Form E70-900E (AUG 2006)
EQUIPMENT MANUAL
File: EQUIPMENT MANUAL - Section 70
Replaces: E70-900E/JUL 2004
Dist: 3, 3a, 3b, 3c

THERMOSYPHON OIL COOLING

TABLE OF CONTENTS

INTRODUCTION

Rotary screw compressors typically used in industrial refrigeration require large quantities of oil be injected into the compressor. This is done to seal the rotors, lubricate the bearings and cool the discharge gas. Since these oil-flooded compressors were first introduced, various methods have been employed to cool the oil supplied to the compressor. Each of these methods of oil cooling can be categorized as either direct or indirect.

Direct cooling of the oil involves the injection of liquid refrigerant directly into the compressor rotors or the compressor discharge stream prior to oil separation. With this method, the oil and refrigerant mix resulting in the oil being cooled.

Indirect cooling of the oil 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 temperature as the cooling medium - has gained wide acceptance due to numerous advantages including:

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

The purpose of this paper is to provide general information and guidance on the design of piping for thermosyphon oil cooling systems. It should be noted that the design approaches, sizing criteria etc. 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. This paper is intended as an aide in the actual design and evaluation of these systems by qualified professionals. 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 (i.e. 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 cooler(s). As will be explained more fully below, the transfer of heat from the oil being cooled to the refrigerant maintains the density differential necessary for flow.

The basic equipment required for a thermosyphon oil cooling system consists of:

1. **A source of liquid refrigerant at system condensing temperature and pressure.** This liquid source should be located in close proximity to the compressor unit/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 cooler(s). This elevation difference is commonly referred to as “available liquid head”. A minimum liquid head greater than 6 feet (1.8 meters) above the cooler centerline generally results in satisfactory performance. The minimum head required may be greater based on actual system design. See Appendix A for a detailed calculation.
2. **An oil cooler heat exchanger.** In the illustrations contained in this paper, oil coolers are depicted as shell and tube heat exchangers with the refrigerant on the tube side. Other types of heat exchangers (i.e. plate type) can also be used in thermosyphon oil cooling systems.

Figure 1 depicts the basic thermosyphon system when no heat is being transferred from the oil to the refrigerant (as would be the case before a compressor has been started). Liquid refrigerant, at condensing temperature, fills the tubes 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.

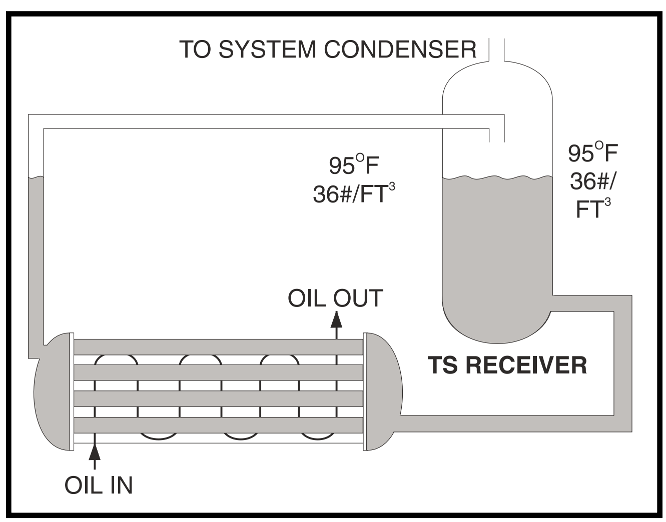


Figure 1. Thermosyphon Oil Cooler - System Not Operating.

When the compressor package is operating, hot oil (above the refrigerant temperature) flows through the shell of the oil cooler. Heat will flow through the tube 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 tubes will boil as it absorbs its latent heat of vaporization from the oil. Note that 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/vapor mixture that returns from the oil cooler is separated in the supply vessel. The vapor is vented to the inlet of the refrigerant condenser where it is reliquified.

