INDUSTRIAL REFRIGERATION

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THERMOSYPHON OIL COOLING

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Introduction

Rotary screw compressors in industrial refrigeration can require the injection of large quantities of oil into the compressor, which seals the rotors, lubricates the bearings, and cools the discharge gas. Since first introducing these oil-flooded compressors, designers employ various methods to cool the supply of oil to the compressor. Each of these methods of oil cooling are either direct or indirect.

Direct cooling

This method of oil cooling involves the injection of liquid refrigerant directly into the compressor rotors or the compressor discharge stream before oil separation. With this method, the oil and refrigerant mix, which results in the oil cooling.

Indirect cooling

This method of oil cooling involves the use of a heat exchanger and a cooling medium such as air, water, glycol, or a refrigerant. With these methods, only the oil supplied to the compressor is cooled. The discharge gas from the compressor remains at a higher temperature, usually equal to the oil reservoir temperature.

Thermosyphon oil cooling - an indirect cooling method that employs boiling refrigerant at saturated condensing temperatures as the cooling medium - has gained wide acceptance due to numerous advantages. These include:

- No compressor power increase or capacity decrease as with direct liquid injection systems.
- No risk of water or glycol contaminating the oil or refrigerant charge in the event of an oil cooler leak.
- The cooling medium (refrigerant) is non-fouling, resulting in higher rates of heat transfer and longer oil cooler life.
- Oil heat is rejected to to the ambient through the refrigerant condenser. No additional heat rejection equipment, such as a separate closed-circuit water or glycol cooler, is necessary.

The purpose of this paper is to provide general information and guidance on the design of piping for thermosyphon oil cooling systems.

Note: Design approaches, sizing criteria, or other system design aspects presented herein are not absolute. These are methods that have, from experience, proven successful.

Like other aspects of industrial refrigeration system design, thermosyphon oil cooling systems require careful engineering evaluation in order to ensure proper operation. Qualified professionals can use this paper as an aide in the actual design and evaluation of these systems. The ultimate responsibility for the proper design of a thermosyphon oil cooling system rests with the party responsible for the refrigeration system design.

Principle of operation

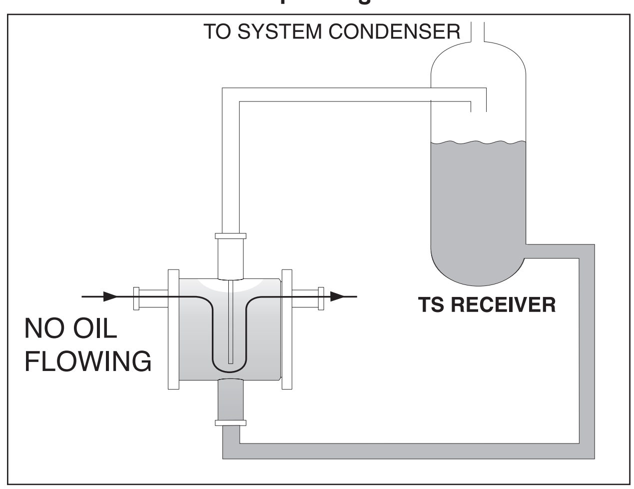
The thermosyphon principle refers to the circulation of a fluid where the motive force for fluid flow is provided not by the addition of mechanical work, like a pump, but rather by gravitational forces and a difference in fluid densities between two vertical legs of the flow loop. In a thermosyphon oil cooling system, this principle of fluid density differential is used to circulate coolant (refrigerant) through the oil coolers. As is explained later in this section, the transfer of heat from the oil being cooled to the refrigerant maintains the density differential necessary for flow.

A thermosyphon oil cooling system requires the following basic equipment:

- A source of liquid refrigerant at system condensing temperature and pressure. Place this liquid source in close proximity to the compressor unit or oil cooler to minimize pressure losses in the piping. The liquid level in the refrigerant source vessel must be maintained at an elevation above that of the oil coolers. This elevation difference is commonly referred to as “available liquid head”. The minimum head necessary may vary based on actual system design. See *Appendix A* for a detailed calculation.
- An oil cooler heat exchanger. It is possible to use other types of heat exchangers such as welded plate, plate and shell, or shell and tube.

Figure 1 depicts the basic thermosyphon system when there is no transfer of heat from the oil to the refrigerant, as is the case before a compressor starts. Liquid refrigerant, at condensing temperature, fills the tubes or shell of the heat exchanger and the rest of the thermosyphon loop to the normal liquid level in the supply vessel. Because the density of the liquid refrigerant is the same in all parts of the loop, there is no flow of refrigerant through the oil cooler.

Figure 1: Thermosyphon oil cooler - system not operating



When the compressor package is operating, hot oil (above the refrigerant temperature) flows through the refrigerant side of the oil cooler. Heat flows through the walls from the higher temperature oil to the lower temperature refrigerant causing the oil to become cooler. At the same time, some of the refrigerant in the cooler boils as it absorbs its latent heat of vaporization from the oil.

The configuration of the oil cooler heat exchanger is such that the refrigerant vapor created in the oil cooler can easily escape and flow back to the supply vessel. The liquid and vapor mixture that returns from the oil cooler separates in the supply vessel. The vapor vents to the inlet of the refrigerant condenser where it reliquifies.

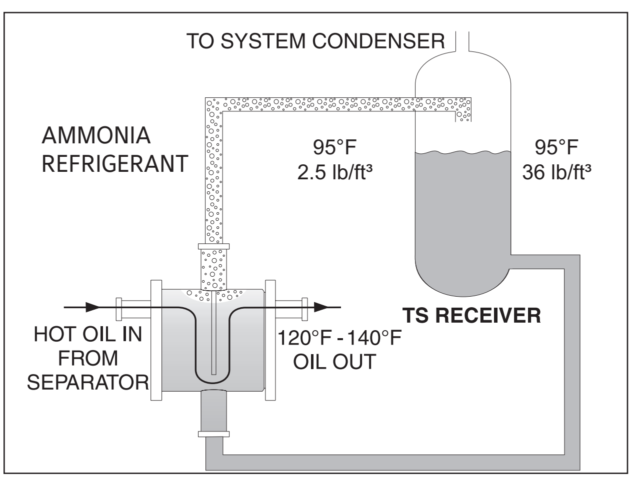
Determine the rate of refrigerant vaporization in the oil cooler by dividing the heat rejected from the oil by the latent heat of vaporization for the specific refrigerant and operating temperature. In order to ensure that all heat transfer surfaces are wetted by the refrigerant, thermosyphon oil cooling systems are designed so that more refrigerant flows through the oil cooler than is actually vaporized. Assume the refrigerant design flow rate to be four times the rate of vaporization. This is commonly referred to as a 4:1 overfeed rate.

Note: When explaining the operation of a thermosyphon system, it is necessary to distinguish between the two vertical legs of the piping loop. In this paper, the term vertical liquid supply line refers to the vertical run of piping by which refrigerant is supplied to the oil cooler, and the term wet return line refers to the vertical run of piping by which refrigerant returns to the supply vessel.

Figure 2 depicts the basic thermosyphon system in operation with a 4:1 overfeed rate. The refrigerant in the vertical liquid supply line is all liquid, and its density is the same as it was before the compressor package was operating. The refrigerant in the wet return line is a mix of three parts liquid and one part vapor (by mass). Because the density of refrigerant vapor is much less than the density of the liquid, the two-phase mixture in the wet return line has a density that is considerably less than that of the refrigerant liquid in the vertical liquid supply line.

The difference in fluid densities, when multiplying by the height of the vertical legs in the thermosyphon piping loop, yields a pressure differential.

Figure 2: Thermosyphon oil cooler - system operating



This pressure differential is the force that drives the flow of refrigerant in the thermosyphon loop. Like any fluid flowing in a piping system, the refrigerant in the thermosyphon loop experiences frictional forces which oppose the flow. The greater the refrigerant flow rate, the greater the magnitude of the pressure losses due to these frictional forces. The refrigerant flow rate stabilizes at a point where the pressure drop due to friction losses in the thermosyphon piping loop is exactly equal to the pressure differential supplied by the difference in fluid densities and the height of the vertical legs in the piping loop.

When designing a thermosyphon oil cooling system, one usually begins with the design oil cooler heat rejection and an assumed refrigerant overfeed rate, say 4:1. The refrigerant piping is then designed so that total friction losses in the piping loop at this flow rate are approximately half the pressure differential provided by the differences in fluid density and the available liquid head. This approach adds a margin of safety in the design to allow for errors in the estimate of the available pressure and the pressure loss in the piping and valving. A detailed calculation example is given in Appendix A.

In an actual system, the available liquid head is fixed and the oil cooler heat rejection varies depending on the compressor's operating conditions. At lower rates of oil cooler heat rejection, the refrigerant flow is less. It is a characteristic of thermosyphon oil cooling systems that higher oil temperatures result in more vaporization of refrigerant, greater density differential, and higher refrigerant flow rates. In other words, the more that oil cooling is necessary, the harder the thermosyphon system works to provide it.

Thermosyphon systems

There are numerous ways to design the piping in a thermosyphon oil cooling system. Below are some examples of common practice. However, this by no means covers all the possible geometries that work. Alternate piping schemes may be better suited for a particular application. It is the responsibility of the system designer to determine the best scheme to pursue for their system.

