the ejector system, but by the performance of the precondenser. This precondenser exchanger is typically a severe fouling service in many, if not most, refinery vacuum towers.

Steam Turbine Vacuum Surface Condensers

The vacuum in the turbine exhaust surface condenser is provided by two small ejectors, #1 and #2 (Fig. 23.3). Note that there are two small jet discharge condensers: A & B. They are both inside a single shell, divided into two sections by a horizontal baffle (rather than the vertical baffle I've shown for clarity). A delta P of 20 mm Hg for each condenser A & B is assumed. Each jet is developing a compression ratio of about 4.3. This design has been universal for 150 years.

Improving vacuum in a surface condenser a modest amount may increase the work extracted from each pound of turbine motive steam by 10%. A small amount of air is always going to leak into the surface condenser and the turbine itself. This air is drawn off by the two-stage ejectors shown in Fig. 23.3.

One check to make on a surface condenser is to see how hard air is blowing out of the vent. However, a good deal of the time, I find no air is blowing out of the vent. Rather, air is being sucked into the vent! Meaning, if I put a piece of cardboard over the top of the vent, the cardboard sticks to the vent, held in place by the vacuum in the stack. The cause of this reverse air flow is that the shell side partition baffle shown in Fig. 23.3 is leaking. This allows the air from the vent stack to recirculate back into the suction of the #2 jet. Periodically, the pressure builds up on the B side of the condenser, and air is blown out intermittently from the vent. This problem results in:

1. Overloads the #2 jet.
2. Increases the discharge pressure of the #1 jet.
3. Increases the pressure in the surface condenser.
4. Reduces the amount of work that may be extracted from each pound of the turbine motive steam.

Function of Condenser "B"

To correct this problem, one should first consider Fig. 23.3 more carefully. What is the function of condenser "B"? The answer is that it has no function. It's an engineering error. Venting the discharge of the #2 ejector directly to the vent stack will reduce its discharge pressure from 780 to 760 mm of Hg. This will result in a small reduction in the surface condenser pressure. However, if the baffle between sections A & B in the ejector system condenser is leaking, venting the second stage condenser directly to the atmospheric vent stack, results in a large reduction in the surface condenser pressure. Either way, the result is increased turbine efficiency.

The observation that condenser "B" has been standard in steam turbine surface condenser design for 150 years, but represents an engineering error, has lately been recognized by several of my major clients, who have eliminated it in their plants in Louisiana.

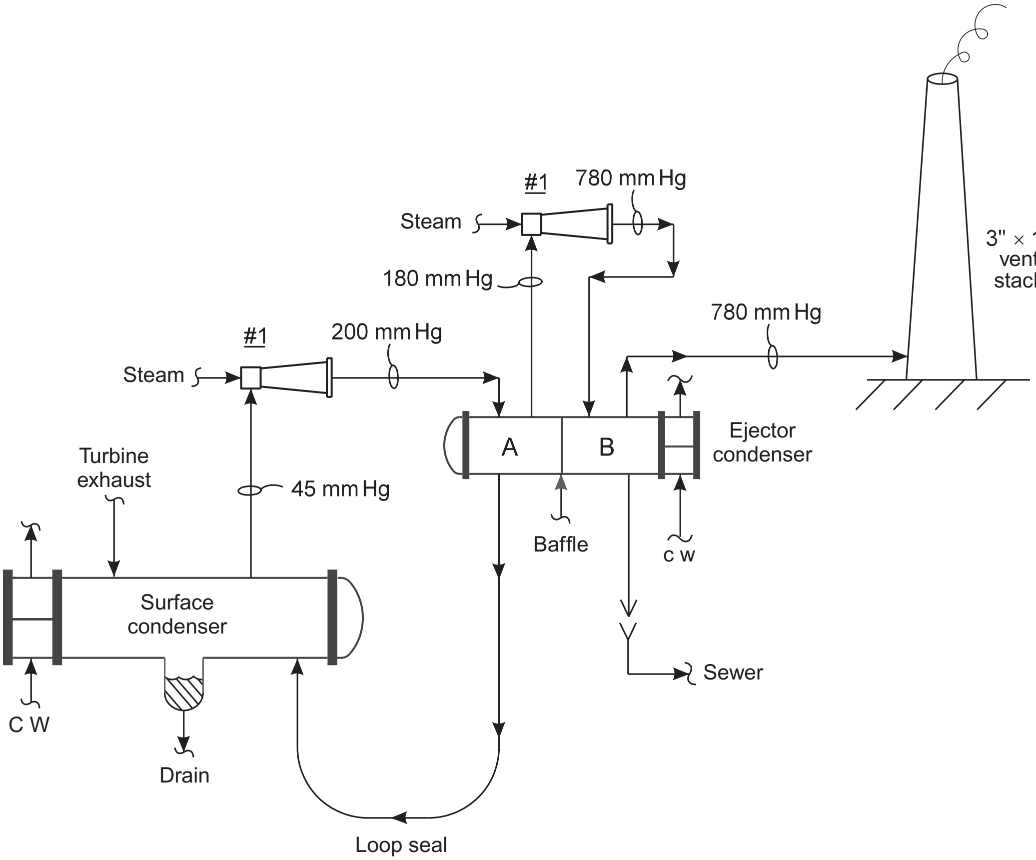

FIG. 23.3 The baffle in the injector condenser is actually horizontal rather than vertical. This is a typical steam turbine exhaust surface condenser system.

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Upgrading Vacuum Tower Performance

This chapter summarizes a case history of an improved vacuum tower performance at the Good Hope Refinery, where I was the Tech Service Manager from 1980 to 1983. Our objective was to increase gas oil recovery from vacuum tower residual product.

Reducing Resid Gas Oil Content

Each barrel of gas oil left in vacuum tower bottoms is a major loss. However, many plants currently do not sample the vacuum tower bottoms stream for fear of a fire at the sample point. The auto-ignition temperature of the vacuum resid is 320°F. The sample point on the run-down stream is at 450°F. The plant Safety Department may prohibit the use of such a sample station.

However, we can get a sample without any hazard even though at temperatures below the 320°F auto-ignition temperature, the resid will solidify.

The sample can be obtained at the bottom's pump discharge temperature of 680°F using a steel bottle (see Fig. 24.1). Allow the bottle to fill with resid. Block it in and wait for the bottle to cool. Then, unscrew it and send to the lab.

What would seem to be the simplest way to reduce gas oil lost to vacuum resid would be to cut back on the wash oil flow rate (see Fig. 24.2) above the grid from say 5000 reduced down to 3000 BSD. That will not yield 2000 BSD of incremental heavy gas oil product. First, about 80% of the wash oil evaporates in the grid. Thus, reduced wash oil lost to vacuum tower resid would be only 400 BSD, not 2000 BSD. Also, this will promote coke formation, due to lack of wetting of the grid. Over a period of months, the quality of the gas oil product will deteriorate, and the flash zone pressure will increase.

The metals in the gas oil will increase and these higher amounts of metals will ruin the catalyst in the FCU feed gas oil hydrotreater.

Increasing Flash Zone Temperature

We could raise the heater outlet from 760°F to the design of 780°F, if the heater is not overfiring and the FD air blower is running below rated speed, and the burner flames are not near the tubes. But, the higher flash zone temperature will increase cracked gas flow, which can overload the vacuum system. The tower pressure would increase as a result and gas oil recovery will drop.

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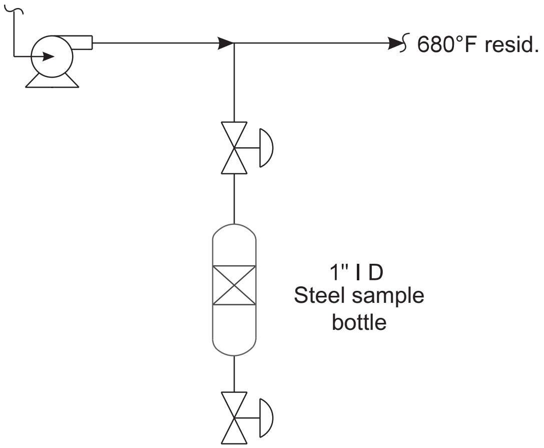

FIG. 24.1 Sample bottle.

Another option is to increase the steam through the vacuum heater passes concurrently with the higher outlet temperature. The reduced oil soaking time in the tube due to the steam would offset the hotter outlet. Formation of cracked gas would then decrease. Also, the coil steam would reduce the hydrocarbon partial pressure in the flash zone.

But that places a larger load on the first stage ejector (see Fig. 24.3). However, the extra steam in the heater passes will suppress the average coil temperature by promoting more vaporization in the heater tubes. The lower temperature will also reduce the rate of cracked gas formed in the tubes.

The extra steam may cause an overload of the first stage ejector capacity and raise the vacuum tower flash zone pressure. Only a plant test could prove what is the best course of action.

Commissioning Spare Steam Ejector

Another alternate is to commission the spare first stage ejector. But, when tried (see Fig. 24.4), the vacuum breaks and the tower top increases by 20 mm Hg. Why? Testing the jet by itself showed that it ran fine. The steam nozzle was replaced anyway, but that made no difference to the performance.

We tried again to start-up the idle jet. However, the new first stage jet was on an elevated 80 ft platform by itself. It was connected to the tower overhead vapor line by 120 ft of 16″ of bare piping. As soon as the outside operator cracked open the inlet gate valve (see Fig. 24.4), the local vacuum pressure gauge jumped from 36 to 48 mm of Hg and continued to trend higher.

The operator observed that it happened every time. However, we noted that the suction line could be full of water. When the operator opened valve "B," water would be drawn into both the old and the new ejectors, which bogs down both jets. One would need to drain down the 16″ line going to the new jet before opening valve "B." However, the 2″ valve (i.e., "C") just upstream of valve "B" cannot be simply opened, because the

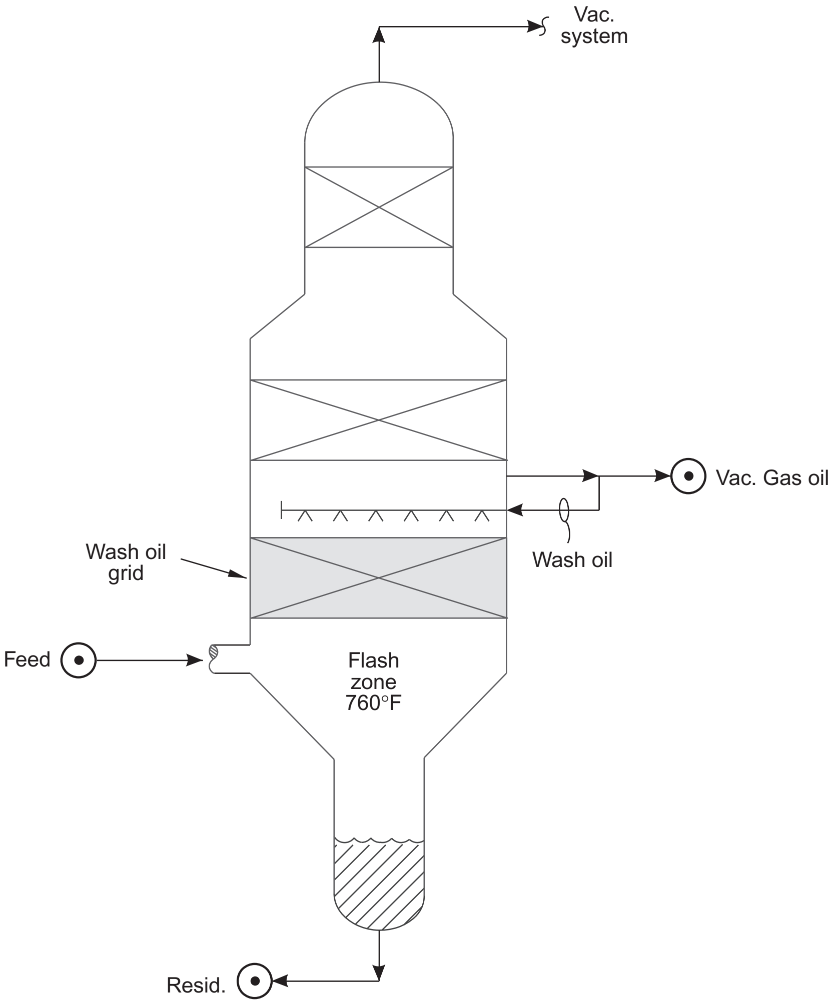

FIG. 24.2 Wash oil protects grid from coking.

100 foot long, 16″ line was under a vacuum. If opened, it would draw air into the jets and the vacuum would break.

What we did was to run a hose from the 2″ drain down to a bucket of water 50 ft below the "C" drain valve (see Fig. 24.4). For this to work, the hose has to be filled (i.e., primed) with water first. Then the 16″ line can be drained before opening valve "B."

The 2″ drain should have been piped into the seal drum. It was a design error.

Silicas in Steam Nozzle

The vacuum had suddenly gotten worse a few months back. A common cause is silicate deposits in the steam nozzle (see Fig. 24.5). The steam quality may be bad, contaminated

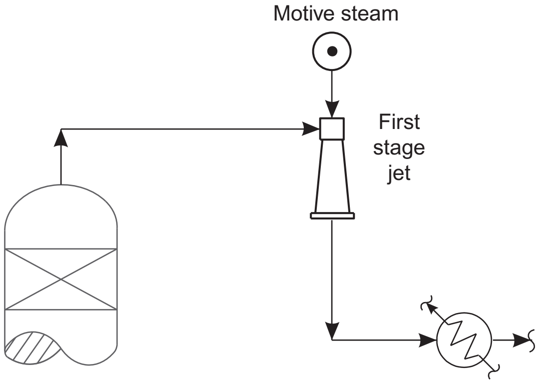

FIG. 24.3 Extra steam to vacuum heater overloads first stage jet.

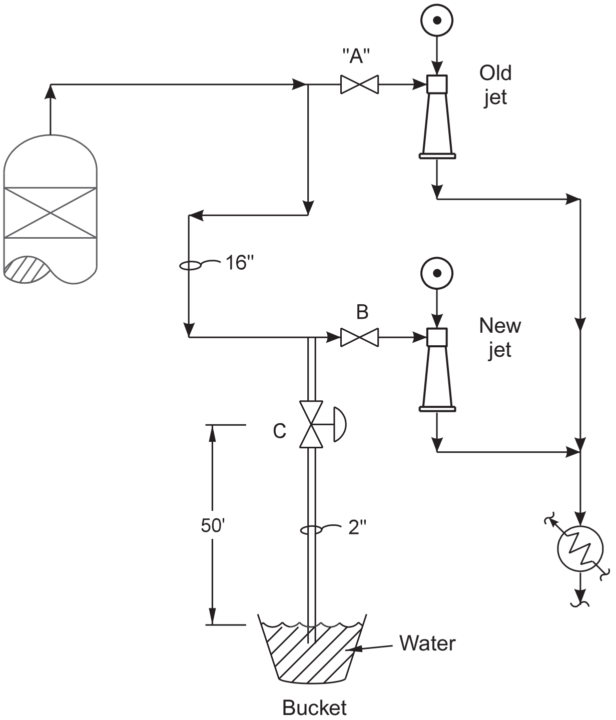

FIG. 24.4 New jet required drain line for start-up purposes.

with carry-over from a kettle waste heat boiler. The entrained moisture could be 10,000 ppm of TDS (Total Dissolved Solids) mainly silicates, which precipitate out in the steam nozzle. On the back of the jet, there is a plug that can be used to clear the steam nozzle with a wire brush. This cannot be done while the jet is running. But once the new

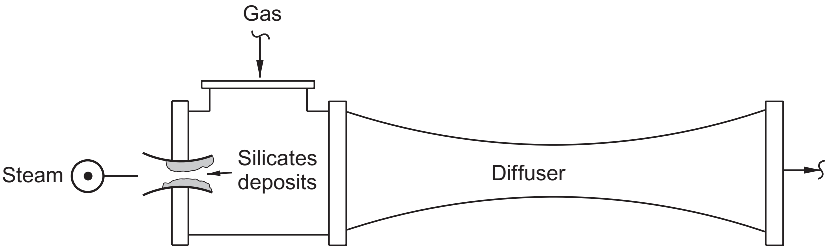

FIG. 24.5 Silicates fouling is the most common cause of poor ejector performance in refineries.

ejector is on-line, we can clean the old jet. This plug is intended to check the ID of the steam nozzle. But it can also be used as a clean-out access to the steam nozzle. Note that larger ejectors may not be supplied with this connection. In this case, the motive steam chest would need to be removed to check and clean the nozzle.

Wet Steam

Water in steam degrades jet performance and damages the ejector. A "Steam Separator" will prevent the problem and can be ordered from:

- Anderson Company
- Clark Reliance

The condensate drains through a bucket trap. They work rather like a centrifugal separator.

We can observe the effect of wet motive steam to an ejector. The temperature of the ejector discharge line drops by 15–30°F within a few feet of the discharge of the jet. If the steam blowing out of a bleeder is invisible for an inch or more, the steam is dry. If the steam blows out white, it may still be reasonably dry (i.e., a few percent moisture). If water accumulates at a low point drain every few minutes, the steam is likely wet enough to adversely affect vacuum. We once wrapped 30 ft of a 2″ steam supply line with rags and insulation, and gained several mm of Hg.

Cooling Water Flow

We observed that the cooling water flow to the first stage vacuum condenser was gradually falling and we chemically cleaned the tubes to remove scale, but it failed to help.

The problem was not fouling, but air which is evolved out of the water at the condenser outlet. The evolved air chokes off the flow of water through the tubes. As the water rises 80 ft to the elevated surface condensers, it loses head pressure and liberates air. The dissolved air content of the cooling water ought to be constant, but why was the loss of cooling water was getting worse?

Investigation showed that the suction piping of the cooling water pumps was 7 ft above the water level in the sump. That means that the suction piping to the water pump is under a vacuum of 3 psi or 12 psia. There are 2.3 ft of water in a psi. If small holes develop in the carbon steel suction piping due to corrosion, the air is sucked into the pump's suction. At the 60 psig pump discharge pressure, the air dissolves in the water. In that way, the dissolved air content of the cooling water can increase by a factor of four. Further, the problem could get worse with time.

One solution is to obtain a roll of duct tape and wrap the suction line. This also will help the pump discharge pressure. Once, we used duct tape to fix an air leak on the steam ejector diffusor. (Note: While duct tape does a good job of sealing up a vacuum leak, using it for more than a temporary patch is not advised. If the metal is thin enough to develop a hole, it could be uniformly thinning, and it could collapse on itself.)

Tube Material Selection

We had to retube one of the first intercondensers due to wear, and as a result, had issues with that condenser. Condensers often need to be retubed at least once during their operational life. Normally they work better after new, clean tubes, have been installed. In this case, the condenser worked fine for the first few months. But, in fact it should have worked even better due to the material change. The stainless tubes were replaced with carbon steel tubes. The carbon steel tubes appeared to be a better option. Not only were they readily available and cheaper, but the carbon steel also has a better heat transfer rate when compared to the stainless tubes. The better heat transfer rates should give the condenser more capacity.

But, carbon steel tubes are a bad choice. The carbon steel tubes in this service are exposed to cooling water that is rich in oxygen on the one side, and oxygen containing vapors on the other side. Both of these cause the carbon steel to oxidize and rust. While the tubes do work well at first, it does not take long for them to rust over, which impedes heat transfer. That rust buildup can actually choke off the cooling water flow over time, which further hurts performance. Carbon steel tubes generally don't last more than a year or two before developing major cooling water leaks.

This also explains the increase in hotwell condensate rates and the reduction in condenser performance.

Not only do the carbon steel tubes hurt performance, but because they fail so quickly, it means they need to be replaced much more often. This requires additional work and time, so the carbon steel ends up not being as cheap as first thought. Stainless steel tubes, brass tubes, or duplex stainless tubes are preferred in vacuum condensers.

Drain Leg Leak

To identify air leaks, look at the off-gas composition for unusually high levels of oxygen and nitrogen.

An air leak will cause the third stage ejector pressure to be poorer than design. In this case, the leak was in the second condenser's condensate drain leg, about 10 ft above the condensate drum.

But how would air ever leak into the condensate drain leg, as those drain legs are filled with liquid? However, they are also at a subatmospheric pressure down to the liquid level in the condensate receiver. If the line was to develop a hole, it would pull air into the system, not leak out as one would experience with a line that was under a positive pressure. But if the air was being pulled into those lines, why did it not flow down into the hotwell with the condensate?

The reason is that the flow through the seal legs is low compared to the line size, which are designed to vent back up to the condensers. Any air that gets pulled into a leak would rise toward the ejectors, which operates at a low pressure. If the leak is large enough, the air being pulled in can actually prevent the condensate from draining and cause the condenser to flood.

Summary

In conclusion, we found that the 1050°F minus content of our vacuum residue had dropped from 18% to 10%. This had reduced our vacuum resid production rate by about 2000 BSD, with a concurrent increase in vacuum gas oil yield of 2000 BSD. The delta value was worth $40,000/day.

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Vacuum Ejectors as Wellhead Compressors

South of Laredo, Texas, in the vast empty space of Zapata County, is the land that I love. Superficially, it's not an inviting landscape. Deep gullies, desiccated and eroded hillsides, withered grass, and desperate trees.

But deep under the clay and gravel surface, there sleeps a vast treasure. The Wilcox Formation. Natural Gas. Almost pure methane. Free of hydrogen sulfide and carbon dioxide. At a depth of 9500 ft, the gas has been resting and waiting for me at a pressure of 2800 psig for a 100 million years.

This is not entirely true. The high pressure gas has already been recovered by the Production Division of GHR Energy. My division, consisting of me and Juan Hernando, is in charge of Secondary Recovery in the Zapata Gas Field. After 1 or 2 years, when over 90% of the natural gas has been produced, and wellhead pressure has dropped from 2800 to 200 psig, I am put in charge of the largely depleted wells. I am the caretaker for hundreds of these wells that produce only 5% of the gas flowing to the Laredo Compression Station.

One hot afternoon, Juan and I drove up to a depleted well, with a wellhead pressure of 200 psig. As the pipeline gathering system pressure was 600 psig, Juan suggested that we install a rented Ingersoll-Rand, two-stage reciprocating, natural gas driven, wellhead compressor. As on 100 other wells, this would compress low pressure wellhead gas for delivery to our Laredo Compression Station, about 40 miles distant.

As we discussed the rental and operating charges for the wellhead compressor, to decide if the value of the recovered gas was worth these costs, I noticed that near our depleted well, there was a new well, flowing with a larger volume of gas, with a wellhead pressure of 2500 psig.

"Juan, I got an idea. Let's not rent a compressor and use a lot of fuel gas to run it. Let's use the gas pressure from the new well, to compress the gas from our old well."

"Señor Norm, how does that work?"

"We'll use a steam ejector. But, instead of motive steam, we'll use 2500 psig natural gas from the new well."

"No comprehendo, Señor Norm. Que está 'Ejector'? How does that work?"

"Okay Juan. You listen and I'll talk. It's a long story."

"Si! Si! Norm. But we have much time."

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Vacuum System—Steam Ejectors

As a child, I had toilet training. One of the things that I noticed during my training was that when it was windy outside, the water level in the toilet bowl became unstable. As there was no wind inside our bathroom, I couldn't see the relationship between the toilet water level instability and a windy day?

Years passed. One day, as I was drinking soda through a clear plastic straw, I blew across the top of the straw. I saw that the soda was drawn up into the straw, above its level in the bottle. I wondered why?

More years passed. I graduated from University, married, and purchased a house. I began my career in bathroom maintenance, specializing in toilet malfunctions. A neighbor suggested that my current problem could be that the toilet's roof vent might be plugged (see Fig. 25.1).

I had not been aware that toilets had roof vents to prevent vapor lock in the drain line to the sewer. Investigation showed a bird had built a nest on my toilet's roof vent. As I removed the nest, I grasped my childhood observations on the connection between the wind and the wavering toilet bowl water level.

As the wind blows across the top of the roof vent, an area of low pressure is created in the vent pipe and in the toilet drain line connected to the sewer. The level of water is drawn up in the drain line. That water comes from the water level in the toilet bowl (like my soda

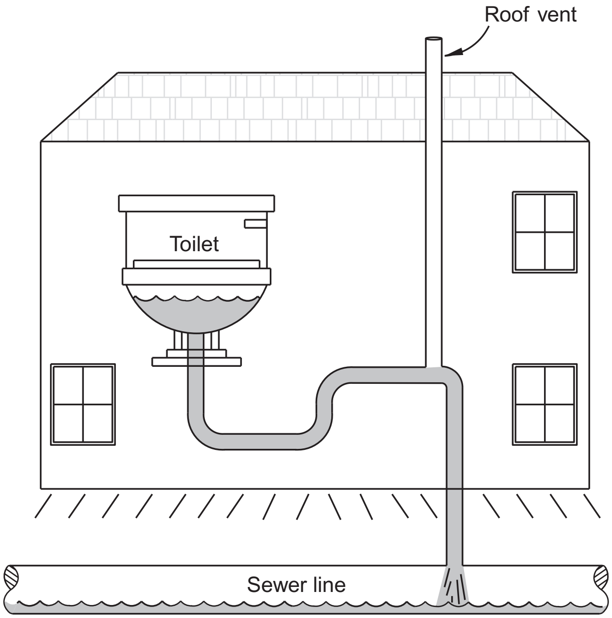

FIG. 25.1 Wind blowing across vent causes water level in toilet to rise and fall.

straw experiment, see Fig. 25.1). When the wind gust passes, the pressure in the vent line returns to atmospheric pressure. The water level in the toilet bowl then returns to normal.

Wind velocities of 20 or 30 MPH (or 30–40 ft per second) will lift a water level a fraction of an inch. But suppose the air velocity was 12,000 ft per second (8000 MPH or 10 times sonic velocity). Then, the water level could be lifted by 20 ft. When a fluid (air, steam, or natural gas) flows with a lot of velocity, past an opening, the pressure in that opening is going to go down. That's why, as the wind blows across the top of a chimney fireplace, draft increases.

Converting Heat to Velocity

A car tire has a pressure of 35 psig. If you depress its needle valve, the air will feel cool. The air is blowing out at a rate of around 1000 ft per second (sonic velocity at atmospheric pressure). The majority of the energy to accelerate the tire's air comes not from the tire pressure but from the temperature of the expanding air. This is called an "Isentropic Expansion." Basically, this means that a portion of the heat content of a vapor, in the form of sensible heat, is converted to velocity.

Ordinarily, if a gas like air blows through a simple hole, with a pressure reduction from 35 psig to atmospheric pressure, it will exit from the hole at sonic velocity. That's called critical flow. However, if the hole is shaped like the nozzle shown in Fig. 25.2, then the exit velocity of the gas from the nozzle can be greater than sonic velocity. The reduction in pressure at the outlet of this nozzle will proportionally be much greater. Not 1 in. of water pressure drop, but 10 or 20 ft of water pressure drop. That is, the nozzle is creating an area of reduced pressure or a partial vacuum.

The pressure of the atmosphere, at sea level, at my home in New Orleans, is equal to 34 ft of water. If the high velocity nozzle I've described was pulling the water up in my soda straw by 17 ft, I would then say the vacuum developed by the nozzle was equivalent to 50% of atmospheric pressure:

$$\tfrac{1}{2} \cdot 14.7 = 7.3\,\text{psia}$$

Or, if I was measuring the vacuum in "Inches of Mercury," where atmospheric pressure is 0 in. of Mercury at sea level, then my vacuum pressure gauge would read 15″ Hg of vacuum.

Or, if I was measuring the vacuum in mm of Hg, where atmospheric pressure at sea level is 760 mm Hg, then my vacuum pressure gauge would read 380 mm Hg of pressure.

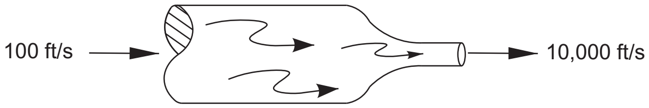

FIG. 25.2 Acceleration of vapor thru a nozzle. An isenthropic expansion.

In the United States, we use the inches of mercury system. The rest of the world uses mm of Hg to measure vacuum where:

- Full vacuum = 0.0 mm of Hg
- Atmospheric pressure at sea level = 760 mm of Hg

Steam Ejector as a Compressor

A steam ejector is used in process plants and power stations to produce vacuum. It is among the most widely used items of process equipment.

Vacuum ejectors are used to produce:

- Road asphalt
- Lube oils
- Waxes
- Electricity
- Orange juice
- Desalinated water

Much of my time as a process engineer is devoted to improving the vacuum in steam turbine surface condensers, with the objective of extracting more turbine power from each pound of motive steam.

There are two types of vacuum ejectors:

a. The simple diverging jet, shown in Fig. 25.3.
b. The more complex diverging-converging ejector, which I'll discuss later.

The diverging jet will typically develop a compression ratio of about three to one. For example, the inlet pressure to the jet might be 200 mm of Hg. The atmospheric pressure in Denver (the Mile High City) is 700 mm of Hg. Therefore, the compression ratio is:

$$700 \div 200 = 3.5$$

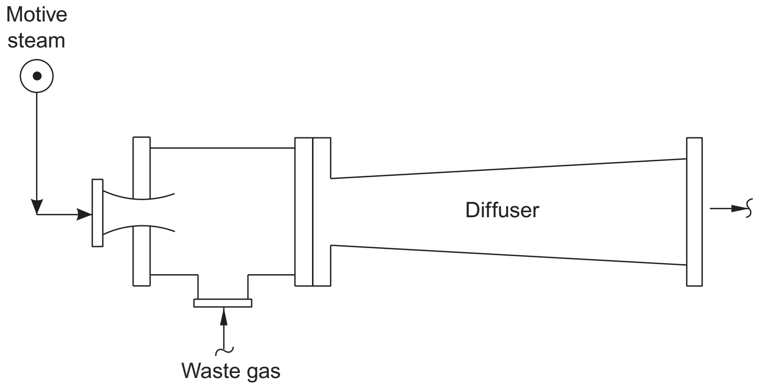

FIG. 25.3 A diverging or hogging steam ejector.

The motive fluid for an ejector is most often steam. But I have worked with ejectors using:

- Air
- Natural gas
- Nitrogen
- Refinery wet gas (mostly propane and butane)
- Water
- Diesel oil

Operators call the jet in Fig. 25.3, a "Hogging Jet." It is less efficient than a converging-diverging ejector and is typically employed in intermittent services, where the cost of the motive fluid is not important. The motive steam flows into the steam nozzle at perhaps 100 ft/s. Due to the shape of the nozzle, the steam accelerates to about 10,000 ft/s. The energy to accelerate the steam is derived mainly from the temperature of the steam. What I've observed is that if the motive steam is:

- 150 psig
- 370°F

As it comes out of the steam nozzle, it is flowing at:

- 200 mm Hg
- 150°F

The waste gas (Fig. 25.3) that is being evacuated is drawn into the ejector's mixing chamber. Typically, three or four pounds of motive steam is needed to suck in one pound of waste gas.

The mixture of waste gas and the motive steam flows into the discharge section, which is called the diverging section of the jet, because the cross-sectional area of the discharge pipe gets bigger. Since the cross-sectional area gets bigger, the vapor velocity will slow. That is, the kinetic energy, or velocity, or the momentum of the vapors, is converted into pressure. I call this the velocity boost. And just like any other compressor, the heat of compression causes the vapors to heat up. What I've observed is the vapors heat up by about 50–70°F.

Most hogging ejectors discharge to the atmosphere. Others discharge to a condenser, where the steam is condensed, and the residual compressed gas flows into a low pressure fuel gas system.

Converging-Diverging Steam Ejectors

Some ejectors develop a high compression ratio by relying on the "sonic boost" (see Fig. 25.4). As the high velocity steam flows through the ejector—starting out at 10,000 ft/s, it gradually slows in the diverging sections of the diffusor. The vapors flow from

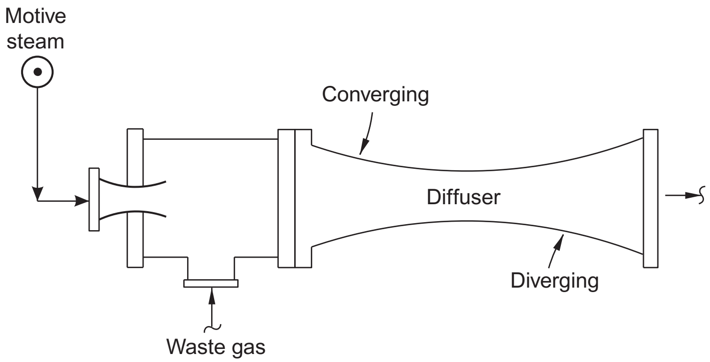

FIG. 25.4 Converging-diverging ejector.

a sonic velocity, to a subsonic velocity. This creates a pressure wave, which is called the "*Sonic Boost*."

Most operators who have worked with steam ejectors have heard the jets periodically making a "surging" or "hunting" sound, and associate this sound with a poor vacuum. The sound is an indication that the steam ejector is periodically losing its "Sonic Boost."

In a converging-diverging ejector, most of the compression is a result of this sonic boost. As the vapors (motive steam plus waste gas) leave the diffusor throat, their velocity is reduced as a result of flowing through the diverging section of the jet, where the cross-sectional area increases. At some point, the vapors, which have been traveling above the speed of sound since leaving the diffusor throat, drop below the speed of sound. This transition from supersonic to subsonic causes a pressure wave to develop. Kind of like a plane breaking the sound barrier, which generates a "*Sonic Boom*." It's typically referred to as the critical mode of ejector operation.

The sonic boost will compress the flowing vapors a lot. For example, from 30 to 200 mm Hg. This compression will heat up the diverging section of the diffusor body.

Loss of Sonic Boost

For the sonic boost to develop, the transition from sonic to subsonic velocity has to occur downstream of the diffusor throat. Factors that push back this transition zone (see Fig. 25.4), back to the diffusor throat, will cause the sonic boost to be lost suddenly and completely.

The sonic boost transition zone is pushed back because of back-pressure in the downstream condenser. The things that cause increased back-pressure from the condenser are as follows:

- Increased condenser pressure drop.
- Condenser fouling and loss of heat transfer efficiency.
- Increased cooling water temperature (i.e., summer time).
- Back-pressure at the condenser discharge because a downstream ejector is malfunctioning.
- Condensate back-up from the condenser seal legs.
- Air leaks in the condenser.

The other reason for the transition zone from sonic to subsonic is too far back in the diffusor is that the vapor velocity entering the diffusor is too slow. Still above sonic, but not as fast as it should be. Possible causes are as follows:

- The motive steam pressure is too low.
- The motive steam is wet.
- The steam nozzle is worn.
- The steam nozzle is leaking around its threads, where it screws into the ejector body.
- The steam nozzle throat has silicate deposits.
- The steam nozzle position is not adjusted correctly.
- The jet is overloaded with gas.
- There is a hole in the diffusor.

Natural Gas Recovery

In Laredo, I used this concept I had learned about vacuum ejectors in refineries. Juan and I worked out the design to recover the gas from the old well, using the high pressure gas instead of motive steam from the new well.

Well “A” was the old depleted well, with a wellhead pressure of 200 psig. The gathering system line pressure was about 600 psig. The 200 psig gas could not flow into the gathering system without compression. The potential gas flow from the well was only 200,000 SCFD and the installation of a reciprocating wellhead compressor would be hard to justify.

The other well, “Well B,” was the new, high pressure well, with a wellhead pressure of 2500 psig. The gas flowed through the wellhead choke, into the 600 psig gathering pipeline at a rate of 2,000,000 SCFD.

Juan and I purchased a hogging type (i.e., single stage, diverging—Fig. 25.3) ejector. We used the natural gas from well “B” (at 2500 psig) as motive gas to boost the smaller amount of gas from well “A” (at 200 psig) up to the pipeline pressure of 600 psig.

Don’t plan to visit this installation the next time you’re in Zapata County, Texas. Within a year, the pressure of well “B” had declined below that required to act as motive gas for the ejector.

GHR Energy went bankrupt. Juan Hernando returned to his ranch in Webb County. And I went back to troubleshooting refinery vacuum towers and condensing steam turbines back in Louisiana and in Enid, Oklahoma.

26

Steam Turbine Vacuum Surface Condensers

I began my career in 1965, as a process engineer for the American Oil Company in Whiting, Indiana. I applied myself diligently, but never advanced in my position. Likely, I devoted too much attention to troubleshooting process problems, and too little to interacting with management. Thus, in 1980, I resigned and secured a position as Technical Service Manager at the Good Hope Refinery, near New Orleans. I reported directly to the owner: Mr. Jack Stanley.

A few days after I started work, my phone rang, “This is Jack – K-805 won’t come up to speed. Fix it!” Click.

K-805 was the main combustion air blower on our 85,000 BSD FCU (fluid catalytic cracking unit). It was running slow. Thus, the blower’s discharge pressure was too low to supply air to the catalyst regenerator which resulted in a reduction in FCU feed capacity. I already knew the cause of the slow air blower speed:

- The steam turbine driver was producing only 80% of its normal horsepower.
- The low horsepower output from the turbine was due to the high turbine exhaust pressure to the vacuum surface condenser (see Fig. 26.1).

I calculated that to restore turbine horsepower, I would have to improve vacuum in the surface condenser from its current 90 mm Hg (26.5″ Hg) to 50 mm Hg (28″ Hg). But how? (Note: At sea level, atmospheric pressure equals 0.0″ Hg, 14.7 psia, or 760 mm Hg.)

Vacuum Condensers

James Watt did not invent the steam engine, as we were taught in high school. Watt invented the Barometric Condenser, which increased work extracted from steam by 300%. Watt’s design exhausted steam from a steam engine into a vacuum. Especially if the motive steam is at a low pressure, like 30 psig, the steam engine exhaust pressure is really important to maximize the amount of horsepower extracted from each pound of steam.

At my Sulfuric Acid Regeneration Plant in Texas City that I operated in 1974, the blower was driven by a steam turbine that exhausted to a barometric condenser. I’ve shown a sketch of it in Fig. 26.1. It was a trouble-free piece of equipment, unlike other items in that plant that I still recall with loathing after 43 years.

Understanding Process Equipment for Operators and Engineers. https://doi.org/10.1016/B978-0-12-816161-6.00026-6

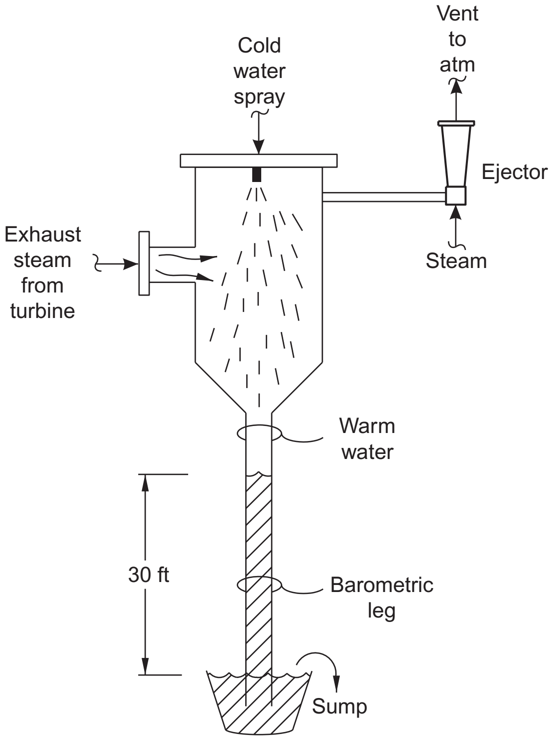

FIG. 26.1 Old style barometric condenser.

The exhaust steam flowed into an elevated chamber, where cold water was sprayed to condense the steam. The warm water and condensed steam were drained by gravity into a sump. Air leaks were extracted by a steam ejector. The pressure of the surface condenser was a little higher than the vapor pressure of water at the condenser vapor outlet temperature.

If the barometric condenser operated at a pressure of 100 mm Hg, and the atmospheric pressure in Texas City was 760 mm Hg, then the condenser would have to be elevated by:

$$\frac{(760-100)}{760}\cdot 34' = 30\ \text{ft} \tag{26.1}$$

I have assumed that atmospheric pressure at sea level is equivalent to 34 ft of water (14.7 psia or 29.97 in. of mercury, or 760 mm of Hg). The drain line is called a barometric leg, hence the name "Barometric Condenser."

Surface Condenser

Often, older refinery vacuum tower ejector systems are retrofitted to convert their barometric condensers to more modern surface condensers which usually do not work as well.

The origin of the term "*Surface Condenser*" arises from the fact that "*Barometric Condensers*" do not have any heat exchanger surface area, but surface condensers have internal heat exchange surface area.

Shell and tube heat exchangers all have internal surface area. So, the real meaning of the term surface condenser is that it has two outlets as shown in Fig. 26.2:

1. A liquid drain, typically from a boot in the bottom of the shell.
2. A vapor outlet for noncondensables, located on the top of the shell, or on the side of the shell, but inside an internal baffle used to collect the noncondensable vapors (i.e., an "Air Baffle").

Problems with the design, installation, and maintenance of this internal air baffle result in one of the largest wastes of energy in the history of mankind.

If the motive steam pressure to a condensing steam turbine is very high, perhaps 1500 psig, the vacuum developed in the surface condenser will only improve the amount of electricity, or horsepower, or work extracted from the steam by a few percent.

If the motive steam pressure to a condensing steam turbine is low, perhaps 100 psig (as was the case at the Good Hope FCU discussed below) the vacuum developed in the surface

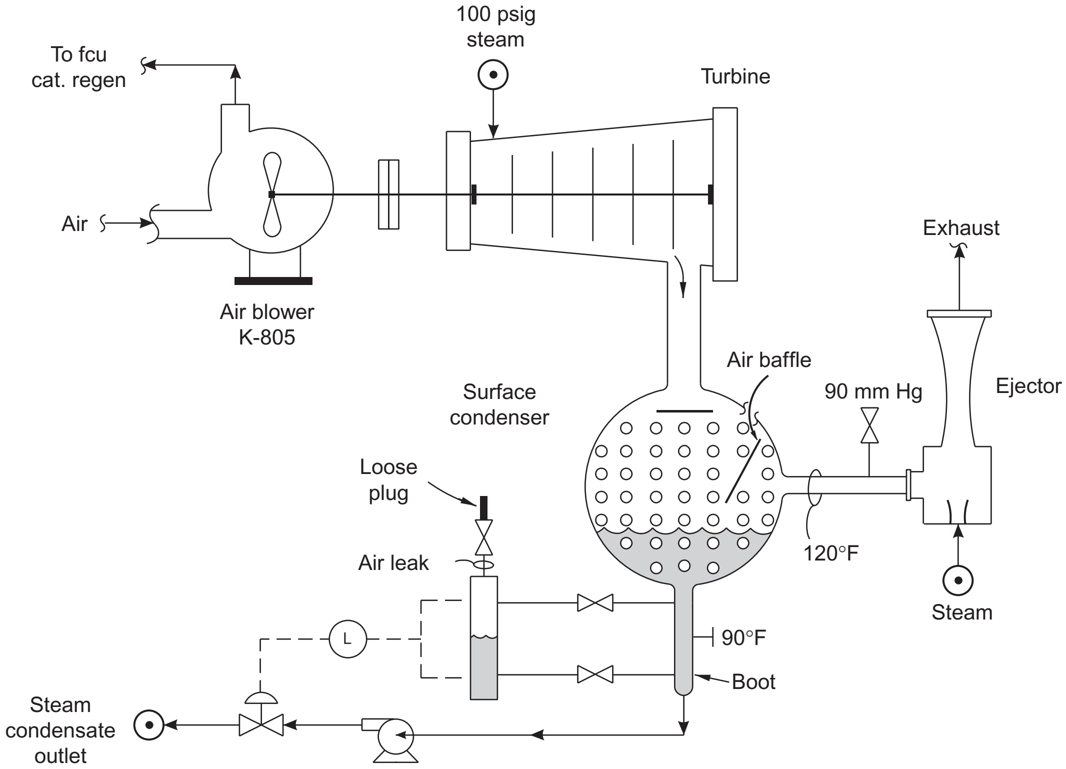

FIG. 26.2 Effect on condensate back-up in surface condenser slows FCU air blower speed.

condenser may improve the work extracted from the steam by 30%. For the case of low pressure motive steam, a reduction in the surface condenser inlet pressure of 2 or 3 in. of mercury (50 or 75 mm of Hg) (or 0.1–0.2 BARA) is a really big deal.

Correction for Barometric Pressure

A typical plant vacuum pressure gauge IS NOT corrected for atmospheric pressure. When I was younger, I didn't realize this and made some serious errors when field troubleshooting vacuum systems. You should avoid this mistake by using either:

- A mercury manometer
- An electronic digital absolute vacuum manometer

I use a "Meriam—Model #M101-GN0015" that cost $550 (United States) in 2015. Mercury manometers are no longer permitted outside of the lab in the United States.

When using a plant vacuum pressure gauge, regardless of the type, you'll have to make a manual adjustment as follows:

- *Step One*—The gauge is calibrated for atmospheric pressure (29.97 in. of Hg) at sea level.
- *Step Two*—Determine atmospheric pressure for your location either by checking the weather report or your lab manometer. Atmospheric pressure varies a lot with elevation and with weather. Colder weather increases atmospheric pressure because the air is denser.
- *Step Three*—Correct for your working elevation. For each 100 ft above grade, atmospheric pressure will be reduced by 0.11 in. of Hg.
- *Step Four*—To determine full vacuum at your location, take the difference between 29.97 and the ambient pressure at grade. Let's say ambient pressure in Denver, Colorado (a city in the United States at a high elevation) is 28 in. of Hg.

$$29.97 - 28.00 = 1.97'' \text{ Hg}$$

If you have climbed up a ladder with 100 rungs (i.e., 100 ft),

$$1.97 + 0.11 = 2.08$$

- *Step Five*—Subtract the 2.08 from 29.97:

$$29.97 - 2.08 = 27.89 \text{ in.of Hg}$$

That will be the reading for full vacuum on your pressure gauge.

- *Step Six*—If you wish to convert your reading ("*x*") to mm of Hg (absolute), as I always do:

$$(27.89 - x) \cdot (25.4) = \text{mm of Hg (absolute)}$$

$$\text{There are } 25.4 \text{ mm per in.}$$

Once I forgot to make this correction while working at a refinery in Canada at a high elevation. I reported to my client that vacuum in their surface condenser was really bad. Based on my analysis, they made a lot of costly changes, to no avail. The vacuum had been really okay all along.

Vacuum Limits Due to Vapor Pressure of Water

The best vacuum that can be developed in a surface condenser is limited by the vapor pressure of water at the condenser *VAPOR* outlet temperature. Likely half the vacuum systems in the world are limited by this one simple and easily observed parameter. In Table 26.1, I have tabulated the vapor pressure of water at typical condenser vapor outlet temperatures.

In stating this limitation, I am assuming that some free water is draining from the liquid outlet nozzle. I'll illustrate this common problem. Let's assume I have a condenser with a vapor outlet of 100°F. The vapor outlet composition is 200 pounds an hour of steam and the liquid outlet is 800 pounds an hour of water. According to Table 26.1, the condenser is operating at a vacuum of 49 mm Hg.

Now, I turn on a second jet in parallel with the first jet. Ejector capacity is doubled. So, the jets are pulling twice as much vapor flow from the condenser. The vapor flow has doubled to 400 pounds per hour. The liquid outlet flow has dropped to 600 pounds per hour. But, the vacuum is still the same 49 mm Hg, assuming that the vapor outlet is still 100°F.

The only way to improve the surface condenser vacuum, when limited by the surface condenser vapor outlet temperature, is to lower this vapor outlet temperature, by improving heat transfer efficiency in the surface condenser.

FCU Air Blower Horsepower

Now, refer to Fig. 26.2. Note that the pressure in the surface condenser vapor outlet (120°F) corresponds to the vapor pressure of water (90 mm Hg) at this vapor outlet temperature. This observation gave me a sense of confidence in my technical analysis.

I expected that the vapor outlet temperature should be the same or a bit lower than the water boot temperature. Lower because the vapor passes over several extra cooling tubes underneath the air baffle, as shown in Fig. 26.2. Typically, the effect of this air baffle is to

Table 26.1

Vapor Pressure of Water, mm Hg	
80°F	26
90°F	36
100°F	49
110°F	66
120°F	88
130°F	125
212°F	760

reduce the vapor outlet 10–20°F, below the water boot outlet temperature. But in this instance, the vapor outlet was 120°F, and the water boot temperature was 90°F. Opposite of my expectations!

Career Enhancement

The tone of Mr. Stanley's phone call had upset me. It seemed hostile. I had worked in the plant for only a few days and felt uncertain of my position.

Confronted by the above temperature profile, which contradicted my understanding of how vacuum surface condensers worked.

"Perhaps," I thought, "there's a big pressure drop across the air baffle that's causing the vapor outlet to be 30°F below the liquid outlet?"

So, I decided to check the pressure in the water boot with a vacuum pressure gauge.

I always try to make field measurements myself to ensure accuracy. I decided to connect my vacuum pressure gauge to the top of the level glass on the boot. I recall placing my 6″ pipe wrench on the ¾″ steel plug, to remove the plug from the top gauge glass vent valve.

However, I didn't need the wrench because the ¾″ threaded plug was *LOOSE*!

Then, I tightened the plug with my fingers, and:

- The air leaking into the level glass (and the level control bridle—see Fig. 26.2) stopped.
- The air pressure at the top of the glass went DOWN.
- The level in the glass went up.
- The boot level control valve opened rapidly.
- The water temperature in the boot went up (due to less condensate back-up).
- The temperature at the vapor outlet went down (i.e., the condenser had been suffering from condensate back-up).
- The vacuum in the surface condenser improved from 90 to 50 mm Hg.
- The turbine speed increased.
- The FCU air blower discharge pressure increased.
- The operators lined up the K-805 air blower to the FCU catalyst regenerator.
- FCU feed rate was restored.

Had I effected this improvement at American Oil, my former employer, I would have gotten a warm hand shake from the operators. But that would have been the extent of my reward.

At the Good Hope Refinery, where I worked directly for the owner, this was a big event. Mr. Stanley thanked me personally for my work. We became friends and have remained so to this day.

Well, that was 38 years ago. But it's kind of like your first date with Gloria in High School. There are some things you never forget.

Condensate Back-Up and Blowing the Condensate Seal

This story I have just related, illustrates the common problem of condensate back-up. Related to this is blowing the condensate seal, when steam blows out of the condensate drain nozzle. Also, an extremely common process limitation. However, regardless of the particular problem, the key to success is personal and direct field observations, combined with fundamental application of Chemical Engineering principles. None of my stories required purchasing of new technology, computer modeling, or proprietary process equipment. The stories are old, but the process technology is ageless.