The rate of refrigerant vaporization in the oil cooler can be determined 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 however, thermosyphon oil cooling systems are designed so that more refrigerant flows through the oil cooler than is actually vaporized. Typically, the refrigerant design flow rate is assumed 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 “downcomer” refers to the vertical run of piping by which refrigerant is supplied to the oil cooler, and the term “riser” 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 downcomer is all liquid, and its density is the same as it was before the compressor package was operating. The refrigerant in the riser 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 riser has a density that is considerably less than that of the refrigerant liquid in the downcomer.

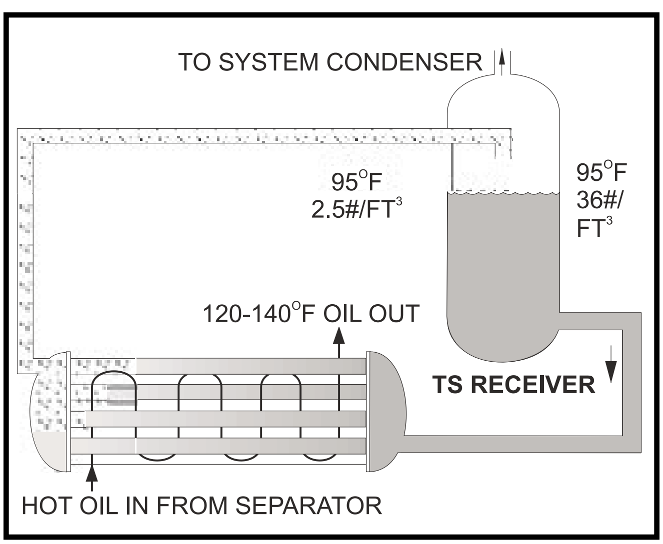


Figure 2. Thermosyphon Oil Cooler - System Operating

The difference in fluid densities, when multiplied by the height of the vertical legs in the thermosyphon piping loop yields a pressure differential. This pressure differential is the force that drives the flow of refrigerant in the thermosyphon loop. As does 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 will stabilize 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 1/2 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 will be fixed and the oil cooler heat rejection will vary depending on the compressor’s operating conditions. At lower rates of oil cooler heat rejection the refrigerant flow will be 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 needed, the harder the thermosyphon system works to provide it.

THERMOSYPHON SYSTEMS

There are numerous ways that a thermosyphon oil cooling system can be piped. Below are some typical examples of common practice. However this by no means covers all the possible geometries that will 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 his or her 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. The connections on the thermosyphon receiver (which can be either a horizontal or vertical pressure vessel) are located such that liquid refrigerant will fill this vessel up to a certain point and then overflow into the system receiver. The thermosyphon receiver must be elevated relative to the system receiver, and an equalizing line must connect these two vessels to ensure that liquid refrigerant will flow freely to the system receiver.