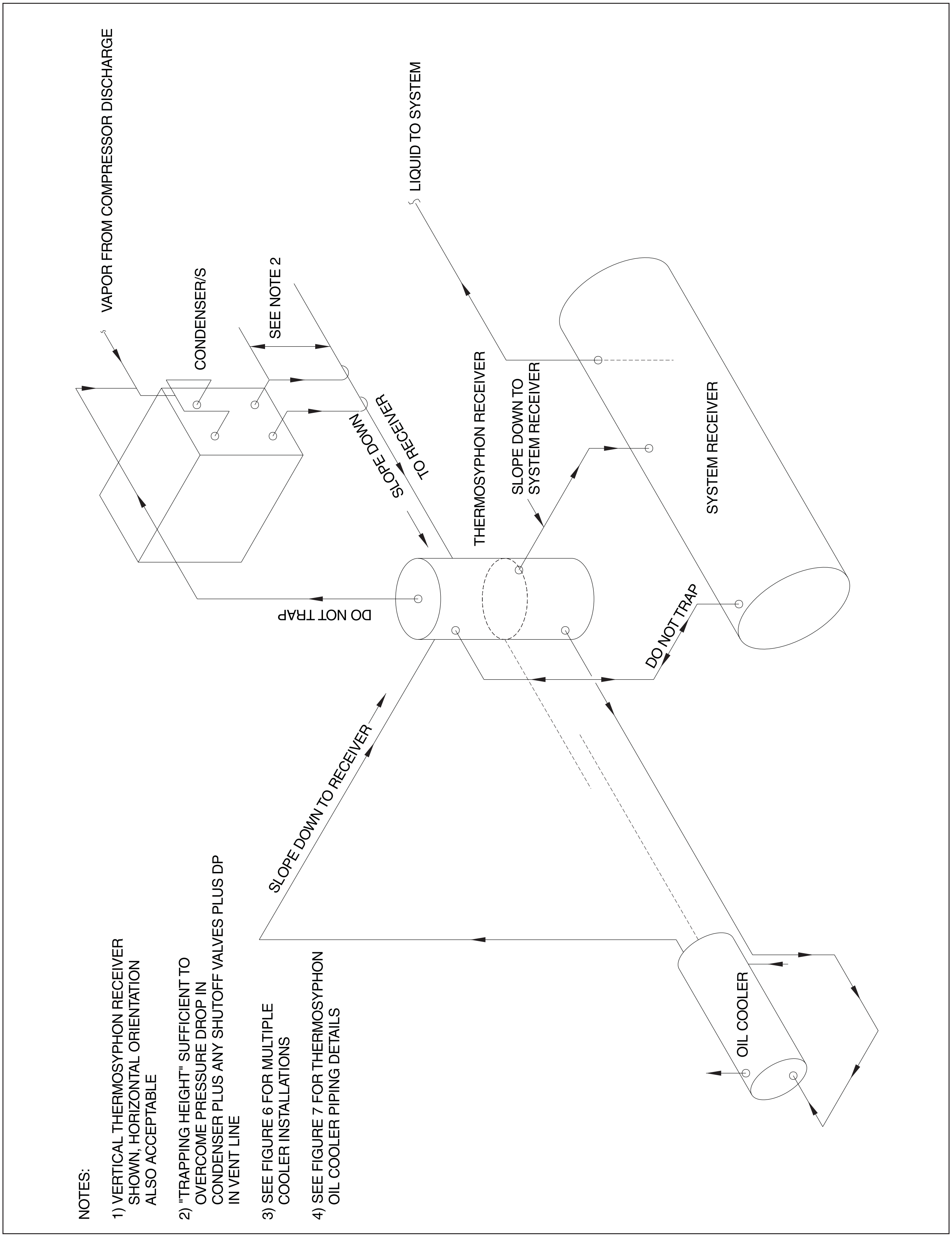
Flow-through thermosyphon receiver

In the arrangement shown in Figure 3, all of the high-pressure liquid refrigerant leaving the condenser flows through the thermosyphon receiver on its way to the system receiver.

Liquid refrigerant, draining from the condenser, flows by gravity to the thermosyphon receiver. Locate the connections on the thermosyphon receiver, which can be either a horizontal or vertical pressure vessel, such that liquid refrigerant fills this vessel up to a certain point and then overflows into the system receiver. Elevate the thermosyphon receiver relative to the system receiver, and connect these two vessels using an equalizing line to ensure that liquid refrigerant flows freely to the system receiver.

The liquid refrigerant supply to the oil cooler exits the thermosyphon receiver well below the level of the overflow connection to the system receiver. This ensures an uninterrupted 5 min supply of refrigerant liquid to the oil cooler regardless of system demands.

Figure 3: Flow-through thermosyphon receiver



When the compressor package is operating, hot oil (above the refrigerant temperature) flows through the oil cooler. As heat transfers from the higher temperature oil to the lower temperature refrigerant, the oil cools while some of the refrigerant in the oil cooler boils. During compressor operation, the vertical liquid supply line supplying refrigerant to the oil cooler contains liquid refrigerant, while the wet return line exiting the oil cooler contains a mixture of refrigerant liquid and vapor. Gravity causes the denser refrigerant liquid to flow downward to the oil cooler, displacing the less dense liquid and vapor mixture and pushing it up the wet return line and back to the thermosyphon receiver. The liquid and vapor mixture must enter into the vapor space of the thermosyphon receiver but at a level only slightly higher than the maximum liquid level.

In the thermosyphon receiver, the vapor that returns from the oil cooler separates from the liquid and vents to the inlet of the refrigerant condenser where it once again condenses.

The refrigerant in both the thermosyphon and system receivers is saturated, and its temperature is equal to the condensing temperature. Likewise, the refrigerant in the oil cooler is saturated at the same temperature. This refrigerant boils as it absorbs heat from the oil, but its temperature does not change. This means the oil cools to within an approach temperature of the refrigerant condensing temperature. Generally, this approach temperature is in the range of 15°F to 35°F (8.3°C to 19.4°C). For example, with a condensing temperature of 95°F (35°C), the temperature of the oil exiting the cooler is approximately 110°F to 130°F (43.4°C to 54.4°C).