27

Practical Use of Thermodynamics

“Earthling! You have been selected to be transported to Og, our home planet.”

“Why me? You should select a younger human. I’m 77 years old.”

“Your selection was not based on your extreme longevity. The selection algorithm chose you correlated for your technical content. Specifically, your stored knowledge relating to employment of steam in processing residual deposits of liquified hydrocarbons, extracted from sedimentary formations under the surface of your home planet – earth.”

“Fine! I’ve spent 57 years studying steam. What would you like to know?”

“On Og, we have selected a variety of different fluids to provide process functions – ammonia, methane, carbon dioxide, as well as water. But on your planet, Earthling, you use water in its vaporized state, to the exclusion of other fluids. Explain your preference for vaporized water.”

“It’s a long story. Do we have time?”

“The distance to Og is 12 earth light years. We will be traveling at 93,000 earth miles per second.”

“That’s 24 years. We’ve got lots of time,” I said.

Steam in Vacuum Ejectors

“My main interest with steam is its widespread use in petroleum refinery steam ejectors. I’ve written a book on this subject [1]. In Fig. 27.1, I’ve shown a sketch of a converging-diverging steam ejector. Most of the electricity which we generate on earth relies on use of the ejector. About 30% of clean products are recovered from black crude oil by use of the steam ejector. Most of our giant rotary engines, which we call steam turbines, also rely on this steam vacuum ejector. The vacuum ejector is a gas compressor, but without any moving mechanical parts.”

“What then compresses the gas?” the visitor from Og asked.

“The kinetic energy, or velocity, or the speed of the steam. I call it the ‘Venturi Principle’.”

“Venturi? Also an Earthling?”

“Yes. The high pressure steam at 170 psia and 400°F shown in Fig. 27.1, enters the steam nozzle at 200 ft per second. Due to the shape of the nozzle, the steam exits from the steam nozzle at 10,000 ft per second and typically a pressure of 0.4 psia (20 mm of Hg, absolute). That increase in the velocity of the steam represents a great increase in the kinetic energy or the momentum (mass times velocity) of the steam.”

“Yes. The pressure of the steam is converted to velocity. I have observed this myself when high pressure water escapes from a small aperture or nozzle. A clever earthman – Bernoulli – discovered this principle.”

Understanding Process Equipment for Operators and Engineers. https://doi.org/10.1016/B978-0-12-816161-6.00027-8

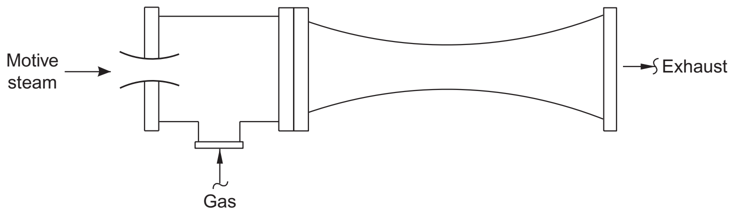

FIG. 27.1 Converging-diverging ejector.

“Sorry, but you are 98% wrong. Water is an incompressible fluid. Steam is a compressible fluid. For a compressible fluid like steam or air, the majority of the increase in kinetic energy or velocity does not come from the vapor’s pressure, but from the enthalpy, or the heat content of the vapor itself. For steam, the heat in the steam provides about 98% of the energy to accelerate the steam. The rest comes from the pressure of the steam.

“It’s really obvious. If I want to generate electricity for a city using steam from a boiler, I’ll supply the BFW (Boiler Feed Water) through a 12″ water supply line. If I generate electricity for a city by means of hydroelectric power, I need an entire river. That’s why we use coal, oil, gas, and U-235 to make steam to spin turbines in a magnetic field. In this manner, heat is converted into electric power.”

“The problems with generating electricity from steam are:

- To convert much of the heat content of the steam to power, the steam needs to be condensed at a good vacuum. For example, condensing 400 psig steam under a good vacuum, compared to just exhausting steam to the atmosphere, would increase electricity production by about 40%.
- Second, to achieve a good vacuum, air leaks and residual noncondensable gasses such as CO_2, need to be extracted from the exhaust steam condenser with an ejector system (see Fig. 27.2).
- Finally, if the motive steam is wet, in that it contains moisture, the ability of the steam to develop a good vacuum in the ejector is reduced.

“Here’s the problem. It’s the kinetic energy of the steam that compresses the air leaks. The faster the steam blows out of the steam nozzle, the more compression work it can do on the noncondensable air leaks. As the steam slows down through the diffuser (see Fig. 27.1), its velocity or kinetic energy or momentum, is converted into pressure in accordance with Bernoulli’s principle. But the kinetic energy of the steam is derived from its heat content. If the steam is very wet, then the water evaporates into steam as it comes out of the steam nozzle. That robs the steam of some of its heat content or enthalpy, which would otherwise be used to convert to kinetic energy.

“That is, moisture slows the steam as it leaves the steam nozzle. Slow steam can’t compress the noncondensable gas as efficiently as high velocity steam. The compression ratio inside the diffuser (see Fig. 27.1) is reduced. Since the discharge pressure of the ejector is fixed by the condensing pressure in the downstream condenser, the pressure at the suction of the ejector must increase.”

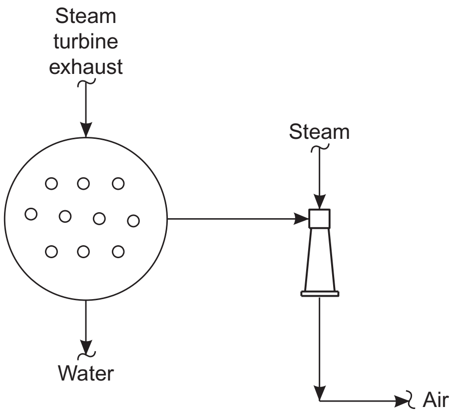

FIG. 27.2 Condensing steam turbine.

"So, Earthman, any water in the motive steam to an ejector will degrade its performance?"

"Not really! What I've observed is that one or two % moisture is not important. But much more, will definitely increase the ejector's suction pressure. Since we have several decades of travel together, I'll tell you a true story that happened in a Pemex Refinery in Monterrey, Mexico."

Property of Saturated Steam

"This plant had a vacuum tower servicing a 150,000 BSD crude unit. The tower top pressure was 35 mm Hg (0.7 psia), which is quite high. The design was about 15 mm Hg.

"The steam supply was 170 psia steam superheated to 500°F. Use of superheated steam reduces the mass flow of motive steam to the ejector. Superheated steam is less dense than saturated steam. So, Pemex had installed a 'De-superheating Station', to reduce the superheated 170 psia steam, to its saturation temperature of 365°F. The de-superheating station injects a water mist into the superheated steam. As the mist evaporates, the steam will cool due to the conversion of temperature to latent heat of evaporation of the water injected.

"In practice, 20°F or 30°F of superheat, does not appear to affect the performance of an ejector. But 135°F (i.e., 500°F − 365°F = 135°F) certainly does.

"To make sure that the steam was properly de-superheated, the unit engineer Pedro Desoto, had set the de-superheater temperature control (Fig. 27.3) to operate at 360°F, or 5°F *below* its saturation temperature at the flowing steam pressure."

"I looked at the condensate inlet control valve (TRC) and it was open 40%. As the setpoint was 5°F below the saturation temperature of the steam I expected the control valve would be 100% open. But, when I checked with Carlos, the panel control operator, he said, "Senor Lieberman. The valve previously used to run on auto at 100% open. But last week, I switched it to manual and closed it off to 40%. And, the vacuum went from 40 mm Hg to 35 mm Hg, which is muy bueno."

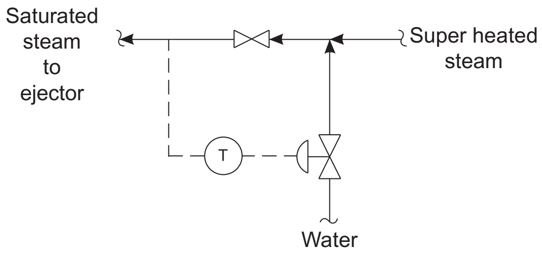

FIG. 27.3 De-superheater.

And I asked, "Why didn't you close it more?"

And Carlos said, "Because my supervisor told me not to make any more changes to the process without a written directive from our unit process engineer, Senor Pedro."

"So, I told Carlos to close that valve to 1%, and the vacuum improved from 35 mm Hg to 20 mm Hg. Afterwards, Pedro told me that the increased gas oil recovery from No. 6 (vacuum resid) was 4,000 BSD at $10 per barrel or $40,000 per day."

"Yes Earthling. I understand the importance of knowing the properties of steam. I understand that condensate injection cannot reduce steam below its saturation temperature, and that moisture in the ejector motive steam will raise the ejector inlet pressure and degrade vacuum. But what is the meaning of 'barrels' and 'dollars'?"

Field Observations

"Sometimes, in spite of my great age, younger people do not believe me, when I tell them that the velocity of steam that drives a turbine, or causes an ejector to develop a vacuum, comes not from its pressure, but from the temperature, or heat, or enthalpy of the steam. So, to prove this, I ask them to:

- Check the pipe temperature of the steam supply line to the ejector, and compare it to the temperature of the ejector's diffuser inlet (Fig. 27.1) (using an infrared temperature gun).
- Check the steam supply line temperature to a steam turbine, compare it to the temperature of the turbine's steam chest.

"In both cases, the reduced temperature observed is due to the conversion of heat into velocity. A famous book, 'Heat; a Mode of Motion,' [2] was written on this subject about 150 years ago. It's my favorite book."

"Visitor from Og, if you want to quantify the relationship between heat, kinetic energy and work its best to study the 'Mollier Diagram' and review the meaning of an Isentropic Expansion that I learned in Thermodynamics."

"Ah yes, Earthman! You should have said that you are describing a process that takes place at constant Entropy. On my home planet, Og, I will take you to visit our 'Temple of Entropy.' Entirely constructed from Lithium Bromide pellets."

Steam Turbines

"I can also relate a story about how steam turbines are run more efficiently by exploiting the concept of entropy and enthalpy in steam.

"I had decided that I could correct a plant's steam balance by increasing steam turbine efficiency. The plant had a large number of steam turbines that used 400 psig motive steam and exhausted the steam to 100 psig. The excess 100 psig steam was vented to the atmosphere. To stop the wasteful venting, I needed to extract 30% more work from each pound of the 400 psig motive steam by exploiting the difference between an isentropic expansion of the steam and an isenthalpic expansion. I had been trained on this technique in my youth, 57 years ago, when I studied Thermodynamics at University.

"Fig. 27.4 shows what is called in the refining industry, a 'Topping Steam Turbine'. Note that I have shown the governor and the two steam nozzles in this figure."

"Yes Earthman, I understand clearly."

"There are two ways to control the speed of a steam turbine:

- The evil way—Which keeps the heat content or enthalpy of the steam constant.
- The good way—Which keeps the entropy, or the potential of the steam to do work, constant, but reduces its heat content.

"The evil way relies on throttling the steam flow using the turbine governor and the good way uses the steam nozzles. (I have attached a Mollier Diagram for reference—Fig. 27.5)."

"Yes Earthman! The good way uses the steam nozzles in the turbine's nozzle block to control the flow of steam in an isentropic manner, and avoids the isenthalpic expansion of the motive steam. We know all about this on Og. But we use ammonia or methane as the working fluid, as we have very little water on Og.

"Incidentally, Earthling, can your body extract energy directly from starlight? We have no free oxygen on Og. Perhaps you can use metal ions or hydrogen sulfide vapors to sustain your metabolism?"

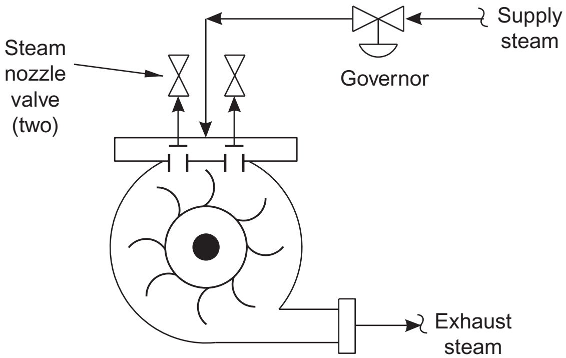

FIG. 27.4 Topping steam turbine.

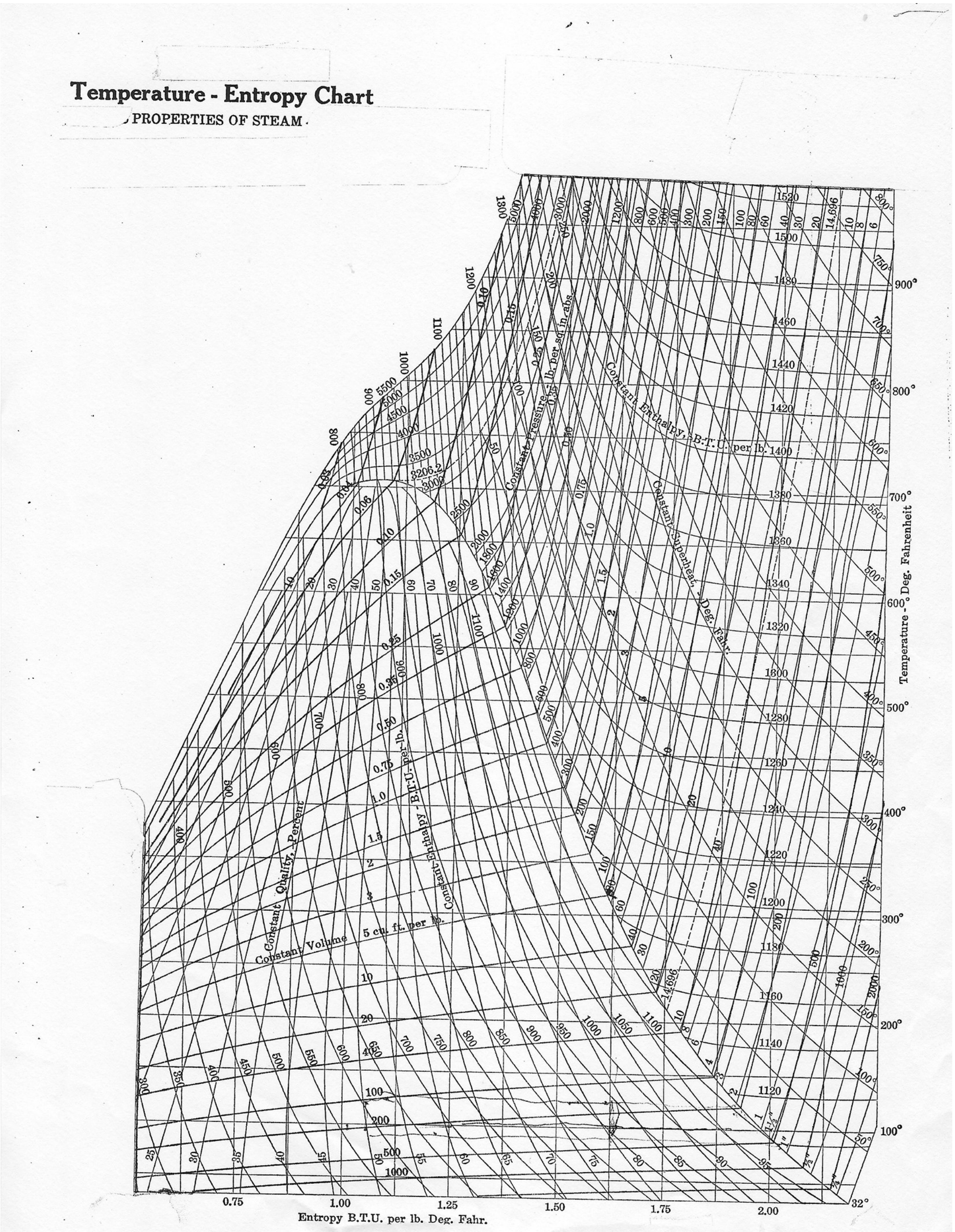

FIG. 27.5 Temperature-entropy chart.

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28

How Steam Does Work in Engines, Turbines, and Vacuum Ejectors

Reciprocating Steam Engine

Early in my education, in junior high school, I learned that the Industrial Revolution in England was initiated by the invention of the reciprocating steam engine, shown in Fig. 28.1. It seemed simple enough to understand. The 400 psig saturated steam, generated in a boiler, pushed against a piston head. On the other side of the piston was atmospheric pressure (i.e., 0 psig).

The motive 400 psig steam, having completed its work of pushing against the piston, would then exhaust to the atmosphere, as follows:

- The 400 psig steam supply inlet valve would shut.
- Simultaneously, the exhaust steam valve would open and vent the 400 psig steam to the atmosphere.

The steam, as it flowed into the cylinder's exhaust valve, was the same pressure and also the same temperature, as the motive steam supply. That is, the 450°F saturation temperature.

The Creole Queen

Last year, I went for a ride on the Creole Queen, an excursion steam boat that plies the Mississippi River in New Orleans. My friend Bob and I toured the engine room, which was open to visitors. During our tour, Bob asked me, "Norm, certainly the steam is doing work on the piston. Work is a form of energy. Since both the motive and exhaust steam pressure and temperature are both at the same conditions (400 psig and 450°F), where does that energy come from?"

"Bob, it comes from the pressure difference between the two sides of the cylinder. Between the head-end and the crank-end," I explained. I had learned that in junior high school.

"But Norm, that pressure difference, on its own, is not a source of energy. I guess you can say that the 400 pound steam is pushing the high pressure side of the piston, and causing the piston rod to move. But where is the steam getting the energy from to push the piston?

"I can see," Bob continued, "that the piston rod is moving the gear drive on the paddle wheel, which is pushing the Creole Queen upstream against the Mississippi's current. That takes energy that must be coming from the motive steam."

"You're right Bob."

"Yeah. But it sure ain't coming from the steam pressure or the steam temperature?"

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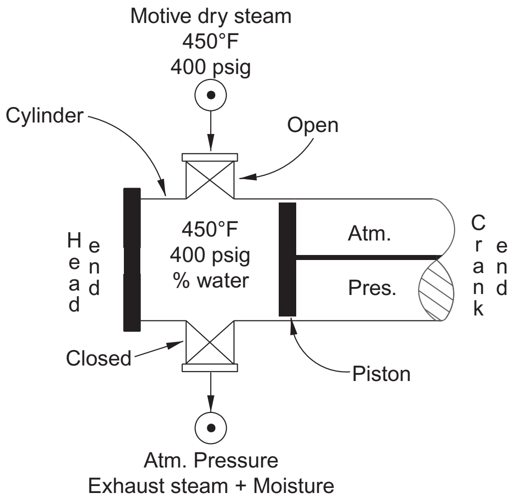

FIG. 28.1 Reciprocating steam engine during power stroke.

Thermodynamics

The energy in steam can take several different forms:

1. Temperature—sensible heat
2. Pressure—potential energy
3. Velocity—kinetic energy
4. Latent heat—vaporization or condensation

I've noticed as I age, things I thought I understood, I really never properly grasped at all. The older I get, the more mysterious ordinary things become. Perhaps it's a consequence of examining the physical world ever more closely? And Bob's question opened a whole new realm of uncertainty regarding one of my favorite subjects—thermodynamics.

Clearly, Bob was right. None of the energy produced from the Creole Queen's engine was extracted from the motive steam temperature or pressure.

"Bob," I observed, "in a steam turbine, it's the velocity or the kinetic energy of the steam, that drives the turbine wheel."

"Norm, we're not talking about a steam turbine. We're discussing the reciprocating steam engine on the Riverboat and the origin of the energy that drives it."

"Well Bob, that energy is the last item on my list. Item No. 4—Latent Heat."

Mollier Diagram

When we returned to my home in New Orleans, I extracted from my files, an aged Mollier Diagram (Fig. 28.2). "Bob, as the steam flows into the cylinder, between the piston and the cylinder head (Fig. 28.1), it exerts an expansive force against the piston head. The work of expanding steam pushes the piston rod, which spins the paddle wheel."

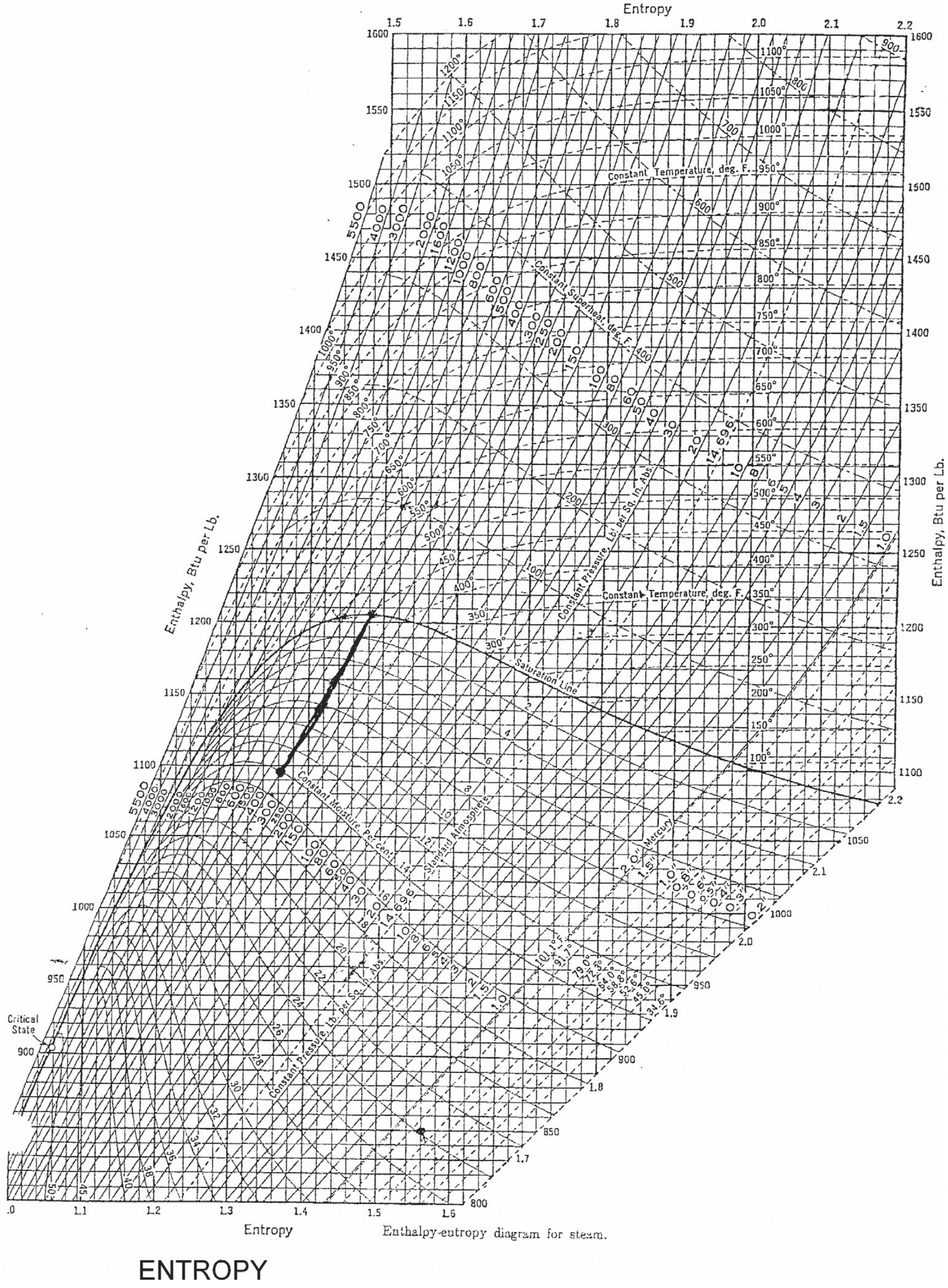

FIG. 28.2 Mollier diagram.

"Right, okay. But where's does the energy come from to exert this expansive force or work?"

"Bob, as the steam expands at a CONSTANT PRESSURE, the steam will start to partially condense. The steam is still saturated at 450°F, but its heat content, or enthalpy, is reduced, because it is doing work as it pushes the piston. Some of the steam vapor has changed to liquid water.

"Or, it's applying a force over a distance as the steam expands and pushes the piston. And:"

$$(\text{Force}) \cdot (\text{Distance}) = \text{Work}$$

"Norm, how much work is the steam doing?"

"Well Bob, let's look at the Mollier Diagram. You can see that 14 weight of the saturated 400 psig steam has condensed. The heat content of the steam has been reduced from:

- 1205 btu/lb
  To
- 1095 btu/lb

"About 110 btu/lb has been lost from the motive steam."

"I got it Norm. That lost heat has been converted to WORK. But how much work?"

"Well Bob. There's around 2500 btu's in each Horsepower. So:

$$(110) \div (2500) = 0.044\,\text{hp}/(\text{lb STM})$$

"I asked the Chief Engineer on the Creole Queen—you know, that drunk looking guy—how much steam they were raising in their boiler. He said 12,000 pounds per hour."

"Well that means they were generating:

$$(0.044) \cdot (12{,}000) = 528\,\text{hp}$$

"Is only 14% of the steam condensed, Norm?"

"No Bob. Likely it's more than that, due to ambient heat losses, friction, valve losses, and other mechanical inefficiencies."

"You know, Norm, I guess I never really understood before how those early steam engines invented in the 1700s, actually worked."

"Me neither. I thought I did, but I didn't figure it out until just last year."

"I wonder why they didn't explain it to us properly in high school, in New York?"

"Most likely, Bob, because Mr. Steinback, the science instructor, didn't know himself."

"Well Norm, you're never too old to learn."

Steam Turbines

I learned in 1980 how steam turbines work during a 5-month strike in Texas City, when I worked as a Scab. I knew I was a Scab, because when the operators returned to work, they gave me a t-shirt that said "Rat Scab."

Steam turbines work differently than the steam engine I've described. A reciprocating steam engine converts the heat content of the steam directly to work by pushing on the piston.

In a steam turbine, the heat content of steam is first converted to kinetic energy or momentum. It's the kinetic energy of the steam derived from its enthalpy or heat that drives the turbine wheel. I used to think that it was the pressure of the steam that spins a turbine. Let's calculate how much of a turbine's horsepower output is derived just from the steam pressure.

Steam Pressure Converted to Horsepower

A typical steam turbine will produce 1 hp for each 10 lb/h of 1000 psig steam. How much of that 1 hp is derived from the steam pressure?

- *Step 1*—There are 2.3 ft of water in a single psi.
- *Step 2*—$(2.3\,\text{ft}) \cdot (1000\,\text{psi}) = 2300\,\text{ft}$.
- *Step 3*—There are 770 ft-lbs (i.e., potential energy) in each btu (heat).
- *Step 4*—$(10\,\text{lbs of steam}) \cdot (2300\ \text{ft}) \div (770\,\text{ft-lbs}) = 30\,\text{btu}$

The heat content of 1 hp is equal to about 2500 btu. Therefore:

$$(30\,\text{btu}) \div (2500\,\text{btu/hp}) = 1\%$$

That is, only 1% of the energy to generate power from a steam turbine comes from the steam's pressure.

Or, I would need 100 pounds of water at the same pressure as a single pound of steam to produce the same amount of work.

A plant that generates hydroelectric power requires the flow from an entire large river. A plant to generate electricity from steam requires a 10 in. water supply line.

Momentum of Steam

The water pressure in my house in New Orleans is 40 psig. After it comes out of the nozzle of my garden hose, this pressure has been converted to a velocity of around 70 ft per second (this is a calculated number, not including any frictional losses). If I squirt the water against the wheel of my up-ended bicycle, I'll convert the momentum of the water, derived from its pressure, into rotational energy.

If I performed the same experiment with 40 psig steam, the velocity hitting my bicycle wheel would be about 700 ft per second. This extra velocity had been derived from the enthalpy of the steam.

A steam turbine works in the same way. The steam flows out of the steam chest and through the steam nozzles. Depending on the exhaust steam pressure leaving the turbine case, the steam blowing out of the nozzles can be 1000 or 10,000 ft per second. If the steam from the turbine case exhausts into a 30 psig low pressure steam system, it might exit the nozzles at 1000 ft per second.

If the steam from the turbine case exhausts into a vacuum surface condenser, operating at 27 in. of mercury (75 mm Hg), it might flow out of the nozzles at 10,000 ft per second.

Either way, 98+% of the energy to accelerate the motive steam comes from the heat of the steam—not its pressure.

Function of the Turbine Governor

It's hard to see the position of the Governor in the field. To estimate its position, I'll compare the pressure of the motive steam to the pressure in the steam chest. You will rarely find a pressure connection on the steam chest. However, I use the water drain line from the steam chest for this purpose. If the motive and chest pressures are within 5–10 psi, then the governor is 100% open.

The purpose of the Governor Valve is to control the steam chest pressure. If the turbine is running above its set or required speed:

1. The governor will close.
2. The steam chest pressure will decline.
3. The flow of steam through the nozzles in the steam chest will decline.
4. The mass and the velocity (i.e., the momentum) of the motive steam striking the turbine wheel will diminish.
5. The turbine will slow to its set speed.

The steam pressure drop through the governor reduces the ability of the steam to do work, as the governor does not efficiently convert the heat content of the steam to kinetic energy.

Port or Hand Valves

The hand valves are used to save steam:

- *Step 1*—Closing one hand valve will reduce the flow of motive steam impacting the turbine wheel.
- *Step 2*—The turbine slows, which causes the governor to open.
- *Step 3*—The combined effect of Steps 1 and 2 is to increase pressure in the steam chest.
- *Step 4*—The amount of steam, and hence the velocity of the steam, blowing through the still open nozzles, increases. This restores the turbine speed with less steam.

The operator should continue closing hand valves until the governor is wide open. A rough rule of thumb is that for each hand valve closed, 10% of the turbine's steam usage is saved.

Another way to save steam is to slow the turbine. Each reduction of 10% in turbine speed will save roughly 25% of the motive supply steam.

At Texas City, while working for Amoco, I spent 12 h one night in 1980, adjusting steam turbines. I saved 20,000 pounds per hour of 400 psig steam. I thought Larry Durland, the plant manager, would reward me with a gold star, or a Certificate of Merit, or something…

But all I received was a warning letter never to alter a steam turbine operation without the permission of the maintenance manager.

Isentropic Expansion

The high velocity steam that impacts the turbine wheel causes the turbine to spin. This velocity comes from the heat or enthalpy of the motive steam, as shown in Fig. 28.2, the Mollier Diagram. If we are using 400 psig saturated motive steam and exhausting the steam to the atmosphere, we would drop straight down on the Mollier Diagram (i.e., along the line for constant entropy) and read:

- Motive steam = 1205 btu/lb
- Exhausted steam = 965 btu/lb
- Heat converted to kinetic energy = 240 btu/lb

As there are about 2500 btu per horsepower, this means each pound of steam would produce:

$$240 \div 2500 = 0.096\,\text{hp}$$

Turbine Efficiency

Actually, the above calculation is not quite correct. It assumes 100% efficiency. More likely, based on a few measurements I've made at a fertilizer plant in Enid, Oklahoma last year, efficiency of about 70%–80% should be assumed.

On the other hand, the amount of work extracted from a pound of steam can be improved by:

- Exhausting the steam to a condenser under vacuum.
- Running with the governor valve more open.
- Superheating the motive steam.
- Increasing motive steam pressure.

I suggest the reader calculate from the Mollier Diagram (Fig. 28.2), the incremental work that can be recovered by exhausting the steam to a good (26″ Hg) vacuum.

Vacuum Ejector

The largest part of my work as a process consultant is involved in design and troubleshooting vacuum ejectors. A vacuum steam ejector is like a two-stage compressor but with no moving parts. What then compresses the off-gas from a vacuum tower or the air leakage from a turbine's surface condenser?

It's the velocity of the motive steam.

The steam flows to the steam nozzle at 100–200 ft per second. It exits from the steam nozzle at a velocity of 10,000–12,000 ft per second. Ten times sonic velocity. One or two percent of the energy to accelerate the steam comes from reducing the steam pressure from 150 psig to 26 in. of the Mercury vacuum. Most of the rest of the energy to accelerate the steam comes from the steam's temperature.

In 1965 when I was the new process engineer on a crude unit, working for American Oil, I noted that the supply steam temperature to the vacuum tower jets was 380°F. Yet, only a few inches after the steam entered the front end of the jet, the steam had cooled to 100°F. The steam had suddenly lost:

$$\left(380^{\circ}\mathrm{F} - 100^{\circ}\mathrm{F}\right) \cdot (0.55) = 154\,\mathrm{btu/lb}$$

where 0.55 is the specific heat of steam.

The steam had not lost any energy. The 154 btu/lb had been converted to velocity of the flowing steam.

Momentum Converted to Pressure

As the steam travels through the jet, it slows. About half the kinetic energy of the steam is converted back into heat. That is, the jet discharge temperature increases to 200°F. The other half of the jet's kinetic energy is converted into work, compressing the inlet off-gas from 26 to perhaps 10 in. of Mercury.

[Note: 26″ of Hg = 100 mm Hg absolute, 10″ of Hg = 500 mm Hg absolute.]

The increase in pressure is not uniform across the length of the jet. A small portion of the pressure increase occurs upstream of the diffusor throat. About 70% of the increase in pressure occurs after the diffusor throat, as the steam velocity slows from sonic to subsonic. This is called the "Sonic Boost." This Sonic Boost can be lost due to several malfunctions:

- Wet motive steam
- Excessive gas rates
- High discharge pressure
- Steam nozzle erosion

When the sonic boost is first lost, the jet makes a surging or hunting sound, as the majority of its compression work is lost.

Summary

Regardless as to how steam does work in a variety of machines, it is not the pressure of the steam that is supplying most of the energy. It is the enthalpy or heat of the steam that is spinning a turbine wheel, or compressing a gas in a vacuum jet, or pushing the piston in a steam engine. The original text, published by Dr. Tyndall in the 1830s that described how a steam engine works, was called "Heat: A Mode of Motion." In the 20th century, this subject was renamed "Thermodynamics." This wonderful book is still in print—in paperback no less! I have a hardback copy of the third edition on my desk as I write these words.

29

Centrifugal Pumps

"I have told you before not to use 100 psi on the seal flush. Twenty pounds is enough."

"But that's too low. The pump discharge pressure is 80 psi. The seal flush pressure needs to be above the discharge pressure by about 20 pounds."

"That's completely wrong. The pump's seal flush pressure must be a bit higher than the suction pressure. Not the discharge. Especially since the pump is in reduced crude service and the flushing fluid to the seal is diesel. You're losing a lot of valuable diesel product to the reduced crude."

"Really, Lieberman. How much? Can't be a lot?"

"It depends to a great extent on the design of the mechanical seal. For an older pump with an old style component seal:

- 3–4 GPM per in. of shaft diameter, when the pump is running.
- Half that amount when the pump is idle.
- Twice that amount if there is both an inboard and outboard seal.
- For more modern pumps with cartridge seals, use one-third that amount.
- If the seal flush pressure exceeds its design pressure by a large amount, because it's running above the discharge pressure, rather than just above the suction pressure, the amount of seal flush may get really high according to:"

$$\text{Seal oil flow} \approx (P_{\text{seal}} - P_{\text{suction}})^{1/2} \tag{29.1}$$

where:

- P_{seal} = Seal oil pressure downstream of the seal flow adjustment valve, psi.
- P_{suction} = Pump suction pressure downstream of the suction strainer, psi.

"Mr. Lieberman, what you say only makes sense to me, if the pressure inside the entire pump case is the suction pressure, rather than the pump discharge pressure. But, the pressure of the liquid inside the pump case is certainly the discharge pressure. That's because the liquid has been pressured-up by the pump impeller (see Fig. 29.1)."

"That's not right for a centrifugal pump. What you're thinking does apply for a positive displacement pump. However, 95% of the pumps in our plant are ordinary centrifugal pumps, where the function of the impeller is not to increase the process liquid pressure, but accelerate the flow."

"Sir! Could you explain this to me? I thought I understood how my pumps worked. But now I'm confused."

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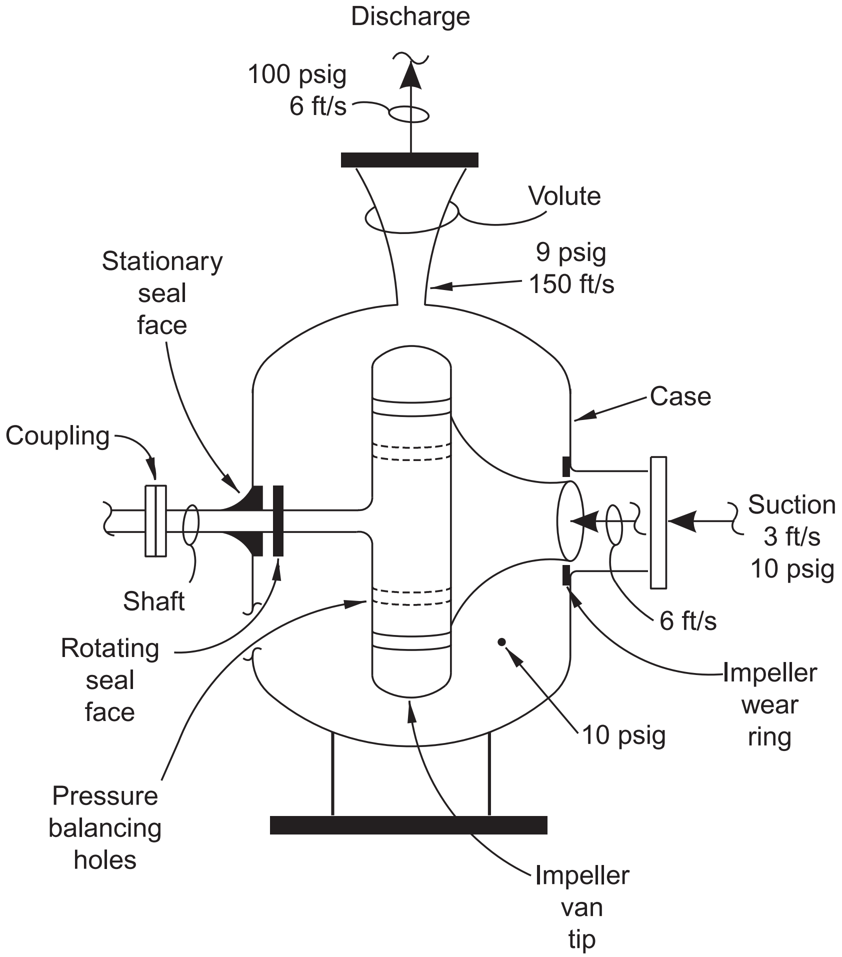

FIG. 29.1 Pressure inside the pump case is the pump suction pressure. Pressure increases as liquid flows thru volute.

Velocity, Head, and Pressure

"I once performed an experiment when I was 12 years old. I had been told that the water pressure in our bathroom was 40 psig and I wanted to measure it. So, I took our bathroom scale into the backyard. Next, I adjusted the hose's nozzle to get the biggest pressure possible from the hose. Then, I directed the jet of water straight down onto the scale."

Much to my disappointment, the scale reading did not increase to 40 pounds, but only to about six pounds. But then I noticed that the jet of water was concentrated in a circle of only a half inch diameter.

One-half inch is equal to an area of:

$$(0.5)^2 \times \pi/4 = 0.2\,\text{sq.in.} \tag{29.2}$$

So I then divided the 6 pounds reading on the scale by 0.2 $(\text{in.})^2$:

$$6\,\text{lbs} \div 0.2\,(\text{in.})^2 = 30\,\text{lbs}/(\text{in.}) = 30\,\text{psi} \tag{29.3}$$

I figured that the missing 10 psi ($40 - 30 = 10$) must be due to losses, because the water had to flow a mile from the water tower on Veteran's Blvd., to my house.

I could understand that loss of 10 psi. But, what I found confusing was that the water flowing from the garden hose nozzle was not at a pressure of 40 or 30, but at 0 psig, or atmospheric pressure. So I wondered from where, was the 30 psi pressure that I had measured on my bathroom scale coming from?

Conversion of Velocity to Flowing Pressure

Daniel Bernoulli came from Switzerland. According to Dan, there were several ways that energy could appear:

- Heat—thermal energy
- Velocity—kinetic energy
- Head—potential energy
- Pressure—effect of gravity
- Electricity

Directing a jet of water from a garden hose straight up converts the high velocity (kinetic energy) of the water into feet of head (potential energy). At sea level in New Orleans:

$$D_{\text{H}} = 0.178 \cdot V^2 \tag{29.4}$$

where

- D_{H} = Ht. of water, inches
- V = Velocity of water, ft/s.
- The 0.178 is the theoretical conversion factor for converting the kinetic energy or velocity of water (ft/s) to inches of water head (i.e., potential energy).

If water shooting out of the end of a garden hose at 30 ft/s was directed straight up, it would rise to:

$$0.178\,(30)^2 = 160\,\text{in.} \equiv 13\tfrac{1}{2}\,\text{ft} \tag{29.5}$$

Actually, the water would not quite rise to 13½ ft because of frictional losses.

Function of a Pump Impeller

"Mr. Lieberman. You've drifted off the subject of pump impellers."

"Not really Donald. Let's refer to Fig. 29.1 again. A typical suction piping velocity might be three feet per second. Then the liquid flows into the eye of the pump's impeller and accelerates to say six feet per second. That acceleration of the flow from 3 to 6 feet per second requires energy."

"Does that acceleration energy come from the pump's impeller?"

"No. It comes from the potential energy of the liquid, or from the head of liquid."

"Is that what you engineers call the '*Required NPSH' of the pump*?"

"Yes."

"And if the required NPSH isn't smaller than the head of liquid available in the vessel I'm pumping out of, then I need to raise the vessel liquid level. Otherwise, my pump's going to cavitate," Donald said.

"Yes. You'll tear up the pump's mechanical seal due to vibration."

"This is getting interesting, Mr. Norm. So what happens to the liquid after it flows into the eye of the impeller?"

"The liquid is spun around inside the impeller. As I've shown in Fig. 29.1, the liquid accelerates from six feet a second to a hundred feet a second as it discharges from the vane tip of the impeller."

"Norm, where does that energy to accelerate the liquid come from? Also from the available NPSH?"

"No. It comes from the electric motor, or steam turbine, spinning the impeller. According to Dan Bernoulli, electricity, which is a form of energy, can be converted to kinetic energy, which is what's happening inside of the pump's spinning impeller.

"The pressure of the liquid discharged from the impeller is a little bit lower than the pump suction pressure. The impeller discharge pressure, is that pressure inside the entire pump case. Just to make real sure that the pump's seal doesn't see any pressure much above the pump suction pressure, there are holes (pressure balancing holes) drilled through the impeller, as I've shown on Fig. 29.1."

"Yeah Mr. Lieberman. I've seen those holes before. Jerry told me those holes were there to keep the pressure on both sides of the impeller equal. Is that right?"

"Yes Donald, Jerry is correct. But, that pressure he's referred to is the pump suction pressure, not the discharge."

"Mr. Norm. None of this makes any sense. How does the pump discharge pressure increase from the 10 psig suction, to 100 psig discharge? I'm still confused."

Converting Velocity to Head

"Let's look down into the discharge nozzle of an un-flanged pump. The discharge flange is six inches, but the opening down inside the pump case is a lot smaller. Go ahead and look, Donald."

"Yes Sir, it looks like it's about a 1½ inch opening."

"It follows then (from Fig. 29.1), that the velocity at the 6″ discharge flange has dropped from 100 ft per second, down to 6 ft per second:

$$100\,\text{ft/s} \cdot (1.5/6)^2 = 6\,\text{ft/s} \tag{29.6}$$

"And the reduction in velocity has been converted, Donald, into…"

"Hey! I got it! It's just like a roller coaster car slowing down as it goes uphill. The speed of the car is converted to elevation, or what you call feet of head. Or, what that Swiss guy Bernoulli, calls Potential Energy. But that don't happen until the liquid's out of the pump itself. That's why the seal flush pressure only needs to be a little bit higher than the pump's suction pressure. The discharge pressure really don't affect the pressure that the pump's mechanical seal has to hold. And even if the seal flush pressure did build up a bit behind the impeller, that pressure would bleed itself back to the pump suction through those pressure balancing holes drilled in the back side of the impeller. Yeah man! I got it."

Relationship of Head to Discharge Pressure

"Mr. Norm, you keep talking about 'head'. That doesn't mean anything to me. As an operator, I only see pressure—not head. Doesn't the centrifugal pump create a certain amount of discharge pressure?"

"Well, yes and no, Donald."

"Could you make up your mind? Does a centrifugal pump increase the pressure of the liquid it's pumping?"

"A centrifugal pump, which is pumping a non-viscous product, develops a certain amount of feet of head, at a given volume of flow, regardless of the density or specific gravity of the liquid pumped. To be more clear…"

"Mr. Lieberman, you couldn't be less clear. But go ahead. Talk slow and I'll listen."

"Okay Donald. To convert from feet of head—which is what the pump creates—to pressure—which is what you're interested in, we use the equation:"

$$\mathrm{DP} = \frac{(D_\mathrm{H}) \cdot (\mathrm{SG})}{2.31} \tag{29.7}$$

where:

- DP = Increase in discharge pressure over suction pressure, psi.
- D_H = Increase in feet of head developed by pump, feet.
- SG = Density of liquid at flowing temperature compared to density of cold water at 60°F.

"So a pump that's pumping diesel will put up more discharge pressure than a pump that's pumping naphtha 'cause the diesel's denser than the naphtha?"

"Correct, Donald."

"Yeah. Okay. I got the specific gravity idea. But how about the, 'nonviscous' stuff? What's all than mean?"

"Look Don. Ninety-plus percent of the stuff we handle in a refinery is low viscosity. Meaning less than fifty centistokes. In our plant, the only high viscous stuff is cold heavy Venezuelan crude, or cool vacuum tower slop wax. Anything that has a viscosity of more than 50–100 centistokes wouldn't be handled with a centrifugal pump, but a different type. A positive displacement pump."

"Let's just stick with the centrifugal pump for now, Mr. Norm. You can explain about 'Positive Displacement Pumps' later. But how about what you just said about 'At a given volumetric flow.' What's that mean?"

Pump Curve

"Let's refer to Fig. 29.2. The horizontal axis of the chart is flow in GPM. The vertical axis on the left side of the chart is head in feet. The vertical axis on the right side is required horsepower. It doesn't matter if you're pumping water with an S.G. of 1.00, or of gasoline with an S.G. of 0.75. The pump will still produce the same feet of head as long as the GPM or flow is constant."

"Okay. But how about that horsepower curve? I guess that's the same for water and gasoline also?"

"No," I answered, "It's confusing. The motor horsepower, or the amount of amps the motor will pull, is a linear function of the specific gravity (S.G.). If the S.G. increases by 20%, so will the motor amps. The pump curves are developed using water in a test rig. Even though a pump may be designed for diesel with an S.G. of 0.85, the horsepower shown on the right hand vertical axis of the pump curve is likely for water."

"Man, that is real confusing."

"Yes it is."

"But still Mr. Lieberman, I'm real glad you explained this to me. It explains why my P-403 pump trips off on high motor amps, when I switch from water to caustic, unless I throttle back on the discharge of that pump first."

"Right. Donald the S.G. of the water is 1.00. The S.G. of the caustic is 1.20. At the larger S.G., you're going to be limited by motor horsepower. Not the pump's capacity."

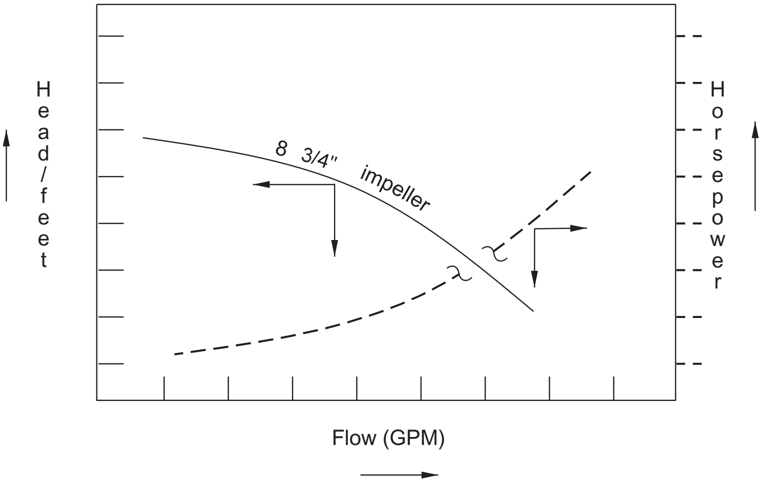

FIG. 29.2 Pump performance curve, amps are proportional to motor horsepower.

Positive Displacement Pumps

“Mr. Norm, it’s near the end of my shift and I’m going to the gate soon. But I got one more question. Out in the tank field, we got a crude pump that I’ve been told is different from all our centrifugal pumps we got on our unit. I looked at it a while ago. It looks the same as our centrifugal pumps. What’s the difference?”

“I get confused by this also, Donald. Both types of pumps do appear the same. But that pump in the tank field is likely to be a positive displacement pump. The important difference is:

- Centrifugal pumps develop the same feet of head for a particular flow rate.
- Positive Displacement pumps do not run on a curve. They put up as much head or discharge pressure as they need, to overcome the pressure of the system they’re pumping into. If the pump needs to push 100,000 BSD of crude into a tank, its discharge pressure might be just 25 psig. If the pump needs to push 100,000 BSD of crude to your crude unit, its discharge pressure could be 150 psig.”

Seal Oil Flow to Idle Pumps

“Good explanation Mr. Lieberman. I’m going to re-set the seal flush pressure on all my pumps tomorrow. Just like you told me. A bit above the pump’s suction pressure. Does that go for the spares also, that I ain’t running?”

“No! For example, take your resid pumps, that use diesel for seal flush. When you shut the pump down, Donald, wait a few minutes for the diesel oil to flush through the pump. Then block the diesel seal flush in. Even though the pump is not running, the seal flush flow will still be around 60% of its flow, when the pump was in operation. Every barrel of diesel you put into resid is costing the refinery around ten bucks. Maybe more. Okay Donald, so let’s not waste seal flush on pumps that are not in operation.”

“Man, they should have told me that when I hired in. I’ve been leaving the seal flush lined-up to all my idle pumps for twenty years. Someone should’ve told us operators before not to do that. What a waste!”