Figure 4: Elevated system receiver

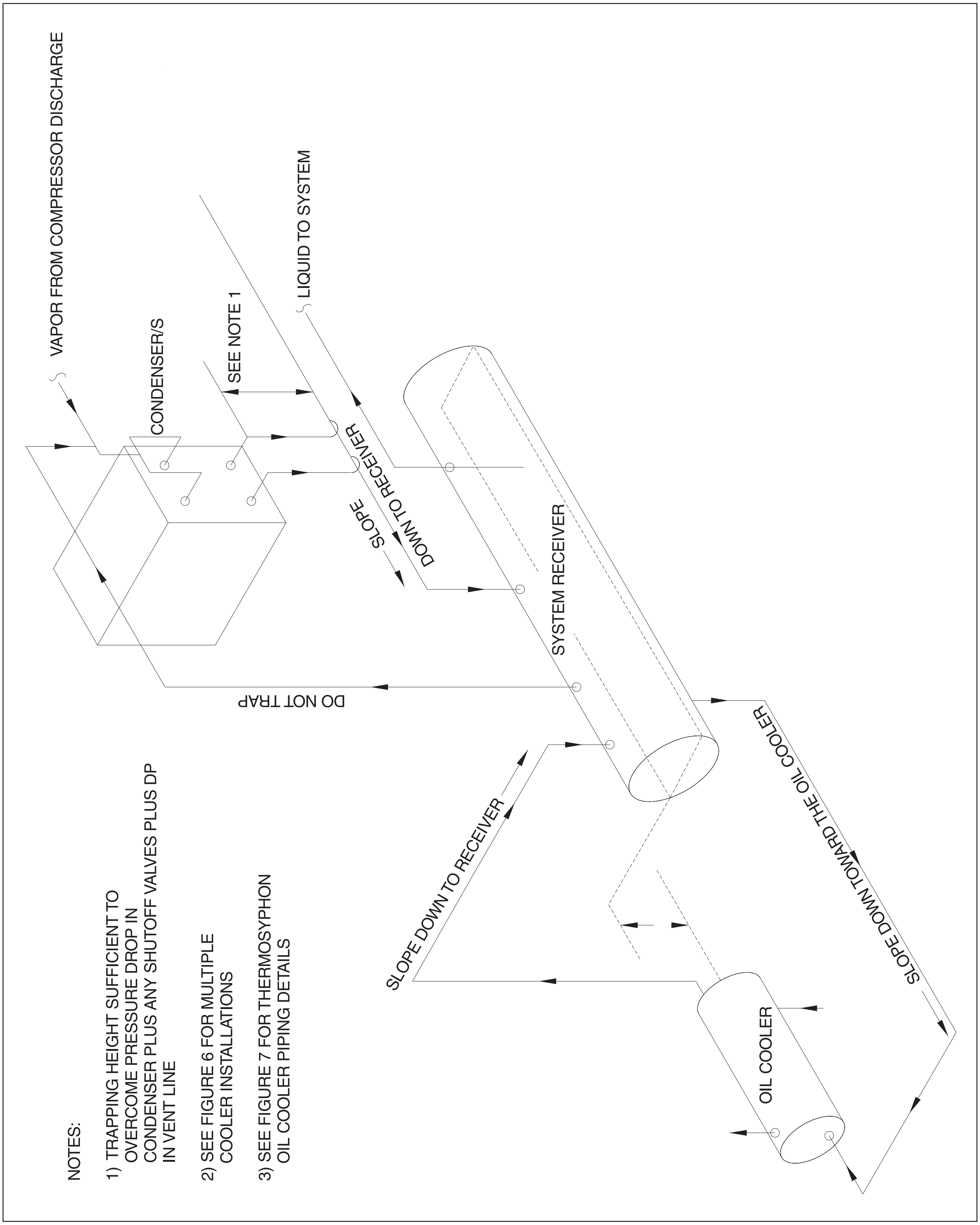
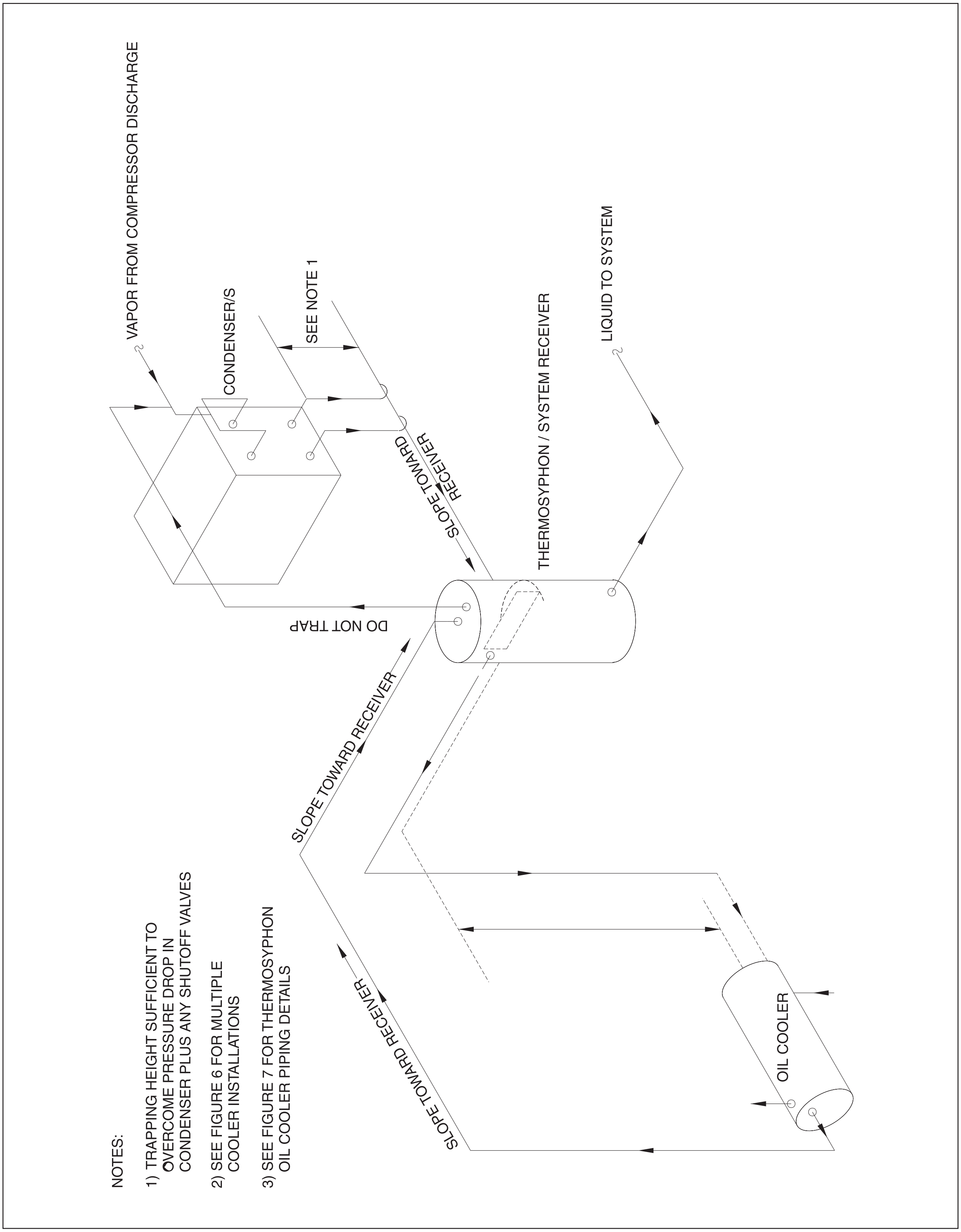


Figure 5: Combination thermosyphon/system receiver



System receiver as thermosyphon receiver

When it is possible to install the system receiver at sufficient elevation (relative to the oil cooler), the system high-pressure receiver can also serve as the refrigerant supply vessel for thermosyphon oil cooling, as shown in Figure 4.

Liquid refrigerant, draining from the condenser, flows by gravity to the system receiver. Locate the outlet connections on this vessel such that the refrigerant liquid supply to the oil coolers is physically lower than the liquid supply to the system. This ensures an uninterrupted 5 min supply of refrigerant liquid to the oil cooler regardless of system demands. To accomplish this, take refrigerant for oil cooling off the bottom of the receiver and use a dip tube for the main system liquid line. Project the nozzle on the receiver for the thermosyphon liquid supply up into the receiver to prevent dirt being drawn into the supply line and eventually into the oil cooler.

If you cannot elevate the system receiver sufficiently, an alternative approach is shown in Figure 5. In this arrangement, the system receiver is a vertical vessel. In the top of the vessel is a weir dam over which all the liquid from the condenser is forced to flow. The weir dam forms a reservoir of liquid refrigerant to feed the thermosyphon oil cooler. The excess liquid that spills over the weir into the bottom of the receiver feeds the rest of the system.

Take off the liquid feed to the thermosyphon oil cooler from the receiver at the bottom of the weir dam. Size the volume of the weir dam to provide at least 5 min of liquid retention. The thermosyphon return line must enter the receiver in the vapor space just above the liquid level in the weir dam.

Multiple thermosyphon oil coolers

A large percentage of the refrigeration systems using thermosyphon oil cooling involve multiple compressors sharing a common thermosyphon supply vessel. A disproportionate share of thermosyphon problems seem to occur with these multiple-compressor systems. However, the operating principles for a thermosyphon system with multiple oil coolers are the same as those for a system with a single oil cooler

Properly accounting for the total heat load on the oil cooling system is the first key to a successful design. Remember that at some operating conditions, compressor oil cooler heat rejection is greatest at full load, while at other operating conditions, heat rejection is greatest at part load. Consider every compressor on the oil cooling system and every conceivable state of compressor loading to find the worst case condition for total oil cooler heat rejection. If additional compressors are added to the system in the future, you must size common parts of the oil cooling system to handle the heat rejection from these future compressors.

It is possible to design a multiple-compressor system with separate supply and return lines to each thermosyphon oil cooler. In which case, this piping would be designed as for a single-compressor system. Generally however, thermosyphon oil cooling systems serving multiple compressors use common refrigerant liquid supply headers and liquid/vapor return headers. See Figure 6 for the piping setup of a multiple compressor system using common supply and return headers.

⚠ Caution

Consideration must be given to the order in which compressors cycle on and off relative to the order in which the oil coolers are fed refrigerant. When a compressor cycles off, the thermosyphon oil cooler for that compressor fills with liquid refrigerant. This can temporarily starve other oil coolers further downstream resulting in nuisance compressor shut downs on high oil temperature.

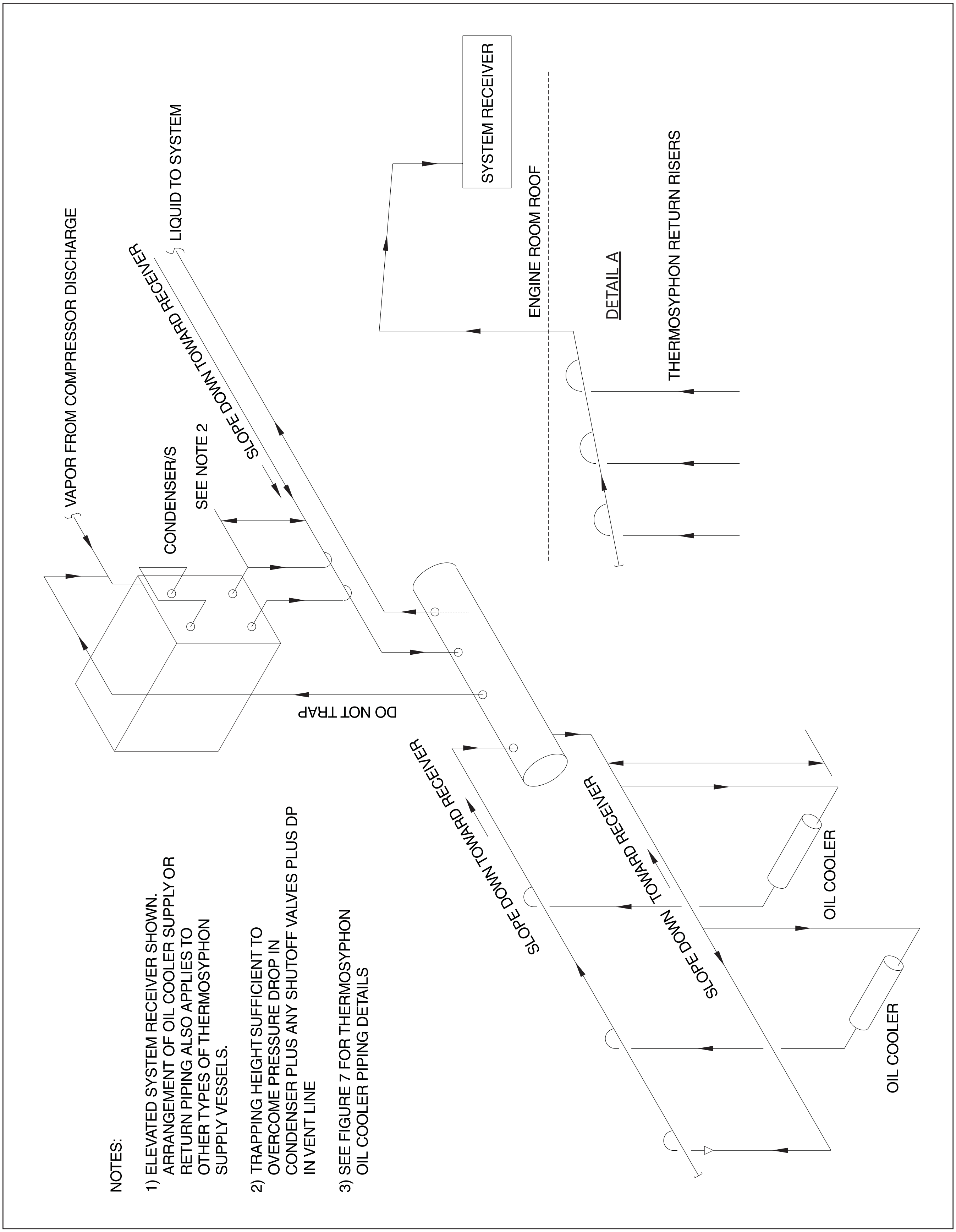
If in the designer's judgement, compressor order cycle could be an issue for a particular system, then it is possible to run separate liquid feed lines as necssary to avoid nuisance compressor shut downs. Alternatively, locate the compressors that run the most at the beginning of the supply header.

A thermosyphon system serving multiple compressors is very likely to have a liquid supply header that runs horizontally through the engine room. Vapor bubbles form in this header as the saturated liquid refrigerant it contains absorbs heat from the warm engine room and vaporizes. Make sure to vent these bubbles back to the supply vessel before they can combine to form a large vapor bubble that would interfere with the free flow of liquid refrigerant to the oil coolers. The easiest way to accomplish this is to pitch the liquid header so it rises 1/4 in. per ft in the direction of flow, moving away from the supply vessel. If the "tail end" of the liquid header then connects to the return header, any vaporized refrigerant can easily return to the supply vessel. Ensure the vertical liquid supply line carrying liquid refrigerant to the individual oil coolers exits the bottom of the supply header. The end of the supply header cannot exceed the elevation of the supply vessel.

It is also necessary to pitch the horizontal return header carrying the two-phase refrigerant mixture back to the supply vessel. Pitch the header so it falls 1/4 in. per ft or more in the direction of flow, moving towards the supply vessel. Ensure the wet return line carrying refrigerant liquid and vapor from the individual oil coolers enters the top of the return header.

A variant of this system is the situation shown in Detail A of Figure 6. In some applications, it is impossible to get all the return header above the supply vessel. In this case, the return header can be split in two sections with a vertical wet return line in the middle. However, in this situation, you must slope the return header in the engine room upward 1/4 in. per ft or more in the direction of flow toward the vertical wet return line. This allows vapor bubbles in the wet return line in the engine room to return to the supply vessel. Slope the part of the return header that is above the supply vessel down toward the supply vessel as before.

Figure 6: Multiple thermosyphon oil coolers



Typical thermosyphon oil cooler piping details

The previous diagrams have purposely left out valves required for proper operation in order to show the piping more clearly. Figure 7 shows the minimum number of valves needed to operate the system.

Service isolation valves are sometimes used in the refrigerant supply and return lines to the oil cooler, and are appropriate for installations where multiple thermosyphon coolers connect to a common supply vessel. Generously size these valves and ensure they are a low pressure drop design. Do not use globe valves. Additionally, install a sight glass in the vertical leg of the refrigerant supply and return lines of every oil cooler to facilitate trouble shooting the system if necessary.

Provide oil drain valves at the low points to the supply vessel and the refrigerant side of the thermosyphon oil coolers. Over time, oil accumulates in these locations and interferes with the efficient operation of the thermosyphon system. Periodic removal of oil is required.

Pressure vessel codes generally require the installation of a pressure safety valve on the oil side of the thermosyphon oil cooler. This pressure safety valve may discharge to the compressor package's oil separator vessel.

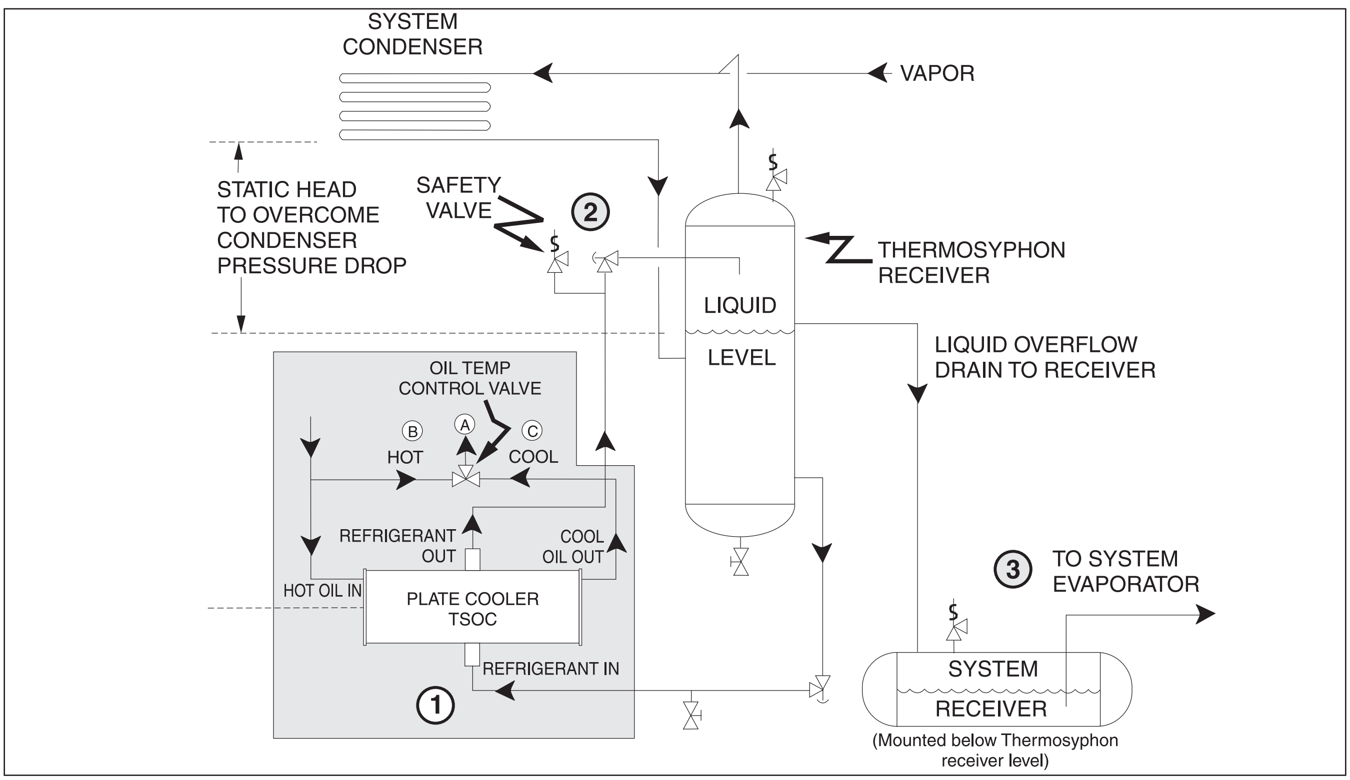
Oil temperature control

The temperature of the oil leaving the thermosyphon oil cooler is generally about 15°F to 35°F (-9.4°C to 1.7°C) above the condensing temperature. For installations where the condensing temperature or the engine room temperature is not expected to drop below about 65°F (18.3°C), oil temperature control is not generally required. Without control, the oil temperature simply floats with the condensing temperature. If considering a thermosyphon oil cooling system without temperature control for non-FRICK compressors, contact the compressor manufacturer to advise the minimum allowable temperature for oil supplied to the compressor. For FRICK compressors, the minimum oil temperature at start-up is 50°F (10°C).

Older discussions of thermosyphon oil cooling sometimes mention controlling the supply of liquid refrigerant to the oil cooler as a method of oil temperature control. This method is rarely seen in the field today and is not recommended. By far the most common method is direct control of the oil temperature with a cooler bypass oil mixing valve.

Control of oil temperature with a bypass mixing valve on the oil side affords inexpensive, automatic control. At higher condensing temperatures (95°F [35°C]), the bypass remains closed and the oil temperature floats with the condensing temperature. At low condensing temperatures, some hot oil bypasses around the cooler and mixes with cooled oil to maintain a constant oil supply temperature of approximately 130°F (54.4°C) to the compressor.

Figure 7: Thermosyphon oil cooler piping



System dynamics

All previous discussions of thermosyphon oil cooler operation have assumed steady state operation. Real world refrigeration systems seldom operate at steady state conditions, but rather experience many dynamics including startup, shutdown, load changes, hot gas defrosting of air units, ice maker harvesting, condenser capacity changes, and other conditions.

The system dynamics most likely to affect the operation of thermosyphon oil cooling systems are those that cause intermittent interruptions in system refrigerant flows or changes in system condensing pressure (recall that the thermosyphon oil cooling system operates at the system condensing pressure). An adequate supply of refrigerant liquid in the thermosyphon receiver permits a system to sustain operation during brief interruptions in system refrigerant flow.

Rapid decreases in system condensing pressure can lead to blowout of the refrigerant in the oil cooler circuit. Oil cooler refrigerant circuit blowout normally occurs when there is a large need for system hot gas, for example, defrosting of air units or harvesting ice makers. It can also be caused by a large, step increase in condenser capacity, for example, a condenser fan or pump cycling on. In either case, it is the sudden decrease in system condensing pressure that causes this to occur.