“Yeah Don! Walk in the light, while there is light.”

Safety Note

The heavier the material, the lower its auto-ignition temperature. Gasoline has an auto-ignition temperature of 480°F. Asphalt, when spread over a porous surface, like your clothes, auto-ignites at 300°F.

That’s why a seal leak on heavy vacuum gas oil pumps is so dangerous. It’s much safer to get a sample of naphtha at 350°F than vacuum resid at the same temperature.

Benefits of Excess Available NPSH

Supplying extra “*Available Net Positive Suction Head*,” tends to suppress formation of bubbles, and hence of cavitation, inside a pump’s impeller, by reducing recirculation inside the impeller. This will improve the reliability and safety of the pump, by extending the operational life of the pump’s mechanical seal. Also, seal maintenance costs will be greatly reduced and largely offset the capital investment for increased vessel elevation. One new modern double-mechanical seal may cost $50,000 (United States) to purchase.

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Centrifugal Pumps: Required Starting NPSH

One common problem in starting a centrifugal pump is establishing flow to the pump's suction. That is, the pump cavitates as soon as the pump's start button is pushed. When faced with this problem, the operator may:

- Raise upstream vessel liquid level.
- Open the discharge-to-suction spill back line (see Fig. 30.1).
- Spray cold water on the suction line and pump case.

The following story illustrates the problem that occurred on a crude distillation unit flash drum in a mid-west US refinery.

Start-Up Problems

I was discussing this problem with a young friend of mine, Howard.

"Mr. Norm, we learned about pump net positive suction head (NPSH) today at University. First, you got your required NPSH. You read that from the pump curve (Fig. 30.2). As the flow goes up, the required NPSH increases.

"For the available NPSH:

$$\text{Available NPSH} = (P_1 + P_A) - (\text{VP}) \cdot 2.31 \div \text{SG} \tag{30.1}$$

where:

- NPSH = Feet.
- P_1 = Pressure at pump suction (psig).
- P_A = Atm pressure, psi.
- VP = Vapor Pressure of liquid at pump suction, psi.
- SG = Flowing specific gravity.

"All you do is provide enough Available NPSH so that it's equal to the Required NPSH at the max design flow rate of the pump," said Howard. "You can get enough of that NPSH by either elevating the drum above the pump it is serving, or sub-cooling the liquid that is being pumped. Perhaps by spraying icy cold water on the pump's suction line. That'll cool the liquid below its saturation temperature."

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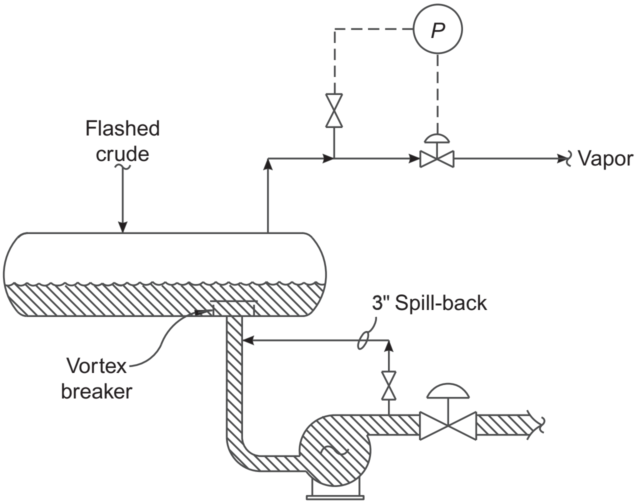

FIG. 30.1 Typical pump installation.

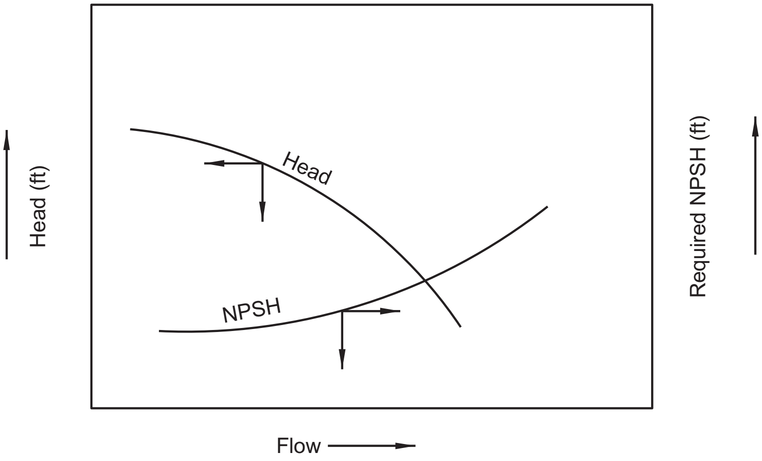

FIG. 30.2 Manufacturer's pump curve showing required NPSH.

Nozzle and Piping Losses

"There are three factors that increase required NPSH beyond that specified on the manufacturer's performance curve," said Howard.

"One of these factors is nozzle losses, and the other is piping friction. For low viscosities, less than 20–30 cSt, the nozzle exit loss is:

$$D_{H_1} = (0.3) \cdot (V)^2 \div 12 \tag{30.2}$$

where:

- D_{H_1} = Loss of NPSH due to acceleration through the nozzle, feet.
- V = Nozzle exit velocity, in ft per second.

"The 0.3 coefficient assumes no vortex breaker on top of the nozzles." (see Fig. 30.1)

"If there is a vortex breaker, you might increase the 0.3 to a 0.5 coefficient in Eq. (30.2).

"There are also suction piping losses due to friction. You can calculate the piping losses using this equation:

$$D_{H_2} = \frac{4}{\mathrm{ID}} \cdot \frac{D_L}{62.3} \cdot \frac{(V)^2}{27.7} \tag{30.3}$$

where:

- D_{H_2} = Piping loss per hundred equivalent feet, psi.
- D_L = Density of fluid, lb/ft^3.
- V = Line velocity, ft/s
- ID = Pipe diameter, in.

You can convert pressure loss from psi to feet by:

$$(D_{H_2}) \cdot (2.31) \div \text{Specific Gravity} \tag{30.4}$$

"I've assumed an ordinary roughness factor inside the pipe in this calculation."

"But, how do I know the piping equivalent feet, before the plant piping is designed?" Howard asked.

"We don't know. As a guess, I use the actual distances between the pump and vessel, and double it," I answered.

"But you've guessed also at the roughness factor inside the pipe. I think that roughness is used to calculate the friction factor. If you've guessed at the friction factor, you've guessed at the suction piping losses. That doesn't seem professional! Engineers aren't supposed to guess," said Howard.

The Bhopal Factor

"You're right Howard. People die when we're not careful. You've heard about the disaster in Bhopal, India, in 1984?" I asked. "Over 10,000 people were killed."

"Was the cause of those people being killed…was that because of a pump not working…like it cavitated? Like it didn't have enough NPSH?" asked Howard.

"Well yes, but only indirectly. First, I'll have to tell you a bit more about cavitation. It's not all that simple."

Symptoms of Cavitation

"Howard, what have you been taught are signs of cavitation in a centrifugal pump?"

"The pump discharge pressure is erratic and low. Pump flow is also erratically low. Mainly, the pump will make a rattling sound, like shaking a bunch of bolts in a can.

Also, operating a pump in a cavitating mode will damage it. The impeller or case will be damaged? But, as long as the required NPSH is a little bit less than the available NPSH, cavitation will be totally avoided."

"Howard, most pumps that I find cavitating, do not have an erratic discharge pressure, unless the suction line or draw-off nozzle is partly restricted. Certainly, discharge pressure may be only 50%–70% of the performance curve head. But the discharge pressure is stable, and the pump runs quietly.

"I call this, 'marginal cavitation,' in the sense that if I raise the pump suction head by just a few feet (i.e., one or two psi), the discharge pressure may increase from 120 to 180 psig. In marginal cavitation, a pump appears to be running on an inferior performance curve. A bit more available NPSH, restores it to its proper performance curve.

"The problem is that even when a pump's calculated available NPSH exceeds the curve required NPSH by a few feet, fluid as it flows through the pump's impeller may still partly vaporize. The bubbles of vapor can damage the pump's mechanical seal, soft carbon face. This, with time, will cause the seal to leak," I added.

"I see. But how can that be prevented?" Howard asked.

"Well Howard, you have to design the pump suction for extra NPSH, beyond that minimum specified by the pump performance curve."

"But, Norm, how much extra NPSH is needed to make sure the seal isn't damaged due to cavitation?"

"The pump seal on their Bhopal pesticide plant leaked repeatedly. So, the pump was by-passed, and they used nitrogen pressure to push the toxic reactant into the pesticide reactor. But, in by-passing the pump, they also by-passed their feed tank high temperature alarm. So when they accidentally got water in the feed tank, it reacted with one of the pesticide chemicals and formed a deadly gas. But, without the high temperature alarm, the plant operators didn't realize there was a problem. If the pump had been designed with extra available NPSH, this wouldn't have happened."

"But is that the only reason for extra available NPSH when designing a new centrifugal pump?" Howard asked.

Starting NPSH Requirements

"There is a more fundamental reason for supplying available NPSH in excess of the manufacturer's NPSH tabulated on the pump performance curve. Let's take a look at Fig. 30.3. Initially, when an operator starts a centrifugal, motor driven pump, he 'primes' the pump by:

1. Fully opening the suction valve.
2. Partly opening the vent on the top of the pump case.
3. Opening the discharge valve just a little bit. Meaning to turn open (that is counterclockwise) the discharge valve about one-sixth of a full revolution.
4. Push the motor start button.
5. Then, open the discharge valve the rest of the way, as fast as possible, but as limited by not losing the pump *suction* pressure (P_2—Fig. 30.3)."

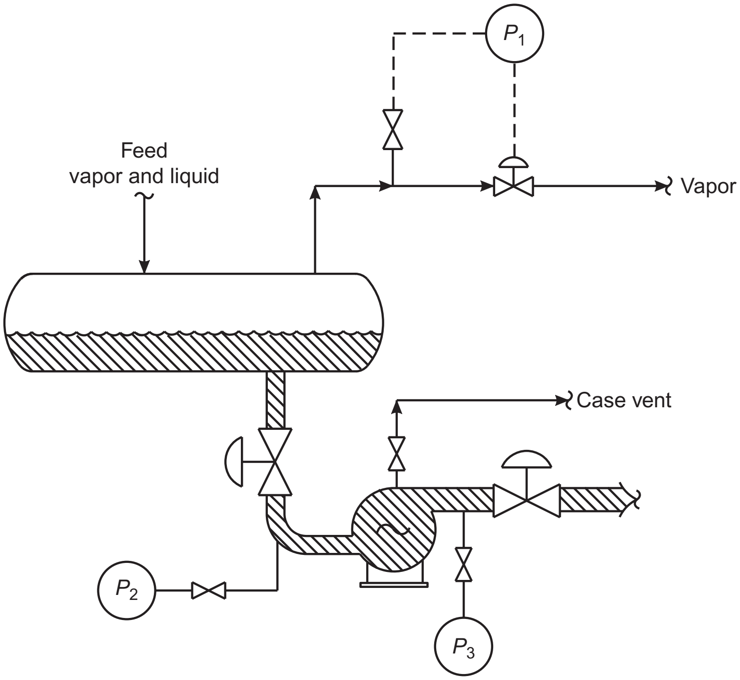

FIG. 30.3 A sudden increase in drum pressure temporarily increases available NPSH for start-up.

"But, wouldn't it be better to open the valve slowly?" asked Howard.

"No," I answered. "Running a centrifugal pump at a very low rate causes excessive vibrations, which is bad for the pump's bearings and seals."

"But I learned at University that it's also bad for the pump if it cavitates. So it seems to me that it's best to watch the discharge pressure. And, if the discharge at P_3 (Fig. 30.3) becomes erratically low, open up the discharge valve slower."

"You're right. If the pump cavitates, it will damage the pump's bearing, but especially the mechanical seal. Most of the pumps used in process plants are, 'Self Flushed Pumps.' That means the seal flush fluid comes from the discharge of the pump itself. If the pump cavitates, the seal flush flow stops. Then, the seal loses lubrication and cooling. But, better to watch the suction pressure. By the time discharge pressure slips down, the pump is already cavitating and seal damage is occurring. By watching the suction pressure, you can avoid cavitation and seal damage before the problem starts."

"Does that mean that there is always a pressure gauge located on pump suctions?"

"Unfortunately, no, Howard."

"Then how can the plant operators watch the suction pressure?"

"Process pumps almost always have a spare pump piped in parallel with the main pump. Watch the discharge pressure of the spare pump that is not in operation (with its suction valve open and discharge valve shut). This will be the same as the suction pressure of the pump that is running."

"But, why is it that pumps cavitate so often when they're being started, but not so much when they are running?"

"Pumps cavitate while being started because of their, '*STARTING NPSH REQUIRED*'," I explained.

"Is that the same as required NPSH? Or is that something different?"

"Look at Fig. 30.3. When a pump starts, stagnant liquid in the suction line has to be accelerated to maybe 3 ft per second. Here's the point:

"Centrifugal Pumps Push, They Do Not Suck."

Calculating Starting NPSH

Anyone who has commissioned a fixed speed, motor-driven centrifugal pump will have noted the need to increase the flow slowly. This is not a problem on a variable speed turbine drive. Then, the turbine speed is increased slowly.

The problem is the need to supply energy to accelerate the fluid in the line feeding the pump. If the suction line is short, this is a problem. For example, if I switch to the spare pump, 90% of the suction piping is already flowing. I can then start the pump with the discharge valve wide open.

However, if the suction line is long, then I have to provide energy to accelerate the mass of liquid in the suction line. This energy does not come from the pump. It comes from the static head of liquid in the suction line. That is, it comes from the available NPSH. To summarize, centrifugal pumps require three sorts of NPSH:

1. The NPSH as shown on the pump curve (Fig. 30.2).
2. Hydraulic losses in suction piping and nozzle exit losses.
3. Acceleration of liquid in the suction line.

I've called this last item, which is often larger than the first two, "The Starting NPSH Requirement." It's a function of:

- The length of the suction line.
- The speed at which the plant operator opens the pump discharge valve.

Increasing Process Flow Rate

"But," Howard asked, "suppose a pump is running close to its design available NPSH. Then, the operator decides to increase the flow by 20%. The required NPSH will increase a lot because the velocity in the suction line will also increase by 20%. Then the pump will cavitate, the seal flush will stop, and the seal will be damaged due to lack of lubrication."

"The engineer," I explained, "needs to take these factors into account when deciding on the elevation of vessels. Failing pump mechanical seals are expensive, but also dangerous."

Increasing Available Starting NPSH on a Crude Unite Preflash Drum

"First we'll increase the drum set point pressure by 10 psi. Next, open the discharge valve (see Fig. 30.3) by ¼ of a turn. Then start the pump, and open the discharge valve in a few minutes all the way.

"The suction pressure will increase by 10 psi. It would seem that the vapor pressure of the flashed crude will increase by the same 10 psi. According to Eq. (30.1), the increased vapor pressure of the flashed crude will off-set the higher suction pressure.

"However, at first, due to the 10 min residence time of liquid in the drum, the higher vapor pressure liquid, formed at the vapor-liquid interface, will not migrate into the pump suction for 10 min. If one can get the pump up and running quickly, a temporary surge of available NPSH will result.

"It's a start-up trick that I've used. Best to design the equipment properly, and allow for starting, as well as running NPSH requirements. Often, while operators are learning this trick, they repeatedly cavitate the pump, and damage the seal."

Practical Steps to Minimize Cavitation

"Howard, to summarize, these are some things to help protect mechanical seals from damage due to cavitation:

1. Place a pressure gauge with a red mark on the pump suction. Train the operators to open the pump discharge valve as fast as possible, consistent with the suction pressure not falling below this red mark.
2. Raise the suction drum pressure rapidly a few psi, just before starting a pump. Then reduce the pressure very slowly once the pump is operating.
3. During normal operations, increase flows really slowly.
4. Run near maximum safe drum level.
5. Spray cold water on pump case and suction line to subcool liquid.
6. Running two pumps in parallel requires less NPSH—but is often difficult to control.
7. Injecting a subcooled slip-stream into the pump suction will increase available NPSH.
8. A discharge spill-back to suction line, diminishes available NPSH, and increases the required NPSH.
9. For steam turbine driven pumps, increase turbine speed slowly.
10. Maintain pump suction screens and keep them clean."

Summary

Common process engineering design practice often fails to take into account the need to accelerate the liquid in the pump suction line. For shorter suction piping, this is of little consequence. Also, the problem of pump cavitation can be greatly mitigated by slowly opening the discharge valve. However, for most applications, supplying an extra amount of available NPSH makes it simpler, and easier, for centrifugal pumps to be started. And, of equal import, the extra NPSH will increase the pump's mechanical seal reliability and reduce the maintenance costs of the plant's centrifugal pumps. Likely, the incremental investment cost for increasing vessel elevation to supply this, "Starting NPSH Requirement," will be offset by seal maintenance savings.

Examples of NPSH Circulations

The following calculations were originally presented as a seminar for the Monroe Refinery in Pennsylvania by Elizabeth Lieberman.

Starting NPSH Requirement

The need to accelerate the fluid in the suction of a pump is called the starting NPSH Requirement (see Fig. 30.4). To calculate this starting NPSH requirement let's assume:

- Suction line = 100 ft
- Eight inch line
- Fluid = water @ 62 lbs/ft^3
- Initial velocity is zero
- Final velocity is 10 ft/s

Answer

- Mass in line is 2170 pounds (see Fig. 30.5)

$$\text{Change in momentum in line is } 10\,\text{ft/s times } 2170\ \text{lbs} = 22{,}170\ \frac{\text{ft lbs}}{\text{s}}(\text{m})$$

If the operator increases the flow rate uniformly over a period of 100 s, the change in momentum is as follows:

$$22{,}170\frac{\text{ft lbs}}{\text{s}} \div 100\ \text{s} = 222\frac{\text{ft lbs}}{\text{s}^2}(\text{m})$$

One pound mass equals one pass force times s^2/ft. Therefore, the force needed to accelerate the water in the pump's suction is as follows:

$$222\,\text{lbs}\,(\text{f})$$

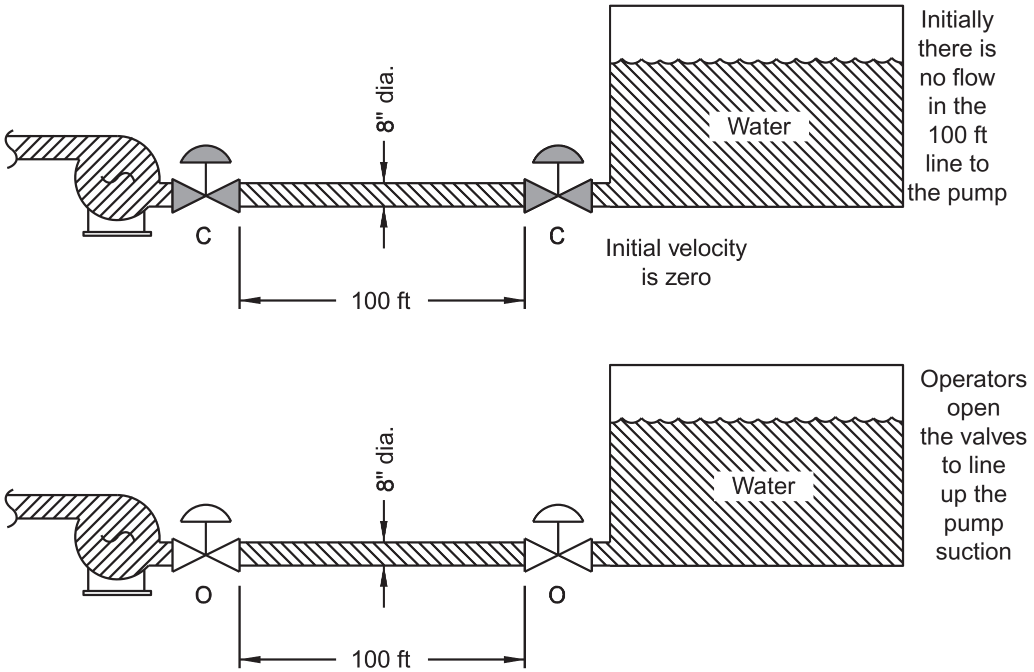

FIG. 30.4 Pumping water from a tank.

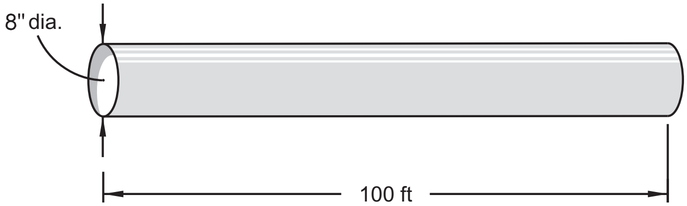

FIG. 30.5 Volume of 100 ft, 8″ diameter feed line.

The area of the 8 in. suction line is 50 sq. in. Therefore, the starting pressure is as follows:

$$222\,\text{lbs (f)} \div 50\,\text{in.}^2 = 4.4\,\text{psi}$$

or

$$4.4\,\text{psi times } 2.31 = 10.1\,\text{ft of water}$$

Fluid to be pumped is water at $\rho = 62$ (lb/ft^3).
Initial velocity is zero.

Pump suction line is 100 ft long and 8 in. diameter.
Final velocity is 10 ft/s.
Assume line is full of water when there is no flow.
First step is to calculate mass of water in line.

Volume of Line

$$\pi r^2 \times 100\,\text{ft} \qquad \text{where } r = \frac{4}{12}\text{ft}$$

$$\pi\frac{16}{144} \times 100$$

$$\rightarrow 0.1111\pi \times 100$$
$$\rightarrow 0.3490 \times 100 \qquad \rightarrow 34.9\,\text{ft}^3$$

If the 100 ft of 8 in. diameter pipe has a volume of
$34.9\,\text{ft}^3$
Then when filled with water at $\rho \equiv 62$ (lb/ft^3)
The mass of water in the line is:

$$62 \times 34.9$$
$$= 2164.16\ (\text{or approximately } 2170\ (\text{lb}))$$

Hence the weight of water to be accelerated in the line from 0 to 10 ft/s is 2164 (lb).

- Suppose the operator increases the flow rate uniformly from 0 to 10 ft/s over a period of 100 s.

  The acceleration in the line would then be

$$\frac{10\ \text{ft/s} - 0\,\text{ft/s}}{100\ \text{s}}$$

$$\rightarrow \frac{10}{100}\ \frac{\text{ft}}{\text{s}} \cdot \frac{1}{\text{s}}$$

$$\rightarrow 0.1\,\text{ft/s}^2$$

∴ The force required to accelerate the liquid in the line from 0 to 10 ft/s is

$$2164\ (\text{lb}) \times 0.1\ \text{ft/s}^2 \qquad \text{Force} = \text{mass} \times \text{acceleration}$$
$$\rightarrow 216.4\ (\text{poundal} \cdot \text{lbwt})$$

- The internal cross-sectional area of the pipe with an 8 in. ID is

$$\pi(4/12)^2$$
$$\rightarrow \pi(16/144)$$
$$\rightarrow 0.349\,\text{ft}^2 \text{as seen earlier.}$$

- But $0.349\,\text{ft}^2$

$$\begin{aligned}&\rightarrow 0.349 \times 144\,\text{in.}^2\\ &\rightarrow 50.25\,\text{in.}^2\\ &\approx 50\,\text{in.}^2\end{aligned}$$

Thus the required starting NPSH will be given by:

Force required to accelerate the liquid in the line from 0 to 10 ft/s divided by the open cross-sectional area of the pipe.

$$\frac{216.4}{50}\text{poundal}\cdot\text{lbwt} \qquad \frac{[\text{M}][\text{L}][\text{T}]^{-2}}{[\text{L}]^2}$$
$$\rightarrow 4.328\,\text{psi} \quad [\text{M}][\text{L}]^{-1}[\text{T}]^{-2}$$

Note 2.31 ft of water ≡ 1 psi

∴ The required starting NPSH = 4.328 psi × 2.31 (ft of water)/psi

→ 9.997 ft of water

Required Starting NPSH → 10 ft of water, to avoid cavitation on start-up.

Suction Pressure Control

I was working in Colombia in January of 2018, starting up an oil field vacuum, water degassing tower. Waste water from a crude tower production well was saturated with corrosive CO_2. I had designed a packed tower 10′ ID and 30′ T-T to flash off the CO_2 at a vacuum of 25″ Hg (125 mm Hg).

The bottom half of the vessel was sized for a water hold-time of 5 min, to protect the bottom water pumps from cavitation due to low level. And, to prevent the water level from backing up into the packed bed, which could cause flooding.

As was my usual practice, I indicated on my design vessel sketch, the bottom level tap 6″ above the vessel's bottom tangent line. I showed that the height of the vessel skirt was located 13′ above the elevation of the center line of the bottoms pumps. The objective was to provide the pumps' required NPSH, which was 12′, at a minimum liquid level in the tower.

I've designed hundreds of process vessels using this criterion. But in this case, something had gone wrong. The bottom level tap was located 10 ft above the tangent line. To control the bottom level, the Colombian operators had to hold the water level perilously close to the packed bed. Too high a level would cause the packed bed to flood.

As the lower level tap was so badly mislocated, I decided to abandon the use of level control entirely and rely on "Suction Pressure Control" for the bottom's pump.

My client (Ecopetrol) always designs pumps with a pressure transmitter at the pump suction, to alert unit operators about impending pump cavitation due to loss of liquid level. I connected this pressure transmitter output to the Level Control Valve on the pump's discharge line.

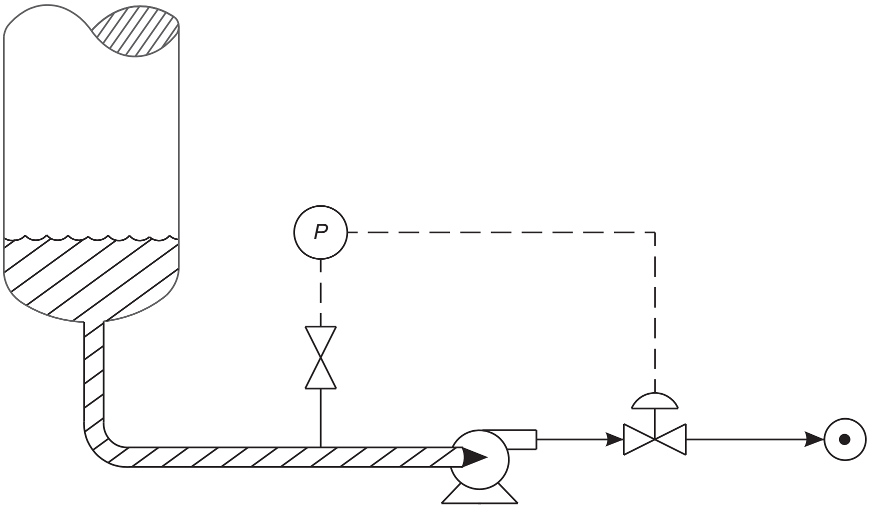

FIG. 30.6 Pump suction pressure control eliminates discharge level control valve and level taps on pressure vessel.

As the required pump NPSH was 12 ft, I decided to set the pump suction pressure to provide six extra feet of extra head. That is, 18 ft of head. One inch of Hg is equal to about 1.2 ft of warm water. Therefore:

$$18' \div 1.2 = 15'' \text{ Hg} \tag{30.5}$$

The vacuum degassing tower operated at 25″ Hg vacuum, my set-point pressure would be:

$$25'' \text{ Hg} - 15'' \text{ Hg} = 10'' \text{ Hg vacuum} \tag{30.6}$$

Actually, I lied about the above calculation. What I really did was to manually lower the suction pressure until the pump cavitated a little. Then, I set the suction pressure about 5″ Hg higher. All done without reference to the actual tower water level, which was below the lower level tap, and hence not visible.

I checked the ambient temperature –102°F. Not a cloud in the sky. I recall thinking, “I’m getting too old for this stuff. But, I do love it.”

Does this mean level control for pump’s suction is an engineering error? Yes! Use suction pressure control—at least for vacuum tower service (see Fig. 30.6).

In addition to the oppressive heat, I ran into another problem on this job. The pump’s suction screen was located downstream of the sensing point for the pressure transmitter. So, when the suction screen plugged during that night, the pump began to cavitate due to loss of suction pressure.

The next morning, I revised the suction screen with larger openings. The lesson is that when retrofitting a pump to suction pressure control, the suction pressure sensing point must be located downstream of the suction screen and never upstream.

Pump Suction Screens

I was starting up the new well water vacuum deaerator I had designed for the oil field drilling operation in Colombia. The bottoms pumps were rated for a low head and 60,000 BSD. I had thought the recycled well water was clean. But, this was a mistake. The pump's suction screen located in the 12″ intake line was a cone covered with a fine mesh. It became over 90% plugged after only 12 h of operation.

But why do we have pump suction screens? Are they really necessary?

Maybe yes. Maybe no. It all depends.

First, each centrifugal pump has a "Maximum Free Passage" specification. It's the clearance between the impeller and the pump case. That is typically a few tenths of an inch. I would specify that the openings of the suction screen at perhaps half of this value or a bit less.

A more critical value is the maximum particle size that the pump's mechanical seal can tolerate. The tolerable particle size is likely to be very small. This criterion only applies for "self-flushed" pumps, which means that the seal flush fluid is a tiny slip-stream drawn from the pump discharge line and then through a restriction orifice.

My plan was to change the pump suction screen, by replacing the fine mesh with a cone with quarter inch holes. This ¼″ hole being smaller than the pump's maximum free passage shown on the pump data sheet.

To protect the pump's double mechanical seal (an expensive $60,000 (United States) item) from having its seal faces scratched with particulates, I disconnected the discharge water flow to the seal. Then, I connected to the half inch fitting, via stainless tubing, potable water which was available at the control room, which appeared to be clean.

Incidentally, this problem could have been avoided by using a "Packed Pump" rather than the more expensive Double-Mechanical Seal for a water pumping service. Packed pumps will unavoidably suffer from leaks along the shaft, and thus are not suitable for hydrocarbon or chemical plant service. However, when just pumping clean water, the extra expense and the extra maintenance complexity of a mechanical seal serve no purpose.

After several months, there has been no report of an increase in the pump's suction screen pressure drop, so I suppose all's well that ends well.

31

How Compressors Work

The people I work with in refineries are most often honest, ethical, dedicated, and confused. It is the complexity of the process equipment that is the origin of the confusion. And, the most confusing sort of process equipment is compressors.

There are three sorts of compressors that I have worked with:

- Centrifugal
- Axial
- Reciprocating

To determine your own CI (i.e., Confusion Index), please take the following test:

- Question #1—A motor-driven centrifugal compressor which is running at a constant speed, at a constant volume of gas flow and temperature at the suction, and at a constant suction and discharge pressure. The molecular weight of the gas now increases from 20 to 40. What happens to the motor's amp load?
- Answer #1—There is no answer. Both the suction and discharge pressure (i.e., the compression ratio) cannot be kept the same. Why? Because the machine is developing a fixed amount of head or feet. The pressure ratio is a function of the molecular weight of the gas. So, if the molecular weight goes up, so must the compression ratio. Either the suction pressure is drawn down or the discharge pressure is pushed up, or both.
- Question #2—Same as above, except now I will permit the suction pressure to drop due to the larger molecular weight. The discharge pressure is still kept constant. What now happens to the motor amps?
- Answer #2—The amps go up. Not because the heavier, higher molecular weight gas is harder to compress. Just the opposite is true. Fifty-eight molecular weight butane requires 15% less amps to compress than 16 molecular weight methane, for a certain volume of gas. The amps on the motor increases, because the compression ratio (discharge pressure divided by the suction pressure) has increased, even though the compressibility factor (Z) has dropped by 15% for butane, as compared to methane.
- Question #3—A reciprocating compressor, running at a fixed speed with a motor driver, handling a fixed volume of gas, and with both the suction and discharge pressures constant, is compressing methane (16 MW). It now switches to compressing propane (44 MW). What happens to the motor amps?
- Answer #3—The amp load goes down by about 10%. Why? Because "Z" the compressibility factor of the gas goes down by 10%.
- Question #4—For a variable speed centrifugal compressor, if I need to keep both the suction and discharge pressure constant, and the suction volume and suction temperature are constant, and the molecular weight of the gas is increasing, should I:

https://doi.org/10.1016/B978-0-12-816161-6.00031-X

  A. Slow down the compressor.
  B. Speed up the compressor.
  C. Varying the speed has no effect.

- Answer #4—Slowing the compressor will off-set the effect on greater gas molecular weight or gas density. This can be done with a variable frequency AC motor driver (i.e., VFD).

If you have missed all four questions, your "Confusion Index" is 100%. Congratulations. Best to read on.

Centrifugal vs Axial Compressors

The only Axial Compressor I have worked with was a giant Air Compressor at the Marathon FCU in Garyville, Louisiana. Along the shaft there were dozens of spinning blades. These blades accelerated the air.

Along the interior of the compressor case, there were also dozens of blades that were stationary. These blades converted the kinetic energy of the air, derived from the spinning shaft, into feet of head. To convert from feet of head to pressure increase:

$$\text{Increase in air pressure (in psi)} = K \cdot \text{Hp} \tag{31.1}$$

where:

- K = a coefficient proportional to the air density that converts feet of polytropic head to pressure
- Hp = feet of polytropic head

The important thing to understand about both axial and centrifugal compressors is they both develop the same feet of head, regardless of the density of the gas being compressed. If the air gets hotter, because it's summer time in Louisiana, then the discharge pressure from the Axial Air Compressor drops off, but the feet of head developed remains constant. Thus, if the air blower discharge pressure was too high in the winter, we could:

- Throttle on the suction.
- Slow the blower's speed.
- Get the axial air compressor's blades really dirty, as happened at the Marathon plant in Garyville.

Centrifugal Compressors

Fig. 31.1 shows a picture of a centrifugal compressor's rotating assembly. Instead, of blades spinning around a shaft, there are wheels. The wheels impart kinetic energy to the gas from the rotational energy of the wheels. As the gas escapes from each wheel, it flows into a circular stationary chamber called the "Stator." Inside the stator, the gas slows. And, the kinetic energy imparted to the gas from the wheel is converted into feet of head. To convert

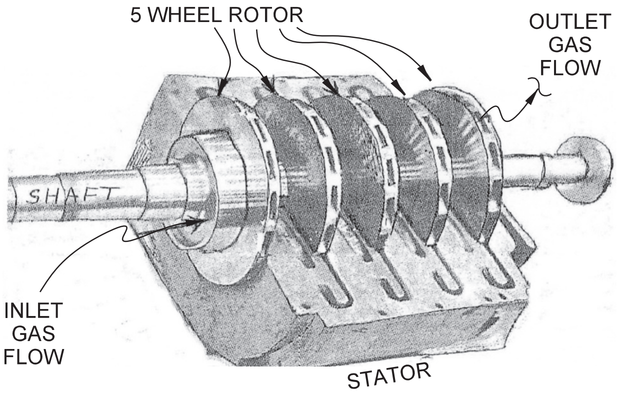

FIG. 31.1 $\Delta P \sim \rho v$ Hp. *Hp*, polytropic head.

from feet of head to pressure, we use Eq. (31.1). My point is that the feet of head developed does not change, regardless of the molecular weight or density of the gas being compressed.

To generate more feet of head to increase the compressor's discharge pressure, we could:

- Make the compressor spin faster.
- Increase the number of wheels.
- Increase the diameter of the wheels.

Alternately, we could generate more compressor discharge pressure by:

- Increasing suction pressure.
- Reducing suction temperature.
- Increasing gas molecular weight.
- Or, reducing the flow of gas.

Centrifugal Compressor Performance Curve

Fig. 31.2 is a centrifugal compressor performance curve. Like a centrifugal pump, when we throttle at the discharge of the compressor, the flow is reduced and the feet of head (i.e., the discharge pressure) increases. If we throttle too much on the discharge of the compressor, we will reach the "surge" point.

Surge is very bad. The gas flow stops and reverses direction. The shaft and the spinning wheels slide backwards. The end of the shaft slams up against its thrust bearing, which is deformed a little with each surge. And, the spinning wheels get closer and closer to the stationary parts of the pump case.

On February 8, 2016, I had a Cardiac Arrest. No pulse, no breathing, no heartbeat. To use the appropriate medical term, I was dead. Luckily, my wife, Liz, called 911 and within 8 min, I had been electro-shocked back to life.

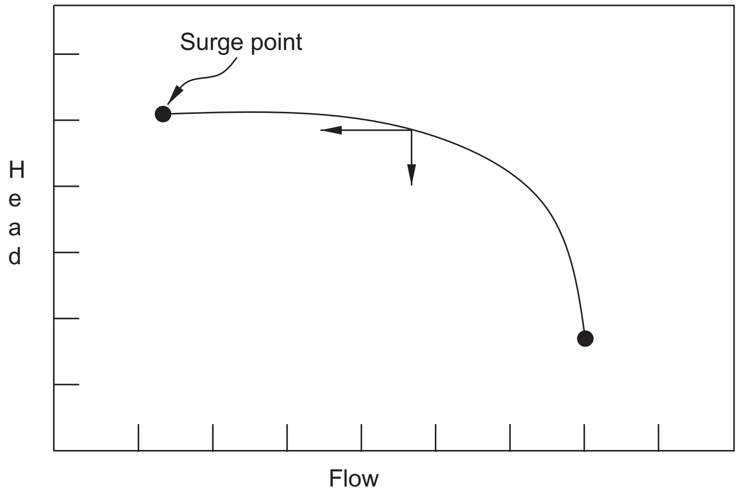

FIG. 31.2 A centrifugal pump performance curve.

Same with surge. You had better do something fast, before the compressor wheels touch the stationary elements and destroy the machine. You too may only have 8 min.

Stopping Surge

I became an expert on this subject on my Alkylation Unit refrigerant compressor in Texas City in August 1975. My options were as follows:

1. Decrease discharge pressure.
2. Increase suction pressure.
3. Decrease suction temperature.
4. Increase the molecular weight of the process gas.
5. Increase flow.

Whatever can be done to increase the density of the vapor at the suction of the machine should be done. Whatever can be done to decrease the discharge pressure should be done. In Texas City, I immediately opened the discharge-to-suction spill-back valve (see Fig. 31.3) to increase flow.

But, as there was no cooler on the spill-back line, the suction temperature became hotter and hotter, which made the surging worse, as the inlet vapor density dropped.

Frank Citek, my East Plant Manager, provided useful guidance. “Lieberman! Stop that damn surge! If you wreak that machine, you’re fired!”

So, I shut down the compressor and also the world’s largest (26,000 BSD) alkylation unit (½% of all gasoline produced in the United States). In Chapter 32, I discuss a surging incident at my Texas City Alkylation Unit in greater detail.

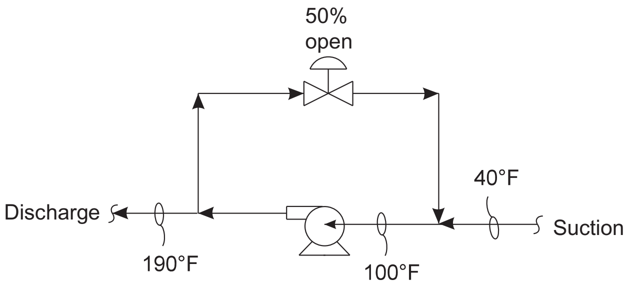

FIG. 31.3 Opening spill-back valve too much causes compressor to surge.

Benefit Suction Throttling of Centrifugal Compressors

Let's assume I have a fixed speed motor-driven centrifugal compressor as shown in Fig. 31.4. I must keep my pressure at *P*-1 constant, to keep the pressure constant in the reflux drum of the FCU fractionator. But as the molecular weight of the wet gas has increased from 30 to 33, gas density will also increase by 10%. Since the discharge pressure is constant and also the feet of head developed by the compressor is constant, and the flow of gas is constant, the suction pressure (*P*-1) is drawn down which I can't tolerate.

What to do? I have two choices:

- #1—Open up the spill-back valve shown in Fig. 31.4. This will cause the compressor to produce less feet of head (Fig. 31.2). The suction pressure will increase. Which is what I want to happen?

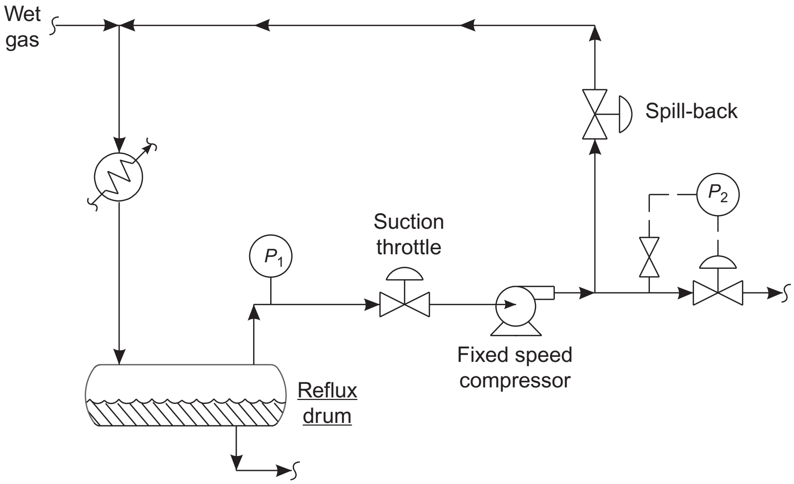

FIG. 31.4 Spill-back vs suction throttle control of *P*-1 pressure.

- #2—Close-off on the compressor suction throttle valve to hold back-pressure on the reflux drum (*P*-1). This does ***NOT*** decrease the compressor suction pressure, but only holds back-pressure on the reflux drum.

Number two is the preferred option. It's rather complicated. But that's the nature of Process Equipment, and why we need dedicated men and women operating plants.

Shape of Compressor Curve

Looking at Fig. 31.2—the horizontal axis is flow; the vertical axis is head. We normally design centrifugal compressors to operate on the flat portion of the curve, where flow is going to drop quickly and head will increase slowly, as we throttle the compressor discharge.

To understand how to operate a fixed speed, centrifugal compressor is a critical and complex undertaking. If we throttle on the compressor discharge, two things are going to happen:

- Thing 1—The head will go up by about 10%.
- Thing 2—The flow will go down by about 20%.

Work, or motor amps, or horsepower are proportional to the feet of head.

$$\text{Work proportional to Head} \cdot \text{Flow}$$

Throttling the compressor discharge causes the pressure at *P*-1 (Fig. 31.4) to increase, as the pressure drop across the discharge valve increases. But, to keep the pressure at *P*-1 at our set-point pressure, the spill-back valve, which is controlling the pressure at *P*-1 at a constant pressure, has to close.

Let me add this to help you understand. At equilibrium (i.e., steady state), the compressor suction pressure is not affected by opening or closing the suction throttle valve.

If you dam up a river at equilibrium, at steady state, once water overflows the dam at its normal flow rate, the height of the river upstream of the dam increases, but the level of the river downstream of the dam is not affected by the dam.

In our case, the effect of suction throttling causes the spill-back valve to close. Since work is proportional to flow multiplied by head, the motor amperage goes down, which is good.

The Antisurge Valve

If you will look at the shape of the compressor curve, you will see that suction throttling to save motor amps pushes the compressor closer to surge. Which is fine, as long as the compressor doesn't surge and self-destruct.

To protect against destructive surge, we program the spill-back valve shown in Fig. 31.4 not to close too much; to maintain a total minimum flow to the compressor.

That's why the spill-back valve is normally referred to as the antisurge valve.

Wouldn't it be simpler to avoid all this complexity? Wouldn't it be best not to have the suction throttle valve and the spill-back valve? Would it not be best just to vary the compressor speed to maintain the pressure at *P*-1 constant? Certainly, yes!

That is why we have "Variable Frequency Drives" and steam turbines. I imagine in 2059, some young engineer reading my book will laugh at us. Laugh at the suction throttle valve and the spill-back. But for now, I'll leave you with a parting thought.

Closing the suction valve saves energy, but pushes the compressor closer to surge. It's like getting married. You give up part of your freedom to avoid that lonely feeling when you come home to an empty apartment.

Reciprocating Compressors

Recips are pretty easy to understand. I've shown a recip in Fig. 31.5, along with its performance curve. The curve is a plot of pressure vs volume. The volume is not the flow through the compressor. It's the volume of gas inside the compressor cylinder, between the piston head and the cylinder head. What confused me when I studied the CARNOT CYCLE in school was that I thought the horizontal axis on the curve was the volume of flow through the compressor. Not true! It's the volume of gas inside the piston which changes as the piston moves.

Recips are double acting machines. The piston compresses gas as it moves in both directions. I will ignore this. Let's just pretend gas is only compressed as the piston moves toward the cylinder head. There are four steps involved:

- *Step 1—Compression*—The piston starts off from as far away as possible from the end of the cylinder. This position is called BOTTOM DEAD CENTER. The piston moves toward the cylinder head with both the intake and discharge valves shut. The gas inside the cylinder, between the cylinder head and piston is compressed.
- *Step 2—Discharge*—The pressure inside the cylinder head has risen to the discharge line pressure, plus a little extra. The pressure inside the cylinder, now being bigger than the pressure in the discharge line, pushes gas into the discharge line through the discharge valve. The suction valve is still shut. The piston continues its travel until it reaches TOP DEAD CENTER. The piston stops when it has covered about 1/3 of the area of the discharge valve. If it covered the entire area of the discharge valve port, gas could not be pushed into the discharge line.

  The gas volume trapped between the piston at top dead center and the cylinder head is called the "Starting Volumetric Clearance." Not that the intake valve has yet to open.
- *Step 3—Expansion*—The piston now retreats toward BOTTOM DEAD CENTER. The gas trapped in the Starting Volumetric Clearance expands and the pressure in the cylinder declines. Note that both the intake and discharge valves are still both closed. The gas expands and the pressure drops, until the pressure in the cylinder drops a little below the pressure in the intake line.

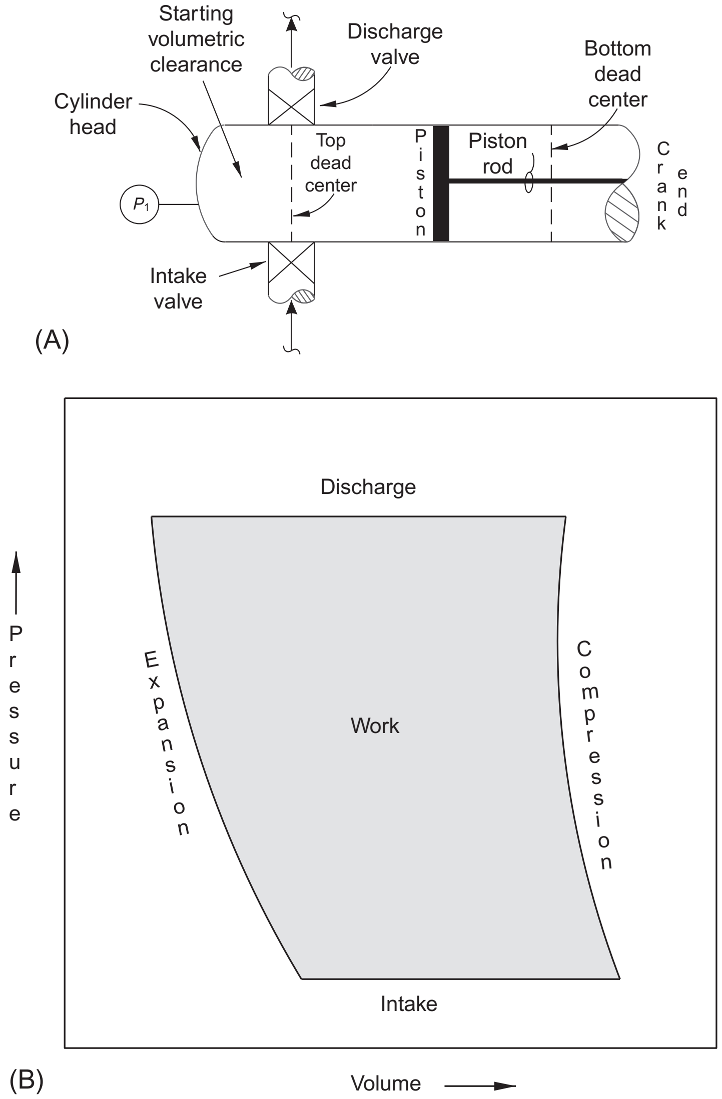

FIG. 31.5 (A) Reciprocating compressor. (B) Carnot cycle curve for reciprocating compressor. Volume of the gas between the piston and cylinder head. Pressure measured at *P*-1.

- *Step 4—Intake*—The gas pressure in the intake line now forces open the intake valve. New gas is sucked into the cylinder from the intake line until the piston returns to Bottom Dead Center.

Carnot Cycle

The picture in Fig. 31.5 is the Carnot Cycle. The area inside the Pressure vs Volume curve I have shown is variously called:

- PV compression work
- Ideal compression work
- Carnot cycle work

By measuring the pressure continuously in the cylinder, and the volume of gas inside the cylinder, based on the position of the driver's crank shaft, it is possible to actually plot the curve shown in Fig. 31.5. This plot is called either:

- Indicator Card

  Or
- Beta Scan

Temperature Profile

Regardless of how gas is compressed, it heats up. The bigger the compression ratio (i.e., Discharge Pressure ÷ Suction Pressure) and the more inefficient the compressor, the more gas heats up. Compressing air from 30 to 150 psig might heat the air from 60°F to 220°F. This would be typical. A normal compression ratio for a single piston might be three or four to one.

Rod Loading

Piston rods push and pull on the piston. The load on the rod is:

$$\text{Piston area}\,(\text{Discharge pressure} - \text{Suction pressure})$$

where:

- Rod loading = pounds
- Piston area = square inches
- Pressures = psi

The temperature rise across the machine then is a direct measure of this rod loading. As the temperature rise increases, the rod loading may exceed the rod's strength and cause the rod to break.

Of course, an increase in the reciprocating compressor discharge temperature, because the intake temperature has increased due to high ambient or cooling water temperature on the upstream cooler, is not a measure of an increase on the rod loading. Lack of understanding this principle cost one of my clients $1,200,000 a year between 1939 and 2017.

A Final Word

I've written this manuscript to tell a story about how it's important to understand how process equipment works. I was consulting recently for a plant whose bottleneck was performance of reciprocating compressors. Their problem was excessive discharge temperatures.