In normal, steady state operation, the refrigerant in the thermosyphon oil cooling circuit is saturated and its temperature is equal to the saturation temperature at the system condensing pressure. When the condensing pressure suddenly drops, the refrigerant in the thermosyphon oil cooling circuit is at a temperature higher than the equivalent boiling pressure. This causes a portion of the liquid refrigerant in the thermosyphon loop to flash to a vapor. The refrigerant expands dramatically as it evaporates and the rapidly expanding vapor can push the remaining liquid refrigerant out of the oil cooler, the vertical liquid supply line, and the wet return line.

Following this evacuation of liquid refrigerant from the heat exchanger, the liquid returns due to gravity and normal thermosyphon operation resumes. Until the liquid refrigerant returns to the oil cooler however, no oil cooling occurs. It is quite possible for oil temperatures to become high enough to cause a safety shutdown of a compressor package before the oil cooling system can recover from a blowout.

Note: During an oil cooler circuit blowout, liquid refrigerant can be propelled at high velocity by the expanding vapor. This can impose severe hydraulic shocks on the piping system.

Maintaining stable condensing pressures minimizes the occurrence of oil cooler circuit blowout. This may require defrosting fewer air units at a time, employing more sophisticated condenser capacity controls, or other measures. Finally, increasing the elevation of the thermosyphon receiver and oversizing the vertical liquid supply line supplying liquid to the oil coolers tends to suppress flash boiling in the oil cooler following a condensing pressure decrease and lead to a more rapid recovery.

System sizing

The sizing tables and graphs are all based on 95°F (35°C) condensing and a 4:1 recirculation ratio for R-717. For higher condensing temperatures, increase all sizing to maintain proper refrigerant flow. Proper operation of thermosyphon systems depends upon refrigerant flow entirely by gravity head. Unaccounted for pressure drop in refrigerant lines and valves can prevent refrigerant flow. If any line sizing is questionable, increase to the next larger size and increase the static head of liquid supply to the oil cooler. A more precise pressure loss calculation is available in *Appendix A* and can help resolve questionable line sizes.

The first step in sizing a system is to determine the total oil cooling heat load from the compressor oil cooling heat of rejection (OCHR) tables. For this example, assume an oil cooler heat load of 653,000 Btu/h on an ammonia system condensing at 95°F (35°C).

Step 1: Sizing the Thermosyphon Receiver (See Table 1).

Under the appropriate refrigerant, proceed down the column to the required heat load and select the vessel with a capacity rating that equals or exceeds the load.

12 in. x 6 ft 500,000 Btu/h
653,000 Btu/h - - - - - Example Heat Load
16 in. x 6 ft 875,000 Btu/h - - - - - Select

Step 2: Sizing the Liquid Feed Line to the Oil Cooler and the Return Line to the Thermosyphon Receiver. See Figures 8, 9, and 10 for R-717.

A) Liquid Feed Line to Oil Cooler

See Figure 8 for R-717 heat loads to 1000 kBtu/h. Follow along the horizontal axis to the required heat load and read up to the nearest line curve, not to exceed:

0.10 psid/100 ft, R-717 (Figures 8, 9, and 10)

Example: 653,000 Btu/h

2 in. - above 0.10 psid/100 ft - - - - - Too Small
2 1/2 in. - 0.058 psid/100 ft - - - - - Select
3 in. - 0.019 psid/100 ft - - - Larger Than Required

B) Return Line from oil cooler

Return to Figure 8. Follow the same procedure as Step 2A but do not exceed:

0.04 psid/100 ft, R-717 (Figures 8, 9, and 10)
Example: 653,000 Btu/h

2 1/2 in. - 0.058 psid/100 ft - - - - - - Too Small
3 in. - 0.019 psid/100 ft - - - - - - - - - Select

Step 3: Sizing the Return Vent Line from the Thermosyphon Receiver to the Condenser. See Figure 11.

Calculate the mass flow rate (lb/min) as follows:

A. Convert the oil cooler heat load in Btu/h to Btu/min

$$\frac{\text{Oil Cooling Load}}{60} = \text{Btu/min}$$

B. Divide the Btu/min load by the enthalpy difference of the refrigerant phase change at the system condensing temperature.

$$\frac{\text{Btu/min}}{\text{H}} = \text{lb of Refrigerant / min}$$

H = 483.2 Btu/lb (R-717 at 95°F)

H = 488.5 Btu/lb (R-717 at 90°F)

See Figure 11. Follow along the vertical axis to the calculated mass flow rate, read across to the slanted reference line and down to the return vent line size.

Example: 653,000 Btu/h

$$\text{A)} \quad \frac{653{,}000 \text{ Btu/hr}}{60} = 10{,}883.33 \text{ Btu}$$

$$\text{B)} \quad \frac{10{,}883 \text{ Btu/min}}{483.2 \text{ Btu/lb}} = 22.52 \text{ lb/min}$$

Select 2 1/2 in. or 3 in. to minimize pressure drop and assure system flow.

Table 1: Thermosyphon receiver sizing*

Shell Size	R-717 OCHR (Btu/h) maximum
8 in. x 6 ft (vertical)	210,000
12 in. x 6 ft (vertical)	500,000
16 in. x 6 ft (vertical)	875,000
20 in. x 6 ft (horizontal or vertical)	1,400,000
24 in. x 5 ft (horizontal or vertical)	2,000,000
30 in. x 5 ft (horizontal or vertical)	2,600,000

*Based on 5 min liquid supply, 95°F (35°C) condensing.