The inlet temperature to the compressor would increase from 70°F to 100°F during the day. This activated the high temperature alarm, set at 250°F. The operators would then reduce the capacity of the compressors, which raised the suction pressure. This reduced the compression ratio which reduced production capacity.

But, did higher gas inlet temperature have anything to do with ROD LOADING? My client should have calculated rod loading based on suction and discharge pressures and piston area. Not on the temperature at the compressor discharge.

Safety Note

A centrifugal compressor surging may become a safety hazard in a matter of minutes or in a matter of hours, depending on:

- Older machines built in the 1950s are more tolerant to surge.
- Slower speed machines (4000–5000 rpm) are more tolerant to surge.
- Machines with fewer wheels (3 or 4) are more tolerant to surge.
- A check valve on the compressor discharge dampens down the force of the surge.

A variable speed machine may often be brought out of surge by increasing its speed. That is, by developing more polytropic head.

32

Why Centrifugal Compressors Surge

Vincent Cappidono emigrated from Sicily as a small child, as part of the creator's plan to make my path in life difficult. Cappy was the Chief Operator at the No. 2 Alkylation Unit at the giant American Oil Refinery in Texas City. In 1974, when I was 32 years old, I was placed in charge of No. 2 Alky which, at 24,000 BSD product rate, was the world's largest sulfuric acid alkylation unit.

"Cappy," I instructed one day, 4 months after becoming the unit superintendent, "our feed charge tank level is really low. We'll have to cut the C_4 olefin feed back to minimum."

No response and no action followed.

"Sorry Cappy. Did you hear what I just said?"

"Yeah! I heard you just fine, Mr. Lieberman."

Twenty minutes passed without Vincent Cappidono stirring from behind the control room kitchen table.

"Look Cappy, we got to…!"

"Okay, okay, Lieberman. Bobby," Cappy called out to the young consul operator, "cut C_4's back to 9,000, and the Iso-butane make-up back a couple thousand barrels too." And then Cappy added, with a trace of tension in his voice, "Bobby, you gotta keep your eye on that refrigeration compressor discharge-to-suction recycle valve. Don't let it open up too much."

Cappy glanced nervously at the wall clock, "Still two hours to shift change Bobby."

"Yeah Mr. Cappidono, but that refrigeration compressor spill-back valve is already starting to open. And the compressor suction pressure has already dropped by a whole pound. Maybe I can put a thousand barrels of olefin feed back in for a while? Let 'B' shift handle it?"

"No Bobby. You just heard what Mr. Lieberman told us? Got to do what the boss says! Ain't that right?"

What had occasioned all this hostility and resistance? It wasn't like Bobby and Cappy were getting a share of the sales revenue of the alkylate product. But, I was about to find out!

"There she goes. You hear that?" Cappy said excitedly. "Now we're in for it." I could see the tension in his face.

A deep, loud, groaning sound, lasting several seconds, followed by a powerful bang, coming from the direction of our giant 5000 horsepower isobutane recycle refrigeration compressor (see Fig. 32.1), resonated through the control room door.

"What's that sound?" I asked. "What's wrong? What's happening?"

Understanding Process Equipment for Operators and Engineers. https://doi.org/10.1016/B978-0-12-816161-6.00032-1

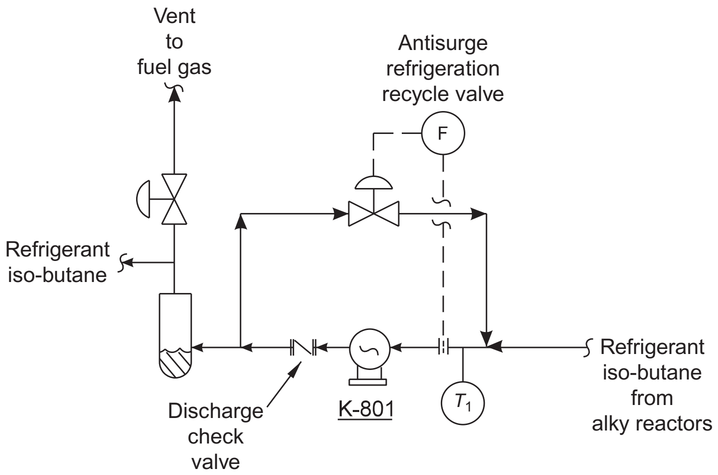

FIG. 32.1 Opening antisurge valve too far overheats compressor suction and causes compressor to surge.

"The refrig compressor is surging. We shouldn't cut that much feed out that-a-way!" observed Chief Operator Vincent Cappidono.

"What you want me to do now, Mr. Cappidono?" Bobby asked tensely. "The refrigerant recycle valve's going crazy."

"Oh, I guess that Mr. Lieberman's gonna direct us operators, Bobby. He being the Operations Superintendent and all."

The intermittent groaning roar, each followed by a vicious bang, continued unabated. I had heard during my previous 10-year career with American Oil, as a Process Design Engineer, that surge would eventually lead to a catastrophic failure of a centrifugal compressor. Eventually, a piece of a rotor wheel would break off, blast through the compressor case, and quite likely, hit me in the head. Most certainly, the destruction of K-801, the alkylation unit isobutane recycle refrigeration compressor, would be for me, a career-ending event.

Meeting Girls on Airplanes

Surge is similar to Aerodynamic Stall. Let me explain.

Let's say you are sitting on a plane next to an attractive woman. The first step is to say, "Hi! I'm Donald." And she will say, "Hi! I'm Linda."

Then say, "Linda, do you fly very often?"

And Linda will say, "Well actually, I do."

Then say, "Linda, have you ever wondered what makes planes fly?"

Linda will likely respond, "Well no Donald, I never thought about that."

I then suggest you draw a picture of the cross-section of a wing (see Fig. 32.2).

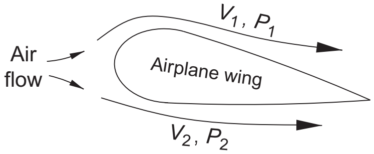

FIG. 32.2 Lift is created by converting air pressure to velocity.

Aerodynamic Stall and Surge

"Linda. Due to the cross-section of the wing and more importantly, the angle of attack (i.e., the angle of the wing as it is pushed through the air by the plane's engines), the velocity of the air (V_1) above the wing, is greater than the velocity of air beneath the wing (V_2). That is, the air accelerates above the wing, to a greater extent than beneath the wing.

The energy to accelerate the air, does not come from the plane's engines. It comes from the air itself. To be more precise, it comes from the barometric pressure of the air. Since the acceleration of the air is greater across the top of the wing (V_1), than below the bottom of the wing (V_2), then the pressure at P-1, will be less than the pressure at P-2.

If we take this differential pressure ($P_2 - P_1$), times the area of the wings (A):

$$(P_2 - P_1) \cdot (A) = \text{Lift}$$

That is aerodynamic lift. As the plane flies slower, this lift decreases. Also, less dense air, at higher altitudes, also diminishes lift. When the lift is less than the weight of the plane, the plane will fall out of the sky, crash and burn. That's called "Aerodynamic Stall.""

At this point, Linda will pretend to fall asleep unless she is also a process engineer. So this approach does have a certain—if limited—potential for a successful outcome.

Causes of Centrifugal Compressor Surge

When a centrifugal compressor surges, unlike the plane, it continues to spin. The gas flow first stops, and then flows *BACKWARDS* through the rotor (see Fig. 32.3).

The spinning wheels on the rotor's shaft do not so much increase the pressure of the gas, but mainly accelerate the gas. Then as the gas slows down in the stationary elements, which are part of the compressor case, the velocity of the gas provided by the spinning wheels is converted to *FEET OF HEAD*.

To convert feet of head (Hp) into differential pressure (DP) (i.e., discharge minus suction pressure), we multiply "Hp," feet of head, times the gas density (D_V):

$$\text{DP} = D_V \cdot \text{Hp}$$

The feet of head is proportional to the:

- Speed of the machine.
- The number of wheels.

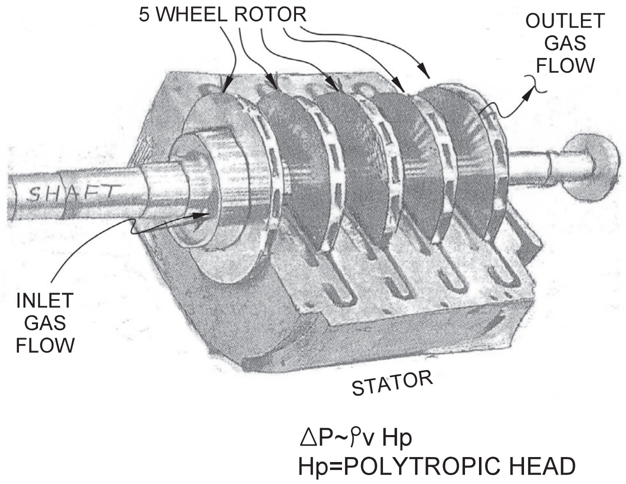

FIG. 32.3 Compressor fire stage rotor.

- The diameter of the wheels.

The density of the gas is proportional to the gas:

- Molecular weight.
- Compressibility factor.
- Pressure.
- Absolute temperature (°R or °K).

Factors that promote surge are as follows:

- High discharge pressure.
- Low suction pressure.
- Low molecular weight.
- High suction temperature.
- Low flow.

Polytropic Head

Hp denotes polytropic head which is a function of the compressor's wheels and stages. As we are not determining an actual compression ratio, but only considering directional effects, I have not differentiated between head and polytropic head. The head developed in a constant speed, centrifugal compressor, at a fixed volumetric flow rate of gas, is constant, regardless of the changes in the gas molecular weight or temperature.

If the vapor flow rate is reduced, the minimum flow antisurge recycle valve (see Fig. 32.1) will open to maintain the suction flow above the surge point flow. But the recycle flow will increase the temperature at the suction of the compressor (*T*-1), if it is not first cooled. This will make the situation worse by reducing gas density. Similar to Donald forcing his attention on Linda, when she has pretended to fall asleep.

Effect of Surge

If the differential pressure developed by the compressor (DP) is less than the pressure difference between the discharge and suction (Fig. 32.1), then the gas flow will stop and begin to flow backwards through the rotor's wheels. That's what makes that "surging" sound.

The reversed direction of the gas flow will forcefully push back the rotor. The shaft then impacts the thrust bearing (that constrains the axial movement of the rotor's shaft). Therefore, each surge slightly deforms the thrust bearing. This causes the spinning wheels to get closer to the stationary parts of the compressor case (Fig. 32.3). When a spinning wheel touches a stationary part, a piece of the wheel will break loose, pierce the compressor case, and possibly hit me in the head.

Function of the Discharge Check Valve

The loud, banging sound, sometimes heard in conjunction with surge, is very distressful but not bad. At my Alky Unit in Texas City, this banging sound was the compressor discharge check valve slamming shut (Fig. 32.1) as a consequence of the reversed gas flow. This dampened the flow of the reversing vapors and also the force of the surge pushing back against the rotor. I suppose then, that such a check valve should be included in the design of all centrifugal compressors.

Preventing Surge

Typically, a flow transmitter is located on the suction of a compressor to assure that gas flow is high enough to stay above the low flow minimum limit. If the flow falls below this limit, the discharge-to-suction "antisurge" valve will open. However, if the discharge pressure is above design, or if the gas molecular weight is below design, or if the suction temperature is above design, or if the suction pressure is below design the machine may surge anyway. This may happen even if it is operating above its minimum design gas flow to prevent surge.

I used to think that for a variable speed compressor, such as the three air blowers on the sulfur recovery unit in Texas City, speeding up the steam turbine drive would stop surging. Often it would. But, sometimes it caused the air blower to surge worse. I checked the family of performance curves and there were areas on the curves where a higher speed increased the minimum air flow needed to avoid surge. One thing I learned for certain about these air blowers. Turning on an extra blower when not needed was a good way to get all three air blowers to begin surging.

Rotors that are fouled will have an increased tendency to surge, as they will develop less polytropic head than a clean rotor. Spraying a naphtha mist into the suction flow (typically one weight percent) retards fouling, especially in hydrogen recycle gas compressors.

Surge is more destructive for machines that:

1. Run at higher speeds—a low speed being 3000–4000 rpm.
2. That have multiple wheels on the same shaft, that is, more than three or four wheels.
3. That operates with low molecular weight gas, and hence most likely require a high polytropic head.
4. Is of relatively recent vintage—older compressors, built before the 1960s, were of a more rugged design.
5. Subject to rotor fouling deposits.

My Alky Unit in Texas City

In spite of Cappy and Bobby's doubts, I knew exactly how to handle the surging problem of K-801, my isobutane recycle refrigeration centrifugal compressor. I phoned Henry Zipperain, the old shift supervisor on "B" shift.

"Hey Zip, what you doing?"

"I'm helping my old lady with the grandchildren, Norm. What's up?"

"Well Zip, would you mind coming out to the plant a bit early?"

"Like when, Norm?"

"Like now!"

Zip arrived 20 min later. Just in time. The plant manager, Mr. Durland, had heard the surging machine, and decided to investigate in person. "Lieberman," he screamed above the roar of the surge, "do something! Just don't stand there. You're going to destroy the compressor!"

"Yes sir. I'm on it."

Old Zip first opened up the vent to fuel gas from the compressor discharge refrigerant receiver, which lowered the compressor discharge (Fig. 32.1). Next, he opened the by-passes on the feed coolers (for both C_4 olefins and isobutane makeup) to increase the refrigerant isobutane vapor flow to the compressor suction.

Also, he then pinched back on the compressor discharge-to-suction recycle gas valve with the comment, "I been telling you, Mr. Norm, we need a cooler on this recycle." This step also increased the compressor suction pressure by about 1 psi.

Zip next raised the isobutane recycle depropanizer reboiler outlet by 5°F. "Norm, this will heavy-up the iso-recycle a bit, by reducing its propane content after a while."

Finally, Zip increased the isobutane recycle rate. This carried a little more sensible heat into the reactors, to make more refrigerant vapors, without the extra heat of reaction from the olefins reacting with the isobutane to make alkylate.

Within half an hour, the surging stopped, Mr. Durland went back to the Admin Building, Bobby and Vincent Cappidono went off shift, Zip went for a smoke, and tranquility returned to No. 2 alky unit.

Controlling Vapor Recycle Temperature

A month later, the OCAW Union went out on strike in Texas City. Zip, I, and the other supervisors were therefore running the Alky Unit. One night, Zip said, “Norm we gonna fix that superheated compressor vapor recycle stream. We gonna inject liquid iso-butane recycle into that eight inch vapor line, through a one inch spray nozzle. It’ll evaporate and cool that superheated recycle. We’ll just use screwed pipe and fittings so we can get the job done before the end of the shift.”

And we did. It worked just fine to prevent surge, when the antisurge spill-back valve was opened too far.

Summary

I have had several similar occasions since my experience in Texas City in 1974, when my alky isobutane refrigeration went into surge. I always react in the same manner. That is, I vent the discharge to the flare or fuel. The fastest way to get out of surge is to reduce the centrifugal compressor discharge pressure. The only reasons I avoided rotor damage in the incident I related in Texas City, was that this was an older, slower compressor, with a relatively small number of wheels (i.e., three). Also, the discharge check valve dampened the force of the reversing vapor flow.

33

Refrigeration System

In my desk drawer, I have a tube of yellow, translucent, very viscous liquid. It's been there for 49 years. It's "C-20 Polypropylene." A gasoline additive used by Texaco in the 1960s at a rate of 1 cc per gallon. A top cylinder lubricant for engines.

American Oil, my employer, made the C-20 additive for Texaco by reacting the propylene from Cracking Unit LPG produced at the El Dorado Refinery. The catalyst system was solid pellets of Aluminum Chloride, plus a small flow of liquid Iso-propyl-chloride. To meet Texaco's viscosity spec, the reaction was carried out in the liquid phase at 200 psig and 40°F. The lower the reaction temperature, the higher the viscosity of C-20 product.

American Oil's El Dorado Refinery was an absolute run-down plant in Arkansas. The sole reason American Oil didn't shut down the plant was because it was their only refinery that produced Viscous Polypropylene.

Refrigeration Limit

Jack Horner, the V.P. of American Oil's 12 refineries, had decided to abandon the El Dorado plant and build a new Viscous Polypropylene Facility in the Sugar Creek Missouri Refinery, unless capacity in El Dorado could be increased by 50%. This target was given to Larry Durland, the manager of El Dorado. Mr. Durland asked the American Oil Engineering group in Chicago for help. The assignment trickled down to me, a junior process engineer.

"Lieberman," my supervisor, Bill Duvall, instructed, "get down to El Dorado. Do a quick review. Come back pronto. It's a waste of time. Dr. Horner says we can't spend more than $40,000 on the expansion. The last time we estimated the plant expansion, it was $6,000,000. We sure ain't going to do a 50% expansion for $40,000. Fly economy and take a bus to O'hare. Taxis are expensive. Remember meal allowance is $20 a day."

The El Dorado Refinery

I arrived at the plant, to find the gate house empty. The gate, which was falling off its hinges, was closed with a piece of string. I opened the gate and followed the gravel path past the crude unit, FCU, the rusty alky unit, finally to a cinder block building labeled, "Poly-Propylene Control Room." I introduced myself to Les Raiford, the sole operator.

"Mr. Norm, we're limited by refrigeration capacity. We keep our reactor temperature at 40°F. If it gets warmer, polypropylene viscosity gets too low to meet the Texaco Spec."

"Les, can't you run the refrigeration compressor faster and circulate more refrigerant?"

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"No. It's a motor driven compressor. Runs at 3600 rpm. We got it fully loaded. The spillback valve is blocked in. I'll draw you a picture of the refrigeration loop (see Fig. 33.1). Ain't nothing else to do."

We walked through the unit. Les pointed to the refrigerated reactor, compressor suction drum, the centrifugal compressor, the air-cooled condenser, and the refrigerant receiver drum. "Mr. Norm, that refrigerant drum came from Krupp. From the Germans after World War I. Part of reparation payments in 1919. My daddy told me about it."

"Les, what do you use for refrigerant?"

"Propane. We make it off of the reactor effluent depropanizer. Just plain LPG like that comes in 5-gallon containers you buy for camping."

"Ever think of using something else?"

"Nope! The refrigeration compressor is designed for propane, with a few percent of butane and ethane."

The City of El Dorado

The sign above the check-in desk at the hotel was definitive:

- No out of town checks.
- No credit cards.

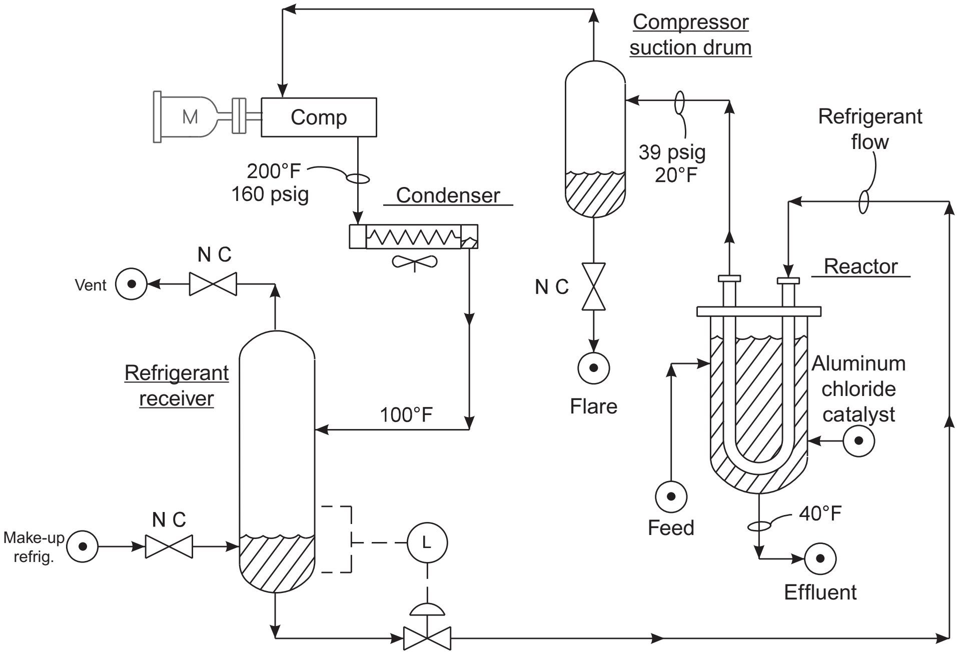


FIG. 33.1 Propane refrigerant.

The restaurant next door had a hand painted sign, "Carl's Diner – Where Colored Folks Meet to Eat." In the refinery itself, the Plant Maintenance Division was divided into:

- Pipe fitters
- Machinists
- Electricians
- Colored laborers

American Oil, in 1968, actually designated a distinct group of employees as "Colored Laborers." There were two dilapidated elementary schools—one for blacks and one for whites. Both existed in close proximity, but isolated in their own environments. The local Chinese restaurant served blacks, whites, and me. But, did not accept credit card payments.

Refrigerant Composition

During my initial inspection with Les Raiford, I noticed that the compressor motor driver was running far below its FLA point (i.e., Full Limit Amp Load). At home, motors trip-off because they overheat from excessive load. In the plant, motors trip-off when they reach a rated amperage load: the FLA point.

The refrigerant compressor motor was oversized. This is not a waste of energy. Doubling a motor size from 100 to 200 BHP only increases power consumption by 3%, if the load of the equipment being driven is not altered. Also, incremental costs of motors are small.

I had learned as a young engineer (in 1968, I was 25) that compressors are limited by either:

1. Driver power
2. Suction volume
3. Maximum discharge pressure

The relief valve setting on the refrigerant receiver (see Fig. 33.1) was 260 psig or 100 psi above the current compressor discharge pressure. As the amp load was well below the FLA point of the motor, I was quite sure I was only limited by the compressor's suction volume.

The compressor suction temperature was fixed at 20°F, based on the need to keep the reactor at 40°F. Vapor volume is proportional to:

1. Number of moles
2. Temperature (°R or °K)
3. Pressure (PSIA or BARA)

As #1 above had to increase, and item #2 was fixed at 20°F, I had to increase the compressor suction pressure, item #3.

I finished my giant portion of pork fried rice, paid the $6 bill, and went to walk back through the quite streets to my vintage hotel.

Increasing Refrigerant Volatility

"Les, if the Cat Cracker is down, what do you do for feed to the Polypropylene Unit?" I asked the next morning.

"Mr. Norm, could you watch the board for a minute? I got to go to the toilet at the Cat. Could you tell Mr. Durland, we sure could use us a toilet?"

Twenty minutes later, Les explained, "I got about two days of storage in my Bullet for Cat P-P's. After that, we got to import pure propylene. Comes by rail car. Cat P-P's is 60% propylene and 40% propane. The imported stuff is mostly pure propylene, with a bit of ethane. Only need to run about 60% of our regular charge rate of the pure propylene to make the same amount of the viscous polypropylene product for Texaco. I unload the tank cars into that Storage Bullet down yonder."

"Les, what's the pressure in the Cat P-P's Bullet?"

"About 160 psi."

"What's the pressure in the pure propylene Bullet?"

"About 220 psi. It sometimes gets higher, 'cause we get extra ethane in the purchased propylene. More ethane, more pressure. I just gas-off that extra ethane from my depropanizer reflux drum. No big problem."

Les Raiford looked out the glass front of the old control room at the black laborers loading drums of toxic AlCl pellets into the spare reactor.

"Mr. Norm, what are you thinking about? You're thinking that we can use the imported propylene feed for refrigerant?"

"Yes."

"We talked about this before. Me and John Henry on our graveyard shift. But we couldn't see why propylene would be a better refrigerant than propane?"

"It isn't," I explained. "it's just that the propylene you import is about 37% more volatile than pure propane:"

$$(220) \div (160) = 137\%$$

"That means the same volume of flow to the compressor, at the same suction temperature, will have 37% more weight of refrigerant and be able to remove 37% more heat." (Note: I am neglecting the small difference in molal latent heat of vaporization between propane and propylene. Also, I have neglected that this calculation should be at the 20°F compressor suction temperature, and not at the ambient temperature in the storage bullets. But I prefer to relate the story as it happened.)

"You know Norm, I was in the Army in Korea in 1950. I fought at the Inchon Landing. We could try this out now. I can connect our imported propylene storage bullet to our refrigerant make-up line (see Fig. 33.2), and vent the refrigerant to fuel gas at the same time. That'll increase the propylene content of the circulating refrigerant."

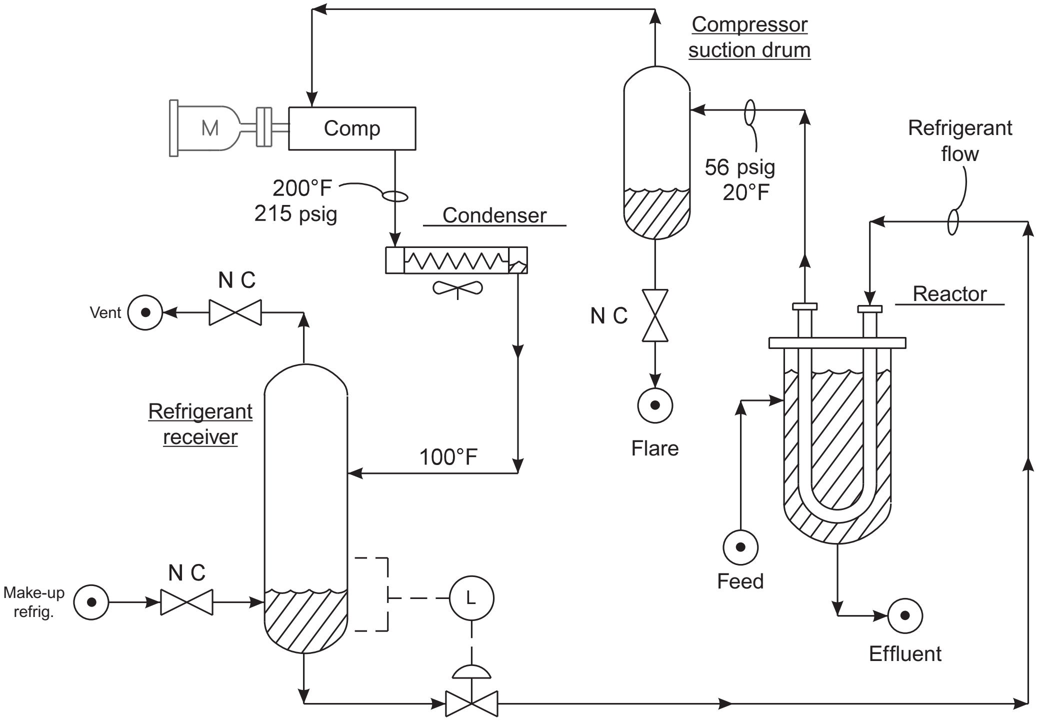


FIG. 33.2 Propylene refrigerant replaces propane refrigerant.

One Experiment is Worth a Thousand Calculations

"Were you in the first wave at Inchon?" I asked.

"Nope. The Marines were. I was in the follow-up wave. Them Commies just ran away. Not much of a fight."

"Well okay. Let's go slow. I'll monitor the compressor suction pressure. We'll keep the reactor temperature constant. If we can raise the compressor suction pressure by a few pounds an hour, that will validate my theory."

"And if this works, don't forget about our toilet. And Norm, I'm also thinking about the compressor discharge pressure. Not now. It's spring. But we get 100°F days in southern Arkansas. Our discharge air cooler might have a problem with condensing all that propylene refrigerant?"

Condenser Limitations

"Les. Do you know what a swamp cooler is?"

"Sure. It's when you spray a mist of clean water into dry air. The water evaporates, and cools off air by like 10°F."

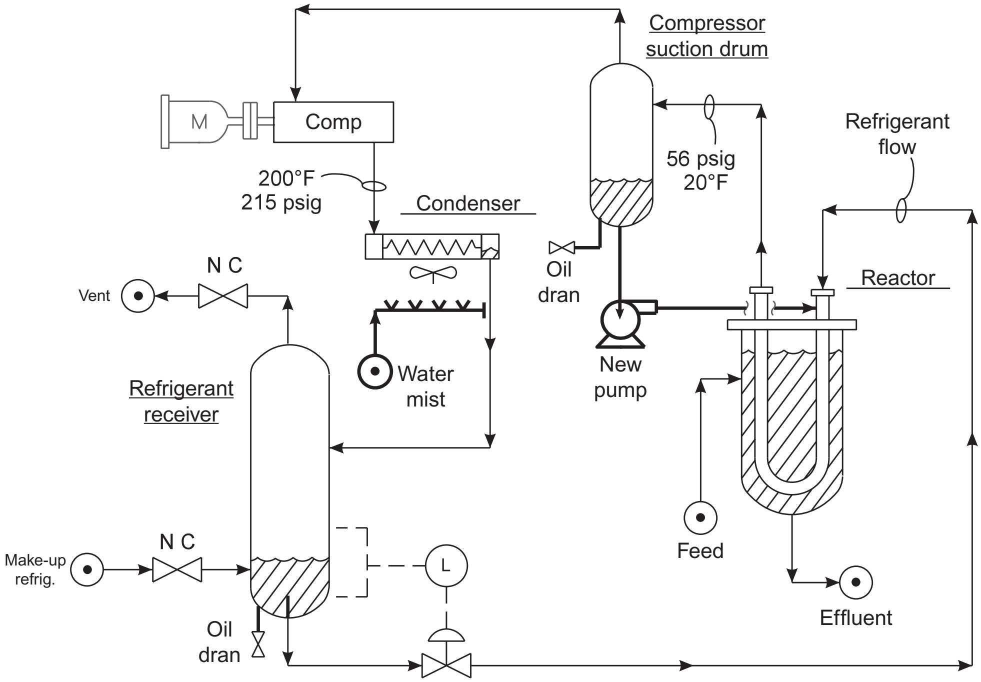


FIG. 33.3 Additional improvements.

"Right. We can purchase some mist nozzles from Walmart (see Fig. 33.3) and set them about three feet back from the F.D. air cooler fans. Not to get the fins on the tubes wet, but to humidify and cool the air."

"I don't know. You're gonna need a couple dozen of them nozzles. And with all that PVC pipe and valves, and hiring guys to assemble it – you're looking at maybe $500 easy."

"Hey, don't worry. I got $40,000 to work with. Dr. Horner, the V.P., has already approved it."

Refrigerant Carry-Over

"One of our problems, Norm, is refrigerant carry-over from our evaporator. Liquid gets carried into the Compressor Suction Drum, which we wind up draining to the flare. At the higher rates, you're planning, this may become a big problem."

"Well, since the compressor suction drum is slightly elevated above the reactor, we could drain it back to the evaporator?"

"No. It ain't elevated enough for that to work."

"Okay, then we'll install a small pump. Don't forget I have $40,000 to work with (This was in 1968. Pump prices have increased ten-fold since then.)"

Fouling

"Another problem we got, Norm, is fouling on the outside of the tubes in the evaporator, and inside the tubes on the air cooled condenser. I bet that that's bad for heat transfer efficiency on both exchangers."

"Fouling?" I asked. "Can't be. The refrigerant is completely clean."

"Well Mr. Norm, it's the compressor lube oil that leaks into the circulating LPG that causes the fouling. Coats the tubes with oil."

"I've seen this problem before. Let's just install a couple of low point drains on the compressor suction drum and on the Refrigerant Receiver (Fig. 33.3). Just blow-down both drains once a week, or once a day, to keep the refrigerant clear of lube oil. That's what the filter does on your home air conditioner freon circulation at the condenser outlet line."

Forced Refrigerant

About 4 months after the changes shown in Fig. 33.3 were implemented, the engineer at the FCU, Noel, suggested that the pump I had installed to transfer liquid refrigerant out of the Compressor Suction Drum could be run continuously, to convert the reactor cooling coil from once-thru refrigerant flow, to forced circulation.

I returned to El Dorado to try Noel's idea. Instead of evaporating 99+% of the refrigerant, I adjusted the circulation rate, so that only half the refrigerant was vaporized.

I calculated that heat transfer coefficient (i.e., "U") of the reactor cooling bundle had increased from 60 to 90 btu/h/ft^2/°F. This resulted in a further 20% increase in Viscous Polypropylene production.

I did not give Noel credit for his idea. I'm still a fan of forced circulation to reduce percent vaporized in evaporators and reboilers. And, I still take credit for other people's ideas as long as they work.

Conclusion

Over the next 3 months, these changes I've described were made. Viscous Polypropylene production increased by 40%. Les Raiford's son, who was studying Chemical Engineering at LSU, called to congratulate me on a successful project.

A year later, Dr. Horner, V.P. of Refining, decided to shut down the El Dorado Refinery anyway. All the operators were discharged—blacks and whites, without discrimination.

Larry Durland, the Plant Manager at El Dorado, was promoted to the 360,000 BSD refinery in Texas City, as Plant Manager. He retired in 1978.

A new Viscous Polypropylene Unit which I designed was constructed at the American Oil Refinery in Sugar Creek, Missouri. But Texaco stopped using the additive in their gasoline a few months later. The Sugar Creek Refinery was also abandoned shortly thereafter.

And I, the author of this true story, worked at American Oil until 1980. I was not promoted or rewarded for my idea of optimizing the polypropylene refrigerant composition. But maybe I have been rewarded, just not by American Oil.

The names in this story have not been changed to protect the guilty.

34

Reciprocating Compressors

I was working in the gas fields in Laredo, Texas, for Good Hope Energy, the second largest producer of natural gas in the 1980s, behind Exxon. I reported to the owner of Good Hope, Jack Stanley. Jack owned Good Hope Energy personally. There were no stockholders or directors. Just Jack.

"Norman. Get down to our compression station in Laredo. Gas flow to Aqua Dulce is low. Gas is selling for $6.00 per million BTU's. It's cold up north, and demand and prices are peaking. Let's cash in. Get with Juan Hernandez. He'll pick you up at the airport. Drive around Zapata and Webb counties. Speed up those recips to pump more gas. Get going, Lieberman."

"Mr. Stanley, it's not all that simple. Reciprocating compressors have several complex features and suffer from a wide variety of malfunctions. I've written a book published by PennWell [1] on the subject. Would you like me to explain?"

"No. Tell me when you get back."

Starting Volumetric Clearance

"Juan," I observed as I climbed into the company truck at the airport, "it's damn cold today."

"Si, Señor Norm. The cold has shut down many of our gas field reciprocating compressors. The wet gas has frozen the fuel gas supply lines. Muy malo! The fuel lines are full of ice."

"Not exactly. The lines are plugged with natural gas hydrates. Which form at temperatures well above the freezing point of water. We'll blow the lines clear. But my concerns are broken valve plates and most importantly, excessive 'Starting Volumetric Clearances' between the piston and the cylinder head at 'Top Dead Center'."

"Ah. Mi amigo. What mean's this 'Starting Volumetric Clearance'? No comprendo."

"Juan, let's drive down to Hebronville Compression Station and I'll explain how reciprocating compressors work. You drive and I'll talk. To understand the critical significance of a recip's starting volumetric clearance, I'll have to explain first the 'Carnot Cycle'."

"Si! Si! Professor Carnot. Muy importante."

Carnot Cycle

Fig. 34.1 is the Carnot Cycle Curve for theoretical compression work. The vertical axis is the pressure of the gas inside the cylinder. The horizontal axis is the volume of gas

Understanding Process Equipment for Operators and Engineers. https://doi.org/10.1016/B978-0-12-816161-6.00034-5

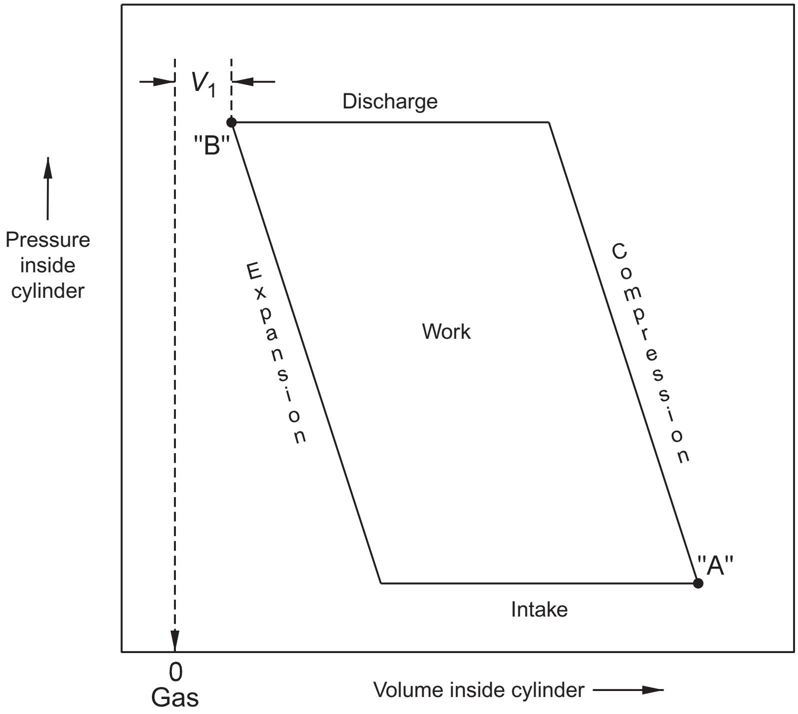


FIG. 34.1 The carnot cycle, "V_1" represents the "Starting Volumetric Clearance."

inside the cylinder. The area enclosed by the entire curve is called PV work or theoretical compression work.

Fig. 34.2 is a simplified view of a reciprocating compressor. In my discussion of how piston movements relate to the Carnot Cycle, I'll make several simplifying assumptions:

1. Reciprocating compressors are "Double-Acting" compressors. Gas is compressed by the piston as it moves toward the head of the cylinder and also as the piston moves to the crank-end (i.e., the end closest to the drive shaft). I'll ignore the crank-end compression for simplicity.
2. I'll also ignore gas leakage around the piston rings and around the valves and other mechanical inefficiencies. Recip's are, in practice, subject to a lot of mechanical inefficiencies, which often are very large.
3. I'll ignore valve pulsation and frictional losses, which are often also very large.

Looking at Fig. 34.2, let's describe the parts of the recip [2].

- *Piston*—Slides back and forth inside the cylinder. It's pushed by the piston rod. To keep gas from leaking between the piston and cylinder, there's a piston ring. Just like on your car engine.

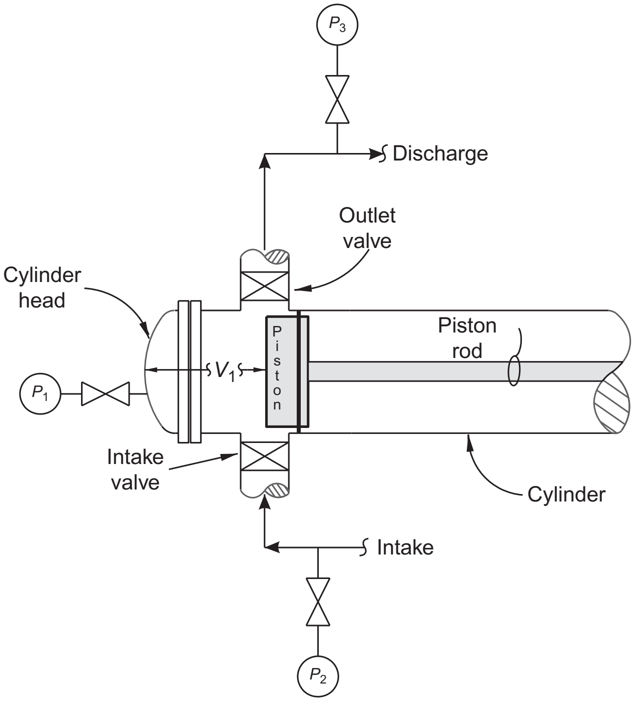


FIG. 34.2 Components of a reciprocating compressor.

- *Valves*—These are spring-loaded check valves. They are moved from open to closed, and from closed to open, by differential gas pressure. The springs push on plates. If the gas pressure inside the cylinder is larger than the intake line pressure, then the inlet valve plate is pushed by the gas pressure into a closed position.

If the gas pressure inside the cylinder is smaller than the discharge line pressure, then the outlet valve plate is pushed by the gas pressure into a closed position.

Let's refer back to Fig. 34.1. Point "A" is when the piston is as far back in the cylinder as it can get. This point "A" is called "Bottom Dead Center." The volume of gas inside the cylinder (i.e., the horizontal axis of the Carnot Cycle Curve) is at its maximum volume.

The piston now is pushed by the piston rod toward the cylinder head. Both the inlet valve and the outlet valve remain closed, because the pressure inside the cylinder at "*P*-1," in my Fig. 34.2, is more than the pressure at *P*-2 (Intake) and less than the pressure at *P*-3 (Discharge).

As the piston moves toward the cylinder head and compresses the gas:

a. The pressure inside the cylinder at *P*-1 goes up. That's why this part of the Carnot Cycle is called compression.
b. The amount of gas inside the cylinder remains the same, but its volume decreases.
c. The gas gets hotter because the piston is doing compression work on the gas.

Starting Volumetric Clearance

Here's my main point. The piston stops moving *before* it passes the outlet nozzle on the cylinder.

The piston is quite long. Typically, 6″–8″. If it moved much past ⅓ of the diameter of the discharge port opening, it would block off the outlet nozzle. So, the piston stops moving at about ⅓ of diameter of the outlet nozzle. This is point "B" on my Carnot Cycle curve (Fig. 34.1). It's called "Top Dead Center."

The volume of gas trapped between the piston and the cylinder head at Top Dead Center is called the *"Starting Volumetric Clearance."* This volume of gas determines the "Volumetric Compression Efficiency" of the recip. That's why it's important. The volumetric efficiency is a measure of how much gas is compressed for each stroke of the piston.

During the compression part of the Carnot Cycle, both the inlet valves and outlet valves shown in Fig. 34.2 are closed.

When the pressure at *P*-1—meaning the pressure inside the cylinder—reaches the discharge line pressure at *P*-3, the outlet valve springs open (see Fig. 34.2). The piston continues moving toward the cylinder head just like before. But now, the pressure inside the cylinder remains constant (i.e., the *P*-1 pressure). Gas is pushed out of the cylinder by the piston, through the outlet valve, and into the discharge line.

This is called the discharge portion of the Carnot Cycle. During this portion of the cycle:

- The inlet valve is closed.
- The pressures at *P*-1 and *P*-3 are identical.

> The piston actually moves from bottom dead center to top dead center in just a fraction of a second.

Recip Valves

"Por favor, Señor Norm," Juan asked, "before you explain further, tell to me what 'valve' you are talking about? Are there some kind of valves inside the reciprocating compressor cylinder? How do they work? What are they supposed to do? Explain about these valves first, before talking more about Professor Carnot and his cycle."

"Okay. Let's refer to Fig. 34.3. A typical valve is a heavy, round cylinder. Maybe 6″ high and perhaps 4″–8″ wide. There are two plates inside the valve. One is pushed by the springs. The other is stationary. When the two plates are pushed together by the springs, the valve is closed, and gas flow stops. If the gas pressure on top of the valve in Fig. 34.3 is bigger than the gas pressure underneath the valve, then the plate is pushed down. Now, the gas can flow through the ports, and the valve is open. That's the discharge valve.

"If you rotate the valve by 180°, it's the same as the intake valve."

"Muy bien, Señor Norm. Yo comprendo. Then what happens inside the cylinder after the piston reaches its top dead center position?"

"Yes, Juan, I'll continue my explanation of the Carnot Cycle, while you study Fig. 34.1."

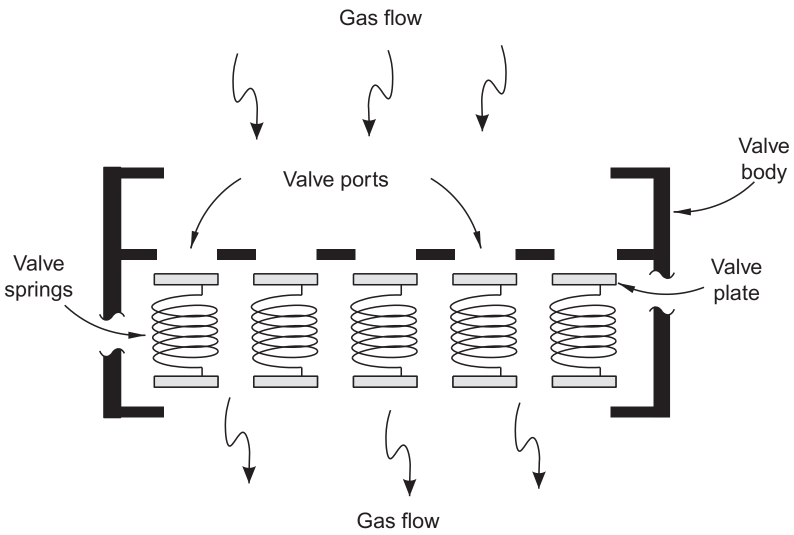


FIG. 34.3 Components of a reciprocating discharge valve. Gas pressure overcomes spring tension and pushes valve plate down and open. Intake valve is reversed orientation. This is a process representation and not mechanical descriptive.

Gas Expansion

The natural gas trapped between the cylinder head and the piston at point "B" is the pressure of gas in the discharge line. When the piston reverses its travel and starts to move back toward point "A" (bottom dead center), the pressure inside the cylinder will start to drop and the outlet valve (Fig. 34.2) will shut. The inlet valve will remain closed.

"Si! Si! Señor Norm. Both valves are closed because as the gas trapped in the starting volumetric clearance expands, the pressure at P-1 is bigger than P-2 (the intake line), and smaller than P-3 – the discharge line."

"Right. The gas continues to expand as the piston retreats, until the gas pressure inside the cylinder falls to the intake line pressure. That is the Expansion portion of the Carnot Cycle (Fig. 34.1). Now the pressure at *P*-1 equals the pressure at *P*-2.

"Juan, note that the bigger the starting volumetric clearance, that more gas has to expand before the pressure inside the cylinder equals the intake line pressure. Certainly, new gas cannot flow into the cylinder until the pressure at *P*-1 drops to the pressure at *P*-2. Therefore, increasing the Starting Volumetric Clearance inside the cylinder head (Fig. 34.2), will reduce the rate of gas compressed by the recip."

Intake

"With the gas pressure inside the cylinder having declined to the intake line pressure, the compressor intake valve shown in Fig. 34.3, now springs open. New gas now flows from the wellhead into the compressor. The gas flow continues, until the piston retreats to its starting point at bottom dead center (pt. A, Fig. 34.1).

"As soon as the piston begins its travel back towards the cylinder head, the intake valve springs closed. Of course, the outlet valve is still shut."

"Norm, does the shaded area on the chart have a special name?"

"Si, Juan. It is called either:

- Ideal Compression Workor
- PV Workor
- Carnot Cycle Work"

"But a question, Norm. Since some of our wellhead compressors are driven by electric motors, and run at a constant speed, how could we reduce the recip's capacity, if it's pumping too much gas?"

"Juan. On the back-end of the cylinder head, where you have the pressure gauge, just unscrew this gauge. Then, take the one liter steel bottle that's in the truck. Screw it into the 1½″ connection on the cylinder head. This will increase the starting volumetric volume by one liter worth of gas volume. Increasing the starting volumetric volume by one liter, reduces the compression capacity."

"So Norm. To gain more compressor capacity, and pump more gas for Señor Stanley, the owner, I should shut all the 'Adjustable Head End Clearance Pockets' by turning the valve handle in a counter-clockwise direction. This will reduce the starting volumetric clearance and increase the volumetric efficiency of the recip."

"That is all correct, mi amigo."

Sources of Compression Inefficiency

"Juan. I should tell you, that there are a lot of problems with the valves that reduce their actual performance, such as:

- Valve velocity losses (friction).
- Valve pulsation problems.
- Valve leakage.
- Broken valve plates.
- Excessive spring tension.
- Bad piston rings.

"These problems are best identified by running an 'Indicator Card' or a 'Beta Scan' test. I've written a book published on this subject by PennWell, 'Troubleshooting Natural Gas Processing' [1]. Would you like to purchase a copy, Juan?"

"No, Señor Norm."

Going Home

I spent 2 days working with Juan in the gas fields in both Zapata and Webb counties. Mainly, ice hydrates had frozen in the compressor fuel gas lines. Juan and I disassembled the ¾″ gas supply lines from the well to blow out the ice plugs and return the compressors to service.

At one well, the ice plug suddenly blew out of the disconnected pipe. Propelled by 400 psig of gas pressure, I saw the ice plug race harmlessly past my face.

But then, the 10 ft of ¾″ steel pipe snapped back against my legs with great force. I was knocked off my feet. Juan drove me to the hospital in Laredo. The next day, I flew home to New Orleans, having been delivered to the airport in a wheel chair, but after having increased gas flow by about 10%.

A week later, back at work, Mr. Stanley greeted me. "Lieberman. Where were you? Did you meet some girl down in Laredo? Don't expect me to pay you while you're partying. Better get back to the Hebronville Compression Station. There're problems with the glycol dehydration unit. Exxon may cut us off because of the wet gas. Get going."

Piston Rod Loading

The world's largest producer of rubber is Arlanxeo. They produce synthetic rubber from butadiene by polymerization. The company is owned by Saudi Arabia. Their largest plant is in Orange, Texas.

The plant started production in 1939 just in time for World War II. With the Japanese overrunning Malaya, Vietnam, and Indonesia, there was a desperate need for synthetic rubber to replace the lost sources of natural rubber from the British and Dutch rubber tree plantations.

Last year in 2017, I taught a seminar in Orange for Arlanxeo to their operators and engineers. After each session, I visited the plant to advise how to improve their operations.

Butane Circulation Limit

Their main bottleneck to increasing synthetic rubber production was the limited circulation rate of the butanes. The bottleneck to butane circulation was the capacity of three, two-stage reciprocating compressors. This bottleneck was especially severe in July and August.

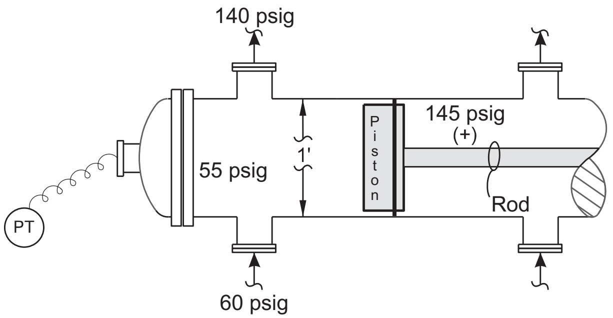
Orange, Texas, is an awful place in the summer. And these conditions result in a 90°F plus cooling water supply temperature, which causes the butane circulation reciprocating compressors to trip-off due to high discharge temperatures.