Figure 8: R-717 TSOC pipe sizing, 0 kBtu/h to 1000 kBtu/h - OCHR

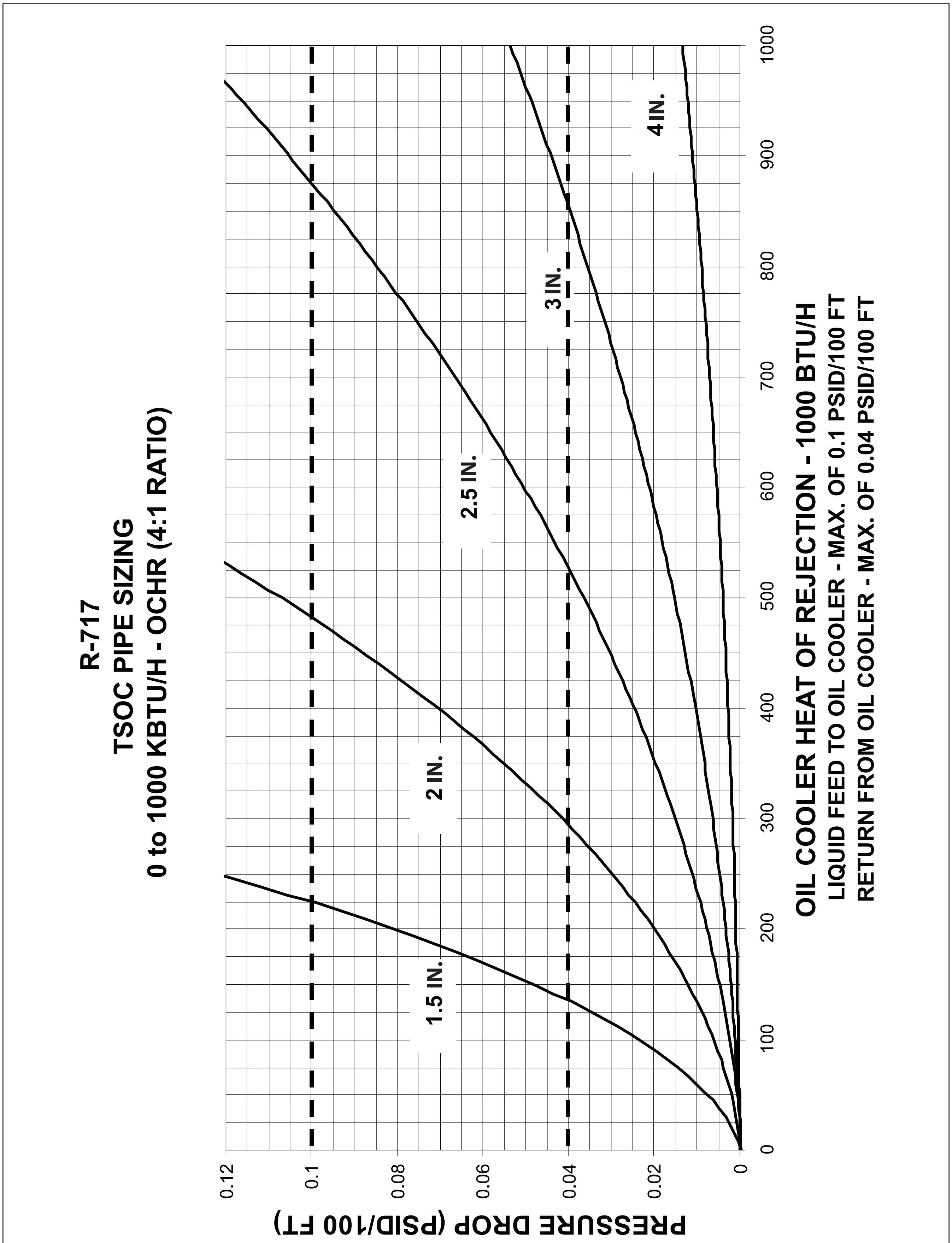


Figure 9: R-717 TSOC pipe sizing, 1000 kBtu/h to 3000 kBtu/h - OCHR

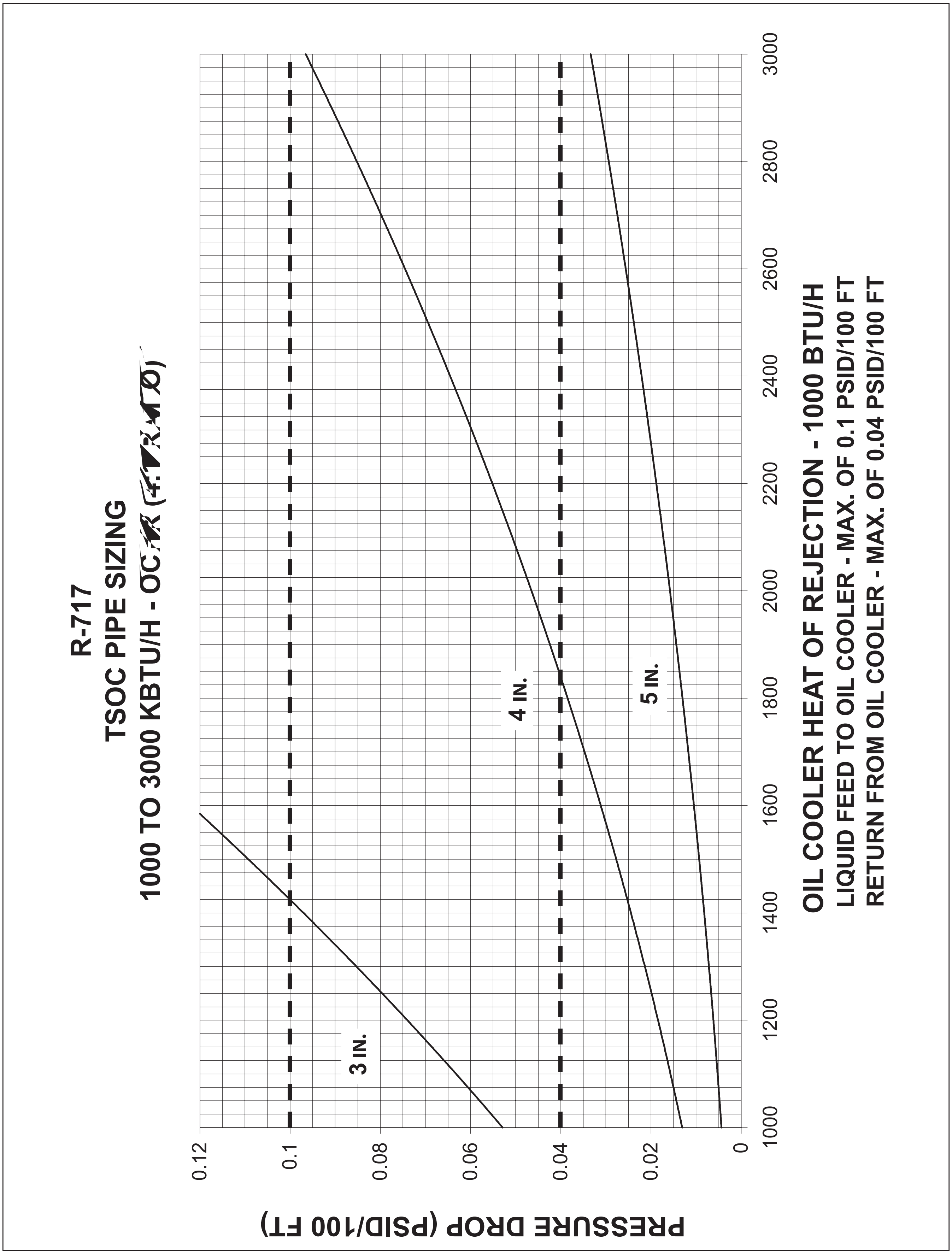


Figure 10: R-717 TSOC pipe sizing, 3000 kBtu/h to 5000 kBtu/h - OCHR

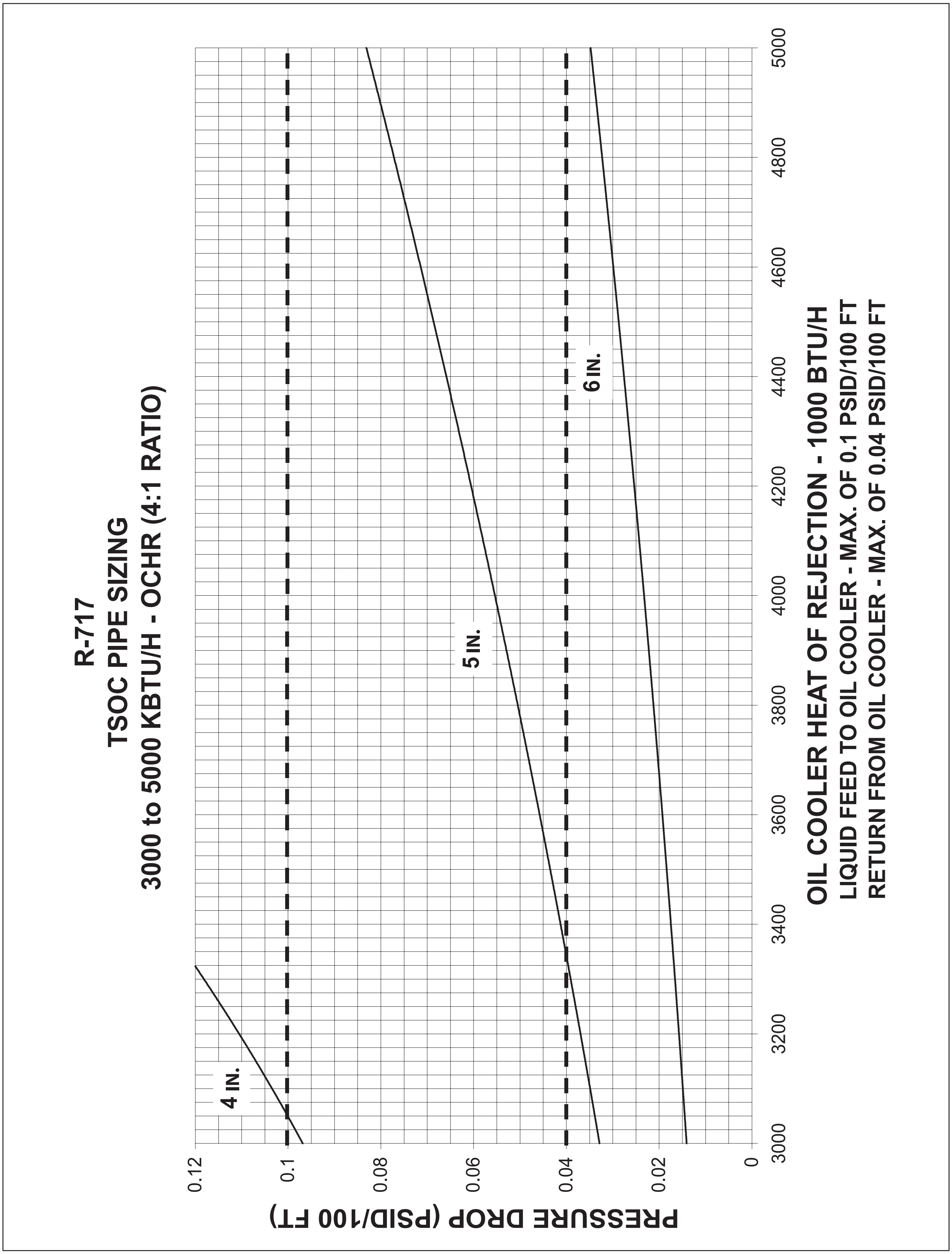
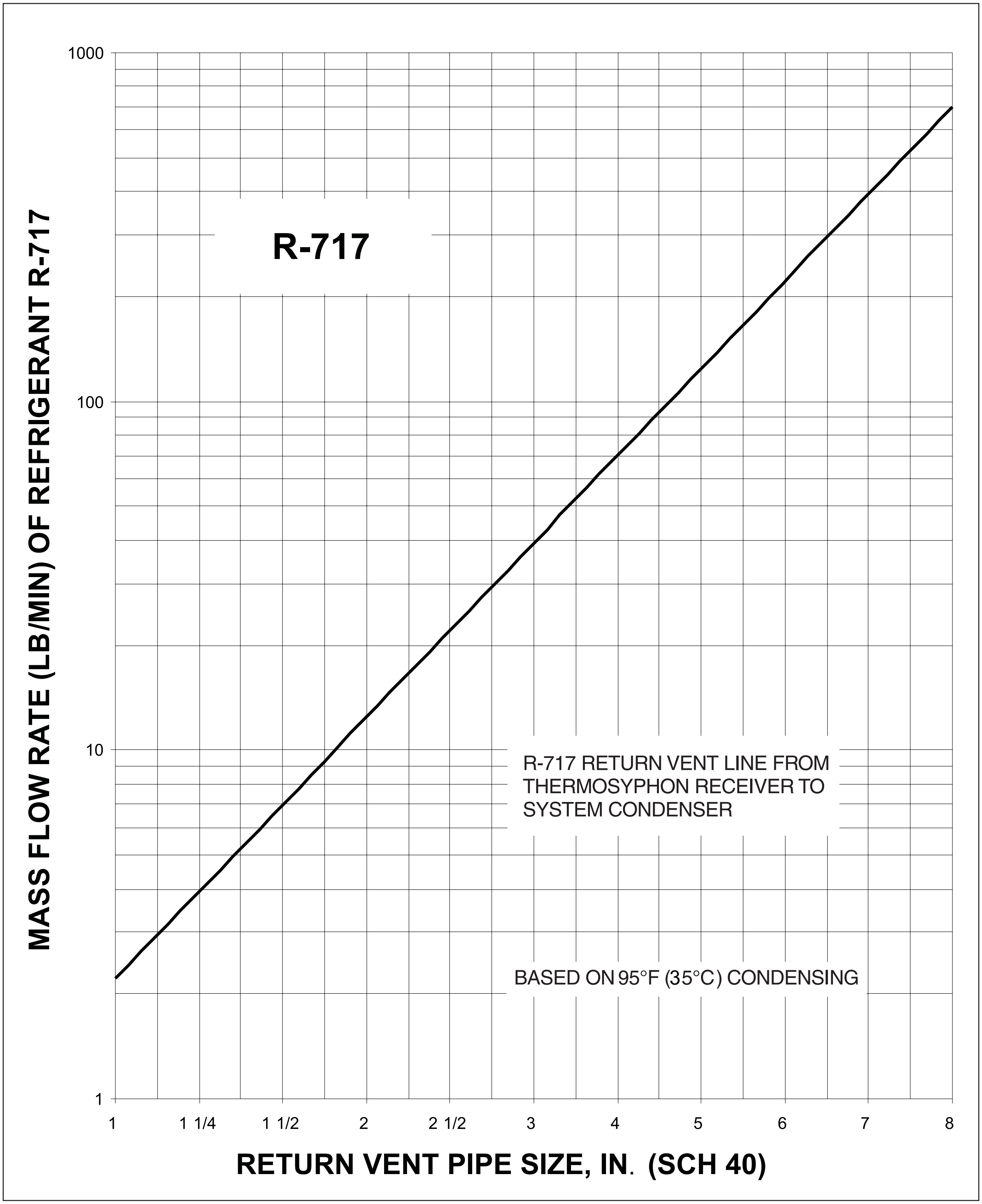


Figure 11: R-717 return vent line sizing



Appendix A - Required minimum liquid head height calculation

The sizing example given previously provides reasonably sized piping. However, if the graphs yield a size that seems questionable, you can make a more rigorous calculation as outlined below. In fact, after selecting sizes from the graphs, it is still beneficial to calculate pressure losses in the piping and the minimum head height.

The basic calculation process is to first determine the initial pipe sizes from the sizing graphs.

1. For a given oil heat rejection rate, determine the refrigerant flow rate through the piping.
2. After the flow rate is known, determine the pressure loss in the liquid supply line using the Moody diagram.
3. Determine the pressure loss in the return line, which is a mixture of gas and liquid, from the charts in this appendix.
4. Then, add the pressure losses in the thermosyphon supply and return lines, plus an estimate of the cooler pressure loss to get the total loss in the piping. Compare this number to the available liquid head to ensure that the selected piping is adequate.

For this example, assume that the thermosyphon oil cooling load is 425 MBH for an ammonia system at 95°F (35°C) condensing. From Figure 8, a 2 in. liquid supply line to the oil cooler results in a pressure loss of 0.08 psi/100 ft. Because this is below the limit of 0.10 psi/100 ft, the 2 in. pipe is acceptable. However, for the return line, the 0.08 psi/100 ft exceeds the limit of 0.04 psi/100 ft. A 2 1/2 in. pipe must be used for the return line from the thermosyphon oil cooler.

The next step is to determine the actual flow rate at an assumed 4:1 recirculation ratio from equation 1.

Note: Figures 8 to 10 are based on a 4:1 recirculation ratio.

By definition, the refrigerant flow rate is four times the evaporation rate which you can determine from the oil cooler heat rejection and the latent heat of vaporization. Determine the latent heat of vaporization or h_{fg} from the FRICK *Engineering Data and Tables pamphlet (E20-4G/J66)* at 95°F (35°C) condensing temperature.

$$\text{Refrigerant Flow Rate} = \frac{\text{4 x Oil Cooling Load}}{\text{Enthapy of Vaporization}} = \frac{\text{4 x 425,000 Btu/h}}{\text{483.2 Btu/lb x 60 min/h}} \qquad (1)$$

$$\text{Refrigerant Flow Rate = 58.6 lb/min}$$

With the flow rate known, calculate the pressure loss in the liquid supply line using the Moody diagram and the Darcy-Weisbach formula (see an undergraduate engineering text on fluid mechanics). In order to use the Moody diagram, perform the following steps:

1. Determine the velocity of the refrigerant in the liquid line. The velocity in the 2 in. liquid supply pipe is defined by equation 2.
2. Obtain the density of the liquid refrigerant and cross sectional area of a 2 in. schedule 40 pipe from the FRICK *Engineering Data and Tables pamphlet (E20-4G/J66)* at 95°F (35°C) condensing temperature.

$$\text{Velocity} = \frac{\text{Refrigerant Flow Rate}}{\text{Liquid Refrigerant Density x Inside Cross Sectional Area of the Pipe}} \qquad (2)$$

$$\text{Velocity} = \frac{\text{58.6 lb/min x 144 in.}^2\text{/ft}^2}{\text{36.67 lb/ft}^3\text{ x 3.36 in.}^2} = \text{68.5 ft/min = 4110 ft/h}$$

3. Calculate the Reynolds number using equation 3. Obtain the viscosity of liquid ammonia at 95°F (35°C) from CoolWare™ software. The Reynolds number does not have units.

$$\text{Reynolds \#} = \frac{\text{Liquid Density x Velocity x Inside Diameter of Pipe}}{\text{Viscosity of Liquid Refrigerant}} \qquad (3)$$

$$\text{Reynolds \#} = \frac{\text{36.67 lb/ft}^3\text{ x 4110 ft/h x 0.1723 ft}}{\text{0.2985 lb/ft - h}} = \text{87,000}$$

4. Determine the relative roughness of the pipe given by equation 4. You can find the absolute roughness for various materials along with a Moody diagram in *Mark's Standard Handbook for Mechanical Engineer*s.

$$\text{Relative Roughness} = \frac{\text{Absolute Roughness}}{\text{Inside Diameter of the Pipe}} \qquad (4)$$

$$\text{Relative Roughness} = \frac{150 \times 10^{-6}\text{ ft}}{\text{0.1723 ft}} = 0.00087$$

Given the Reynolds number and relative roughness, the Moody diagram indicates the friction factor is 0.023. Calculate the pressure drop in the pipe from the friction factor using equation 5. (assume 100 ft of pipe). The result is that 100 ft of 2 in. schedule 40 pipe has a pressure loss of 0.067 psi with 58.6 lb/min of liquid ammonia flowing through it at 95°F (35°C).

There are usually elbows and at least one valve in the liquid supply line. The pressure drop through the fittings and valves can be handled by using equivalent lengths; refer to *Section V* in the *FRICK Engineering Data and Tables* pamphlet (*E20-4G/J66*). A table of equivalent lengths for valves and fittings is given.

$$\text{Pressure Drop} = \text{Friction Factor x Liquid Density x } \frac{\text{Length of Pipe}}{\text{Inside Diamter of Pipe}} \text{ x } \frac{\text{Velocity}^2}{2} \qquad (5)$$

$$\text{Pressure Drop} = \text{0.023 x 36.67 lb/ft}^3 \text{ x } \frac{\text{100 ft}}{\text{0.1723 ft}} \text{ x } \frac{\text{(4110 ft/h)}^2}{2} \text{ x}$$
$$\text{x}\left(\frac{1}{\text{32.2 ft-lb/lbf - s}^2}\right)\text{x}\left(\frac{\text{1 h}}{\text{3600 s}}\right)^2 \text{ x } \frac{\text{1 ft}^2}{\text{144 in.}^2}$$

$$\text{Pressure Drop = 0.069 psig / 100 ft of 2 in. pipe}$$

For our example, assume that there are two 2 in. long radius elbows and one 2 in. angle valve. Reading the table for ferrous fittings results in an equivalent length of 2.3 ft for a welded elbow and 25 ft for a flanged angle valve. Equation 6 provides the total equivalent length of straight pipe:

$$\text{Total equivalent length of pipe} = \text{6 ft of pipe} + (\text{2 ells x 2.3 ft/ell}) + (\text{1 valve x 25 ft/valve}) \qquad (6)$$

$$\text{Total equivalent length of pipe = 35.6 ft of pipe}$$

Equation 7 provides the total pressure loss in the thermosyphon liquid supply line:

$$\text{Total Liquid Line Pressure Drop} = \frac{0.069\text{ psig}}{100\text{ ft}} \text{ x } 35.6\text{ ft} \qquad (7)$$

$$\text{Total Liquid Line Pressure Drop} = 0.025\text{ psig}$$

The next step is to find the refrigerant side pressure drop through the thermosyphon oil cooler. Find this on a heat exchanger rating sheet.

It is now necessary to determine the pressure drop through the thermosyphon return line.

Note: Figures 8 to 10 are based on a 4:1 recirculation ratio and the assumption that all the refrigerant flowing through the pipe is liquid. The pressure losses shown on the vertical axis of the graphs are correct only for the thermosyphon liquid supply line. The return line has a mixture of gas and liquid for which the assumption of all liquid flow in the pipe is not valid.

The calculation of gas and liquid mixture, or what is referred to as two phase flow pressure losses by hand, is very time-consuming and involved. To make the sizing easier, the traditional approach is to select the pipe based on the flow assumed to be all liquid, but use a very low pressure loss per length of pipe limit. This in effect increases the pipe size which compensates for the higher pressure loss when the flow is a mixture of gas and liquid. The 0.04 psi/100 ft limit selected is based on experience gained from actual installations.

For example, when selecting thermosyphon return lines for ammonia systems using a 0.04 psi/100 ft limit, the actual pressure drop in the pipe, although unknown, is not so high that it affects the performance of the system in the field.