Pressure Differential Across Piston

Fig. 34.4 is a sketch of a reciprocating compressor cylinder. Recip's are double-acting machines. There are two sets of valves:

- One set is on the *"head-end."* The head-end is at the end of the compression cylinder.
- The other set is on the *"crank-end."* The crank-end is that part of the compression cylinder near the shaft, driving the compressor's piston rods.

When the piston moves toward the cylinder head, the gas inside the front-end of the cylinder is compressed and then pushed into the discharge line, when the pressure in front of the piston exceeds the discharge line pressure.



- Valve DP = 5 psi
  (Measure at head end) "PT"
- Area = 0.785 ft^2 (of piston)
- DP = (60–5)+(140+5) = 90 psi
- Rod loading = 90 X 0.785 X 144= $\boxed{\text{1018 pounds}}$

FIG. 34.4 Piston rod loading calculation.

When the piston moves toward the cylinder head, the gas pressure inside the back-end of the cylinder is reduced, until its pressure falls below the suction line pressure. At which point, gas from the suction line flows into the back-end of the cylinder.

The ratio of the discharge line pressure, divided by the suction line pressure, is called the compression ratio.

Compression Work

Before I ride my bike, I always reinflate my old bike tires. I compress the air from 14.7 psia (1 bara) to 45 psig (4 bara) for a compression ratio of four. The compressed air gets appreciably warmer. Partly due to the heat of compression from the work I'm expending, and partly due to frictional losses in my bicycle pump.

It's the same with a reciprocating compressor. The butane vapors compressed increase in temperature due to the heat of compression work, plus the frictional losses in the compressor. Meaning, that the temperature rise between the inlet and the outlet of the compressor is proportional to the compression ratio. If the outlet temperature increases, because of the inlet temperature increase, this is *NOT* an indication of an increase in the compression ratio.

Excessive Piston Rod Loading

When I worked in the natural gas fields in Laredo, Texas, I was in charge of a 100 reciprocating wellhead compressors. The most common failure of these machines was piston rod breakage (see Fig. 34.4), due to excessive rod loading.

Rod loading is expressed in pounds of force. To calculate rod loading on a reciprocating compressor:

$$\text{Piston area} \cdot (P_D - P_S) = \text{Rod loading} \tag{34.1}$$

where:

- Piston area in square inches
- P_D = discharge pressure, psig
- P_S = suction pressure, psig
- Rod Loading = pounds force

(see Fig. 34.4)

From the above, it's clear that there is a relationship between rod loading, compression ratio, and the temperature increase of the gas being compressed.

In Laredo, during the hot days of summer, ambient temperatures would increase to 120°F. Compression discharge temperatures would, as a consequence, also increase. The compressors were equipped with high discharge temperature trips. The trips would shut the compressors down to protect the piston rods from failure due to excessive loads.

The first afternoon this happened, I calculated the rod loads from Eq. (34.1) and reset the trip temperature consistent with the higher suction temperature caused by the greater ambient temperature to the compressor interstage air cooler.

In 1985, in the Laredo gas fields, we did not have access to a computer control to calculate maximum permissible rod loads, based on suction and discharge pressures. So, during periods of cooler weather, I had to reset the high discharge temperature trip point lower. A cumbersome practice.

Arlanxeo Rubber Plant

To avoid their reciprocating butane compressors from tripping off due to excessive discharge temperature, which was a consequence of increased suction temperature, caused by increased cooling water temperatures, which was a result of high ambient temperatures, Arlanxeo operations personnel would activate the valve unloaders.

These valve unloaders would disable one or more of the discharge valves shown in Fig. 34.4. The result was to:

- Raise the compressor suction pressure.
- Reduce the compressor discharge pressure.
- Reduce the compression ratio.
- Which reduced the compressor's discharge temperature.

Also, activating the valve unloaders reduced the butane circulation rate and the plant's synthetic rubber production by about 10%, worth $10,000 a day. And this had been going on since 1939—for 78 years!

Computer Control Application

In 1939 there were no computers to calculate the compressor's piston rod loading. So, the instrument design engineer did the best he could to protect the reciprocating compressor from rod failure, by using the discharge temperature as an indication of possible excessive compression ratio and rod loading.

I asked the Arlanxeo process engineer and operations superintendent what part of the compressor they thought might be damaged by exceeding the 300°F discharge trip temperature.

There was no answer.

It all shows the importance of understanding how process equipment works. The alternate is to lose $10,000 a day, every summer, for 78 years.

Molecular Weight Effect on Reciprocating Compressors

I never argue with people. I expect that everyone will always agree with me. For example, I was working for the Coastal Corporation in their refinery in Aruba. My project was to upgrade their Hydrogen Production Facility. My design included substituting a hydrogen-rich purge gas stream as feed to the hydrogen plant, which was used in refinery fuel gas. The purge gas originated from Aruba's diesel hydrotreater unit. Refinery fuel gas came mainly from the plant's Delayed Coking Unit. The compositions were as follows:

	Purge Gas	**Fuel Gas**
% H_2	80	10
Molecular Wt.	8	32

The purge gas, I calculated, being richer in hydrogen than the fuel gas, would make a better feed to the hydrogen plant, as it was already mainly hydrogen.

The hydrogen plant feed gas was compressed from 50 to 400 psig by dual giant, two-stage reciprocating compressors. The total gas flow was 100 MM SCFD of refinery fuel gas.

When the day came to switch over from fuel gas to purge gas, I was getting ready to go home to New Orleans. Mr. French, the plant manager, came into my office with a complaint. "Lieberman. You screwed up."

"What's wrong Sir?"

"Those two recips are not designed to compress purge gas. They're designed to compress fuel gas. As soon as we lined up to compress the purge gas to the Hydrogen Plant, the gas flow rate slipped down from 100 MM SCFD to 50 MM SCFD. You screwed up Lieberman."

"Mr. French. Are you talking to me? I don't make mistakes!"

"Yes Lieberman. I'm talking to you. I just got off the phone with the vendor who supplies the valves for those recips. He told me that those valves are designed for 32 molecular weight fuel gas, not for 8 molecular weight purge gas. That's the reason we can't deliver the required gas rate to the hydrogen plant."

"And your proposed solution, Sir?"

"The vendor says we can replace the valves, for purge gas service during a 3-day outage."

"And the cost?"

"Rather reasonable. $950,000."

I recall sitting at my desk and becoming progressively angrier. I wanted to defend my design. But before I could say anything further, Mr. French continued. "Lieberman, don't you realize that if you reduce the gas molecular weight by a factor of four, from 32 to 8, that the discharge pressure developed by the compressor will fall drastically, and thus the amount of gas that can be compressed will also drop drastically?"

"Well," I thought, "everyone's got to learn." So, I decided not to argue and went home to New Orleans.

Two Months Later

It was a beautiful day in Aruba, when the retrofitted compressors, with their new valves were put back into service. The plan was to start up on fuel gas (32 MW), and then switch to purge gas (8 MW).

I was standing next to Mr. French, watching the flow meter, which lined-out at 100 MM SCFD, the design flow. As the feed flow was switched from fuel gas to purge gas, the indicated gas flow rate dropped to 50 MM SCFD—just like before the new valves had been installed.

Mr. French turned to me in dismay. "Norman! What's happened? The flow has dropped by half. The new valves have not helped."

"No. The flow has not changed. It's still the same 100,000 SCFD. I'll explain."

Effect of MW on Flow Measurement

"The flow is measured through a flow meter. The pressure drop through the flow orifice plate, is measured at the two taps on either side of the orifice plate. The DP is calculated by:

$$\mathrm{DP} = 0.6 \cdot \frac{D_{\mathrm{V}}}{62} \cdot V^2 \tag{34.2}$$

where:

- DP = pressure drop in inches of water
- D_{V} = density of gas, lbs/ft^3
- V = velocity of gas flowing through the orifice plate, ft per second

To calculate flow, multiply the velocity calculated above based on the measured DP, by the hole size (i.e., area) of the orifice plate, which is printed on the orifice plate handle.

If the gas molecular weight drops from 32 to 8 (i.e., by a factor of 4), then the measured pressure drop will also decrease by a factor of four (Eq. 34.2). And, as flow is proportional to the square root of DP, the indicated flow will drop by a factor of two. But actually, the gas volumetric flow has not changed."

Effect of Molecular Weight on Reciprocating Compressor

"But Norm, how then is the recip affected by molecular weight?" Mr. French asked.

"Sir, one effect is to decrease the valve velocity losses. These are the relatively small frictional losses through the recip's inlet and outlet valves. They have a small effect on compression efficiency and vary linearly with molecular weight. Reducing the MW by a factor of four will increase efficiency by only a few percent."

"Of more interest, is the effect on the compressor horsepower requirement. The compressibility factor (Z) of purge gas is close to 1.00. The compressibility of fuel gas is about 0.90. This means that the amp load on the motor driver will increase by 10%, when compressing the lighter, hydrogen rich gas."

"But Mr. Lieberman. Our Hydrotreater Recycle Gas Compressor puts up a lot more DP, and draws a lot more amps, and a lot more flow, when it's compressing a heavier gas?"

"Mr. French. That's a centrifugal compressor, where:

- DP is proportional to polytropic head, multiplied by the gas density."

"Mr. Lieberman, we have wasted $950,000 on the new reciprocating compressor's valves. Why didn't you explain all this to me two months ago? I came to discuss this matter with you then."

"Yes Mr. French. I do remember. But it's not my fault. A discussion is like an argument, and I never argue."

"That's fine Lieberman. I am canceling your consulting agreement with the Coastal Aruba Refinery effective today. Have a nice trip back to New Orleans."

Reciprocating Compressor Discharge Temperature Limitations

The limitation on recip's is often a high discharge temperature alarm, due to:

- Increased compression ratio.
- Valve velocity losses (DP).
- Valve leakage.
- High inlet temperature.

Temperature limitation is not intended to protect any component from excessive temperature. It is to protect piston rods from breaking due to excessive rod loading. Rod loading is a function of:

1. Compression ratio.
2. Piston area.

Temperature rise across a compressor is a function of the compression ratio. Cylinder discharge temperature is a measure of rod loading and the compressor inlet temperature.

To protect a compressor from excessive rod loading, the compressor may trip-off at high outlet temperatures. This will cause a premature trip if:

1. Compression efficiency is less than design due to valve leakage, high spring tension, or valve fouling.
2. Inlet temperature is above the design used to calculate the high temperature trip point.

Conclusion: Recips trip-off due to high ambient temperature. When this happens, reset the trip higher. A high ambient temperature does not increase the rod loadings.

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[1] N.P. Lieberman, Troubleshooting Natural Gas Processing, PennWell Publications, Tulsa, OK, 1988.

[2] The Oil & Gas Journal Technical Manual, “Foreman’s Page”, vol. 2, 43. 1976.

35

Piping Vibrations

I have observed vibrations in a number of different sorts of piping systems including fired heater tube passes. A year ago, I had a consultation assignment in a lube oil refinery in Louisiana to resolve such a problem.

The problem was described by my client as a vibration in their fired heater radiant tubes. But actually, the vibrations had little to do with the heater itself. However, it did provide me a fundamental lesson on what causes vibration in piping systems in general.

The fired heater was atypical. A small circular unit. The tubes were circular in shape, and there was only a single pass. Meaning, all the tubes were in series. The single 10 in. inlet nozzle flowed through the single pass of 6 in. tubes into the 10 in. outlet nozzle. The inlet nozzle elevation was about 50 ft above grade. The outlet was at eye level (Fig. 35.1).

The heater feed was a mixture of 500°F very heavy lube oil and hydrogen. Both streams were comingled 40 ft from the heater inlet nozzle. The calculated inlet velocity of the lube oil-hydrogen mixture was about 3 ft per second. This is much too low to disperse the lube oil liquid, in the relatively large volume of hydrogen.

As the slugs of lube oil entered into the 6 in. heater tubes along with the hydrogen flow, it accelerated to about 9 ft per second. Still, too slow to promote a mixed phase or emulsion flow. Even after being heated from 500°F to 700°F and vaporizing by several percent, the effluent from the heater was still in a slug flow regime, at less than 12 ft per second.

At the effluent of the heater, the 6 in. single outlet tube expanded into a 10 in., 90 degrees elbow. As I stood with my pipe wrench pressed against the 90 degrees elbow, I felt a sharp knock, as if someone was periodically hitting the elbow with a 20 pound sledge hammer every 5 or 6 s. This then, was the origin of the vibrations—slug flow hitting the outlet elbow.

The periodic momentum of the slugs of lube oil liquid was dampened out into a continuous vibration as the blow to the 90 degrees outlet elbow progressed upstream through the 6 in. heater pass.

You can observe this yourself on a long wire fence. Have a colleague periodically strike a strand of the taut wire with a metal rod. Fifty feet from the point of disturbance, you will feel a continuous vibration of the wire, but not the individual blows.

Steam Hammer

I am most familiar with this type of vibration as a consequence of steam or water hammer.

On a long run of steam piping, localized cooling causes an area of low pressure as a consequence of the condensation of steam. An upstream slug of water in the steam piping rushes toward the region of low pressure with great momentum or speed.

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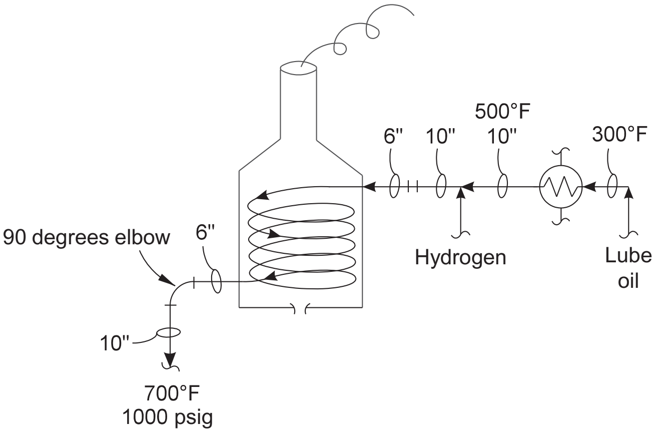

FIG. 35.1 Vibration in lube oil fired heater.

When the slug of rushing water hits an elbow or Tee-junction, it creates the banging sound, which causes the steam piping to vibrate. I've been frightened by steam or water hammer many times, but never personally observed any piping failures as a result.

Avoiding Slug Flow

At the Louisiana lube oil facility, I concluded that the cause of the problem was that the 10 in. feed line was flowing at too low a velocity to induce mixed phase flow between the lube oil and the hydrogen. Likely, if the hydrogen and lube oil were heated together in the upstream lube oil feed preheat shell and tube exchanger, and then flowed to the fired heater through a 40 ft 4 in. line (rather than the existing 10 in. line), the flowing velocity would have been approximately 20 ft per second.

When I worked for Amoco Oil in the 1970s, I was taught that:

- Flows above 25 ft per second would result in mixed phased flow and avoid slug flow.
- Flows below 15 ft per second promote phase separation and slug flow, especially in piping risers.

Effect of Vaporization in Heater

When lighter, more volatile lube oils were processed in the Louisiana Lube Plant, the vibrations were greatly reduced or even eliminated. This apparently was due to the larger percentage of vaporization of the lube oil liquid, which promoted mixed phase flow, and reduced the amount of liquid impacting the 10 in., 90 degrees outlet elbow.

As far as continuing to operate the heater on the heavier lube oil feed stock, several of the tube support hangers were broken. I observed that the tubes were moving an inch or 2 every few seconds in a horizontal plane. A reduction of 20%–30% in the feed rate diminished the intensity of the vibration to a somewhat acceptable degree.

The only successful contribution I have made to date on this project is that the outlet piping vibration is now monitored continuously from the operational control center. There, a high vibration alarm hopefully will alert the operators of impending failure.

Exactly why my client does not shutdown the unit to rectify the fundamental design error of slug flow in the heater's feed piping is difficult to explain to my readers outside of the petroleum refining industry.

Should there be a catastrophic failure at the 90 degrees outlet elbow, the heavy lube oil is well above its approximate 400°F auto-ignition temperature, and there is a reasonable expectation of a hydrogen cloud detonation.

On the other hand, the heater has been vibrating on the heavy lube oil run for almost 2 years and nothing has happened—yet!

It's a gamble.

Piping Supports

Often, vibration of process piping is due to improper supports. The above heater example was obviously not related to a piping support issue, as it occurred only when processing feeds that were heavier, with very small calculated percent vaporizations. The vibrations disappeared on the lighter lube oil feeds that had a higher percent of vaporization.

For piping support vibration problems, I refer my clients to "Stress Engineering"—Metairie, Louisiana, a consultant company dealing mainly with process piping failures in the hydrocarbon processing industry.

Control Valve Vibrations

On several occasions, I have felt and heard control valves making a buzzing sort of vibration. Upon investigation, it was found that the control valve seat (which is changeable) had become loose. The control valve was bypassed and the problem was quickly and easily rectified by the unit instrument technician. The loose control valve seat had, in addition to generating the odd sound, had also caused the flow rate to become somewhat erratic.

You will need to get quite close to the faulty control valve to hear this buzz as it is not particularly loud.

Natural Harmonic Frequency

Several years back, I was working for a small refinery in southern Indiana. They had installed a vacuum seal ring pump that had excessive vibrations. The entire assembly was returned to the manufacturer to be overhauled. But to no avail. Upon return to service, the vibrations persisted.

At this point, the 3600 rpm motor driver was changed to a variable speed (i.e., variable frequency drive). A very slight decrease in motor speed entirely eliminated the vacuum pump's vibrations.

Analysis showed that the piping had been running at its "Natural Harmonic Frequency" at 3600 rpm, which transmitted the harmonic vibrations to the vacuum seal ring pump.

36

Problems in Measuring Process Variables

Understanding what is happening in process units is often difficult enough, without the added complication of questionable pressures, temperatures, flows, and levels. I have been able to solve many long-standing problems by having access to correct data and accurate samples.

Best not to assume that the instrument itself is mechanically and electrically functioning normally. It's a good practice to ask the Instrument Technician so as to assure that the instrument itself is working correctly. A few examples of the problems I have encountered illustrate the complexity of obtaining accurate process operating data and samples. I'll divide the problem into five sections:

1. Pressures
2. Temperatures
3. Levels
4. Flows (vapor and liquid)
5. Samples (vapor and liquid)

Pressures

The house I live in, that has seven bathrooms, my wife's BMW and our 44,000 gal swimming pool, were paid for from making refinery pressure drop surveys. The most common source of error is plugged pressure taps. A pressure gauge reading that slowly creeps up reflects a partially plugged connection. If you have to wait 15 s for the pressure gauge needle to come to rest, that's okay. If it takes 5 min, then the connection is partly plugged and the reading is low.

Fig. 36.1 shows the "Rod-out-Tool" or "Angle-Worm" that's likely available in your plant to unplug bleeders in hazardous services. Every refinery I've worked in has this device available to clear plugged bleeders. Unplugging a bleeder in hot gas oil service with a bent welding rod, not uncommonly, results in premature death. For H_2S service above 100 ppm, a Fresh Air Pak is required. Unplugging bleeders in hydrocarbon and high pressure steam service is potentially very dangerous.

Always place the gauge straight up. Gauges will not read correctly when not in a vertical position. Do not tighten the gauge by twisting by hand. This alters its calibration. Use a

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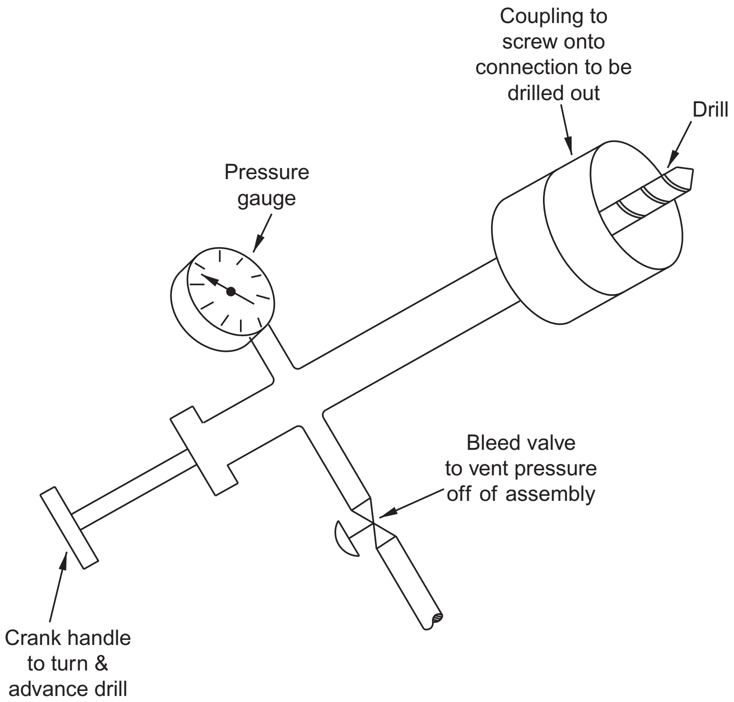

FIG. 36.1 Safe bleeder unplugging device (i.e., rod-out tool).

wrench to twist the stem. When measuring pressures in tar or asphalt service, do not place the gauge in direct contact with the tar. It will plug. I use a small "*U*" shaped fitting filled with diesel as a barrier between the tar and the gauge.

As the objective in conducting a pressure survey is to determine pressure losses due to friction, pressure readings must be corrected for elevation:

$$\mathrm{DP} = (\mathrm{DH}) \cdot (\mathrm{SG}) \div 2.31 \tag{36.1}$$

where:

- DP = psi
- DH = Elevation above grade, feet
- SG = Specific gravity

Effect of De-acceleration

I stumbled across this problem in the Coastal Refinery in Aruba 20 years ago. That is, the discharge pressure from a heat exchanger could be greater than the inlet pressure. In this case, even though both the inlet and outlet pressure taps were at the same elevation, the outlet pressure was somewhat higher. How could this be?

Let's say that the inlet water velocity was 16 ft/s in the 3″ inlet nozzle. If the outlet nozzle size was 6″, then the outlet velocity would be a lot slower (i.e., 4 ft/s). To convert the velocity (neglecting frictional losses) to pressure:

$$\mathrm{IP} = \frac{0.18\left[V_1^{\,2} - V_2^{\,2}\right]}{28} \cdot (\mathrm{SG}) \tag{36.2}$$

where:

- V_1, V_2, velocity inlet and outlet, feet per second.
- SG = Average specific gravity
- IP = Fluid pressure increase due to the de-acceleration

> For the water velocity, slowing from 16 to 4 ft/s, this equates to a pressure rise of 8 psi.

In the above example from Aruba, the service was a heat exchanger in the cooling and partial condensation of a diesel oil hydrotreater reactor effluent. The reduction in velocity, due to cooling and the partial condensation of the reactor effluent, was much larger than the 12 ft/s cited above. The actual pressure rise that I observed on the unit was from 680 to 688 psig. Subtracting de-acceleration effects from this 8 psi increase yielded a pressure drop of 18 psi due to frictional losses. Catalyst fines had partially plugged the exchanger tubes.

Temperature

Most of the temperature measurements I make are with my $85 Walmart infrared temperature gun. Of course, these are skin temperatures taken in places where there are no thermowells and no thermocouples. But, assuming the correct junction and thermocouple wires are installed properly, one would think that the temperature generated from the thermocouple electrical junction would be invariably correct. And, in a sense, they are. But the two problems I have observed still caused the measured temperatures to be 40–60°F below the real process temperature.

The first problem was at a refinery fired vacuum heater outlet in Louisiana. I was certain that the heater effluent temperature recorded was too low, as the downstream temperature at the vacuum tower inlet was 20°F hotter. The heater effluent cools before it enters the vacuum tower due to:

- Vaporization
- Ambient heat loss
- Thermal cracking

I extracted the thermocouple from the thermowell (see Fig. 36.2), planning to insert a glass thermometer into the thermowell to check the temperature locally. However, I immediately saw the problem. The thermowell and of course, the thermocouple, were far too short. Four inches in the 30-in. heater outlet line. As this area of the line was poorly

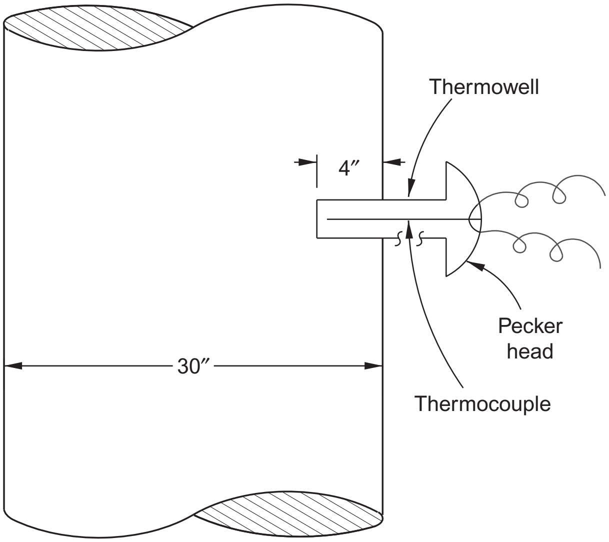

FIG. 36.2 Thermowell too short.

insulated, the indicated temperature was less than the average flowing temperature. During the next turnaround, a 12-in. long thermowell was installed to replace the too short existing thermowell.

I observed a somewhat similar problem at a Visbreaker Soaking Drum in the Coastal Refinery (now owned by Citgo) in Aruba. I was pretty sure that a thermocouple temperature of 810°F was too cold. This temperature was measured halfway up the side of the soaker. The soaker vapor outlet was about 830–840°F. Due to heat of cracking, the lower portion of the soaker must be somewhat hotter. Also, the visbreaker fired heater outlet was 860–870°F. Certainly, the midpoint of the soaking drum must be between 840°F and 870°F, not 810°F.

I recall wrapping a roll of fiber glass insulation around the exterior portion (i.e., the pecker head) of the thermowell assembly (Fig. 36.2). Within 10 min, the indicated temperature had increased by 20–30°F. I had seen this problem before in the overhead vapor line on a Delayed Coker in Texas City. The interior portion of the thermowell was coated with a thick layer of coke. Heat radiation from the exterior of the assembly slightly cooled the thermocouple junction inside the thermowell.

Level

In the old days we measured liquid levels directly with an internal float. I’ve seen just two of these archaic installations in the past 20 years. In a modern refinery, we do not attempt to measure liquid levels directly. We attempt to infer a level based on a differential pressure measurement. Usually, this differential pressure measurement is correct. The problem is that the differential pressure is proportional to both level and fluid density. In the real world, which the process operator and unit engineer must work in, the specific gravity or density of the liquid in a vessel is largely unknown and unknowable for four reasons:

1. Density varies with temperature. For each 100°F increase in temperature, the specific gravity of naphtha is reduced by 5%. For water by 2%.
2. Density varies a lot with composition. For example, heavy naphtha may be 10% denser than light naphtha. This fact was the fundamental cause of the death and disaster at the BP Raffinate Splitter explosion in my former home, Texas City, in 2006.
3. Three feet of water will create the same head pressure as 4½ ft of naphtha. Therefore, a three foot level of water in a level glass may represent a 4½ foot level in a vessel.
4. Almost all process vessels in hydrocarbon service have a layer of foam floating on the liquid in a tower. I have observed foam densities (using radiation) in refinery process vessels, varying from 10 to 45 pounds per cubic foot. The level indicator (which is really just a differential pressure transmitter) is calibrated by the Instrument Technician, based on a foam-free hydrocarbon density.

One thing is for certain. Once the real level in a vessel, whether foam or hot naphtha, rises above the top level tap, no further increase in the indicated vessel liquid level can be expected. The BP operators in Texas City did not know this fact, according to their testimony, when they sued BP for wrongful employment termination after that horrendous explosion and fire at the BP Raffinate Splitter.

Flows

I am just returning, as I write these words, from a visit to the Repsol Refinery in Puertollano, Spain. I worked with a group of very competent engineers. Their group leader, Jose Soto, said, "Norm, our off-gas flow meter from our seal drum does not work. It gives us highly erratic flows, that vary for no apparent reason, by a factor of two or three. This makes it very difficult to troubleshoot our poor vacuum which is undermining our refinery profitability. Muy malo!"

I have had this complaint about vacuum tower off-gas flow meters "not working" for 50 years. But this time, I decided to investigate.

Referring to Fig. 36.3, I observed the installation was proper. The orifice plate was installed correctly (i.e., the orifice plate size on the handle was facing upstream). The glycol pots were properly filled and the orifice taps were clear. Ordinarily, it's best to install an orifice meter in a horizontal run of line, but in "gas" flow, it's not particularly important.

The first thing that I observed was that valve "A" was closed. Valve "C" was also closed, which is correct. Valve "B" was open, which is also correct. But, why was valve "A" closed?

So, I opened it. The instrument technicians purged out the taps and refilled the glycol pots—just to make sure everything was in proper working order. The indicated flow remained totally erratic and very high.

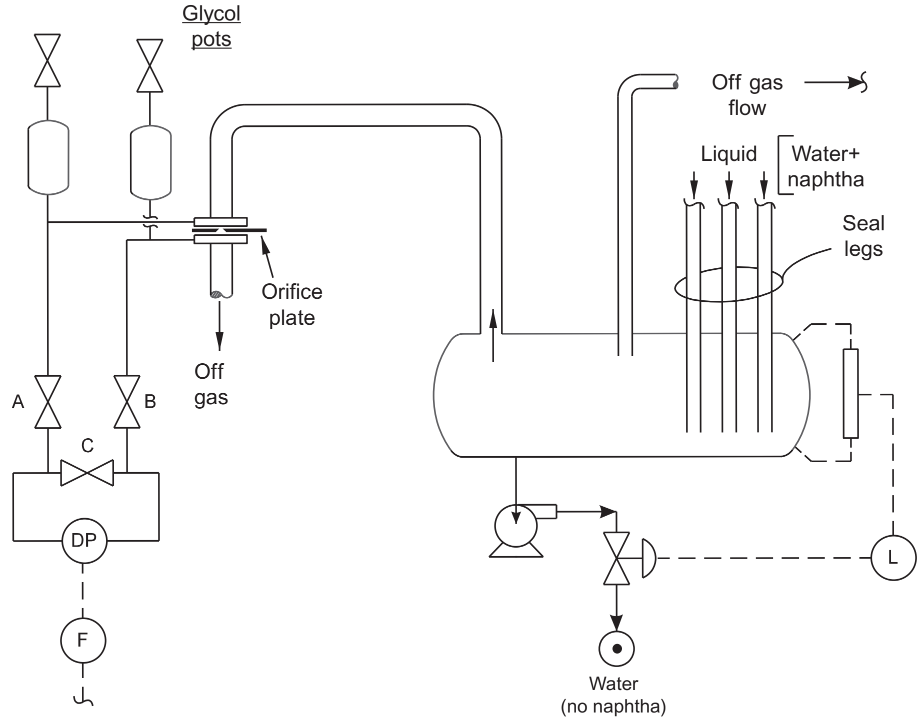

FIG. 36.3 Poor interface level control degraded the off-gas flow meter performance.

Jose said, "Muy malo, señor Norm. It's always like that. It makes no sense."

"But Jose," I asked, "What is the naphtha content of the water from the seal drum?"

"Nada!" (None).

"But Jose," I asked, "What is the rate of naphtha production?"

"Nada. We have no draw-off for the naphtha."

"But Jose," I asked, "where then is the naphtha going that drains into the seal drum with the water from the seal legs?"

"Ah Norman. It must be flowing out with the off-gas. This increases the density and DP of the gas passing through the flow meter. That's why our gas meter reads so erratic and so high all the time. It's not a flow instrument problem at all. It's that we have no indication of the interface level between the naphtha and the water phases in the seal drum. And, no way to drain the naphtha liquid phase out of the drum. It's a process design error. Do things like this happen often in America as well?"

"Sure. How do you think I made enough money to buy a house with seven bathrooms and a swimming pool?"

No orifice flow meter can tolerate a variable vapor-liquid flow and produce a reasonable output. It's not the velocity that varies, but the density that is too variable.

Orifices (contrary to Fig. 36.3) are 99% of the time located in a horizontal run of piping. The orifice taps should be located above the orifice flanges, to retard fouling and water accumulation in the glycol filled lines. Not underneath the orifice flanges—like on many process units.

Samples

My friend Steve has finally earned his law degree. He's now practicing law in Beaumont and gotten out of engineering. Here are three of Steve's mistakes made when obtaining samples:

- Mistake #1—Steve needed a crude unit naphtha sample to design a new crude naphtha debutanizer. The crude tower reflux drum operated at 10 psig and 130°F. He obtained the sample himself in a quart glass bottle. By the time he delivered the sample to the lab, most of the ethane and propane in the naphtha had evaporated. His computer simulation indicated he required a debutanizer pressure of 80 psig. In reality, he needed about 130 psig to condense the light ends. Steve should have gotten a bomb sample, with the bomb completely filled with liquid.
- Mistake #2—Steve needed a wet gas sample from the same drum to size a new centrifugal compressor. He obtained the sample in a bomb. By the time he delivered the sample to the lab, most of the pentanes and hexanes had condensed. He computed the design molecular weight for his new compressor at 38 MW. Had he included the condensed naphtha components, the molecular weight would have been 52. As a result, the motor driver on the new compressor was undersized and tripped out on high amps. Steve should have had the sample heated in the lab prior to passing it through the gas chromatograph for analysis.

  In both cases, my friend should have checked that his samples were at their calculated bubble point (naphtha) or calculated dew point (wet gas).
- Mistake #3—Steve asked an operator to obtain a sample of vacuum tower residue at 450°F in a sample can. The operator dropped the can and the resid auto-ignited. The auto-ignition temperature of hot tar is 320°F, or even lower. We're not allowed to get samples above their auto-ignition temperature in open containers. Incidentally, the auto-ignition temperature of gasoline is typically 450°F.

Steve will make a wonderful lawyer. He will probably get rich, especially with his process engineering background. I'm sure many of my clients could profit from his legal services. He's already billing at a higher hourly rate than I do. He will likely purchase a home with eight bathrooms and buy his wife a Mercedes.

37

Cleaning Process Equipment

There are five types of process equipment that are cleaned on a routine basis:

1. Distillation towers: both packed and trayed.
2. Heat exchangers: shell and tube sides.
3. Fired heaters: tube side—coke removal.
4. Air coolers: exterior finned tubes (this also applies to the convective tube bank of fired heaters).
5. Centrifugal compressor rotor wheels.

I plan to discuss both on-line and off-line cleaning techniques based on my experience, without reference to the large amount of published claims from a myriad of vendors, who offer chemical cleaning services.

Packed Distillation Columns

Towers that are packed with beds of dumped type packing (rings or saddles) are difficult to clean, based on my observations after restreaming. For example, I had an 8′–6″ ID scrubber, packed with 3″ aluminum Pall rings. During a turnaround, I decided to wash off the water soluble sulfate salts, as the scrubber had developed a tendency to flood in the past year. I had the 24″ top manway removed and the top of the 25 ft high bed washed down with a 3″ fire hose.

When we started back-up, scrubbing efficiency was intolerably poor. We could not absorb the SO_3 vapors with the weak sulfuric acid circulating solution. A terrible plume of sulfuric acid escaped into the humid air over Texas City. Even my wife called from home to complain about the dense white acid plume. A shutdown was the only possible response.

This time, after water washing from *"The Top-Down,"* I entered the tower to inspect the rings. The packing was quite clean on the top few feet of the bed. But, as we excavated the packing, removing the rings in 5-gal buckets from the tower, I could see the problem.

Channeling! That became progressively worse as we dug down into the 25 ft deep bed. Obviously, the vapor flow had followed the path of least resistance and flowed through the areas of clean rings and compromised the vapor-liquid contacting efficiency.

I had each bucket of rings dumped onto the ground and carefully water washed. After we had a pile of very clean rings, they were reloaded back in the tower via buckets. But, all this handling had deformed many of the thin gauge aluminum rings. This reduced the open area of the rings. So, when we started back-up, the scrubber flooded, because the liquid phase could not flow easily through the crushed aluminum packing.

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Washing Packed Towers From the Bottom-Up

In 1985 I was working in the natural gas production fields south of Laredo, Texas. My problem one day was a 3′–0″ ID glycol dehydration absorber tower was flooding. Natural gas is saturated with moisture. It has to be chemically dried with glycol, a desiccant, to a few ppm of water, before being transported via pipeline. The absorber was packed with 2″ raschig rings. From experience, I had learned that such towers flood due to acid-soluble corrosion products such as iron sulfide. So, I decided to order a truck, equipped with a circulation pump, to meet me out in the desert at 9:00 p.m. It was July, and daytime temperatures at the well site had been 110–115°F. Best to do the job in the cool of the evening.

The acid I ordered was 10% inhibited hydrochloric acid (HCl). Currently (2019), I would use either Citric Acid followed by a neutralizer or a "ZYME" solution, as HCl is very corrosive.

The truck of HCl was waiting for me at the well site when I drove up. The driver was already hooking up his acid circulation lines to circulate top to bottom. Remembering my fiasco in Texas City a decade ago, I stopped him.

"Amigo. Hook up the return hose to the top 3″ nozzle. Connect the fresh acid at the 4″ bottom nozzle. We've already washed the tower out with clean water. Comprendo? We're going to circulate from the bottom to the top until midnight."

"But Señor, we'll have to fill the entire 30 ft. of the tower with our acid. Maybe my truck not have enough acid? Maybe it go empty?"

"Maybe you're right. Okay. Let's order another truck of acid, pronto."

That's the problem with circulating from the bottom-up, rather than from the top-down. The entire packed bed must be filled with the cleaning solution. It's expensive, but it sure worked that night in Laredo. Except, I was frightened by a three foot long rattlesnake curled around the base of the absorber.

To summarize, the smaller the diameter of the tower, the more practical it is to circulate from the bottom-up. For a 46 ft ID × 110 ft T-T vacuum tower in Saudi Arabia, this is not all that practical.

Trayed Towers

For towers equipped with ancient "Bubble Cap" trays, chemical cleaning is not possible. Each cap has to be unbolted, and the space between the chimney and the cap cleaned manually. I have tried to clean such trays by skipping this step with very negative results. That is, the tower flooded on start-up.

For perforated tray decks such as:

- Sieve
- Valve
- Grid
- Duplex
- Tabs

Circulating the cleaning solution from the top-down, may have mixed results, depending on the tower diameter, on the circulation rate, and the number, size, and distribution pattern of the holes. Other important variables are the number of trays and the length of the circulation time.

My experience with cleaning trays by circulation from the top-down is mixed. Usually, but not always, the tower internal inspection shows reasonably clean trays. But, on other occasions, the results are quite disappointing.

Cleaning Heat Exchangers Tube Side

To clean the tube side of a heat exchanger, assuming the heating fluid is on the shell side, I proceed as follows (see Fig. 37.1):

- *Step 1*—Close tube side inlet valve.
- *Step 2*—Open the tube side bypass valve.
- *Step 3*—Open the small flushing fluid valve.
- *Step 4*—Wait 10–30 min.
- *Step 5*—Reverse steps 1–3.

The tube side fouling deposits will likely spall off due to the thermal expansion of the tubes. Or, dissolve due to the flushing fluid. Or, melt off. Or, be forced out of the tubes due to the sudden reintroduction of the tube side flow.

I used this technique which I called “On-line Spalling,” in 1969 on the world’s largest crude unit in Whiting, Indiana, with excellent results.

For off-line cleaning of the tube side of an exchanger, tubes are simply hydroblasted. This is simple for straight tubes, but can be tricky for a “*U*” tube bundle.

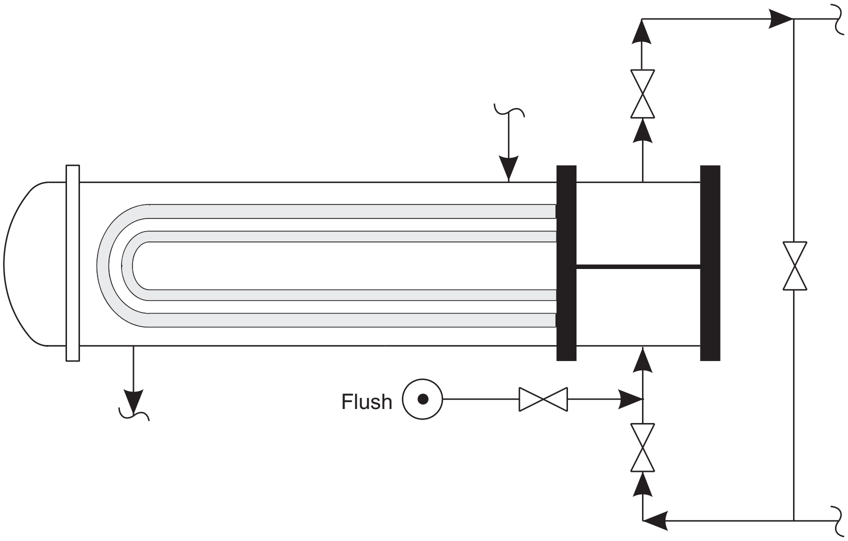

FIG. 37.1 Shell and tube heat exchanger with “*U*” tube bundle.

Hydroblasting "*U*" Tube Bundles

I was once watching a contract maintenance worker hydroblast a "*U*" tube bundle that I had asked to be cleaned. For such a bundle, there is no visual way to see if the return bend has actually been cleared. One has to observe the water flow exiting from the return side of the "*U*" tube.

In this instance, I noted that on 20%–30% of the tubes, only a small trickle of water appeared from the return side. So, I asked the maintenance guy how he was going to clear the plugged return bends of the "*U*" tube bundle.

"Mr. Lieberman, I got to use a special tool on plugged tubes to force water through those bad tubes," he explained. "The problem is the return bend."

"Okay. But how do you remember which of the 500 tubes are plugged?"

"Yeah! I got a photographic memory. I just kinda' remember which is plugged, so I can clear them altogether later on."

I guess there are two possibilities. Either that guy is a mental giant. Or, he had no intention of clearing the plugged tubes. This is a common problem. How does one inspect a "*U*" tube bundle to verify that it has actually been completely cleaned? I would just, when I worked in Texas City, blow air from an air hose through each tube, and see if the air would blow out the end of the return tube. For my 60″ ID alky contactor bundles, this could take an hour or more for each of the "*U*" tube bundles.

Acid Cleaning

I was the acid cleaning king of my alky unit in the 1970s. I would clean, on-stream, the water tube side of one of 20 to 30 coolers every week.

- Step one was to attach a truck of 10% inhibited hydrochloric acid to the cooling water inlet.
- Step two was to attach a source of 12%–15% caustic to the water outlet.
- Step three was to turn on the acid to achieve a pH of one to three at the cooling water outlet.
- Step four was to turn on the NaOH at a rate sufficient to maintain a pH of the water return to the plant's cooling tower of 5½–7½ (preferably 6½).

Too high a pH would cause the plant's cooling water circulation to turn brown and murky, due to the precipitation of dissolved iron salts which fouled exchangers. Too low a pH, would cause the plant's cooling water circulation to become corrosive. Leaks would spring out in the carbon steel cooling water piping return header.

Effective acid cleaning required 15–30 min or less. I used to back-flush the water side before acid cleaning. In more recent times, I believe Citric Acid is used, but I have only used HCl because I only had the phone number in Sugarland, Texas, of the company that sold Amoco HCl.

Shell Side Cleaning

Some heat exchangers have a fixed tube sheet configuration. The tube side inlet and the tube side outlet are on opposite sides of the exchanger's shell. This sort of exchanger does not permit the tube bundle to be extracted for cleaning and should never be used in fouling service on the shell side.

I have tried on several occasions to clean the shell side of a fixed tube sheet exchanger by circulating a wide variety of solvents with limited success. Exchanger pressure drop declined a bit, but heat transfer efficiency did not noticeably improve.

Even when a bundle can be extracted from its shell, it may still be largely immune to cleaning if:

- Tube spacing is 1″ or 15/16″
- Tube pitch is triangular
- Tube size is ¾″
- Bundle is large (i.e., 36″–60″ plus ID)

There are no straight channels through such a bundle. At best, only the outer two or three rows are cleanable. At the Amoco Refinery, such bundles were soaked for a week or two in vats of aggressive chloride solvents, which did eventually, I assume, remove the interior deposits.

To avoid these problems, I design new heat exchangers using:

- 1″ tubes
- 1½ tube spacing
- Rotated square pitch

This configuration provides a ½″ straight passage between the rows of tubes in all directions. Of course, this also diminishes exchanger surface area by 30%–50% for a given shell diameter. But, that's the price we pay for the ability to clean the shell side.

Air Coolers

Having cleaned three forced draft air coolers myself, I have a pretty good feel for how best to clean the exterior finned tubes to restore the cooling air flow. Washing from the top-down is largely ineffective. This happens any time it rains. Anyway, most of the dirt is between the fins on the lower two rows of fins. I suggest you proceed as follows:

- *Step 1*—Drop off the bottom protective screen, and with a broom, sweep the underside of the fins.
- *Step 2*—Assuming the fan is driven by a 3-phase AC motor (240 V), reverse the polarity of the leads, and the fan will run backwards and blow some of the dead moths and willow seeds off of the finned tubes.

- *Step 3*—Soak the fins in a detergent for a few hours. Next, secure a source of clean water with a pressure of 30–40 psig at the fan elevation. Using a ½″ tube, jet the water from a distance of 12″–18″, between the bottom row of tubes. Caution: most fins are aluminum and are easily deformed.

I always electrically lockout the fan motor myself and keep the key. Also, I tie off the fan blade with a rope. It requires at least 1 or 2 h to wash a single tube bundle. Note that cleaning the exterior of the tube bundle will not reduce the amp load on the fan motor. That requires reducing the fan blade angle. Maximum is 23 degrees; minimum is 10 degrees.

Slug Washing Air Cooler Tubes

On an 80,000 BSD Delayed Coker in Los Angeles, I had to improve the performance of the fractionator overhead condenser air coolers. There were 32 of these coolers arranged in parallel. I decided to water wash the eight worst performing tube bundles. I could only do one at a time (without stopping process flow to coolers being washed) due to the limited ability to handle the effluent waste water from the washing. Also, I only had 1 day before I had to fly home. Each cooler would take an hour to wash, including making hose connections. The trick was to select the tube bundles that were most fouled or plugged on the tube side.

To identify these coolers, I checked for those that had the lowest process outlet temperature, which was a possible indication of low flow. Of these, I then checked the air outlet temperature. Those exchangers having both a low air outlet, plus a low process outlet temperature, were the ones most badly fouled on the tube side and required slug washing.

After each bundle was washed, both the air and process outlet temperatures increased, due to the restoration of the process flow rate and heat duty.

Fired Heater Convective Tubes

I would use the same technique for cleaning the studded or finned convective tube bank, if the tubes are accessible from the top of the radiant section. Depending on the fuel being fired, most of these deposits may be vanadium salts from fuel oil, rather than soot. Vanadium deposits can be quite thick (1/4 plus) and form a whitish, smooth deposit, at least on bare tubes.

Unfortunately, only too often, the fins in the convective section have been oxidized due to an after-burn situation. The fins will appear rusty and you can break them off with your fingers. Hydroblasting off the rusty fins is the only option.

Coke Removal From the Inside of Radiant Tubes

The modern, correct way, to remove coke from the interior of radiant tubes is a combination of two methods:

- Pigging
- Burning

“Pigging” is done by forcing a “Pig” through the furnace tube with high pressure water. As I write these words, I am holding such a pig in my hands. It’s about 5″ wide and 8″ long. It was intended to remove the coke from the inside of a furnace tube with an ID of 6″. It is made of hard plastic type material with dozens of metal studs protruding from its surface by ¼″ or so. When pigging a heater tube, progressively larger diameter pigs are used until one just a little smaller than the actual tube ID is used.

I was recently working for a refinery in Detroit on a relatively new Delayed Coker. They had just restreamed a heater after pigging. The heater performance (i.e., the coil pressure drop) indicated that the coke removal had been incomplete. This is normal. Coke removal from the inside of furnace tubes is almost never 100% from pigging alone.

I advised the plant operators to burn out the residual coke using the older “Steam-Air Decoking Procedure.” A mixture of a few percent air dispersed in 150–500 psig steam is injected into the furnace tubes, which have been heated to 1000–1200°F. The residual coke inside the tubes will burn off. CO_2 in the heater effluent is used as a guide to monitor residual coke. When there is no CO_2 in the heater effluent, all coke has been removed. I call this step, “Proof Burning.”

My Detroit client could not follow my advice, because their new unit did not have an environmental permit from the city of Detroit for the emissions resulting from the proof burning step.

Spalling and Hydroblasting

In the 1930s through 1960s, furnace heater tubes were cleared of coke by removal of boxes (called mule-ears) at the end of each pair of tubes. The tubes were then hydroblasted clear of coke. Very few heaters still have this expensive feature, which has been replaced with “*U*” bends, which are welded in place and are not removable.

Hydroblasting was replaced with spalling in the 1960s. Hot tubes were exposed to varying steam flows and temperatures, which caused the coke to spall-off the tube inner diameter due to the change with temperature of the tube ID. The steam would then blow the coke out of the tube. A small amount of air was periodically added to the steam to burn out any residual coke. Ninety percent plus of the coke was spalled out with only a few percent burned. I have always considered this older method to be superior to modern pigging.

Cleaning Heater Tubes’ Exteriors

When firing gas, there are no deposits to clean on the outside tube surface. When burning Industrial Fuel Oil (i.e., No. 6 oil or Bunker Fuel), a ¼″ whitish deposit will accumulate on the tube OD. This deposit is a Eutectic Mixture of 90% plus vanadium and a few percent of sodium. The vanadium by itself does not inhibit heat transfer to any appreciable extent and does no harm. It just makes the tube glow and appear to be overheating.

However, the presence of several percent of sodium forms a corrosive eutectic with the vanadium. At about 1250°F, the mixture liquifies, runs very, very slowly down the tubes, and causes exterior grooves to be cut into the tubes. When the heater is opened, these deposits should be hydroblasted off of the tube's exterior surfaces.

Compressor Rotor

I became an expert on cleaning 10,000 BHP multistage, centrifugal compressor, variable speed rotors in 1987 in Laredo, Texas. The fouling material was a hard, dark gray deposit consisting mainly of:

- Salts
- Drilling mud
- Pipeline corrosion products

These solids were slowly reducing my compressors' capacity (Solar-Centar 10,000 hp gas turbine-driven machines) by fouling of the rotor. From experience, I had learned that if I waited too long, these deposits would break off the rotor. The rotor would become unbalanced and the machine would start to experience excessive vibrations. Then, my on-line cleaning method, which depended on running the machine at a moderate rpm could not be used.

My method was "Nut Blasting." An 18″ × 36″ inspection door was removed and the machine spun at about ¼–½ its normal rpm. Using a sugar scoop, I would then slowly add coarse Pecan nut hulls to the inlet of the first compressor's wheel. After a while (20 min), I would switch to progressively finer nut hulls. This treatment, which would require several hours, was highly effective, and fun. The trick was not to wait until the deposits became too thick, broke off, and unbalanced the rotor's wheels.

Summary

I have limited my description to process equipment cleaning to procedures I have actually done myself or personally supervised. Until I have performed some procedure myself in the field, I really do not understand or remember exactly what to do. But once I have personally participated in some activity, I remember it in detail for the rest of my life. It's like learning how to ride a bike. For that reason, I always try to do things like cleaning process equipment with my own hands at least one time. I tried to indoctrinate the young engineers working for me at the Good Hope Refinery with this philosophy, to which they responded with enthusiasm.

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Learning Process Engineering Principles

Most of what I have needed to learn to achieve my objective of becoming an effective process engineer, I did not acquire in University or Graduate School. I did not learn very much about process engineering employed as a design engineer in Chicago for 16 years. My foundation of process knowledge is built on the times I worked as an operator during the two long strikes in the American Oil Refinery in Texas City in 1974 and 1980. The strikes lasted 4–5 months. As a salaried employee, I worked as a scab. Twelve hour days. Seven days a week. In 1974, on the Sulfuric Acid Alkylation Unit. In 1980, on the Sulfur Recovery Unit that I had previously designed. Had it not been for this opportunity to observe, operate, and repair pumps, turbines, reactors, heat exchangers, and distillation towers, I would not have developed the process insight to the problems that I have today.

The other major source of my process knowledge is derived from childhood experiences. I enjoyed being a child. I continue to regret and resist growing up into an adult. As a child I had an inordinate fascination in such phenomena as:

1. How did the air lift filter pump in my aquarium circulate water?
2. What was the function of the small conical vent on top of our steam radiator that heated our fifth floor apartment?
3. When my mom poured Coke over the rust stains in our bathroom sink, why did the rust dissolve?
4. Why did a drop of water on a very hot, smooth frying pan dance around and not evaporate, while water on a cooler, scratched pan, would instantly boil away?
5. When it was windy outside, why did the water level in our toilet bowl oscillate?
6. Why did water come out of the kitchen tap cloudy, but clear up after standing for 5 min?
7. Why, when my dad opened a warm bottle of beer too fast, did the beer explode violently out of the bottle?

Now I can correlate these childhood observations with refinery process technology:

1. *Air lift pump*—Similar to thermosyphon circulation in reboilers and steam generators.
2. *Radiator vent*—A noncondensable vent installed below the lower pass partition baffle on the channel head of steam shell and tube heaters.

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3. *Removing rust stains with coke*—This is an example of carbonic acid corrosion of iron due to CO_2 contamination of steam, due to the decomposition of carbonates in boiler feed water.
4. *Water on a hot pan*—Same as a heat flux limitation in a reboiler with smooth tubes and high pressure steam.
5. *Erratic toilet water level*—Caused by the draft that wind develops as it blows across the top of a fired heater's stack. Typically, 0.1–0.2 in. of water.
6. *Cloudy tap water*—Air dissolved in water in higher pressure and cooler temperature.
7. *Beer bottle foam-over*—Sudden reduction in a distillation tower pressure causes the tower to flood and carry-over.

Function of a Process Engineer

I quit American Oil in 1980 and went to work as the Technical Service Manager at the Good Hope Refinery. I had a dozen young chemical engineers in my group. They were bright boys, but like me, had not learned much in University that would aid them in their job (Note: In 1980, very few women studied Chemical Engineering. Now it's about 50/50.).

The first thing I explained to the boys as they began work was that our principal job was to solve problems in the plant. Our function was not to:

- Write reports
- Attend meetings
- Interact with vendors
- Do environmental impact studies
- Run computer models
- Provide data for the planning department
- Have lunch with chemical vendors
- Evaluate new refining technology
- Participate in seminars in New Orleans

Our job was to solve plant problems that the operators could not solve themselves.

How to Learn

I didn't learn to drive sitting at a desk in high school. My dad taught me in our brown 1959 Oldsmobile. The operators in our plant were actually "trained on the job." They started working in parallel with an older guy. They followed him around for a few weeks. Then, the experienced operator followed and observed the newer operator for another week to correct any of his or her mistakes.

It's really no different when learning any skill. Most everyone is like me. I can only learn while doing. It's the ancient apprentice concept.

When a new process engineer started work at the Good Hope Refinery in my group, I would give him his first assignment.

"Reid, we're short of reboiler duty on our Crude Unit gasoline debutanizer. Ask Mr. Walker, the area supervisor, to explain the problem to you as he understands it. He will show you where the reboiler is located."

Reid Burt came back to my office the following morning, "Mr. Lieberman, Mr. Walker showed me the reboiler. He says the problem is he can't get enough 150# steam flow to the reboiler. He used to be able to get 14,000 pounds an hour flow, but now all he can get is 9,000. He showed me that the steam inlet control valve is wide open. I made a sketch of the reboiler, but I don't know what else to do? We didn't study steam reboilers at LSU. Should I try to model the debutanizer on the Aspen Simulation Program?"

And I would say, "Reid. Let me get my tools and my bag of pipe fittings. We'll go out together and look the job over in the field. Go get your gloves and hard hat."

Reid and I would then measure the pressures and skin temperatures in the 3″ steam condensate drain line and in the 6″ condensate collection header. And I would explain, as we made the measurements together, the cause of the low steam flow.

"Reid, water is backing up in the reboiler channel head. The problem is vapor lock. The excessive pressure drop in the condensate drain line is causing the water to partly flash back to steam. The evolved steam restricts the flow of water, which backs-up over the tubes in the reboiler. This reduces the heat transfer surface area and reboiler duty."

From this exercise, Reid would learn how to make field pressure and temperature measurements. He would also learn the technical concept of the effect of condensate back-up. I would explain the design error. That is, the omission of a condensate pump to prevent excessive steam evolution in the condensate flow in the 3″ line.

Interfacing With Operators

Up until the 20th century, most technical knowledge was transmitted not through institutions or books, but by the apprentice system. My example cited before, when I said, "Let me get my tools and we'll go out to the reboiler together," was an example of this ancient instructional technique.

Another technique of comparable value is to interface with the plant operators. Especially when it relates to controls (level, flow, temperature, pressure). Working with operating personnel is the best way to understand how control loops and control valves work, and their wide range of malfunctions. Especially, how control loop design interacts with process problems and occasionally creates a "Positive Feedback Loop."

For example, I had no idea of how instrument air is connected to a control valve diaphragm, until it was explained to me by an older operator on my Alky Unit in Texas City in 1975.

Another example of learning from operators is the need for "Starting NPSH" for centrifugal pumps. A concept that I've discussed in detail in this text. Operators all understand the need to increase flow from the discharge of a centrifugal pump slowly, to avoid cavitation. From this observation, I have devised the numerical method for calculating, for design purposes, the starting NPSH requirement for a centrifugal pump. This is the extra elevation of a vessel needed to accelerate the liquid, on start-up, in the pump's suction line.

Learning From Maintenance Personnel

A lot of people, like me, have a problem visualizing a three-dimensional object from a book. I do much better in my comprehension when I can see the actual equipment. A Shell and Tube Heat Exchanger is a good example. I never quite understood how the floating head on such an exchanger was actually attached to the tube sheet using a "Split Ring Assembly," until John Brundrett, the maintenance manager in Texas City, showed it to me during an FCU turnaround.

I had heard a lot about the clearance between the impeller and the pump case, and knew this was important. But until a shop maintenance technician explained this to me, I never really understood it.

I learned how centrifugal compressors work by watching a 10,000 BHP Solar-Centar rotor overhauled in a repair shop in Dallas, while listening to a 2-h monolog from the craftsman cleaning and rebalancing the rotor's three wheels. When I put the compressor back on-line at the Texas, Laredo Natural Gas Compression Station the following week, I had a better understanding as to how the machine actually functioned.

Inspecting Tower Internals

My main field of work as a process design engineer is retrofitting refinery distillation towers and designing new fractionators. I have designed 30″ ID jet fuel steam strippers and a 26′–0″ Delayed Coker Fractionator. I draw the tower internal details to scale on blue lined graph paper at 1 in. per foot. Old process design engineers have a saying, "The Devil is in the Details." As I draw these details, I have a picture in my mind of how this particular feature looked in a tower I had inspected. What had gone wrong with the installation? Are the dimensions I am specifying being adhered to in practice? Have I made a reasonable provision for fouling?

Certainly, I will base the tower ID and the number of fractionation trays on the Aspen computer model output. But, that does not provide insight as to where iron sulfide and salt deposits will require an extra inch of downcomer clearance, or, when extra downcomer bracing brackets are needed.

When I inspect tower internals, I try to understand how the condition of the trays relates to operating problems I have experienced in distillation service. Seeing out-of-level tray decks and slanted weirs correlate with low tray fractionation efficiency, I have learned from the inspections, not to take computer-generated distillation results to necessarily represent the actual distillation tower performance.

Learning From Performance Tests

My favorite way to learn basic process principles is by following a three-step program:

- *Step 1*—Measure a variable in the field, such as the feet of head developed by a pump.
- *Step 2*—From the pump curve, the measured flow, and the density of the fluid, calculate the feet of head that the pump should have been able to develop.
- *Step 3*—Determine the cause of any difference.

I often tell my clients, "I am an expert in refinery distillation technology." But how have I become an expert?

- *Step 1*—I run a computer model of a distillation tower using 80% tray efficiency.
- *Step 2*—Using plant data (i.e., reflux rate, reboiler duty, product analysis), I reduce the tray efficiency in my model to match the observed distillate and bottom products lab analysis.
- *Step 3*—I crawl through the tower to understand why my calculated tray efficiency in Step 2 is only 30%, when the industry standard for this particular service is 80%.

For heat exchangers, I calculate heat transfer coefficient ("*U*") based on observed temperatures and flows. I will then calculate the clean "*U*," based on Heat Transfer Research Institute (HTRI) computer program. Let's say:

$$\text{Observed}\text{``}U\text{''} = 40\,\text{btu/h/ft}^2/^\circ\text{F}$$

$$\text{HTRI Clean}\text{``}U\text{''} = 90\,\text{btu/h/ft}^2/^\circ\text{F}$$

The difference the reader might conclude is fouling. But maybe not. Next, I repeat my field measurements after the heat exchanger is cleaned. The clean observed "*U*" = 60! Now what?

I will then examine the design of the tube bundle for problems such as:

- Excessive shell side clearances.
- Lack of dummy tubes to reduce shell side bypassing.
- Leaking channel head pass partition baffle.
- Tube viscosity higher than I used in my calculations.
- Twenty percent of tubes were plugged as leakers, but the maintenance department did not record this.

Conclusion

Process Engineering is more of a craft than a science. Therefore, instructors who author textbooks and teach engineering classes, who lack field experience in troubleshooting process problems, are ill equipped to fulfill their objective. That objective being to prepare young men and women to understand and solve process plant problems. Unless an engineer understands process equipment malfunctions, especially process control problems, he or she is not ready to design new equipment or function as a plant supervisor.

Modern use of computer technology is neither harmful nor beneficial. It is largely irrelevant to the learning experiences that you and I must have to efficiently operate and design heat exchangers and fractionators.

Although I'm getting old, my rate of learning the craft of process engineering has not diminished with the passage of the years. I learn something new; something exciting and interesting, with every troubleshooting and retrofit design project. As a Process Engineering Division Manager, I also learned something of great importance and of growing relevance. That being, to say to the young engineer who reported to me, "Let me get my tools, and we'll go out into the plant together and work on the problem. Pick up that bag of ¾″ fittings I left in Tommy's office and the new digital vacuum gauge from Bobby."

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Approximate Process Calculation Methods

The ability to do quick, approximate calculations, I have found to be of great aid in understanding how process equipment works. The correlations and data below are intended for speed and simplicity, rather than for accuracy.

Units

V = Velocity, feet per second
D_V = Density vapor, lbs/ft^3
D_L = Density liquid, lbs/ft^3
DP = Pressure drop, psi
D_H = Pressure drop, inches of liquid
PA = Absolute pressure, psia
TA = °F + 460 (i.e., Rankine) (°C + 273 (i.e., Kelvin))
L = Latent heat, btu/lb
SP = Specific heat, btu/lb/°F
DT = Temperature difference, °F
A = Area, ft^2
ID = Pipe inner diameter, inches
H = Height change, ft
(GPM/in) = US Gallons per minute per inch
btu = British Thermal Units
MW = Pounds per mole

I do not use "°C." This stands for Degrees of Communism. I always use "°F," which stands for Degrees of Freedom. For those who have forgotten:

$$0^{\circ}\mathrm{C} = 273\,\mathrm{K}\ (\text{Kelvin})$$

$$0^{\circ}\mathrm{F} = 459^{\circ}\mathrm{R}\ (\text{Rankine})$$

Also, of general interest:

- Temperature in outer space = 3 K
- Adiabatic flame temperature for gas with little excess air = 3400°F
- Water reaches its max density at 4°C and gets lighter after freezing.

https://doi.org/10.1016/B978-0-12-816161-6.00039-4

- Jet engine (i.e., gas turbine) exhaust temperature is typically 1000°F and gets lower as efficiency increases.

Pressure Drop

(see above for term definitions)

$$\text{Thru a hole} = D_H = 0.6\,(V)^2$$

$$\text{Thru a nozzle} = D_H = 0.3\,(V)^2$$

$$\text{100 ft of pipe (Vapor flow)} = \text{DP} = \frac{D_V}{\text{ID}} \cdot \frac{V^2}{400}$$

$$\text{100 ft of pipe (Liquid flow)} = D_L = \frac{D_L}{\text{ID}} \cdot \frac{V^2}{400}$$

$$\text{Change in Elevation} = \text{DP} = \frac{D_H \cdot D_L}{1800}$$

Crest height $= D_H = 0.4\,(\text{GPM/in.})^{0.67}$ (That is, height of the liquid over the top of a weir or of a dam).

Heat

- (ATM Pres (L) – Steam) = 1000 – Latent Heat Per Pound
- (500 psig (L) – Steam) = 800 – Latent Heat Per Pound
- Alcohol (L) = 400
  - (L) Naphtha = 120 – Latent Heat Per Pound
  - (L) Gas Oil = 90 – Latent Heat Per Pound
- SP Hydrocarbon Vapor = 0.6 BTU/LB/°F
- SP Hydrocarbon Liquid (100°F) = 0.5 BTU/LB/°F
- SP Hydrocarbon Liquid (600°F) = 0.7 BTU/LB/°F
- SP Steam = 0.5 BTU/LB/°F
- SP Water = 1.0 BTU/LB/°F
- One Horsepower = 2500 btu
- One KWH = 3100 btu
- 770ft-lbs = 1.0 btu
- Combust 1.0 lb Hydrocarbon = 18,000 btu
- Combust 1.0 lb Hydrogen = 40,000 btu

Common Molecular Weights

Hydrogen = 2
Natural Gas = 20

Cracked Gas = 30
LPG = 50
Light Naphtha = 80
Heavy Naphtha = 120
Kerosene and Jet = 160
Diesel = 200
Gas Oil and Lubes = 280
Air and Nitrogen = 29
Steam = 18
Sulfur = 32
Ammonia = 17

Volume and Weights

$$+100^{\circ}\mathrm{F} = \text{plus } 5\%(\text{Hydrocarbon liquid}) - \text{Increase in volume}$$

$$+100^{\circ}\mathrm{F} = \text{plus } 2\%(\text{Water}) - \text{Increase in volume}$$

$$D_{\mathrm{V}} \text{ changes in proportion to } \mathrm{TA}_{(1)} \div \mathrm{TA}_{(2)} \text{ (Volume vapor vs temperature)}$$

$$D_{\mathrm{V}} \text{ changes in proportion to } \mathrm{PA}_{(1)} \div \mathrm{PA}_{(2)} \text{ (Volume vapor vs pressure)}$$

$$\text{One mole gas} = 379\,\mathrm{ft}^3\ @60^{\circ}\mathrm{F} \text{ and atm pressure}$$

Convert BSD to pounds per hour

BSD LPG = 8 pounds/h

BSD naphtha = 10 pounds/h

BSD jet fuel = 11 pounds/h

BSD diesel = 12 pounds/h

BSD gas oil = 13 pounds/h

BSD asphalt = 15 pounds/h

BSD hot water = 14 pounds/h

My main field of activity is field troubleshooting refinery process problems. I cannot resort to calculations with pen and paper, while trying to uncover a problem climbing up ladders or crawling under fin-fan air coolers. Thus, I have to keep the above relationships mentally readily available for scoping calculations while on-site.

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How to Make a Technical Presentation

You may not like Donald Trump, but when he talks, likely you'll listen. On the other hand, think about all the times you have dozed off while Professor Peterson was lecturing about Gibbs Free Energy, entropy, and enthalpy.

The main objective of making a technical presentation is to have your audience listen. The attendees did not buy a ticket to listen to you speak. They are being paid to be there and are possibly not all that interested. Listening to a technically complex presentation is mentally harder work than delivering the presentation. I should know, having both delivered and received thousands of technical presentations in the last 58 years since I was 18.

Here are a few guidelines that will help you from being "A Talking Head."

Norm's Rules

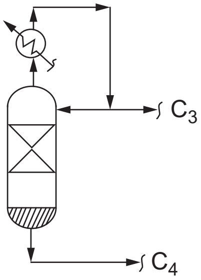
1. Never read your presentation.
2. Never talk while seated.
3. Don't talk behind a lectern. Move around.
4. Interact with your audience. If no one asks questions, ask someone in the group questions.
5. Don't refer to an outline. If you needed an outline, you're not ready to give your talk.
6. Never project columns of data on a PowerPoint presentation.
7. Before your presentation, determine if your visual aids can be easily seen from the back of the room.
8. Before your presentation, determine if you can be heard easily from all portions of the room. Learn to speak LOUD.
9. Presentation should consist of:
    a. 30% silly jokes, human interest stories, and personal reminisces, such as problems with exgirlfriends.
    b. 70% technical material.
10. Pass around visual aids as you talk. Pump impellers and burner tips, for example.
11. Have a name tag with first name in front of every attendee.
12. The most important rule is every visual aid you present must have one—and only one—idea.

Understanding Process Equipment for Operators and Engineers. https://doi.org/10.1016/B978-0-12-816161-6.00040-0

Visual Aids

I attended a PowerPoint presentation pertaining to Stress Corrosion Cracking, delivered by an expert in the field last year in New Orleans. Even though I am very interested in this subject, I felt inclined to kill the speaker after 20 min. Each of his PowerPoint displays had columns of data, or complex sketches, or graphs with a multiplicity of curves. He read his presentation in a monotone and neither asked nor answered any questions. Fortunately, I dozed off after 30 min, but awoke to find the speaker still droning on.

The below figure represents an example of a proper visual aid, which illustrates how the separation I-C_5 and N-C_5 (i.e., pentanes) is aided by reflux. It's hand drawn on a 24″ × 36″ flipchart in bright red, blue, and green colors. The lettering must always be in black. Colors are hard to discern from any distance. Note the "41" in the upper right-hand corner. That's the page number for the handout.



Reflux improves
Separation

Hand Out Preparation

"Hi – I'm Norm Lieberman – your speaker," I say as I greet each person individually, as he or she takes a seat, and I give them their seminar handout, which consists of:

- All the visual aid drawings I will be presenting (i.e., Drawing "41").
- Relevant technical information such as vapor-liquid equilibrium charts, steam enthalpy tables, hydrocarbon vapor pressure curves, which will have too much data to display as I talk.
- My company propaganda stuff.
- Relevant articles I have published in technical journals.

I always have this assembled in a spiral bound book with a clear plastic cover to discourage attendees from discarding my handout. All this is expensive, but I charge it to my clients.

Introducing Your Presentation

Don't start off by introducing yourself or with a joke. A typical start to my seminar is:

"I'm an alcoholic. Hence my interest in making 120 proof whiskey. That is 60% alcohol – 40% water. If I start with wine (13% alcohol – 87% water) and vaporize the wine, the evolved vapor will have 28% alcohol and 72% water. Julie, why does the alcohol concentration in the vapor phase increase as compared to the wine?"

"Because sir, the alcohol exerts a greater vapor pressure than the water. It is more volatile and has a lower boiling point."

"Actually Julie, we could quantify your correct answer by referring to Pg. 9 of our handouts – the vapor pressure vs. temperature of alcohol and for water. Let's all work this calculation through together. However, Julie, I do not appreciate your referring to me as 'Sir.' That makes me sound like an old person. Which I am not. Please call me 'Norm'."

Illustrating Calculations

We all have that terrible picture in our minds of Professor Peterson doing complex calculations on a black board with white chalk in a dimly lit classroom. The first thing I do when I walk into the room where I will be speaking is to check the lighting. I want a brightly lit room to keep everyone alert and to be able to readily see my technical calculations. Of course, I need to arrive an hour before the attendees to get the fluorescent light bulbs replaced.

"Julie," I will say, "kindly illustrate your alcohol-water vapor pressure calculation for the class." I will then hand her a black (never red or purple) marker. "Julie, take your time. Write really big and explain your calculations as you proceed." The attendees will likely know each other and having a coworker explain a problem will help hold their interest.

The Physical Setup

As a presenter, you are in charge. Get to the room early. If you are making a presentation to a group of senior plant operators, make sure there is plenty of coffee. Is there enough seating for everyone's comfort? Can everybody see your drawings from where they are seated?

Is the room temperature control adequate? Is there background noise from air conditioner fans, lawn mowers, construction sites, meetings in adjacent rooms? Better to delay your presentation than to be handicapped by factors beyond one's control.

Can you see everyone's name tag (item #10 above) from where you will be talking? Again, check the lighting. Dim is good for your audience, but bright is what you want from where you will be talking. For a large room, I'll ask an early arrived attendee, "Julie, can you hear me okay from back there?"

Assumptions

For experienced plant operators, I will assume they know what a pump, heat exchanger, or distillation tower looks like, but not the underlying technical principles. For engineers, I will assume that they have a slight familiarity with the basic technical principles, but do not necessarily know what the process components consist of. For plant management, I will assume they know nothing and have a limited attention span.

In general, making a technical presentation to a group of experienced plant operators, pertaining to the process unit they operate, is relatively quite easy.

The level of technical competence of process engineers, operating management, and plant operators is not all that different. I tend to ignore the difference and treat everyone the same in regards to my expectations of their understanding the technical portions of my presentation.

Presenting Equations

For the speaker, presenting equations is difficult. Many attendees routinely "tune out" as soon as an equation appears. It's a response to their experiences in University.

I also have to present equations. But, I try to explain each term by painting a verbal picture of their meaning. For example:

$$\%\text{ of Jet Flood} = \frac{(\text{DP}) \cdot (28)}{(\text{SG})\ (\text{NT})\ (\text{TS})} \cdot (100)$$

- DP is the pressure drop in psi, which you measure by checking the pressures with a gauge, at top and bottom of the tower.
- TS is space between trays, shown on vessel sketch. Typically, 18″ or 24″ or 30″.
- NT is number of trays.
- SG is the specific gravity of the hot liquid on the trays. Typically 1.00 for water or 0.72 for gasoline.
- The "28" factor converts psi to inches of water.

Defining each term verbally, followed by an illustrative example, allows attendees to visualize what the equation represents in the real world, rather than just an abstraction. As I talk, I will reference a large drawing of a distillation tray showing, for example, tray spacing.

The 30% Silly Time

I have spent 25,000 class hours teaching technical seminars since 1983. I have found by trial and error that people need to rest their minds every 20–30 min. Thus, I have a series of stories of a semitechnical nature, one of which I'll share with you.

“Let me tell you,” I start, “by explaining the difference between an Isentropic Expansion and an Isenthalpic Expansion. An Isenthalpic expansion of 400# steam from a 4″ pipe to 10# steam, by flowing into a 16″ pipe, maintains the same velocity of the steam. As it’s the velocity of the steam, or its kinetic energy, that spins a turbine wheel, the ability of the 400# steam to do work has been degraded, but its heat content or enthalpy has been maintained.

“It’s like taking a girl out on a date to an expensive restaurant, where she orders the most expensive item – Asparagus that are out of season. When you take her home, she says, ‘I had a wonderful time; good night.’”

“What have you accomplished? You have wasted a lot of time and money.

“An Isentropic Expansion of 400# steam, allows the steam to expand from the 4″ pipe to 10# steam through a half inch nozzle (i.e., to 11,000 ft/s). Almost all of this large increase in kinetic energy comes from the heat or enthalpy of the steam. The steam’s high velocity can now be used to spin a turbine wheel. That is, an Isentropic Expansion preserves the ability of the steam to do work.

“It’s like taking a girl out on a date to an expensive restaurant when she orders the most expensive item—Asparagus out of season. When you take her home, she says, ‘I had a wonderful time; would you like to come upstairs for a coffee?’”

“Now you’ve accomplished something! ‘A little bit of sugar helps the medicine go down.’”

Kumar, an Indian attendee at one of my seminars, says he’s learned more about thermodynamics from this story, then he had at University from Professor Peterson in a year.

Asking Questions

Never dodge a question. If I know the answer, I will interrupt my presentation to reply in great detail. Questions are the best indication that your presentation is going well. If I don’t know the answer, I will formulate a response during a break, and then during the next session, provide my answer. Or you can always say what I do sometimes, “Julie, good question. I’ve had that same doubt as you have for 45 years, and I still don’t understand it.”

Asking questions of the attendees serves a dual purpose:

- It keeps folks alert and interested.
- It allows the speaker to gauge the level of understanding of the attendees.

Asking question though is hazardous. Five percent or so of attendees will react with hostility and resentment if they don’t know the answer. I’ve seen this from the class evaluation forms I get as feedback from my clients. It doesn’t seem to correlate with age, gender, race, or national origin. I have found though that it’s much less likely to initiate resentment from a really smart operator, manager, or engineer, than it is from a marginally competent attendee. So if you know your group, I suggest you restrict questions to the technically most knowledgeable folks.

Sexist Comments

I have an infrared thermometer to measure the skin temperature of vessels. To illustrate its use, I will select a male and female sitting next to each other. The guy's skin temperature might be 97°F and the lady's skin temperature might be 96°F.

Then I explain that women, in general, use face cream, which acts as a barrier to heat transfer and suppresses their external skin temperature. That we need to take exterior fouling on vessels into account when conducting a plant temperature survey using skin temperature data. This is okay.

But my comment to a young, married couple, showing the same 1°F discrepancy, that this was an indication that they were thermally incompatible, was not appropriate.

Cell Phones

I always ask all attendees to put their cell phones out of sight. If the cell phone is visible, the temptation to check emails and text a response is overwhelming. The best way to handle this problem is to start each day by holding up your own phone and saying, "Look guys! I'm putting it into my briefcase. I'll check it for messages during the first break. Please do the same!"

41

Plant Safety Fundamentals

Refinery safety relies upon two questions:

1. Is the process design intrinsically dangerous?
2. Do shift operators understand how their instruments function?

Safety in refineries is not a matter of attitude. Safety is a product of the knowledge of process engineer and console operators.

Location of Relief Valve

Referring to Fig. 41.1, I have shown two possible locations for the tower pressure relief valve. Which is correct?

The correct choice is position One. If the tower floods, the liquid head pressure in the vapor line will cause the relief valve at position Two to open, due to the static head of liquid in the vapor line, even though tower pressure is not excessive. Large volumes of liquid will then flow into the relief valve header system which hopefully is connected to the flare, and not to an atmospheric vent, as discussed later.

Level Indication

Liquid levels are not measured directly. They are indirectly measured by means of a "Level-Trol" (see Fig. 41.2). The pressure difference (DP) between P_1 and P_2 (psi) is measured and then converted into inches of level (D_H) by use of Eq. (41.1):

$$D_H = (\mathrm{DP})\,(28) \div (\mathrm{SG}) \tag{41.1}$$

where SG = Specific gravity of liquid.

The SG used is called the "Calibration Specific Gravity." It is not typically adjusted for months or years. That is, until the instrument technician recalibrates the level-trol. If the fluid in the level-trol becomes less dense than normal, the indicated level on the panel, relative to the real level in the vessel, will decline, as I've discussed.

Flow Measurement

Flows are indirectly measured by use of a flow orifice plate. The DP across the orifice plate is measured and then converted to flow using Eq. (41.2) (see Fig. 41.2).

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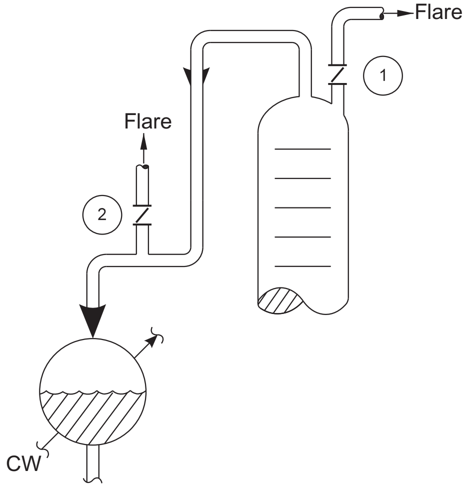

FIG. 41.1 Correct location for relief valve is position #1.

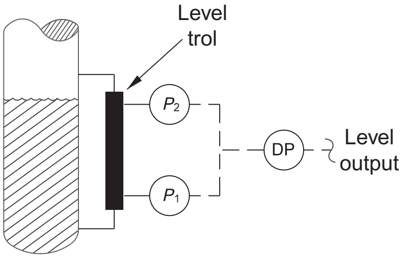

FIG. 41.2 Level is indicated by SG and differential pressure.

$$\text{Flow} = (K \cdot \triangle P)^{½} \tag{41.2}$$

where K = Orifice coefficient.

The factor that many operators do not know is how to correct a flow for a meter being off-zero.

Let's say the meter on the instrument is reading a flow of "*A*." The meter is then bypassed, so that the observed flow should drop to zero, but it only drops to "*B*." To calculate the correct flow, use Eq. (41.3):

$$\text{Corrected flow} = \left[\left(A^2 - B^2\right)\right]^{½} \tag{41.3}$$

In the 1960s, when flows were indicated on a strip chart recorder, we made this correction with a pencil, as the paper strip was already calibrated for the square root function. With a digital display, this correction must be made with Eq. (41.3).

BP 2005—Raffinate Splitter Explosion—Texas City

The following story relates an incident that was a combination of operators not understanding how level and flow instruments worked. And, process engineers not adhering to good design practices.

In 1974 I was waiting for the elevator in the Amoco Oil Building in Chicago, with Gary Elmer. "Norm," Gary said, "if Amoco is in such a rush to get the Xylene Splitter at Texas City built, they should pay me overtime. How about filling out some data sheets to help me for a few days?"

So, to a minor extent, I participated in the design of the Xylene Splitter. After the Isomerization Unit at Texas City was built, the tower was transformed into a Raffinate Splitter which became the most famous distillation tower in America.

You may recall the incident. In 2005 the tower's relief valve opened, as a consequence of a high liquid level. A cloud of naphtha vapor formed because the relief valve was not connected to the flare. It vented indirectly to the atmosphere. The naphtha vapors ignited from a truck engine and exploded. Fifteen contractors were crushed to death in a trailer and 180 injured. BP settled for $10,000,000 per fatality. The price of gasoline rose by 10 cents, as the refinery was the largest producer of gasoline in America.

The understanding in the industry is that this was a consequence of Gary Elmer and me connecting the relief valve vent to the atmosphere, rather than to the refinery flare. The real story is more complex.

Background

I worked as an operating supervisor in Texas City, and as a process engineer in Chicago for Amoco Oil between 1965 and 1981. I designed 7% of the process equipment in Texas City (Cokers, Sulfur Recovery, Light Ends Fractionation, Amine System, Fuel Gas Treating). For the distillation towers I supervised (Debutanizer, Butane Splitter, Depropanizer, Butane-Pentane Splitter), all relief valves were located at the top of the towers. All relieved directly to the atmosphere. The reason I did not object to this design was because it was that way when I had assumed supervision of the process. Also, the towers had been commissioned in 1958 and nothing had happened "**YET**." What would result if a tower flooded due to a high liquid level was not a question that I considered.

Fundamental Causes of Raffinate Splitter Explosion

The documented causes of the Raffinate Splitter failure are as follows:

- Relief valve not connected to plant flare.
- Operator laxity.

However, there are a number of other errors that have not been fully explained (see Fig. 41.3):

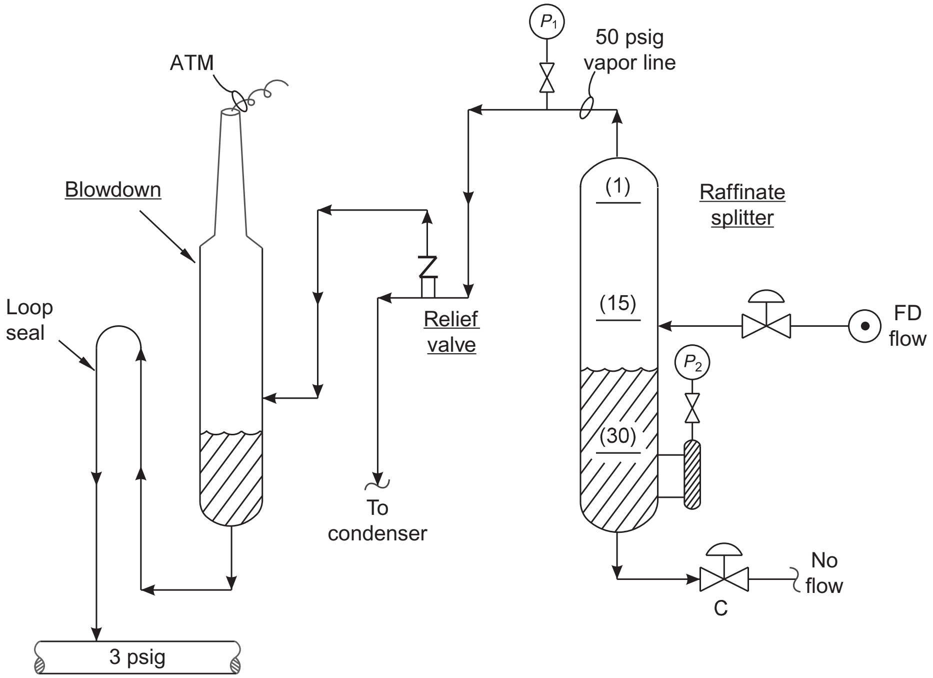

FIG. 41.3 Raffinate splitter—Texas City.

1. Design of the loop seal draining the Blowdown Tower to the 3# condensate collection system.
2. Elevation of the splitter relief valves relative to the top of the tower.
3. Lack of technical support during start-up operations.
4. Operators not understanding the relationship between tower bottom temperature and tower bottoms indicated level.
5. Operators not understanding the concept of a level-trol being "Tapped-Out."
6. Failure to check liquid head pressure at the bottom of the tower.

How Levels Are Measured

We do not directly measure levels in process units. We measure the pressure difference between the top and bottom level taps (Fig. 41.2). To convert from a differential pressure, to level, we use the specific gravity of the liquid (Eq. 41.2). For a given differential pressure, the indicated level that operators observe on the control panel will:

- Go down, as the liquid becomes less dense.
- Go up, as the liquid becomes more dense.

Even though the actual liquid level in the tower has never changed.

During the Raffinate Splitter start-up, the liquid in the bottom of the tower was less dense than normal. Thus, the indicated level observed on the control panel was lower than the actual level. Unfortunately, when the liquid fills the space between the lower and upper level taps, the differential pressure measured between the level taps remains constant, regardless of how high the liquid level becomes (Fig. 41.2). This is called being "Tapped-Out." If the specific gravity of the liquid is 10% higher than normal, the indicated level on the panel will read 110%. But, if the specific gravity of the liquid is 10% lower than normal, the indicated level on the panel will read 90%. As the level rises by 10 ft, or 100 ft, above the top level connection, the indicated level will remain at 90%. Which is exactly what happened at Texas City.

Temperature vs Specific Gravity

The operators did suspect that they had a high level. However, they then commissioned the splitter feed heater. This resulted in tower bottom's temperature increasing. This reduced the specific gravity of the liquid; which reduced the indicated level; which caused operators to draw the false conclusion that their tower's bottom level, which they had begun to suspect was too high, was actually okay.

The hourly operators, who were terminated by BP, testified to the above, as the reason they allowed the tower to overflow. They testified that this was proof that the cause of overfilling the tower was not their negligence, but a faulty level indicator.

The operators were victorious in their litigation against BP (likely because the previous principle was not explained during the proceedings).

Was the Splitter Pressure Excessive

Apparently, the tower pressure itself never became excessive. The operators were bringing in more feed to the splitter than they were drawing off of the bottoms product outlet.

It wasn't that the tower overpressured, it was that it overflowed. If the pressure relief valve had been located at the top of the splitter (as was the case in the towers I operated in Texas City in the 1970s), the overfilling would not necessarily have caused the relief valve to open. But, if the relief valve is located, as in the case of the Raffinate Splitter, many feet beneath the top of the tower (Fig. 41.3), then a head of liquid could exert a head pressure on the relief valve which then may cause the relief valve to open.

Certainly, it was a large volume of saturated liquid, and not just vapor, that blew out of the relief valve, and into the blowdown tower.

False Bottom Flow Indication

The operators had closed off the normal bottoms product outlet level control valve. But why? Because they thought it was leaking. The valve was likely not leaking. But, the flow transmitter was off-zero. Most flow transmitters are off-zero. For example, let's say that the indicated flow is 8 B/H. The instrument is checked and found to be off-zero. The zeroed flow is 4 B/H. Flow meters read as a function of the square root of the measured pressure drop through the flow orifice plate. Therefore, the corrected flow (i.e., compensating for the meter reading off-zero) is (see Eq. 41.3) as follows:

$$(8)^2 - (4)^2 = 64 - 16 = 48 \tag{41.4}$$

$$(48)^{1/2} = 7 \text{ B/H}$$

The meter was reading 1 B/H too high. But if the control valve is shut, and the real flow falls to zero, the operators will see a flow of 4 B/H.

Apparently, the operators in Texas City mistook a meter being off-zero for a large flow. As they were bringing in feed continuously, and the indicated level on their panel was not increasing (but the real level was), they blocked in the splitter bottom's level control valve.

Loop Seal

The Blowdown Stack Loop-Seal is shown in Fig. 41.3. The liquid from the Blowdown Stack flowed into a slightly pressurized (i.e., about 3 psig) hydrocarbon liquid collection system, rather than into an open sewer. To prevent hydrocarbon vapors from continuously backing out of this collection system, into the atmospheric blowdown stack, there was an intervening loop seal. Unfortunately, as the liquid in the blow-down stack flowed upwards in the vertical portion of this loop, the liquid lost head pressure, due to its increased elevation. As this liquid was naphtha at its saturated liquid bubble point, it partly vaporized. The expanding vapor choked off the liquid flow. The result—"VAPOR LOCK."

Vapor lock stopped the flow of naphtha draining from the blow-down stack. The liquid level in the stack backed up over the inlet nozzle. The vapor flowing through the inlet nozzle blew the naphtha up the stack. The naphtha ignited.

Likely, a high point vent on the top of the loop seal—which is a common feature of loop seals used in water plumbing systems—could have prevented the vapor lock.

The Ultimate Question

We can hope that BP and Marathon (the current operator of the refinery) would have learned a lesson from this incident. But that is not the question. Would I have made the same mistake if I had been present? Hopefully, I would have noted that the input flow exceeded the output flow. Then, calculated that the level in the splitter must be increasing. Had the unit's Technical Service Engineer been involved in start-up, he or she should have interceded with operating personnel based on the material balance.

But there is another, more fundamental action that I would certainly have taken to prevent this explosion. In 1984 I was involved in an Alkylation Unit Depropanizer start-up at the Good Hope Refinery. No one was sure of the liquid level. The level gauge glasses were largely useless as the process fluid was water white (not yellow), and the glasses old and scratched.

I placed a pressure gauge (Fig. 41.3) at the top vent of the level glass (P-2) and then checked the head pressure (P-1). This is the pressure difference between the top vapor line and top of the gauge glass. To convert this DP to feet:

$$\text{Feet} = (\text{DP})\,(2.3) \div \text{Specific Gravity of Butane} \tag{41.5}$$

At Good Hope, the measurement indicated we had 40 ft of liquid above the top of the gauge glass. We were "Tapped-Out." The liquid level was pumped down until the indicated level on the panel declined from a dead-steady 80% to an erratic 70%.

In 1974 I demonstrated this method of finding levels on start-up to my operators in Texas City. But by 2006, this simple step had been submerged in the flood of advancing technology, to the detriment of refinery safety.

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Dangers of H_2S, Steam, Nitrogen, and Hot Water

Lieutenant Joe Patrocelli was the Engine Room officer in an Escort Air Craft Carrier in the bloody World War II battle of Leyte Gulf in the Philippines. We worked together at the Amoco Oil Refinery in Texas City on the Amine Regeneration Plant and the Sulfur Recovery Unit.

On graveyard shift, we would exchange war stories. Joe recited naval battles against the Japanese. I would tell tales of death and disaster that I had been associated with in process units. “Norm, being below deck in a carrier under attack by kamikazes is hard. At least guys topside can fight back. You know, we lost a carrier in Leyte.”

“Yeah, Joe. I read about that.”

“Of course, working on a sulfur and amine plant with this bunch ain’t all that safe either. What with the H_2S and SO_2 and the pyrophoric iron. Just last week, Kenny overfilled a sulfur rail tank car. Could’ve killed someone.”

“Joe, refineries are really dangerous places unless you’re careful. I’ll tell you a few stories involving fatalities I have been involved in.”

“Well, Lieberman, we got us nine hours before the day shift shows up. You talk and I’ll listen.”

Coastal—Aruba

A maintenance supervisor (also a retired Marine) was directing a crew of pipefitters changing a spool piece on a rich amine line on an elevated platform. It was a “Fresh Air” job. For some reason, he removed his Scott Air Pak. The rich amine flashed. He died of H_2S inhalation.

- *Remedial Action*—Dealings with equipment in rich amine service is identical to H_2S service, which always requires use of “fresh air.”

American Oil—Texas City

On an Alky Unit, two pipefitters were blinding off a depropanizer reboiler. The tower was floating on plant flare pressure. The sulfuric acid production plant (which used H_2S as feed) had an emergency shutdown, and their H_2S was diverted to the flare. The H_2S backed into the Alky Depropanizer and killed both pipefitters.

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- *Remedial Action*—Blind-off process equipment from the plant flare during a turnaround.

Refinery "X"—Baytown, Texas

A female operator was sent to restart the water pump on the acid gas feed K.O. drum. The drain valve on the pump suction was an ordinary ¾ inch gate valve. It should have been a "Dead-Man's" valve that is closed by a strong spring when not being held open by an operator. Her coworkers found her dead an hour later next to the pump.

- *Remedial Action*—The "dead man's" valve is a spring-loaded valve that must be manually held open. When released, it closes instantly and tightly by itself.

GHR Energy—Norco, Louisiana

A contractor walked into a 6 ft deep pit of hot water from a steam leak on the sulfur recovery complex. He was scalded to death. I had weekend duty as plant manager during this incident. The pit had been excavated to fix the steam leak.

- *Remedial Action*—Unit Supervisors must keep control of construction projects on the process units they supervise.

American Oil—Texas City

Two maintenance men were killed when they entered a hydrocracker reactor. The reactor had been purged with nitrogen. Entering a nitrogen atmosphere does not cause any shortness of breath, or any other discomfort. Only death.

- *Remedial Action*—Add 1% CO_2 to N_2. This will immediately cause shortness of breath. Also, strictly enforce entry permit policy and always provide a "Hole-Watch."

Coastal—Corpus Christi

A contract laborer was killed when a steam turbine, driving a hot asphalt pump (700°F) overspeed and a small connection broke off of the pump. The asphalt sprayed onto the laborer's clothes and auto-ignited. The auto-ignition temperature of asphalt, or vacuum tower bottoms, being 320°F. The turbine's overspeed trip was "wired up" (i.e., disabled with a piece of wire) because it kept tripping off, as a consequence of the governor speed controller malfunctioning. I had noticed before the incident that the trip was disabled, but neglected to report this to plant management.

- *Remedial Action*—Do not run steam turbines unless both the governor speed controller and overspeed trip are both operational. The trip is not a back-up for the governor.

Without the governor functioning, the operators will sooner or later disable the overspeed trip.

Refinery "X"—Beaumont

An operator was killed by boiling water from an eruption from the top of an open coke drum. My investigation indicated that purge steam was accidentally opened into the bottom of the coke drum. My client thought that a localized hot spot in the coke bed caused the eruption.

- *Remedial Action*—Modern coke drum unheading devices should protect against this hazard. Existing units should be retrofitted with a "Delta Valve" on the top head.

American Oil—Texas City

An inspector was crawling across the roof of an alkylation unit's spent sulfuric acid tank. The roof had been reported to be corroded. It was! All they recovered was his teeth.

- *Remedial Action*—The inspector was trying to impress his coworker with his courage. There are some accidents caused by people's belief in their indestructability.

GHR Energy—Norco, Louisiana

The floor or a vertical fired heater was blown down onto two operators who were attempting to light the main burners. The pilot lights were plugged, and they were using an oil-soaked rag as an ignition source. Both were my friends.

- *Remedial Action*—These pilot lights were using fuel gas rather than natural gas. They were plugged with sulfur deposits. Use Natural Gas for pilots, not refinery fuel gas.

Amoco—Whiting, Indiana

Three operators were killed when an Asphalt Oxidizer Tank blew up. Both the steam and nitrogen purge in the off-gas line to the oxidizer incinerator had been deliberately turned off in an attempt to relight the oxidizer pilot light.

- *Remedial Action*—The process is inherently dangerous and unnecessary. Asphalt spec's can be met by better operation of a Vacuum Tower, without resorting to this archaic oxidizer process.

Unocal—Chicago

A coke drum bottom head blew out onto the coke cutting ramp during coke drilling, and killed the operator of the front-end loader removing the coke from the ramp. The bottom

6 ft ID cone of the coke drum was plugged, and 100 ft of water had likely built up behind the pluggage, which gave way suddenly.

- *Remedial Action*—Front-end loaders should never be allowed in coke drum pits, or on the ramps, when coke is being cut. Plugging and blowing out of the bottom head cannot always be avoided. Cut the coke first, and then remove it with the front-end loader.

Tenneco Oil—New Orleans

A dozen contract workers were killed inside an aromatic distillation tower during a turnaround. A contractor had connected the air eductors to natural gas, instead of air. The explosion dismembered the workers, but did little damage to the tower.

- *Remedial Action*—The fittings at the utility stations for air, natural gas, N_2, and steam must all be made different with unique hose connections that are not interchangeable.

Refinery "X"—Norco, Louisiana

The entire crew of an FCU were killed when a free air detonation of a propane cloud blew the cat fractionator down on top of the control room. A water injection had been installed, without a spray nozzle, and the water eroded-corroded through a 6″ pipe elbow in the tower overhead vapor line. I visited this unit recently and read the sad memorial for the lost operators.

- *Remedial Action*—Wash water injection into a pipe should be through a mist nozzle, in a horizontal run of pipe, well upstream of elbows, not through a piece of ½″ tubing. Use de-aerated water, not Mississippi River water that is saturated with oxygen.

Unocal—Chicago

Twenty-six refinery personnel on the fire response team were killed when an LPG amine (MEA) extraction tower failed at a repair weld. The repair weld had not been stress relieved, even though the tower was being used in rich amine H_2S service. The reason the repair weld was not stress relieved was because originally the vessel had not been postweld heat treated.

- *Remedial Action*—All vessels in rich amine service must be postweld heat treated. Also, the use of MEA in refineries with FCU's is inherently dangerous due to the formation of heat stable salts, unless the amine reclaimer is consistently operated which was not the case at the Unocal plant.

A Personal Near Miss

"Joe, I was almost killed myself last year. On a coker fractionator. I had a blind pulled to check the water distribution from a wash oil spray header. The piping contained oil saturated with H_2S. My H_2S alarm went off, but I got myself stuck in the 24″ chimney of the gas oil draw-off tray. I would have expired prematurely if my pal Kumar, a young engineer from New Delhi, hadn't pushed me through the chimney."

"Hell Lieberman, you should never pull a tower blind like that. It automatically cancels all entry permits previously issued by operations. You were really lucky."

"Not only that Joe. I also unscrewed the spray nozzles myself. They were plugged and the wash oil piping could not be water flushed clear of the H_2S saturated gas oil."

"Norm, once you break any connection inside a vessel, this also voids all entry permits. If you wanted to test the spray header distribution, you should have done it outside of the coker fractionator on the ground."

Working for Safety

"You know, Joe, I've thought a lot about safety in process units. The general conclusions that I've made is the importance of having experienced operators and technically competent engineers designing and operating the equipment. Too often, I see younger people placed in charge of operating units, based on their perceived management potential. In some refineries, the really experienced operating supervisors have been overburdened with administrative and record keeping duties. Another problem Joe that I see, is engineering contractors, with little refinery experience, designing process units, and making fundamental errors. Also, a lot of problems develop because plant inspectors have little knowledge or training about metallurgy and corrosion rates.

I guess the real key to safety is what I learned from my mom. Always assume that if anything bad can happen, that it will happen. And that it will happen to me."

"Yeah Norm. It's just like those kamikazes. I was on the deck once during an attack. I felt that every one of them was diving straight at me. I guess working in a refinery is kinda like being in combat. You've got to be alert to stay alive."

43

Safety—Attitude or Engineering?

There is a misconception in the process and refining industry that the key to safety is:

- Awareness
- Alertness
- Attitude

This is the sort of nonsense we have all been exposed to at innumerable safety meetings. Everyone I have ever met in a refinery wants the facility to operate without death and disaster. But only too often, this desire is frustrated by engineering design errors. The 1976 propane explosion at the American Oil Refinery in Texas City is an illustration of a failure in applied technology.