Note: We have condensed a large amount of data into Figure 12. These graphs provide an easy method to determine the two phase flow pressure drop in the return line. They are valid for recirculation ratios from 2:1 up to 5:1 and for 85°F to 105°F (29.4°C to 40.6°C) condensing temperatures.

Returning to our example, the refrigerant flow rate is 58.6 lb/min in a 2 1/2 in. pipe. From Figure 12, the two phase flow pressure drop is roughly 1.5 psig/100 ft of pipe. The liquid line is expected to have elbows and at least one valve. There is little engineering data available for two-phase flow pressure losses through fittings and valves. In lieu of better data, determine an equivalent length, as for the liquid line loss, but multiply this by the two phase flow pressure loss.

Assume that the return line has two long radius elbows and one angle valve. From *Section V* in the *FRICK Engineering Data and Tables* pamphlet *(E20-4G/J66)*, the equivalent length of the elbows is 2.7 ft and the valve is 28 ft. Equation 9 provides the total equivalent length of pipe for the thermosyphon return line, where "h" is the required minimum liquid head height that is solved for. Then, multiply this equivalent length by the pressure loss for straight pipe. Use equation 9 to calculate the total pressure loss.

$$\text{Total equivalent length of pipe} = 6\text{ ft of pipe} + (2\text{ elbows x } 2.7\text{ ft/elbow}) + (1\text{ valve x } 28\text{ft/valve}) \qquad (8)$$

$$\text{Total Return Line Pressure Drop} = \frac{1.5\text{ psig}}{100\text{ ft}} \text{ X } 39.4\text{ ft} \qquad (9)$$

$$\text{Total Return Line Pressure Drop} = 0.59\text{ psig}$$

Determine the total pressure loss in the piping by adding the total pressure loss in the liquid line, the pressure drop through the oil cooler, and the total pressure loss in the return line. Equation 10 provides the total loss.

Note: The pressure loss through the exchanger in this example is 1 psig.

$$\text{Total System Pressure Loss} = 0.0025\text{ psi} + 1.00\text{ psi} + 0.59\text{ psi} \qquad (10)$$

$$\text{Total System Pressure Loss} = 1.62\text{ psi}$$

The total piping pressure drop of 0.87 psig must be overcome by the available liquid head. To calculate the thermosyphon driving pressure, determine the density of the refrigerant in the return line. Calculate this using equation 11. The total flow of refrigerant is divided into 75% liquid and 25% gas for a 4:1 recirculation ratio.

$$\text{Refrigerant Density in Return Pipe} = \frac{\text{Total Flow Rate}}{\frac{\text{Liquid Flow Rate}}{\text{Liquid Density}} + \frac{\text{Gas Flow Rate}}{\text{Gas Density}}} \qquad (11)$$

$$\text{Refrigerant Density in Return Pipe} = \frac{58.6\text{ lb/min}}{\frac{44\text{ lb/min}}{36.67\text{ lb/ft}^3} + \frac{14.6\text{ lb/min}}{0.6517\text{ lb/ft}^3}}$$

$$\text{Refrigerant Density in Return Pipe} = 2.5\text{ lb/ft}^3$$

Equation 12 provides the driving pressure due to the difference in densities of the liquid in the supply and return pipe.

$$\text{Driving Pressure} = \text{Density Difference x Gravity Constant x Height} \qquad (12)$$

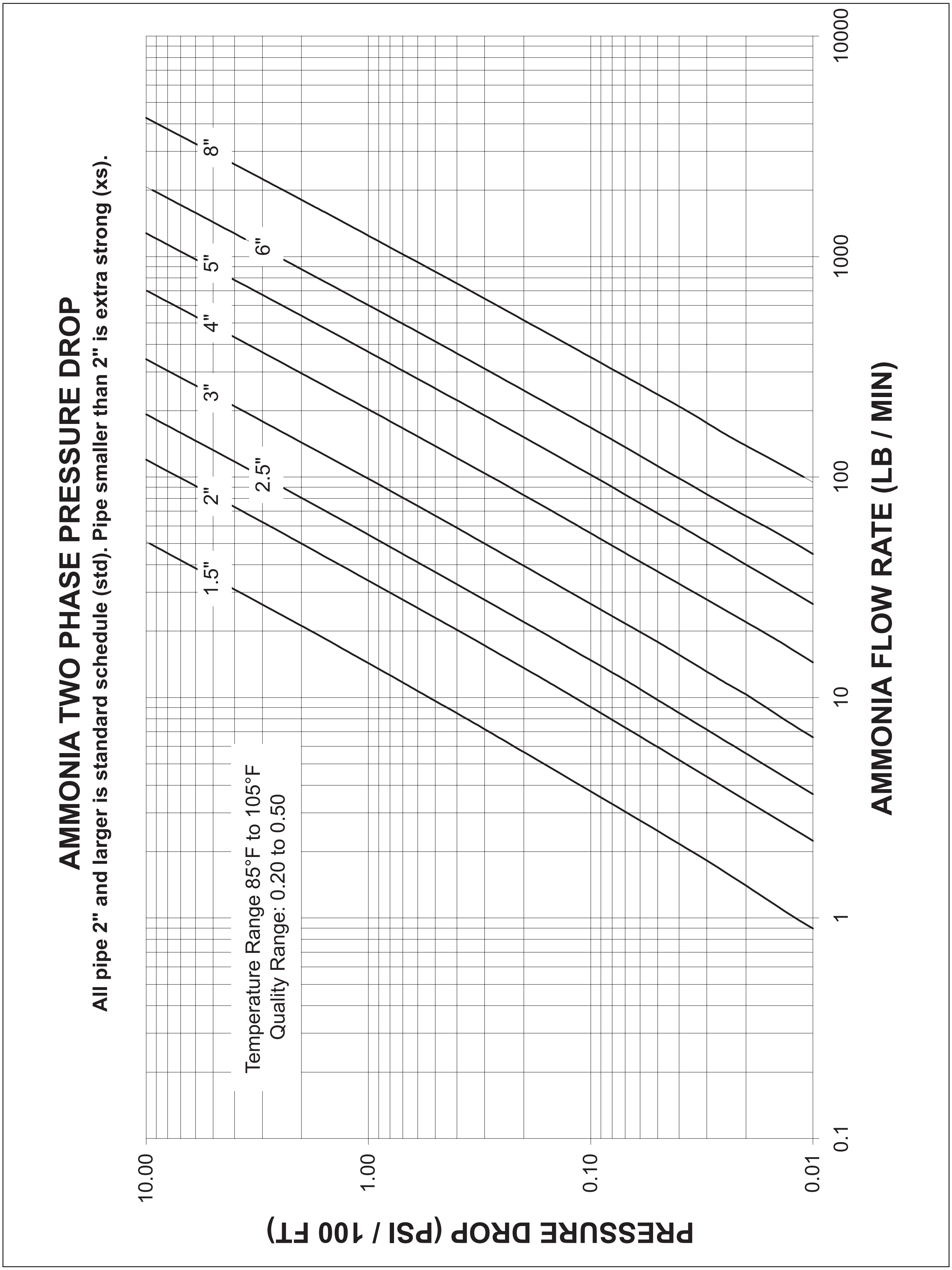
$$\text{Driving Pressure} = \frac{(36.67\text{ lb/ft}^3 - 2.5\text{ lb/ft}^3) \text{ x } 32.2\text{ ft/s}^2 \text{ x } 6\text{ ft}}{144\text{ in.}^2/\text{ft}^2 \text{ x } 32.2\text{ ft-lb/lbf - s}^2}$$

$$\text{Driving Pressure} = 1.90\text{ psi}$$

In this case, the height of the thermosyphon receiver above the oil cooler is 8 ft. The driving pressure to push the refrigerant through the piping and oil cooler is 1.90 psi. The frictional losses are lower at 1.62 psi. In a real system, with these pipe sizes, the flow rate would increase until the frictional pressure loss matched the driving pressure. This means the recirculation ratio is somewhat higher than initially assumed. This design is acceptable.

This is a simplified calculation for ease of presentation. A real system likely has more complicated piping with many more fittings, valves, and other components to consider. However, these techniques extend to any system. As mentioned in the beginning of this paper, ensure only qualified individuals familiar with industrial refrigeration piping practice who can determine the adequacy of these techniques design these systems.

Figure 12: R-717 two-phase pressure drop



May 2020 revisions

Page 6	- °C conversion range corrected

April 2020 Form Revisions

Throughout	- Changed term "downcomer" to vertical Liquid supply line
	- Changed term "riser" to wet return line
	- Removed references to the required 6 ft minimum elevation of liquid level line above oil cooler
	- Removed Graphs 4, 5, 6, 8, and 10
Page 3	- Updated oil cooler heat exchanger point in principle of operation section.
	- Updated Figure 1
Page 5	- Removed 6 ft minimum requirement from Figure 3
Page 7	- Removed 6 ft minimum requirement from Figure 4
Page 8	- Removed 6 ft minimum requirement from Figure 5
Page 10	- Removed 6 ft minimum requirement from Figure 6
Page 11	- Updated Figure 7
Page 13	- Removed R-22 from system sizing section
Page 14	- Removed R-22 information from Table 1
Page 18	- Changed Appendix A title to "Required minimum liquid head height calculation"
	- Removed R-22 information
Page 19	- Updated calculation (8), (10), and (12)

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JOHNSON CONTROLS
100 Cumberland Valley Avenue
Waynesboro, PA 17268-1206 USA
Phone: 717-762-2121 • FAX: 717-762-8624
www.jci.com/FRICK