Sulfuric Acid Alkylation

The Stratco Alkylation process (licensed by DuPont) was developed during World War II by Mr. Stratford and Ward Graham. Butylene and isobutane are reacted to make isooctane at a temperature of about 50°F. The exothermic heat of reaction is removed by an effluent refrigeration loop with a typical composition of:

- 80% Isobutane
- 10% Normal butane
- 10% Propane

The propane is continuously purged from the circulating refrigerant in a depropanizer tower. The overhead propane product from this tower contains an acid ester—a hydrocarbon molecule combined with a sulfur trioxide group (SO_3). This acid ester is entirely noncorrosive as long as it is water free or DRY. But, if it accidentally gets wet, then the acid ester will dissociate into hydrocarbon and weak sulfuric acid. Strong sulfuric acid (90%+) is not corrosive to carbon steel. Weak sulfuric acid (10%–20%) is tremendously corrosive to carbon steel piping. I have personally seen weak acid eat through welds of 6″ piping over a weekend in Texas City in 1974.

Wet vs Dry

According to my Alky mentor, Ward Graham (who passed on in 1980), corrosion in an alkylation unit depropanizer may be entirely suppressed if the depropanizer feed is kept

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dry (free of both dissolved and entrained water). During normal operations, this would be the case except on START-UP of the depropanizer.

I was in charge of the operation of No. 2 Alky from April 1974 through mid-1976. During this interval, the alkylation reaction section was operated without interruption. After all, we produced 0.6% of the total gasoline consumed in the United States. But, the unit's depropanizer was shut down on several occasions to fix tube leaks in the water-cooled overhead condensers. To operate without the depropanizer for a few days was not a problem. Propane accumulated in the circulating isobutane refrigerant stream. After a few days, we could then vent off noncondensables from the refrigerant accumulator drum to fuel gas. The vent gas had 50% butane, as compared to 2% butane in the depropanizer overhead LPG product. So, isobutane consumption was high while the depropanizer was out of service. But, that was the normal operating procedure followed while the depropanizer was off-line for condenser tube repairs. However, on recommissioning the depropanizer, this practice had the potential to introduce corrosive moisture into the tower.

Depropanizer Start-Up Procedure

A leaking tube in the depropanizer overhead water cooled condensers could not introduce water into the tower. The depropanizer operated at 300 psig (20 BAR). Cooling water pressure was 30 psig (2 BAR). The problem of moisture introduction into the tower occurred after we shut the tower down to repair or replace the leaking tube bundle in the overhead condenser. Moisture could then be introduced into the depropanizer in a variety of ways:

1. The tower was steamed out to hydrocarbon free it prior to blinding off (i.e., "spading") the shell side of the condenser.or
2. The tower was de-pressured below the circulating cooling water supply pressure.or
3. When the tower was restreamed, wet field butanes (which contained about 2%–5% propane) was used to start-up the tower to establish a reflux flow at normal 300 psig (20 BAR) operating pressure.

A Design Error

The Alky Unit had not been designed to start-up the depropanizer separately, while the rest of the unit was in operation. As it was possible to operate the unit without the depropanzier running, this should have been considered during the design phase of the project. As a consequence of this omission, in practice, refrigerant isobutane, that contained acid esters, was introduced into the depropanizer while the tower's reflux was saturated with water. The acidic water circulated with the reflux. A rapid rate of corrosion, due to weak sulfuric acid, caused the 4″ elbow on the suction of the reflux pump to fail. I climbed up to inspect the elbow a week after the failure. It looked like an evil person had taken a can opener, and peeled back the elbow, which was no thicker than a lid on a soup can.

The entire contents of the 4′–0″ ID × 15′–0″ T-T reflux drum had emptied. A huge white cloud of LGP vapors drifted across No. 2 alky, over to the No. 3 FCU. The cloud ignited off of the CO Boiler on the FCU and detonated, blowing in the roof of the FCU control room. The entire East Plant of the world's largest refinery was shut down for several months.

Corrective Design Change

After this incident, the unit was retrofitted with a caustic wash (10% NaOH) upstream of the depropanizer, to remove all traces of acid esters from the tower's feed. This, I believe was a correct decision by the American Oil management. However, in the interval before the new caustic wash was installed, I instituted an alternate mode of operations. That was:

- *Step 1*—Steam-out tower prior to start-up.
- *Step 2*—Bring in mixed butane feed for a few hours to establish a reflux drum level and reflux flow using the tower reboiler to vaporize the reflux.
- *Step 3*—Stop bringing in all the wet feed and run only on total reflux for X-hours.

By X-hours, I mean until I *personally* checked to see that the water draw-off boot on the depropanizer reflux drum was utterly free of water. Then, I would wait another shift, before allowing the acidic isobutane refrigerant recycle to be reintroduced to the alky depropanizer.

Summary

The original design error assumed that someone would use a significant degree of engineering insight to realize that the depropanizer must be completely dehydrated before the refrigerant recycle was reintroduced back into the tower. That, I believe, was an unreasonable expectation. Especially, since this requirement was never specified in the unit's original operating instructions.

The management decision to install the 10% caustic wash on the depropanizer feed, which I did not consider necessary at that time, was, I now believe in retrospect, the correct engineering decision. As long as the alkylation unit was operated in a normal, steady state mode of operation, the depropanizer would remain in a dry, and hence a noncorrosive state. However, there were in practice, a number of circumstances that could inadvertently introduce moisture into the tower. The caustic wash was a relatively inexpensive option to enhance the unit safety by extracting acid esters from the depropanizer feed.

My own attitude toward safe plant design was greatly influenced by this incident. I now try to include provisions in my process design work for safe shut-downs, start-ups, and response to upsets and equipment failures. Especially, if water can mix with acidic components.

Basically, I am smarter now.

44

Danger of Improper Process Controls

I am afraid of snakes. But the two things that I fear most are as follows:

- Hydrofluoric acid (HF) used in refinery alkylation units as a catalyst to make isooctane for gasoline.
- A cloud of LPG (propane).

HF acid will penetrate skin and destroy bones. Clouds of LPG (propane or butane) form a white mist, that being heavier than air, drifts across the ground searching for a source of ignition. At which point, it will detonate with a destructive radius of ¼–½ mile. I have been personally responsible for generating four giant clouds of LPG in Texas City in the 1970s.

- A ¾″ nipple broken off while pulling a plug on an alkylation unit contactor by an operator using a 16″ pipe wrench.
- A 2″ drain blew out due to an operator opening it too far when it was partly plugged with hydrates (i.e., hydrocarbon-water solid ice).
- A nonstress relieved weld blew out on an 8″ carbon steel pipe which had been exposed to weak sulfuric acid.
- A 4″ elbow in liquid propane service blew out massively. The elbow was below a huge depropanizer reflux drum, which emptied to the atmosphere.

Only in the last incident, did the vapor cloud find a source of ignition. In the subsequent detonation, the giant American Oil refinery East Plant was devastated. Also, my career at American Oil was adversely impacted.

Alkylation Unit HF Stripper

About 5% of the gasoline produced in the United States is made in HF alkylation units. A by-product of such units is refinery-grade propane produced as the bottom product of the alky unit's HF stripper (see Fig. 44.1). This stripper tower serves two functions:

- First—To strip out ethane to meet the 2% maximum ethane specification for refinery grade HD-5 LPG (i.e., propane).
- Second—To strip HF acid out of the propane and recycle the HF acid back into the Alky Unit Isobutane recycle stream.

The control scheme shown in Fig. 44.1 had been designed by a major contractor whose office is located in the United States, just south of my former home in Chicago, for a mid-sized refinery in New Jersey. I have participated in the operation and design of a

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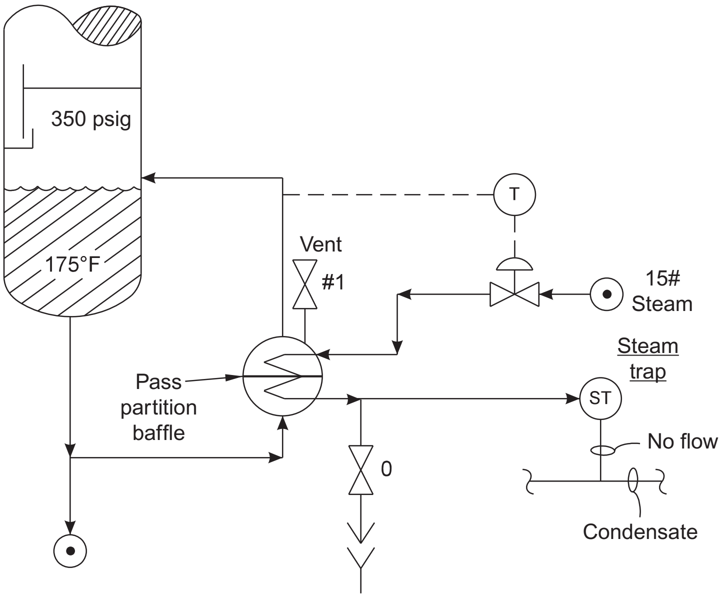

FIG. 44.1 Steam condensate diverted to deck drain.

dozen similar units, and had never questioned the intrinsic safety of the process design or the process controls.

But there is something that I, and the Chicago Engineering Contractor, should have questioned. That being, the 15 psig steam supply to the reboiler. Indirectly, this 15 psig was the cause of the accident which I'll describe in detail.

Carbonates in Boiler Feed Water

Even demineralized BFW has residual carbonates, which decompose into CO_2 when steam is generated in a boiler. Inside the tubes of a reboiler, the steam will condense, but the CO_2 will accumulate as a noncondensable gas. With time, corrosive carbonic acid is formed, which causes leaks in the carbon steel tubes. Venting from the top of the channel head (Valve #1 shown in Fig. 44.1) will not vent off the accumulated CO_2. It will just vent off the 15 psig steam supply. The CO_2 will not be separated from the 15 psig steam until the steam is condensed to water. Basically, this means the noncondensable vent must be located just underneath the channel head Pass Partition Baffle (see Fig. 44.1).

Reboiler Channel Head Pressure

The reader may be puzzled as to why the process controls shown in Fig. 44.1 would be quite reasonable for 50 psig steam, but manifestly dangerous for 15 psig steam. What's the difference?

It has to do with the steam condensate collection header pressure. Typically, this pressure is 20–30 psig. When using 50 psig steam, this is not a problem. The pressure in the channel head is large enough to flow into the condensate line. But for the 15 psig supply steam, allowing for losses across the steam inlet control valve, and the reboiler tubes and outlet nozzle, the condensate pressure is likely to be 10 psig or less.

A loss of reboiler duty due to condensate back-up would result, which could only be restored by the operators draining the steam condensate to the deck. This was precisely the action that the operators in the New Jersey refinery were forced to do to maintain the efficiency of the HF stripper and the required reboiler duty.

Consequences of Tube Leak

The shell side of the reboiler operated at 350 psig. The tube side operated at about 10 psig. The tubes were immersed in liquid propane. The surface area of the reboiler was far larger than required, until a tube leak developed due to the carbonic acid attacking the carbon steel tubes.

The liquid propane, at 350 psig, blew through the leak, flashed, and stopped the flow of the 15 psig steam. With the loss of steam flow, the reboiler heat duty declined. Hydrofluoric acid, no longer being stripped out, fell into the bottom of the tower.

Condensate Drain Valve

Had the steam condensate been flowing into the condensate collection header, the LPG and HF vapors would have been vented from the atmospheric vent located on the steam condensate collection drum, which exhausted at 60 ft above grade. This outcome would have been quite dangerous in itself, as propane is heavier than air.

But, as the steam condensate was being drained to the concrete pad, the liquid LPG and hydrofluoric acid blew out and enveloped the Alkylation Unit in a cloud of propane and HF. The cloud of combustible vapors did not ignite. The operators, alerted by a local combustible gas monitor, donned their HF protective gear, blocked-in the condensate drain valve, and shut down the unit without injury to refinery personnel, and before the toxic vapor cloud could spread to the surrounding residential community. The plant had a rather lucky escape.

However, I recall that in the 1970s, in Texas City, I also escaped from the consequences of an LPG vapor cloud three times. But, my luck ran out the fourth time, when the 4″ elbow on the reflux drum outlet line failed. "God watches over the feeble minded," but only up to some point. Then, we had best take corrective action ourselves, as there is a limit even to divine mercy.

Retrofit Design

To correct this situation, I proposed two alternates to the refinery. Both options would eliminate the dangerous practice of draining steam condensate to the ground.

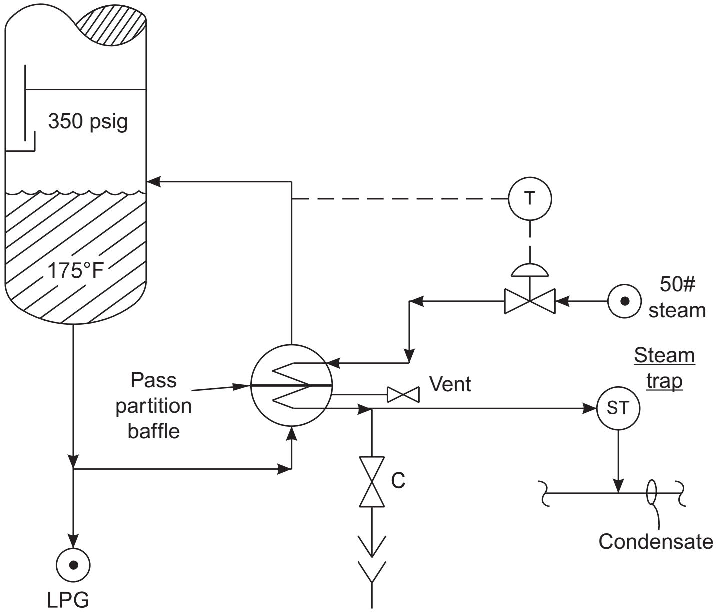

FIG. 44.2 Steam condensate flows to condensate collection line.

Both options (see Fig. 44.2) included relocation of the noncondensable (i.e., CO_2) vent just beneath the channel head pass partition baffle.

The simplest option was to replace the 15 psig steam with 50 psig. This would allow the channel head pressure to overcome the pressure in the condensate collection line. This was the option that was adopted by the refinery.

A more complex option was to install a condensate pump and drum, and pump the condensate. This more expensive option would allow the continued use of the low value 15 psig steam but was rejected by the refinery due to capital cost considerations.

Steam Trap

Both the existing and the new designs used a steam trap and relied on pressuring the condensate into the condensate collection line. For lower pressure steam, it's best to avoid the problem of flashing (i.e., vapor lock) downstream of the trap, by pumping the condensate, rather than just relying on the pressure in the channel head. Further, steam traps often prove to be unreliable. However, if no condensate recovery is required, I will use a steam trap, and drain the condensate into the sewer, but only for nonhazardous services.

Note that the steam condensate cannot be simply drained without holding some back pressure. Just allowing condensate to blow-out to atmospheric pressure, would greatly diminish the reboiler's heat transfer efficiency, due to blowing the condensate seal.

45

Process Plant Safety

The major long-term cost of running a refinery, or I expect, any process plant, is safety. Not the occasional "lost time injury" or occasional fatality. I am referring to incidents such as:

1. The BP, Texas City Raffinate Splitter Explosion—15 killed
2. The Unocal Amine—LPG Absorber—20 killed
3. The Amoco Asphalt Oxidizer Fire and Explosion—3 killed
4. The BP Deep Water Horizon—11 killed
5. Union Carbide—Bhopal—10,000 killed

These were multibillion dollar incidents, all of which I am personally familiar with were the result of operators and process engineers making entirely avoidable design and/or operational errors.

The incident at Bhopal was the result of an effort to avoid loss of production due to the marginal design of a pump's mechanical seal. It destroyed the Union Carbide Corporation and killed 10,000 people.

The incident near my home in New Orleans, cost BP $20,000,000,000 (United States), killed 11 people and spilled 5,000,000 barrels of crude oil into the Gulf of Mexico.

Often, I am retained as an expert witness in these sorts of catastrophes. The one common thread seems to be, that at some critical point in the decision-making process, that a middle-level management person selects the wrong path, because he or she did not understand how the equipment worked, and therefore could not foresee the ultimate consequences of their decision.

I've made that sort of mistake four times at my Alkylation Unit in Texas City. I got away with it the first three times. But, the fourth time, the law of averages prevailed, with a resulting butane vapor cloud detonation. But that was in 1976. I'm much smarter now.

Sulfuric Acid Alky Operation

From 1974 to 1976, I was the operating supervisor of the world's largest alky unit with a capacity of 26,000 BSD of alkylate production. Between 1974 and 1976, four incidents occurred, all on the same isobutane refrigerant depropanizer tower, that provide a cautionary tale worth sharing with younger process engineers and operating supervisors.

I had worked for American Oil for 10 years as a process design engineer before being promoted to my position in 1974 at the Texas City Sulfuric Acid Alkylation Unit. The alky depropanizer processed an isobutane refrigeration recycle stream containing several

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hundred ppm of SO_3 in the form of dry sulfuric acid esters. The service was not corrosive, except during start-up, when traces of water reacted with the acid esters, to produce weak sulfuric acid, which is extremely corrosive to carbon steel.

Somewhat surprisingly, sulfuric acid stronger than 90% is not particularly corrosive to carbon steel. Weak acid of 10% strength can eat through the welds of carbon steel piping in a matter of a few shifts.

Alkylation Unit Design

The American Oil Alkylation Unit in Texas City began operations in 1958. It was an effluent refrigeration unit that used sulfuric acid as a catalyst. The term “Effluent Refrigeration” means that the reactors were cooled to 50–60°F, by the evaporation of the principal reactant, Isobutane.

The Isobutane refrigerant was compressed and recycled back to the alkylation reactors, after passing through a depropanizer. Propane in the alkylation feed would gradually accumulate in the Isobutane refrigerant recycle stream and was continuously removed in the depropanizer to produce LPG as a by-product of the production of alkylate (see Fig. 45.1).

The feed to the depropanizer contained a few hundred ppm of sulfuric acid in the form of Acid Esters. Upon being heated in the depropanizer reboiler, the Esters would decompose into:

$$\text{Ester} \rightarrow SO_3 + \text{Alkylate}$$

The SO_3 was noncorrosive, as long as it was dry. As the alkylation unit refrigeration loop had been in contact with strong (i.e., 90%–98%) sulfuric acid, which is an extremely powerful desiccant, the depropanizer feed *normally* was totally free of water.

However, if the depropanizer should ever be taken off-line for repairs and steamed out to displace butane vapors, on start-up, the tower would be wet for a period of hours or days.

The Nature of Sulfuric Acid

Fresh sulfuric acid is purchased at a strength of 98 wt%. It is entirely noncorrosive to carbon steel. Even at strengths as low as 88%, rates of corrosion are extremely small.

However, upon dilution with water to a strength of a few percent, weak sulfuric acid will eat through the heat-affected zones of welds in carbon steel piping in just a few days, or even a few hours, with potentially catastrophic results. This being based on personal observation during a sulfuric acid carry-over into the effluent water wash treater on a Sunday morning in 1975.

Allowing small amounts of SO_3 to dissolve in water will produce highly corrosive weak sulfuric acid. Thus, the need to carefully and totally dehydrate the alkylation unit depropanizer prior to the introduction of its normal feed of acidic recycle isobutane refrigerant.

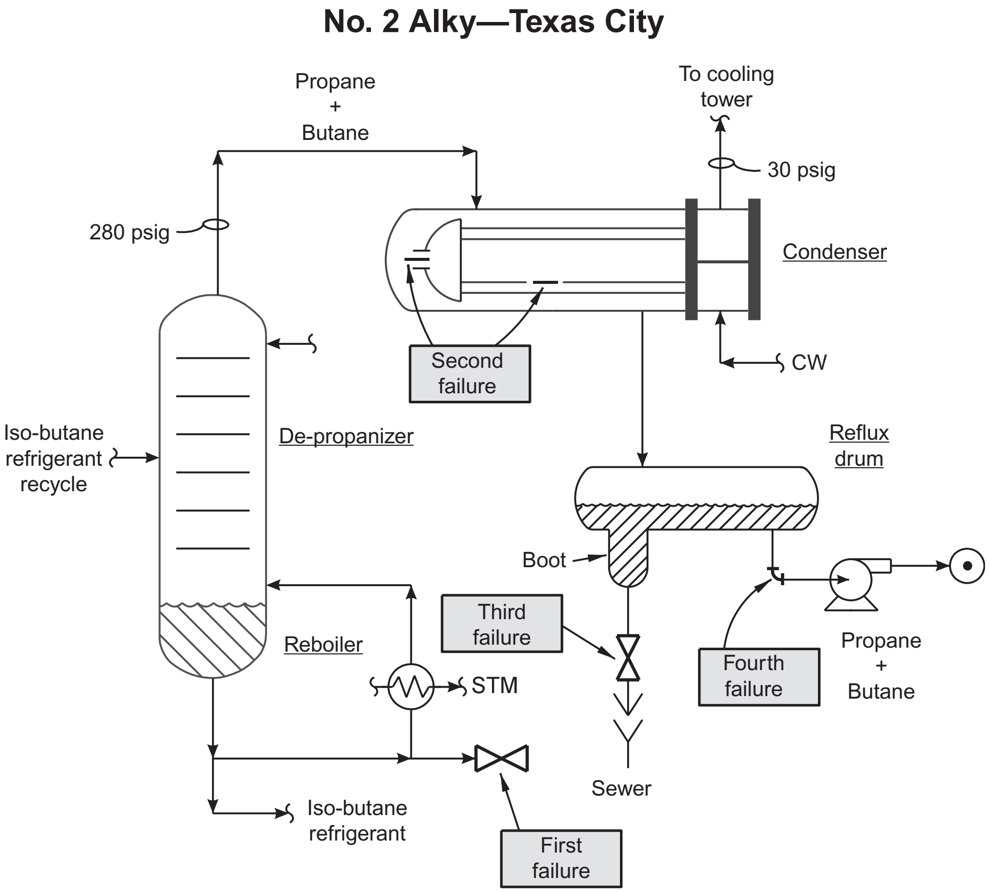

FIG. 45.1 Sulfuric acid alky unit depropanizer—a series of failures ending in a detonation of a vapor LPG cloud.

Correct Start-Up Procedure

The best way to avoid the problem of creating weak sulfuric acid in the alky depropanizer was to:

1. Never to start-up using the Isobutane from the refrigerant recycle loop, even though that was the procedure written in the Operations Manual.
2. Bring in a source of refinery mixed butane from tankage as the depropanizer feed.
3. Shut off the feed, and run the depropanizer on *total reflux*, until absolutely no water appeared in the overhead reflux drum water draw-off boot drain valve.
4. Check the reboiler and tower low point drains for water.
5. Expect that the previous procedures will take 2–3 days, and that personal verification that all three drains are blowing free and dry is essential.

Unfortunately, in 1974, I was unaware of this procedure. I relied upon the good judgment and experience of the alkylation unit's operating personnel and the technical support infrastructure provided by American Oil.

No. 2 Alky—Texas City

The American Oil Refinery in Texas City produced 240,000 BSD of gasoline in 1974 (3% of total US consumption). About 10% of this volume was made on my alky unit.

My training for this appointment consisted of 10 years as a process design engineer in Chicago, and 4 h of instructions from Paul Stelly, the man I replaced.

The First Incident

One day, I became concerned that a tower reboiler drain was plugged. I instructed my unit pipefitter (Little Mac) to clear the drain on the ¾″ bleeder on the reboiler feed line, a few feet above grade. To do this, a ¾″ male plug had to be removed from the female threaded ¾″ bleeder. The plug was difficult to remove, and the determined pipefitter used an 18″ cheater bar for leverage and broke the ¾″ bleeder off of the welded connection on the 8″ pipe.

A huge cloud of isobutane vapors rapidly enveloped the entire unit. The air became chilled in the spreading fog of hydrocarbons.

Fortunately, on a sulfuric acid alkylation unit, there are no fired heaters or gas turbines. Also, at this time, there was no hot work (welding, cutting) in progress in the local vicinity. An older pipefitter (Big Mac) whittled a small piece of wood into the shape of a plug and hammered the wooden plug into the broken off ¾″ nipple. Within 30 min, the chilly white-isobutane fog dispersed, and the incident ended without any repercussions. I had the wooden plug covered with epoxy resin and fiber glass cloth, which I viewed as a reasonable method of repair.

The Second Incident

The overhead condenser of my depropanizer consisted of a horizontal exchanger. Water on the tube side and propane vapors on the shell side. The tube bundle had originally been Admiralty Brass. But the new tube bundles were a big expense, as tube leaks were common. Once 15% of the tubes were plugged, company standards were bundle retubing or replacement. So, I had opted for a less expensive carbon steel tube bundle, instead of expensive admiralty brass.

The cooling water for the depropanizer condensers came from a 12-cell cooling tower on the Alky Unit. One day, as I was observing the cooling tower from a distance, I saw what appeared to be several geysers of water erupting from the cooling tower's distribution decks. I climbed to the top of the cooling tower with Fuzzy Griffith, my day shift foreman. Looking across the cooling tower, we could see a dense white fog rising from the distribution decks.

A gigantic cloud of explosive isobutane mixed with air!

Fuzzy, despite his 60 years, raced back to the control room to shut down the unit and de-pressure the depropanizer. Inspection showed that:

- Only a single carbon steel tube had failed in the propane condenser.
- High pressure (270 psig) propane vapor (contaminated with SO_3) had leaked into the cooling water, flowing through the tube side.
- The jet of acidic water, plus propane vapor, had violently impinged on the floating head.
- A hole, the size of my fist, had been eroded through the carbon steel floating head, permitting the shell side fluid (propane at 270 psig) to flow into the cooling water return line (at 30 psig).

After safely shutting down the unit, I realized why my makeup isobutane requirements had recently been increased by almost 4000 BSD over the past several months. Also, why propane production had declined to essentially zero over the same period.

Third Incident

The corrosion experienced in the alkylation unit depropanizer was a consequence of residual water in the tower's reflux drum. Acting on this assumption, I instructed Randy Ramirez, a young operator, to check the boot drain on the depropanizer reflux drum for water. The drain was a 2″ drain located directly above a sewer and pointed directly down into the sewer grating.

Randy explained to me later that the 2″ drain was partially plugged and that he had to open it three or four turns before he could initiate flow. But, the drain blew out with tremendous force, as 270 psig liquid propane erupted from the half-open 2″ valve, and rebounded off of the sewer grating. Randy admitted that he had become frightened and ran away.

The first I knew about this potential disaster was that the entire alkylation unit was engulfed in a dense white cloud. The cloud of isobutane had auto-refrigerated and chilled the area around the control room, and was spreading across the refinery fence into a residential area.

I turned on the plant fire alarm. But, by the time the fire truck arrived, Fuzzy Griffith had talked to Randy and was in the process of closing the 2″ depropanizer boot drain valve. The breeze from Galveston Bay quickly dispersed the white cloud of cool isobutane vapors.

The Fourth (and Final) Incident

Suspecting that weak sulfuric acid corrosion was the root cause of the depropanizer overhead condenser tube failures, I initiated an inspection program (i.e., Son-Aray) to monitor the carbon steel piping for loss of metal thickness. The inspector, Joe Hensley, reported to me once a month on corrosion rates. In general, these were running at a few mils per month (pipe thickness is typically 250 mils). This implied a substantial, but not a dangerous rate of corrosion. Corrosion rates of 100 mils per year should be regarded with alarm. Corrosion rates of 20 mils a year require continued and careful monitoring.

What I had failed to notice was that Joe was not monitoring the reflux drum 6″ liquid propane outlet line. The elbow from the bottom of the reflux drum outlet nozzle could only be reached with a 20-ft extension ladder, which we did not have on my alkylation unit.

On a Sunday afternoon in June 1976, this elbow blew out. It rather looked as if someone had pried off the top lid of a soup can. An opening, 3″ wide and 12″ long, in the 6″ elbow allowed the 6 ft ID, by 20 ft T-T depropanizer reflux drum, to empty out over a few minutes.

The resulting cloud of LPG vapors drifted across the alky unit toward #3 FCU (the world’s largest Fluid Cat Cracker). Dan Wilson, the FCU Shift Foreman, saw the white cloud of propane approaching, and directed his crew to take cover under the tables and desks.

A moment later, the cloud detonated. The source of ignition possibly being the FCU CO Waste Heat Boiler. The ladders were torn off of the adjacent FCU main fractionator, and the control room roof was blown in. The side of a cooling tower on the Delayed Coker, ¼ mile away, was also blown in.

No one was killed as a result of this detonation. American Oil transferred me from my operating supervisor position in Texas City, to the Chicago Engineering office as, “Coordinator, Process Engineering Development.” In this position, I was responsible for organizing many important meetings in Chicago, and interfaced with other important people like myself.

The Lesson Learned

The pattern of repeated butane leaks should have alerted me that there was something fundamentally wrong with my operation of the alky unit. In retrospect, it was that my operators were not carefully draining water during start-up, from all of the depropanizer low-point drains.

In a more general sense, I had treated each of the related incidents as a separate failure, which once controlled, should best be forgotten. Of course, each incident was just one spark away from catastrophe.

After this final incident, American Oil modified the process to caustic wash the depropanizer feed to remove all traces of the Acid Esters. The refinery is now owned by Marathon Oil, and No. 2 alky has long been dismantled. But the lesson of “Hope for the best, but plan for the worst,” still remains with me as the basic foundation of Process Plant Safety.

Afterword

I visited my old control room last week (i.e., 2018). The unit is gone. But the private toilet I built in 1974 for Julia Tienaman, the first female operator in the Texas City Refinery, is still operational.

Safety Note

The heavier the material, the lower its auto-ignition temperature. Gasoline has an auto-ignition temperature of 480°F. Asphalt, when spread over a porous surface, like your clothes, auto-ignites at 300°F.

That's why a seal leak on heavy vacuum gas oil pumps is so dangerous. It's much safer to get a sample of naphtha at 350°F, than vacuum resid at the same temperature.

Safety Note

The double mechanical seal used on centrifugal pumps is a safety device. It's not for the reduction of environmental emissions. If the high pressure or the high level alarms on the outer seal reservoir pot are not working, the safety benefits for the double mechanical seal are largely lost.

An alarm or trip that is not tested routinely will never work in an emergency.

46

Avoiding Death and Disaster

"Lieberman! The safety meeting is starting. Let's go!"

"I'm not coming. Go ahead without me, Harvey."

"You know, Norm, you have a real poor attitude towards safety. Your lack of safety sensitivity is bad for company morale."

"Harvey, safety isn't a matter of attitude or morale. It's about applying basic technical principles when critical decisions are reached."

"That's not what our Safety Department teaches. We should be 'SMART':

- **S**afety
- **M**inded
- **A**lert
- **R**esponsible
- **T**houghtful

"Being 'SMART' Norm, is the key to safety. It's our new slogan."

"Well, my attitude is that the key to avoiding death and disaster in process plants, is applying common sense and fundamental technical principles at the time critical decisions are made. Let me give you a few examples, Harvey, from my long experience in the Oil and Gas Industry."

"Oh, okay. But be brief Norm. They're serving barbecued shrimp at the safety meeting. Sure you don't want to come? You'll miss Mr. Overbourne's introduction. I know he'll be looking for you!"

BP—Deep-Water Horizon Blow-Out

On April 20, 2010, eleven people were killed and BP lost about 25 billion dollars, 20 miles from my home, when an off-shore well blew-out. The cause was fundamentally not the poor cement job around the well's casing, or the extremely deep water, or the weak oil bearing sand formation. These were just problems inherent in the project. The real cause of the blow-out is shown in Fig. 46.1.

The integrity of the cement around the completed well casing had to be tested to see if it could hold back the formation pressure when put on-line. Attempts to see if the well could be depressured by venting at *P*-1 (the big line) were unsuccessful. It seemed that high pressure (4000 psig) natural gas was leaking into the well, when the flow was supposed to be shut-off. This meant that millions of dollars would have to be spent, and months taken to repair the leak—likely by recementing the well's casing.

Understanding Process Equipment for Operators and Engineers. https://doi.org/10.1016/B978-0-12-816161-6.00046-1

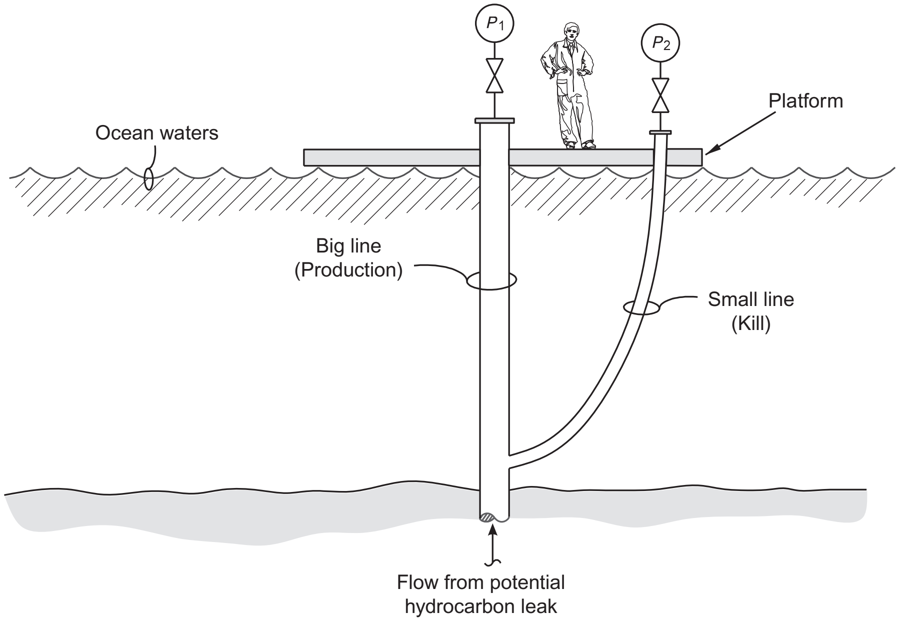

FIG. 46.1 Deep water horizon blow-out 16 killed BP.

So, a couple of the technical guys got together and maybe said, “Let’s try and see if we can depressure the well through the small line, called the ‘Kill Line’” (see Fig. 46.1). And they did, and concluded that the well’s integrity was fine. So they put the well on-line, and the well blew-out and caught fire.

I told this story to my nephew’s 12 year old daughter. “Uncle Norm,” she said, “If both lines started and ended at the same place; and the big line was blowing out gas; and the small line wasn’t, don’t that just meant the small line was plugged with stuff from the well; but the big line was too big to get itself plugged up?”

And this from a girl who thinks Bulgaria is a suburb of Houston.

We, in the oil and gas business need to make decisions with the available information. At the point of decision, I always ask myself, “What’s the worst that will happen if my decision is wrong?” Often, discretion if the better part of valor.

BP—Texas City—Raffinate Splitter Explosion

About 15 or 20 people were killed, hundreds injured, and about five billion dollars lost when a gasoline raffinate splitter blew out a cloud of gasoline, which exploded in Texas City, at a refinery I helped design and operate between 1965 through 1980. It’s true that there were a number of horrendous design problems that contributed to the catastrophe, such as:

- The relief valve was on the vapor line located well below the top head of the splitter tower.

- The relief valve opened into an atmospheric vent (i.e., the blowdown stack), rather than into the flare system.
- Liquid drained from the atmospheric vent (i.e., blowdown drum—see Fig. 46.2) through a loop seal, which vapor locked).

But the real cause of the problem was that the operators didn't understand how their level indication worked. Some of the operators thought they might have a very high liquid level, and that the tower was full. But, when they increased the feed temperature, they noted a reduction in the bottom's level indication, on their control panel. They then concluded that the tower's level indication was okay and continued to bring more feed into the splitter. The tower then overflowed, and pressured-up the relief valve, due to the weight of gasoline overflowing the splitter and pressing down on the relief valve.

The two factors that operators missed were as follows:

1. Hotter gasoline is less dense.
2. The level indicator will read a reduced level when the density of the gasoline is reduced, even though the real level is 100%+.

I learned about this when my Dad taught me to check my car's engine oil. The oil level ran 1″–2″ higher on the dipstick when the engine was hot.

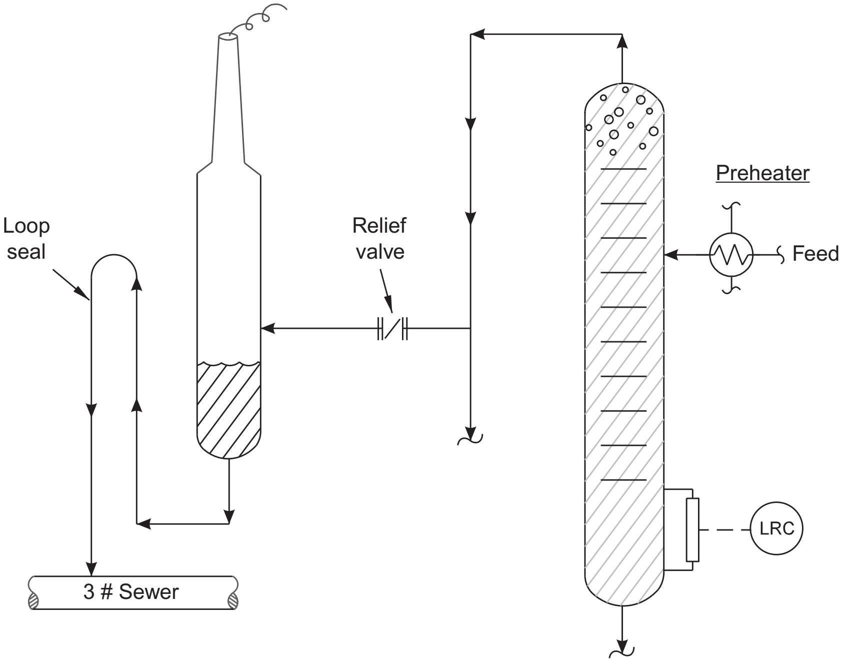

FIG. 46.2 Texas City raffinate splitter—16 killed—Amoco-BP.

Unocal—Chicago—LPG Amine Extraction Tower Failure

Seventeen volunteers of the fire response team, and one operator were killed at Unocal's Chicago refinery (now Citgo), and the alky unit was destroyed, when a C_3–C_4 LPG MEA (monoethanolamine) H_2S extraction tower failed. The vessel ruptured along a circumferential repair weld.

It is quite true that Unocal had been quite negligent in taking measures to avoid the explosion:

- Used MEA, rather than less aggressive DEA, or MDEA.
- Failed to run the MEA reclaimer, to remove heat stable salts from the circulating amine. The heat stable salts promote corrosion.
- Allowed excessive acid gas loading of the MEA solution, which also accelerated corrosion.
- Failed to properly inspect the vessel for cracks at repair welds, during turnarounds (see Fig. 46.3).

But, the fundamental cause of the failure was a bad decision that occurred before the vessel was fabricated. The designer had failed to specify on their design vessel sketch that the extraction tower must be, "*Post-Weld Heat Treated to Relieve Stresses in the Heat Affected Zones of Welds.*"

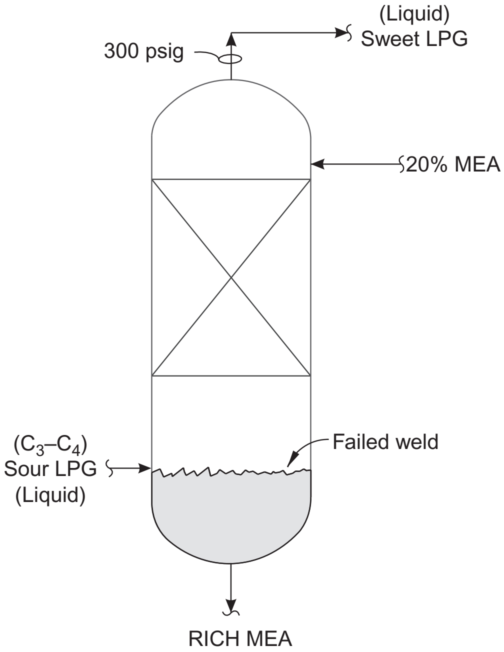

FIG. 46.3 Lack of postweld heat treating results in death and disaster. 18 killed—Unocal.

Thus when a repair weld was made many years later, the repair weld was not postweld heat treated either. What the engineering design contractor had missed was that:

- Amine treating of H_2S extraction is a potentially corrosive service.
- MEA is more corrosive than DEA or MDEA.
- When steel corrodes, one product of corrosion is iron sulfide (HS^-).
- The other product of corrosion is a hydrogen ion (H^+).

Ionic hydrogen will penetrate welds and will accumulate as molecular hydrogen in unstressed relieved welds and cause the welds to fail. Neither Unocal, nor the engineering contractor was found liable for this explosion. But my client, Nooter Engineering, who did the repair weld, was found, quite unjustly, to be at fault.

Shell Norco Explosion

Six operators were killed, the FCU (Fluid Catalytic Cracking Unit) was destroyed, and thousands of homes damaged, when a Cracking Unit blew-up in Norco, Louisiana. The origin of the explosion was a free air detonation of a cloud of propane from a blown-out 8″ overhead vapor line, from the FCU depropanizer. I saw the failed pipe elbow in an inspection laboratory a year after the explosion. It was obvious what had happened (see Fig. 46.4).

Wash water injected into the 8″ overhead vapor line had cut a groove in the outer radius of the elbow. It's true that the water wash nozzle should not have been pointed straight down at the elbow, 2 ft below. It's true that the nozzle should have been a dispersion, or fog-type nozzle, not just a section of ¾″ tubing cut at a 45 degree angle. It's also true that the water wash flow should have been directed in the horizontal run of piping, 10′–20′ upstream of the first elbow.

But the main problem was that the wash water was just plain Mississippi River water that had not been de-aerated. This caused, "*Cavitation,*" as the high velocity water

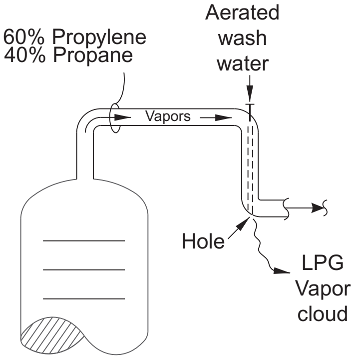

FIG. 46.4 Acidic water impinged on pipe causes erosion and pipe failure—6 killed—Shell.

released oxygen bubbles at the point of impact. Shell should have used de-aerated water as wash water, from the Boiler Feed Water De-aerator Plant.

Apparently, the installation of the ¾″ wash water injection was a retrofit project that had not been reviewed by Shell's engineering department. It was considered to be just a minor piping project to be implemented by a local piping contractor.

I was the expert witness for the lawyers suing Shell on behalf of the homeowners who had sustained very minor property damage. The case was settled by Shell for $300,000,000. The deceased operators' families did not share in the settlement. They were covered by "Workman's Comp," which prohibits such legal action by employees. But the lawyers representing the homeowners received $120,000,000 for their "work."

Union Carbide—Bhopal Disaster

Union Carbide's Pesticide Plant killed over 10,000 people in Bhopal, India. As a consequence, this large, famous, chemical company was liquidated. Bhopal is the worst accident that has ever happened in the process industry.

The cause of this incident has been attributed to a faulty mechanical seal on the Methyl Isocyanate pump. The pump had been designed with a marginal available Net Positive Suction Head (NPSH), which caused persistent cavitation at the entrance to the pump impeller, which caused damage to the seal and frequent seal leaks (see Fig. 46.5).

The resulting methyl isocyanate leaks were an irritant, but not fatal. To avoid this relatively minor problem, plant operators devised a clever solution:

- The pump was shut down.
- The bypass valve around the pump was opened (see Fig. 46.5).

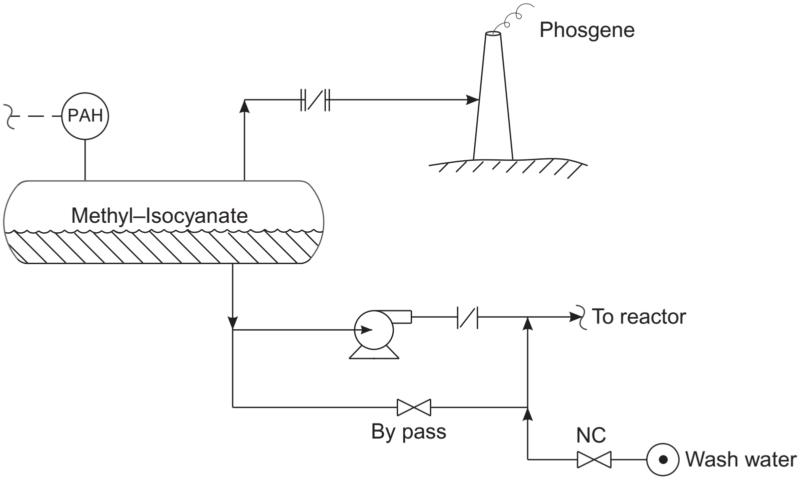

FIG. 46.5 Bhopal, India—10,000+ killed. Union Carbide.

- The Isocyanate vessel was pressured-up by several Bar's, using an existing plant nitrogen purge connection.

As a consequence of the Isocyanate vessel operating at a higher than normal pressure, the vessel pressure alarm would periodically be activated. To avoid the annoyance of the alarm, the alarm was deactivated.

During World War One (1914–18), the Germans introduced poison gas. The most terrible being phosgene. To produce phosgene:

- *Step 1*—Take Methyl Isocyanate.
- *Step 2*—Add water.
- *Step 3*—Mix gently for 5 min.

At the Bhopal Plant, wash water (Fig. 46.5) was used periodically to flush out the feed line to the reactor. With the Methyl Isocyanate pump running, and the pump bypass valve shut, water could never flow back into the vessel.

However, with the pump out of service, and the bypass valve open, water could—and did—one evening, back-up into the vessel. The result was a rapid evolution of phosgene vapors that caused the vessel to overpressure. The relief valve opened, and fatal phosgene gas settled over the city of Bhopal, India. The water injection continued even as people died.

But why?

Because the high pressure alarm had been deactivated, due to running the vessel at a higher than normal pressure with nitrogen. Thus the control panel operator was not alerted to the problem of the excessive vessel pressure.

The real cause of the Bhopal disaster was not a faulty pump mechanical seal. It was not an equipment failure or a design error. It was a failure to follow my Mother's advice:

"Norman, listen. Hope for the best, but plan for the worst."

"Mom, what does that mean?"

"Norman, you have to examine the consequences of your actions. Think things through before you make changes. You're not a child anymore. Bad things happen to people that act without thinking carefully first. Just following all the rules is not good enough. Try to be more like your Uncle Herb. He's an engineer!"

Amoco, Texas City—Death by Nitrogen

Three contractor maintenance workers died from nitrogen suffocation in a hydrocracker reactor. The reactors had been purged with nitrogen prior to issuing entry permits. Two identical reactors had been tested for breathability. The third was full of nitrogen only. The three workers entered the wrong reactor, collapsed, and died. This happened around 1970.

One only feels difficulty in respiration due to ½% CO_2 in air. If no CO_2 is present, regardless of lack of O_2, no respiration discomfort will be experienced.

I once (1982) connected plant N_2 to the instrument air system. I almost killed everyone in the control room. Our air operated instruments displaced the actual air with N_2 emitted by the instrument panels. Fortunately, a low O_2 alarm sounded, and I realized my error.

"God watches over the feeble minded." (Good Hope Refinery, Norco, Louisiana).

Summary

"Lieberman," Harvey said, "what's the point of all these stories? They are certainly tragic – but seem disconnected. Accidents will happen! The best way to minimize accidents, is for all employees to have a positive attitude towards safety awareness."

"No Harvey. The best way to avoid accidents is to understand when to:

- Stress relieve vessels and piping welds.
- Know when equipment or piping has been properly depressured.
- Understanding how level measurements vary with liquid density.
- Not using air saturated wash water without a spray nozzle that impinges on piping.
- Which alarms should never be deactivated.

That is Harvey, to understand how the process equipment actually works and the experience to anticipate what can go wrong, when lessons learned from past experience are ignored."

"Those who do not study history, are condemned to repeat it."

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Our Engineering Function in an Era of Global Warming

As a refinery process engineer, my work brings me into contact with technical and operational management. These individuals determine implementation of energy saving projects such as:

- Training operators to run process units in an energy-efficient manner.
- Power recovery turbines.
- Air preheaters.
- Frequency control for electric motor-driven pumps.
- Recovery of steam condensate.
- Maximizing heat recovery from product.
- Recovery of low BTU waste gas streams.

When I discuss with these decision makers their objectives for operations, I have found their main concerns are [1] as follows:

- Maximizing short-term return on investment.
- Plant safety and operability.
- Maximizing heavy gas oil recovery from vacuum tower residue.
- Maximizing crude rates.
- Avoiding labor relations problems.
- Minimizing maintenance expenses.

I have found a lack of interest in what I would consider longer term objectives:

- Energy conservation.
- Climate change.
- Composition of the atmosphere.
- Global warming.
- Sea level rise.
- Ocean pH decrease.
- Loss of biodiversity.

The individuals who manage and direct petroleum refineries are typically Chemical Engineers in their early 40s to late 50s, married, with grown children. Most do not know the current or past CO_2 levels in the atmosphere, or the rate of increase of ambient global

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temperatures, the rate of sea level rise, or the pH trend of the oceans. If technically trained older colleagues do not know—and do not care to know—these trends, what can we expect younger engineers and operators to understand?

Explanation to Refinery Management Personnel

To explain the serious nature of the problem, which is basically a lack of the timely adoption of sustainable energy options, I have developed a short presentation, which has *NOT* been greeted with any great degree of enthusiasm or interest by refinery management personnel. It isn't so much that my audience does not believe me; it may be that they simply don't care to hear the "Inconvenient Truths."

To start with, I'll explain that when the earth was created 4,200,000,000 years ago, the CO_2 in the atmosphere was about 250,000 ppm (25%). Most of the rest of the atmosphere was then, and still is, nitrogen.

Start of Life

As soon as the earth had sufficiently cooled, for free water to exist, life formed. Naturally occurring molecular substances formed in interstellar space, apparently rather readily form complex molecular structures that can spontaneously reproduce themselves. All that is required is liquid water, carbon, sulfur, nitrogen, phosphorous, calcium, and other common elements. As a source of energy, iron, methane, hydrogen sulfide, and other common chemical elements can be utilized in exothermic reactions. Another relatively minor source of energy is oxidation using hydrogen peroxide produced from the solar radiation striking a molecule of water. The bacteria that used these minor sources of energy reproduced and populated our planet, without altering the composition of the atmosphere to any great extent. That's as far as life on our earth evolved for about 2 billion years.

Photosynthesis

What may make the earth unique, in our particular universe, is not life, which apparently begins relatively quickly and spontaneously given the appropriate chemical environment, but photosynthesis. Whereas life evolved almost instantly on our planet as it cooled, photosynthesis:

$$CO_2 + H_2O + \text{photon} \rightarrow O_2 + \text{glucose} \tag{47.1}$$

required several billion years to develop. Photosynthesis evolved from certain bacteria's ability to use hydrogen peroxide as a source of energy to convert CO_2 to glucose. The catalyst that promoted Eq. (47.1) was chlorophyll. Bacteria that use chlorophyll to manufacture glucose from ambient CO_2 had a tremendous environmental advantage over bacteria that relied on relatively scarce iron, vanadium, nickel, and sulfide ions, as a source of energy.

What Happened to the Oxygen

Early bacteria responded to free oxygen as a deadly and hazardous waste product. And so it was, and still is. Free oxygen does not exist in any great concentration in the universe. It's too reactive. Prior to the advent of photosynthesis, the oxygen concentration in the earth's atmosphere was about 3000 ppm (i.e., 2 mm of Hg partial pressure), as compared to the current level of 210,000 ppm (i.e., 160 mm of Hg partial pressure). The small initial concentration of 3000 ppm O_2 was a consequence of solar radiation interacting with water molecules to produce peroxides and hydrogen in the atmosphere.

Oxygen is a dangerous poison to life. It reacts with strands of DNA. Oxidation of DNA is the leading cause of death (sometimes referred to as old age) in humans. At first, the bacteria were protected from death by oxidation from their own waste products. The excessive oxygen dissolved in sea water where it reacted with the iron salts in the sea and precipitated as reddish iron oxides. Hence, the reddish color of many soils today. But, after a billion years or so, the ocean's iron content was almost entirely precipitated as oxides, and thus free O_2 began to accumulate in the air.

Results of Increase in Atmospheric Oxygen

The bacteria both loved and hated free oxygen. Oxygen was great, in that oxidation reactions, especially:

$$\text{Glucose} + O_2 \rightarrow CO_2 + H_2O + \text{Energy} \tag{47.2}$$

produced immense amounts of useful energy. But, oxygen also damaged the bacteria's DNA, resulting in death. So, the bacteria, after considerable consultation, decided to protect themselves from the deadly effects of oxidation by:

- Clumping together to form colonies to reduce their collective surface area.and then
- Building cell walls around their colonies with semipermeable membranes.and then
- Clumping the cells together to make animals such as sponges and plants like blue-green algae.and then
- Arranging the cells, so that each cell had a special function that helped protect its neighbors from the excessive O_2 partial pressure.and then
- Creating ferns, grass, trees, worms, fish, frogs, alligators, bears, monkeys, and me.

And as all this creation was proceeding, starting about 1 billion years ago, the methane in the atmosphere was oxidized to CO_2 and H_2O. Methane (CH_4) is a much more powerful (23–40 times as much) greenhouse gas as is CO_2. So the methane's oxidation reduced global warming quite a bit. However, the bigger effect was the conversion of CO_2 to glucose (Eq. 47.2).

Most of this glucose was used by the bacteria, trees, and humans as a source of energy, by reacting it with O_2 and releasing the sun's radiant energy stored up by photosynthesis (Eq. 47.1). But some of the organic molecules derived from the glucose were buried before they could react with oxygen. Thus the origin of:

- Natural Gas
- Shales
- Petroleum
- Coal
- Tar Sands
- Off-shore hydrates (frozen $CH_4 + H_2O$)
- Peat

Current CO_2 Atmospheric Concentrations

The green plants were so efficient in creating organic carbon molecules using radiant solar energy, H_2O, CO_2, and Mother Earth was so dynamic in burying the resulting organic sludges, that 99.9% of the carbon molecules ($CH_4 + CO_2$) in the atmosphere disappeared underneath the earth's crust. The loss of CH_4 and CO_2 from the atmosphere allowed more solar heat to be reradiated to space. The earth cooled and froze. The white, reflective surface of the earth reflected even more solar heat back into space. Our little home planet was caught up in a positive feedback loop. The earth would have remained entirely frozen, except for one additional factor: Radiation.

Currently, U-235 only represents 0.7 wt% of the total amount of Uranium in the earth's crust (the rest is U-238). Hundreds of millions of years ago, it was present in the core of the earth (along with radioactive thorium) in higher concentrations. Both U-235 and thorium decay to more stable elements. And in doing so, release heat. This heat causes volcanic activity, which releases bursts of CO_2 into the atmosphere. The sudden release of vast amount of CO_2 broke the positive feedback loop which had been initiated by the plants absorbing CO_2 from the air, to produce glucose. The glucose decayed into organic sludge, which was then buried by sediments washed off the land by rain. Fortunately, the power of photosynthesis to absorb CO_2 from the air, and the volcanic action to release CO_2 from buried hydrocarbons, and the ability of sediments to cover-up organic sludges, reached a rough equilibrium. A natural balance had been achieved.

Reduction of U-235 Heat Release

As the concentration of thorium and U-235 inside the earth diminished with time, volcanic activity gradually subsided. And the ability of the planet to recycle CO_2 from buried hydrocarbons diminished. By 1942 (the year I was born), the concentration of CO_2 in the atmosphere had diminished to about 280 ppm, or 0.1% of its prephotosynthesis concentration of several billion years ago. Tree growth was being suppressed due to the lack of available CO_2 in the atmosphere. The earth appeared to be trending toward a new ice age. For example, the Norse settlement in Greenland, which was dependent on dairy cattle, which began to flourish in AD 1000, had largely vanished by AD 1400, due to our chilling planet.

However, about 200 years ago, Thomas Newcomen in Cornwall, invented the steam engine, which used coal as fuel. The CO_2 in the atmosphere slowly increased from 280 to 300 ppm (by 20 ppm), up to the 1970s, when the atmospheric CO_2 concentration rapidly accelerated from 300 ppm to its current level of 420 ppm, at a shockingly steady rate of 0.5%, or 2.5 ppm per year.

Man's activity in the oxidation of buried carbon is off-setting the effects of photosynthesis, which resulted in the suppression of plant growth. Indeed, the increase of CO_2 in the atmosphere from 300 to 400 ppm has accelerated the rate of tree growth in the Northern Hemisphere by about 10%–20%. A hopeful sign, which is significantly off-setting mankind's own lack of real progress.

CO_2 Distribution

The CO_2 released by the combustion of natural gas, coal, and oil does not entirely accumulate in the air. About 40% dissolves in the ocean [2]. The incremental CO_2 in the ocean due to man's activities is too small to affect the ocean as a whole. But, the ocean is not well mixed. The top few hundred feet mix slowly with the 95% of the deeper oceanic waters. Thus, the incremental CO_2 released by the combustion of fossil fuels accumulates in the upper layers of the ocean. As CO_2 dissolves in water, carbonic acid (H_2CO_3) is produced. This carbonic acid has measurably suppressed the ocean's pH (from about 8.0 pH toward 7.8 pH). The solubility of acidic components such as CO_2 in water is reduced, as the water pH is suppressed. Thus, the rate of CO_2 absorption in the ocean from the atmosphere will slowly diminish with time.

The atmosphere is of course warming due to the combined effects of CO_2, CH_4, and Freon. About 80% of the warming is related to CO_2. The reduction of SO_2 (SO_2 reflects sunlight) emissions into the atmosphere, due to the construction of sulfur recovery plants in petroleum refineries, has accelerated global warming. So far, ambient temperatures have increased in the past half century by around 2°F. Sea water temperatures are also increasing. Water swells up as it warms (1% for each 40°F increase), thus the rising sea levels, as noticed during Hurricane Sandy in my hometown of New York City.

Energy Conservation Projects

A large component of my business activity as a process design engineering consultant was based on energy conservation projects. This was formerly a "hot topic" in petroleum refining. In the United States, incremental energy costs are a function of the price of natural gas. This set the cost, not only of fuel to process heaters, but also the cost of steam, and less directly, the cost of electric power. Natural gas has become so cheap and plentiful in the United States (its price has declined by two-thirds in the past few years), that my clients no longer consider energy conservation projects, on their own, as worthwhile objectives.

This is the real effect of “Fracking.” It increases the rate of production of both natural gas and crude oil. There is a widespread misconception as to the controlling factor of the rate of CO_2 emissions. It is not a function of consumption of fossil fuels, but a function of the rate of production of fossil fuels. All of the oil and gas that is produced will be oxidized eventually to CO_2 and H_2O.

Potential for CO_2 Production

In 2008 the idea of “Peak Oil” was common. Oil futures sold for $147/bbl. I know this for sure, because I purchased some. Currently, oil has dropped (as has natural gas) to a third of its peak value, largely due to increased rates of production, as modern technology, such as directional drilling and improved fracking chemicals, is employed. Projects that I have participated in that also could have enhanced hydrocarbon production are as follows:

- Naphtha diluent used to produce superheavy crude oils in Venezuela.
- Natural gas from Mozambique converted to gasoline and diesel in Mussel Bay, South Africa.
- Fractionation of Gas Oils from Tar Sands in Northern Alberta, Canada.
- Injection of CO_2 into gas field formations in Wyoming.

It’s quite apparent that peak oil production was not reached in 2008, as was feared at that time. Actually, quite a bit less than 1% of potentially recoverable hydrocarbons have been extracted by mankind as fossil fuels so far. That estimate is based on the archaic level of CO_2 in the atmosphere, which has been sequestered by Mother Nature. Not so much as oil, gas, and coal, but mainly in the form of:

- Natural gas hydrates frozen in off-shore deposits.
- Shale Oil distributed world-wide.
- Tar Sands—mainly in Northern Canada.

Luckily, for our planet as a whole, CO_2 concentrations greater than several percent are fatal to mankind, but quite beneficial for plant life. And, quite tolerable to the bacteria which initiated the problem with their invention of photosynthesis.

Positive Feedback Loop

The ultimate problem is not actually CO_2, but CH_4. The real danger is not a warming atmosphere, but the heating ocean. The difficulty arises from two factors:

- The warm surface waters of the ocean tend to float on the colder, denser layers.
- Increased fresh water run-off from melting glaciers is reducing the salinity, and hence the density of the relatively shallow waters in the continental shelves.

These two factors are promoting the sublimation of hydrates in the outer continental shelves of the world’s oceans which means that the frozen hydrates will release accelerating rates of methane. As a mole of CH_4 has 23–40 times the effect, as a global warming agent, as a mole of CO_2, a catastrophic greenhouse positive feedback loop is being created.

Our planet has experienced this problem before. That is, the rapid evaporation of hydrates to methane. Comparatively rapidly, the huge concentration of the reactive oxygen in the atmosphere will oxidize the CH_4 back to H_2O and CO_2, which will then support photosynthesis (Eq. 47.1). Equilibrium, likely at the current CO_2 and CH_4 concentrations in the atmosphere, will, with time be reestablished.

The Problem of Time

The warmer, lower pH surface ocean waters will eventually mix with the 90% of ocean water not affected by mankind's fossil burning activities. Plant life will sequester excessive CO_2 in the atmosphere. Hydrates will reform, and our earth will resume its gradual and natural cooling and warming cycles. I could observe that a desirable, steady-state operation, had been restored. The positive feedback loop will be damped out and equilibrium will be reestablished. Likely, it will only take a few thousand years.

Will humankind be able to wait out this inconsequential period of instability in our planet's environment? Like any process plant upset, stability will either be restored, or the unit will spin out of control, with disastrous, and perhaps, fatal consequences.

But one thing I can say for certain. *Time is not on our side.* The destructive results of the continued procrastination in the development of alternative and sustainable energy resources are now apparent. The effects of increasing CO_2 in the atmosphere are a matter of calculation, not only observation.

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[1] Personal correspondence with Mr. Karthik Rajasekaran, Metairie, Louisiana.

[2] E. Kolbert, The 6th Extinction, Picador Publications, 2014.

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Suppressing CO_2 Emissions

My idea of sustainable energy originates from an argument I had with my mother.

"Mom, I need $12 for new shoes."

My mother examined the worn out soles, "Norman, here's 80 cents. Take the shoes to Mr. Federiko, the shoemaker."

"But mom, I want new shoes."

"Norman," my mother explained, "We're poor people. We can't afford new shoes. We've got to get by with what we have."

It's the same with providing energy for our planet. We reside on a tiny speck of dust in the vast Cosmos. If my mother had ruled the earth, she would have commanded:

"People! We live on a poor planet. We can't afford complicated, expensive new technology. We've got to provide energy with the technology and process equipment that we have today. Children of the earth! Time is not on our side."

So far, we have racked up over 1°C increase since 1975 for land-based temperatures. Within 20 years, we'll add another degree. Two degrees is a lot. The extra heat will promote rapid plant growth and improve conditions for many species on our planet. Global warming will also promote more violent storms, desiccation of the interior of large land masses, and rising sea levels.

Most hydrocarbons consumed are devoted to transportation, process plants, electricity generation, and heating. The scope for improvement is small in these areas. Increased use of vehicles offsets increased efficiency. Fertilizer and plastic production are unlikely to be curtailed.

The big opportunity involves electrical power generation. One-third of the world's carbon consumption is used for this purpose. What then are the alternates to coal-fueled power plants?

- Nuclear—The problem with nuclear power is not technical. Safety, spent fuel rod storage, and plutonium toxicity are all problems with known solutions. The danger is people. Engineers, operators, and managers make mistakes. These errors, when manifested at a nuclear power plant, have created unacceptable consequences in Japan, Russia, and the United States. Thus, expansion of nuclear power in the next 20 years will be difficult.
- Wind and Solar—Electricity generated by onshore windmills and solar (PV) cells, in selected areas of the United States is only 20%–30% more expensive than electricity generated by coal-fired power plant designed to remove hazardous components (but not CO_2) from its flue gas. The problem is that wind and solar electricity must have 100% back-up in the power grid for nights when there is no wind. As most of the cost of

https://doi.org/10.1016/B978-0-12-816161-6.00048-5

generating electricity from coal is capital investment and not fuel, this problem doubles the cost of electricity from onshore wind and solar (PV) panels.

- Energy Storage—Energy can be stored by hydrogen generation, compressed air, water storage, flywheels, batteries, and so on. However, none of the possible methods are economically feasible on a large scale.
- Natural Gas—CO_2 emissions in the United States are falling, as natural gas becomes cheaper. On a BTU basis, natural gas now costs 25%–30% of crude oil. As natural gas emits only half the CO_2 as coal, per KWH of power produced, its increased use in the United States has trimmed CO_2 emissions by a few percent. Note that when natural gas is used to generate electricity in a combined cycle power plant:
  - Gas is combusted in a turbo generator.
  - The hot flue gas (1100°F) is used to generate steam, which is then used in a steam turbine to generate more electricity.

The cost of electric power from the combined cycle plant is only 60% compared to a new coal fired plant. Outside the United States—in Europe, China, and Japan, natural gas costs are closer to crude.

- Hydro, Geo-Thermal, Tidal—These are excellent methods of generating power free of CO_2 emissions. But, these options are available only in small localities, such as Iceland and certain coastal regions.
- Ethanol, Bio-Diesel, Hydrogen, Algae, Switch-Grass, Ocean Waves, Fusion—None of these are of any short-term applicability. Ethanol from corn, in particular, is a terrible fraud.

What then is the solution? Fortunately, there is one factor that has mitigated the problem. Trees! The rate of tree growth has accelerated by as much as 20%—mostly due to extra CO_2 in the atmosphere (from 280 ppm up to 410 ppm) during the past century. If we can only moderate CO_2 emissions and give the world's forests a chance to catch up, perhaps in another 50–100 years, the earth's atmosphere will stabilize at 500–800 ppm, which is a desperately high, but not a fatal CO_2 concentration.

So, let's all get out of our offices, skip the meetings, and use the principles described later to run our units more efficiently. Here are a few ideas that I have extracted from my troubleshooting and operating experiences in refineries. I have followed the Golden Rule in assembling these suggestions:

"If I haven't done it myself, I wouldn't recommend it."

- *Steam Turbine-Driven Pumps*
  - Reduce the turbine speed on the governor until the pump's downstream control valve is running in a mostly open, but still controllable position (reference Fig. 48.1).
  - Close port (hand) valves, until the governor valve is running essentially wide open.
  - Adjust linkages for the governor speed control valve, until the pressure drop between the steam supply line and the turbine steam chest pressure is only a few percent of the motive steam supply pressure.

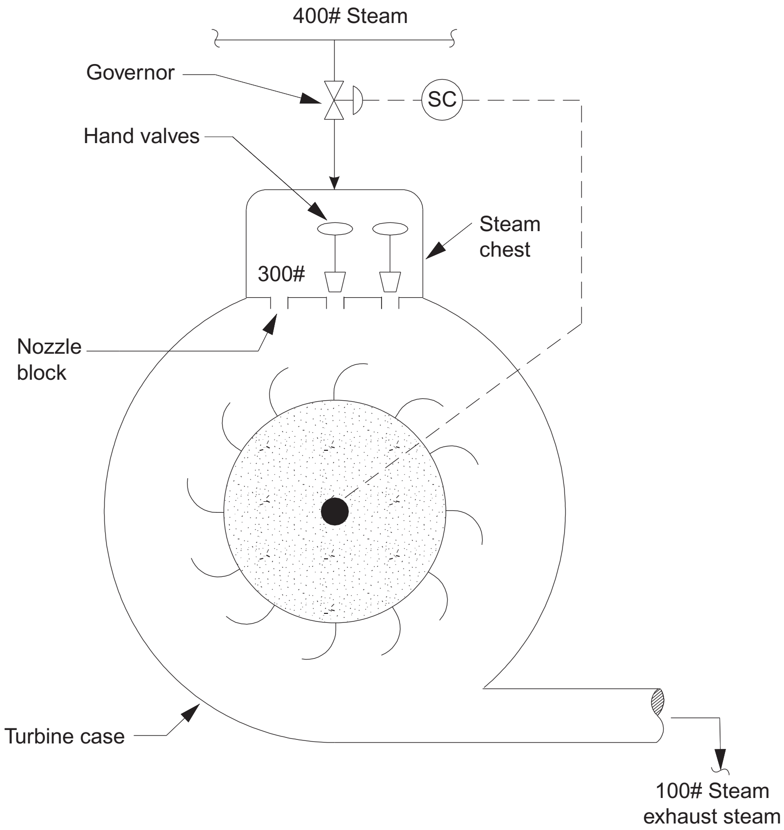

FIG. 48.1 Steam turbine showing hand valves.

- *Motor-Driven Centrifugal Pumps*
    - Renew the impeller wear ring to reduce the impeller to wear ring clearance to design (Fig. 48.2).
    - Renew the interior of the pump case to reduce the impeller vane to internal case clearance back to the design value.
    - Reduce the diameter of the impeller so that the downstream process control valve is operating in a mostly open, but still controllable position.
    - Check that pumps are rotating in the correct direction, as indicated by the arrow on the pump case.
    - Clean the back screen of the motors. This will help the motor run a bit cooler and more efficiently.
    - Do not run spare pumps, if pumps are operating on the flat portion of their performance curves.

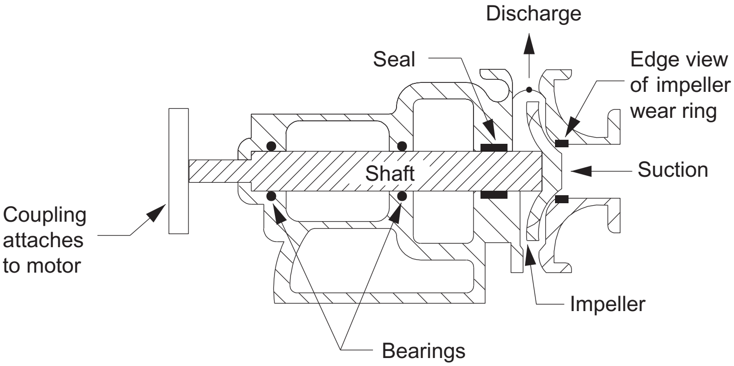

FIG. 48.2 A centrifugal pump.

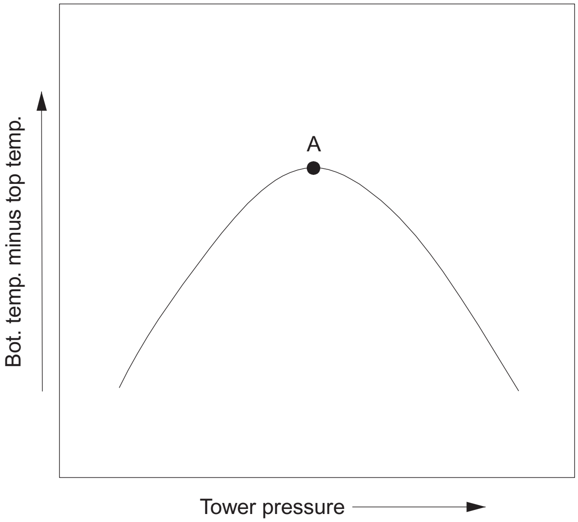

FIG. 48.3 Point A represents the optimum tower pressure.

- *Distillation Tower Pressure*
  - Operate towers at a minimum pressure, consistent with not flooding and with the ability to condense the overhead product. Lowering the tower pressure will enhance the relative volatility of the light to heavy key components.
  - At a constant reflux rate, lower the tower pressure consistent with maximizing the temperature difference between the bottom and the tower top temperatures (Fig. 48.3).
  - Some towers fractionate better at a reduced pressure because the vapor velocity Increases, so as to reduce tray deck dumping or weeping.

- An improvement in separation efficiency will also permit a reduction in the tower's top reflux rate. As the reflux comes from the reboiler, this will result in energy savings by reducing the heat input into the reboiler.

Distillation Tower Tray Efficiency

Tray decks and weirs need to be level. Out-of-level tray decks promote tray weeping. As the vapor will flow through the high point of the tray deck, and the liquid will weep through the low point of the tray deck, vapor-liquid channeling will result (Fig. 48.4). This will degrade tray contacting efficiency, and thus result in more reflux and hence more reboiler duty. One way out of this problem is to use old style bubble cap trays. Bubble cap trays cannot leak, if they are reasonably mechanically intact.

Weirs need to be level. If the crest height is less than the out of levelness of the weir, then the liquid will channel across the tray. Stagnant areas of the tray will not contribute anything to the tray efficiency. The vapor will simply bubble through these stagnant layers of liquid, without undergoing any change in composition. Of course, it's better to level up the weirs. But, if the liquid flow over the weirs is small, I'll resort to the use of picket weirs. Such picket weirs encourage the liquid to overflow the weir in a more uniform pattern. This will largely eliminate stagnant areas on the tray panels.

There has been a lot written about new tray design by the vendors to enhance tray fractionation efficiency. Mostly B.S.! The real answers to enhanced tray efficiency are bubble cap trays and/or picket weirs. Also, maximizing weir and tray deck levelness.

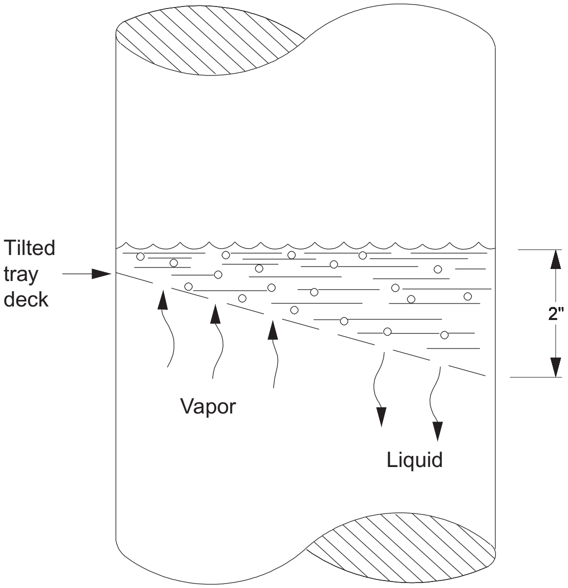

FIG. 48.4 Out-of-level trays cause low tray efficiency.

Heat Exchanger Design

Of the newer, high-tech options for heat exchanger design, the best option appears to be use of helical tube support baffles (Fig. 48.5). These baffles cause the shell side liquid flow to swirl past the tubes in a circular pattern. However, for new exchangers, it's cheaper to purchase a somewhat bigger exchanger, to increase heat recovery. The use of helical tube support baffles is a good retrofit option for increased heat recovery.

Helical baffles serve little purpose if the controlling resistance to heat transfer is on the tube side, as is often the case. If the main resistance to heat transfer is on the tube side of an exchanger, then tube inserts, in the form of springs, are used by some of my clients, but with variable success. I myself have never used these spring-type inserts, which promote turbulence inside the exchanger's tubes. In general, the use of helical baffles to aid in heat transfer efficiency is more common in the process and refining industry, than tube inserts or twisted tubes.

The best way to maintain a good level of heat recovery in a shell and tube exchanger (for liquids) is to maintain high velocities. For example:

- Shell Side—Keep the fluid cross-flow velocity between 3 and 5 ft per second.
- Tube Side—Keep the fluid linear velocity between 6 and 12 ft per second. Much above 12 ft per second, the liquid becomes erosive.

Excessively low velocities lead to laminar flow, and excessive heat transfer film resistance and fouling.

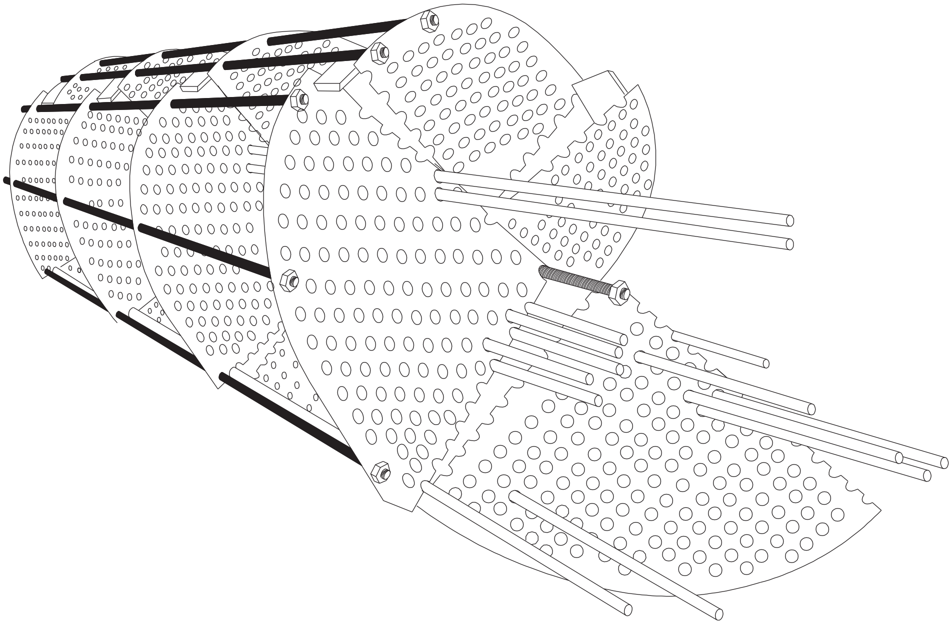

FIG. 48.5 Schematic arrangement showing helical tube support baffles in relation to tube alignment.

Fired Heaters—Excess Oxygen

Adjusting excess oxygen is important. If the heater is being operated at a fixed fuel rate, then the excess air should be adjusted to maximize the heater process outlet temperature (Fig. 48.6).

If, more commonly, the heater is being operated on automatic temperature control, then the excess air should be adjusted to minimize the fuel gas rate, to achieve the desired heater outlet temperature.

Note that I have not made reference to oxygen analyzers or any excess oxygen concentration target in the stack or firebox. Excess oxygen analyzers are not needed for an effective program to maximize energy efficiency. When I adjust excess oxygen, either with the stack damper or with the burner air registers, I do not use, and do not need, an oxygen analyzer.

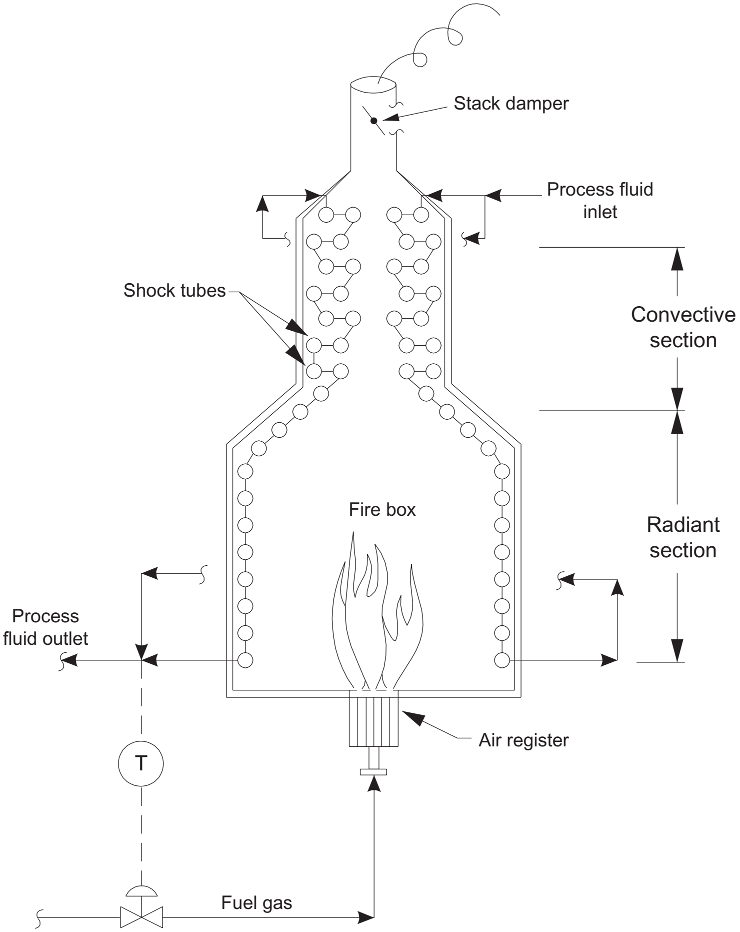

FIG. 48.6 Typical natural draft gas-fired process heater.

Tramp Air Leaks—Radiant Section

Air drawn into the firebox will increase the supply of combustion air. Tramp air in-leakage will not mix efficiently with the fuel. This degradation of the air-to-fuel mixing efficiency will require more excess O_2 for the same degree of fuel oxidation completion. The most common source of tramp air leaks in the radiant section is defective secondary air registers of burners that are not in service. Sight-ports that have fallen off are another potential cause of poor air-fuel mixing efficiency.

More combustion air means more flue gas up the stack which is the major portion of wasted heat in a fired heat.

Tramp Air Leaks in the Convective Section

Cold ambient air will degrade the heat recovery in the furnace's convective section. This happens because of a combination of two malfunctions:

- Excessive Furnace Draft
- Tramp Air Leaks in the Convective Section

A reasonable target for draft in the lower portion of the convective section is 0.05″ H_2O. However, this is only a reasonable target in areas where the wind does not gust at the top of the furnace stack. I have seen moderate wind gusts cause draft to vary by as much as 0.2″ H_2O. Here's the problem. If you are running with a minimum draft (0.05″ H_2O), and the wind stops, the firebox may go positive. Hot flue gas, which may contain high concentrations of SO_2 (which is toxic), can then blow-out of the upper portions of the firebox. This happened to me once in Aruba at the Coastal Plant. Quite a terrifying experience.

Perhaps a minimum safe draft in a windy area is 0.15–0.20 in. of H_2O.

Reducing the draft is typically done by throttling on the stack damper. I was working at the CVR refinery in Kansas recently and reduced the fuel consumption on their natural draft crude heater by 2%. Admittedly not a lot. But, it only required a few hours. Restricting the damper too much will cause a dangerous positive pressure.

Finding Tramp Air Leaks

I have two methods of finding tramp air leaks in the convective section of a natural draft heater. While in operation, I use a silk cloth to see which locations in the convective box will attract the silk cloth. When a heater is out of service during a unit turnaround, I will get an old tire and set it on fire. Or, throw colored smoke bombs into the idle firebox. With the stack damper shut, the convective section air leaks will then be easily observed. Fixing leaks is simple. Use a roll of aluminum duct tape or insulator's mud. Do not expect any dramatic improvement in efficiency. About one or 2% is all that can be expected.

Not all convective boxes leak. On a new heater that I've inspected, the percent oxygen below and above the convective tube banks was 2%. However, for older heaters, the oxygen above the convective tube bank is typically one to 4% higher than the flue gases leaving the radiant box, due to ambient air leaks. Unless you measure the O_2 above and below the convective tube bank, you'll never know the potential energy savings to repair these convective box leaks.

Vacuum Systems

Once, I was making a presentation at a refinery in California. I was trying to sell my services as an energy expert consultant. Halfway through my talk, I noticed that no one was listening.

So, I said, "Look guys. Take me to any part of your plant, and I will save energy within twenty minutes."

The training manager, Mr. Henry, escorted me to their lube oil vacuum tower. I asked the operator on the control panel if he had any problem on the vacuum tower controls (Fig. 48.7).

"Yeah Lieberman. The vacuum tower pressure control's runnin' too far open. That spill-back control valve can't control the tower pressure in a mostly open position. It needs to be 50% open, not 90% open for good control."

So, I climbed up to the three primary steam jets, and blocked-in the process and steam side of the center jet of the three jet system. Steam consumption dropped by one-third (12,000 pounds per hour of 450 psig steam), and the spill-back pressure control valve closed to the optimum of 30%.

"See Mr. Henry, I just saved you 12,000 pounds per hour of steam."

But Mr. Henry was not impressed, "So what Lieberman? Anybody could have done that. We didn't need any energy conservation expert to close off that ejector steam."

And that, Ladies and Gentlemen, is the point. Anyone can effect these changes to save energy, if they will get out of their office, escape from computer-land, and work with the operators in the field.

The simplest way to save 10%–30% of the energy for ejectors is to throttle back on motive steam pressure. Excess steam pressure, above design, reduces the compression ratio that the ejector can develop. Also, the reduction in motive steam pressure will be linear with the reduction in the motive steam flow rate. That is, if I throttle back the steam pressure from 150 psia (135 psig), down to 120 psia (105 psig), I will save 20% of the steam consumed in the ejector. The majority of ejectors I work with suffer from fouled downstream condensers and/or worn steam nozzles. For these jets, the optimum steam pressure is typically 10%–50% below the manufacturer's nameplate motive steam pressure.

Air Coolers

Fin-fan air coolers must be sized for peak summertime operations. Thus, for much of the year, most plants have excessive air cooler capacity. To reduce this excess capacity, many air coolers have louvers on top of the tube bundles. The louvers are closed to restrict

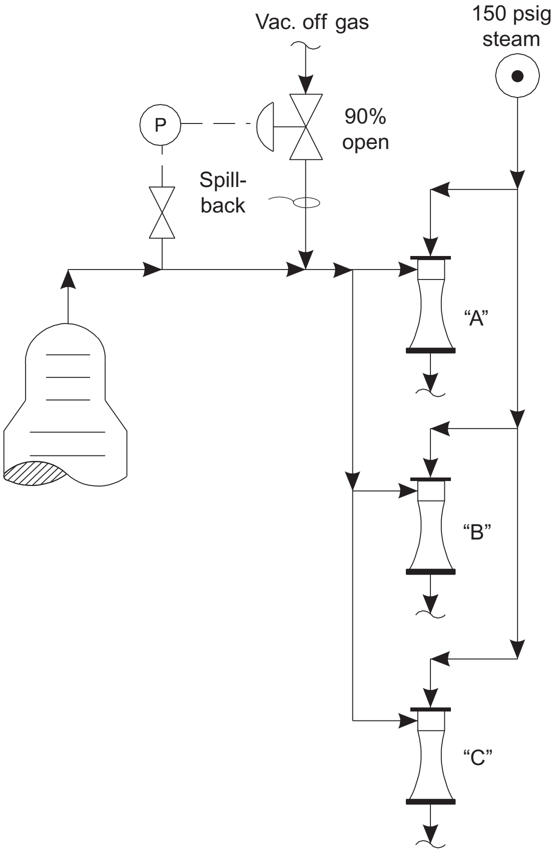

FIG. 48.7 Spill-back pressure control. Too many jets in service.

excessive cooling air flow. However, there is a more energy-efficient way to reduce air flow. That is, reduce the blade angle pitch. I've only done this once. At a Sasol plant in South Africa. I reduced the fan blade pitch from 20–25 degrees, down to 10–15 degrees. I could only make these adjustments to the nearest five degree increment. Two nuts had to be loosened on each blade. The result was small. Twelve percent reduction in the measured amp electric load on the AC motor. But it only required 1 h.

Centrifugal Compressors

As the molecular weight of gas flowing to a centrifugal compressor increases, the gas density will also increase. This will not affect the polytrophic head produced by the compressor. However:

$$\text{Delta P} = (\text{Polytrophic Head}) \cdot (\text{Density})$$

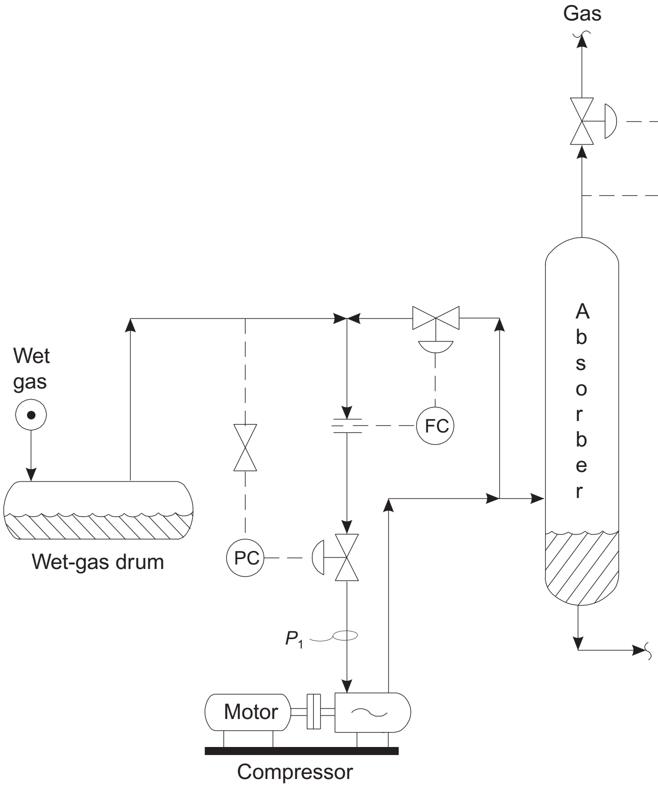

FIG. 48.8 Suction throttling pressure control.

Thus, the compression ratio of the machine will increase, as the molecular weight of the feed gas increases. As a consequence, either the compressor's discharge pressure will rise or the compressor's suction pressure will decline. Either way, the operator will be forced to slow down the compressor to avoid the unwanted increase in compression ratio.

Suppose the machine is driven by an ordinary fixed speed, 3-phase, AC motor. Then what?

The usual operating response is to open the spill-back suction pressure control valve (Fig. 48.8). Which works fine. But, the recycled gas has to be recompressed, which wastes energy. A more energy-efficient response to excessive molecular weight is to suction throttle. It may seem odd, but suction throttling is a more energy-efficient response to controlling the delta P developed by a constant speed centrifugal compressor, than is spilling back gas to the compressor's suction. Try it yourself. Throttle the back-pressure control valve by a few psi on your constant speed compressor. Watch the spill-back valve close. You will see that the amp load on your motor driver will decline. But, don't cause the spill-back valve to close too much. You may cause the compressor to surge and potentially damage the rotor.

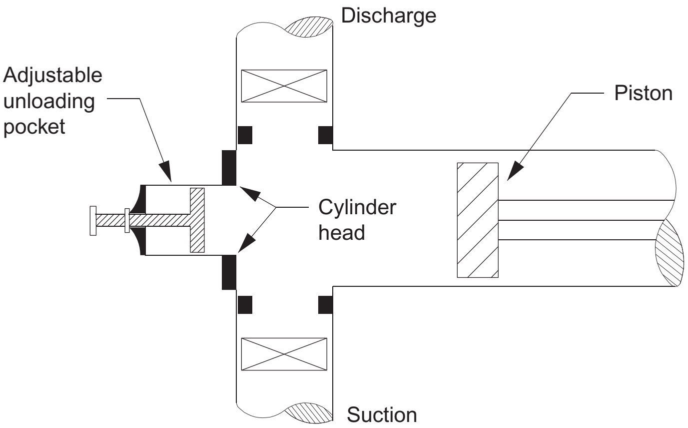

FIG. 48.9 An unloading pocket reduces engine load and volumetric capacity.

The relationship between the energy requirement to run a constant speed compressor, the operation of the suction throttle valve, and spill-back valve, and surge, is a complex subject. I've explained these relationships in detail in my book: *Troubleshooting Process Operations, 4th Edition* – PennWell Publications.

Reciprocating Compressors

The most energy-efficient method to control the flow through a recip is by opening the head-end unloader pocket. That's the large wheel at the head-end of each compression cylinder. Typically, this wheel is not operational through lack of use, so you will need a large valve wrench. Opening the head-end unloader all the way will reduce the volume of gas flow by a maximum of 25% (Fig. 48.9). Reducing net gas flow through your reciprocating compressor, by opening the discharge to suction spill-back valve, is a direct waste of compression energy. Use of the valve unloaders (really disabling the compressor's cylinder valves) is not quite as bad as opening the spill-back valve, but not nearly as efficient as using the head-end unloader pocket. Opening this pocket takes several minutes, and can save up to 25% of compression energy, that was being wasted by the discharge-to-suction gas recycle valve.

Extending Mankind's Future on Planet Earth

I imagine all my readers now believe in Global Warming. The problem is rapidly spinning out of control. In summary, I've listed ideas that you can use to reduce CO_2 emissions:

1. Tray Deck Levelness in Fractionators
2. Use of Picket Weirs
3. Optimizing Pressure in Distillation Towers

4. Heat Exchangers—Shell Side
5. Heat Exchangers—Tube Side
6. Fired Heaters
7. Steam Turbines—Topping
8. Steam Turbines—Condensing
9. Motor-Driven Centrifugal Pumps
10. Fixed Speed Centrifugal Compressors
11. Fixed Speed Reciprocating Compressors
12. Steam Condensate Recovery

Tray Deck Levelness

Ordinary valve, grid, and sieve trays will leak at the low points of the tray decks. The vapor, on the other hand, will take the path of least resistance, and flow to the higher points on the tray decks. The result is vapor-liquid channeling, and therefore low tray fractionation efficiency.

To improve fractionation, the operator increases reflux rate, which then requires more reboiler heat. The resulting incremental heat comes from plant steam, which wastes energy.

Leveling tray decks makes the trays work more efficiently and saves energy, especially when the tower is not heavily loaded and/or the tray deck flow path length is long.

Use of Picket Weirs

Picket or notched weirs are not needed if:

- The weirs are level.
- If the weir loading is greater than 4–6 GPM per in. of weir length.

If not, use of picket weirs to increase weir loadings to 4–10 GPM per in. is going to increase tray fractionation efficiency.

But why?

If the crest height (i.e., the height of liquid over the top of the weir) is zero or very small, on the left side of a tray, then all of the liquid will flow across the right side of the tray. The liquid on the left side of the tray deck will be stagnant.

Liquid that is stagnant cannot alter the composition of the upflowing vapors. Hence tray efficiency is lost and can only be regained by increasing both the reflux and reboiler rates. Which wastes energy.

Optimizing Pressure in Distillation Towers

Lowering the tower pressure will increase the difference between the vapor pressures of hydrocarbons. That is, lower pressure increases relative volatility. The greater the relative volatility, the less reflux and reboiler heat is needed to make the required split. However, lowering the tower pressure too much may cause excessive entrainment, which reduces fractionation efficiency.

Optimum tower pressure can be determined by running plant tests at various pressures. There is no way to calculate the optimum tower pressure for an existing unit. It's dependent on:

- Tray deck levelness
- Weir levelness
- Nonideality of the key components

Typically, at reduced rates, a lower operating pressure is more optimum.

Heat Exchangers—Shell Side

The best heat transfer is always obtained by placing the high viscosity fluid (i.e., cold crude; tar) on the shell side. To enable cleaning of the shell side in tar, or asphalt service, use:

- 1″ tubes
- 1½″ tube spacing
- Rotated square pitch
- 316 ss tubes

This will provide straight, ½″ gaps, to hydroblast the shell sides during a turnaround. Fill the space between the tubes caused by the channel side pass partition baffles with dummy tubes. This is only necessary if the shell side tube support baffle cut is perpendicular to the pass partition baffles.

Make sure there are seal strips that prevent shell side bypassing across the gap created on the top of the shell, by the shell side inlet flow impingement plate. Too often, these seal strips are lost in maintenance activities when the bundle is pulled.

Heat Exchanger—Tube Side

Tube velocities should be kept between 6 and 12 ft/s. Above 12 ft per second, tube erosion may occur.

During a turnaround, make sure the channel head pass partition baffle is tightly secured to the channel head tube sheet gasket. Leave off the channel head cover to check this closure. A small gap will likely erode out into a major bypass stream.

High alloy tubes, which remain smooth, will resist fouling far better than carbon steel tubes. However, do not use 316 ss with carbon steel tube support baffles. This will result in galvanic corrosion of the baffles.

I never use tube inserts. They're great at first, but interfere with tube cleaning during a turnaround. Twisted tubes create shell side cleaning problems. Helical baffles, on the other hand, usually work fine, but are expensive and only effective, if controlling resistance to heat transfer is on the shell side.

Fired Heaters

Using an excess O_2 analyzer by itself is a waste of time and money. Exxon now uses a combination:

1. O_2 analyzer in radiant section (not in the stack).
   Combined with:
2. Combustible analyzer (i.e., CO, CH_4, methanol, etc.) in the radiant section.

These two analyzers are connected to a computer to control combustion air. I do not have any hands-on experience with this relatively new system, but it is reported to work very well by my Exxon students.

Always minimize tramp air leaks in the convective section. Either by minimizing draft (0.05″–0.10″) above the radiant roof tubes, or by finding and fixing tramp air leaks in the convective section exteriors.

All convective boxes leak after the first unit turnaround. A reasonable increase in O_2 between the radiant box outlet and the convective section inlet is 0.5%. Two percent increase in oxygen is excessive and wastes energy. That is, cold tramp air cools off the flue gas in the convective section.

Steam Turbines—Topping

Turbines exhausting to the atmosphere, or to low pressure steam (when the low pressure steam is in excess) often run inefficiently. To save as much as 30%–50% of the motive steam:

- *Step 1*—Slow the turbine down using the governor speed control (*not* the steam inlet gate valve), until the process control valve on the pump being driven is mostly open.
- *Step 2*—Close the horsepower valves (typically two or three are available) one at a time until the governor valve is wide open. Actually, you cannot see the position of the governor. But, if the pressure in the steam chest is within 10–20 psi of the steam supply, the governor is likely 100% open.

  If you do not have horsepower valves on your turbine:
- Option 1—Open steam chest and plug off one or more ports.
- Option 2—Screw in smaller steam nozzles. This is an easy, one-shift job for the machinist.

Condensing Steam Turbines

Turbines that exhaust to vacuum surface condensers offer a huge opportunity to save energy, by reducing the pressure in the surface condenser. For example, let's assume:

$$\text{Motive steam} = 200\,\text{psig}$$

$$\text{Exhaust pressure} = 22\,\text{in. of Hg}\ (200\,\text{mmHg})$$

If we can reduce the turbine exhaust steam pressure to 26 in. of Hg (100 mmHg), steam consumption might drop by 10%–15%. I suggest the reader refers to his or her "Mollier Diagram" and assume an "Isentropic Expansion."

To determine means and methods to achieve the improved vacuum, refer to my book, "Troubleshooting Vacuum Systems," Wiley Publications.

Adding superheat to steam significantly decreases steam consumption for a required amount of turbine horsepower.

Motor-Driven Centrifugal Pumps

Reducing the pump's impeller size is the historic method to save electricity on the motor driver. A reduction of 10% of the impeller diameter saves 27% of the electric power.

I do not use this technique any longer, as it does not eliminate the parasitic energy losses across the downstream control valve.

We should use, "Variable Frequency Drives," and eliminate all control valves on the discharge of pumps. At the Co-op refinery in Mt. Vernon, Indiana, I saw they had three crude pumps operating in series, with not a control valve in sight.

Control valves typically waste 10%–50% of the energy output from the pump's motor driver. Size pumps *all* with full size impellers and VFD (Variable Frequency Drives) motors and forget about the downstream control valve.

For turbine drives installed in the 1930s, there were never control valves on the pump discharge. A 100 years ago, they relied on speed control, just like we ought to do now.

Centrifugal Compressors Fixed Speed

To save energy on variable speed compressors at reduced thru-puts, one simply slows the compressor. On a fixed speed compressor, to reduce the compressor's capacity, the operator has two choices:

1. Open the discharge to suction spill-back.
2. Pinch on the suction throttle valve.

Assuming the compressor is operating in its normal range (i.e., on the flatter portion of its performance curve), choice No. 2, throttling on the suction valve, will reduce the amp load on the motor driver. Choice No. 1, spill-back, increases the amp load on the motor.

BUT, choice No. 2 moves us closer to compressor surge (which will destroy the compressor's rotor). Choice No. 1 moves us away from destructive surge.

Suction throttling will save electric power, but needs to be done with reference to the compressor's surge curves.

Reciprocating Compressors

For a variable speed machine, slow the compressor down to save energy at reduced loads. For motor-driven machines, opening the discharge to suction spill-back valve is a direct waste of electric power. Use of the valve unloaders is also inefficient. You can see this for yourself. When the valve unloader is activated to reduce capacity, the discharge valve cap will still get hot. An indication of wasted work.

To efficiently control the flow through fixed speed recip's, the "Adjustable Head-End Unloading Pocket" is supposed to be used. This will reduce the gas flow with no energy waste. All recips do not have this feature, but can be retrofitted with the unloader. It bolts onto the head-end flange of the recip.

Steam Condensate Recovery

In many plants, steam condensate is dumped into the sewer. A tremendous waste of energy. The reason is vapor lock. The condensate flashes into 5% steam, which expands and chokes off the flow. To avoid that problem, pump the condensate, rather than pressure it into the condensate collection system.

Another alternative is to inject cold water into the hot condensate to suppress steam evolution and vapor lock. Of course, the cold water must have its hardness deposits removed, prior to mixing with the hot steam condensate stream, intended for reuse.

I used this idea to suppress vapor lock on an amine regenerator, lean amine flow, in Aruba many years ago, with excellent results.

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Note: Page numbers followed by *f* indicate figures, *t* indicate tables, and *b* indicate boxes.