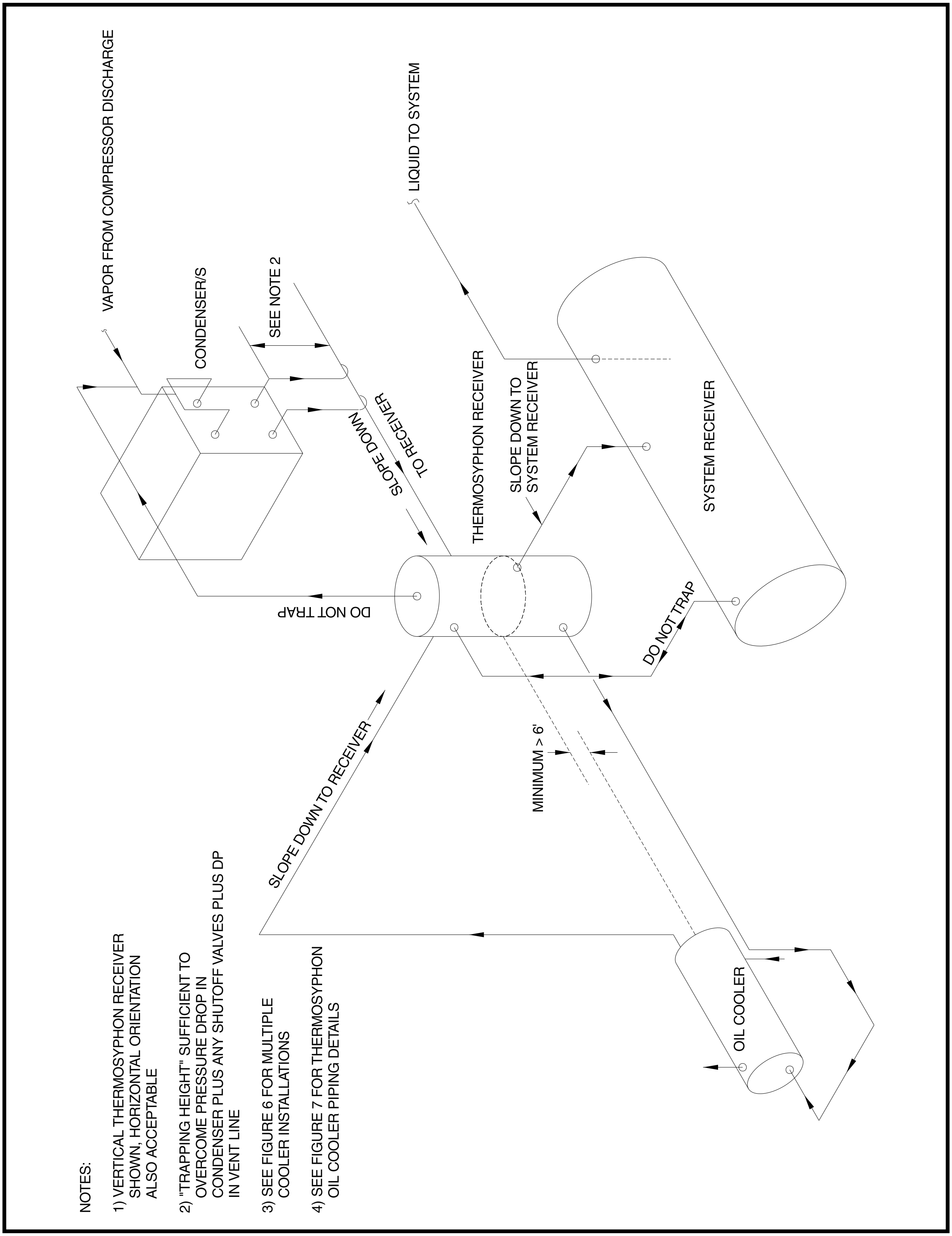


Figure 3. Flow-Through Thermosyphon Receiver

Note that 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 minute supply of refrigerant liquid to the oil cooler regardless of system demands.

As stated above under Principle of Operation, the liquid level in the thermosyphon receiver must be maintained at a minimum elevation greater than 6 feet (1.8 meters) above that of the oil cooler.

When the compressor package is operating, hot oil (above the refrigerant temperature) flows through the oil cooler shell. As heat transfers from the higher temperature oil to the lower temperature refrigerant, the oil is cooled while some of the refrigerant in the oil cooler tubes boils. During compressor operation, the downcomer supplying refrigerant to the oil cooler contains liquid refrigerant, while the riser exiting the oil cooler contains a mixture of refrigerant liquid and vapor. Gravity causes the more dense refrigerant liquid to flow downward to the oil cooler displacing the less dense liquid/vapor mixture and pushing it up the riser and back to the thermosyphon receiver. The liquid/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 is separated from the liquid and vented to the inlet of the refrigerant condenser where it is once again condensed.

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. The oil therefore is cooled to within an approach temperature of the refrigerant condensing temperature. Typically, this approach temperature is in the range of 15 - 35°F (8.3 - 19.4°C). For example, with a condensing temperature of 95°F, the temperature of the oil exiting the cooler would typically be 110°F to 130°F (43.4°C to 54.4°C).

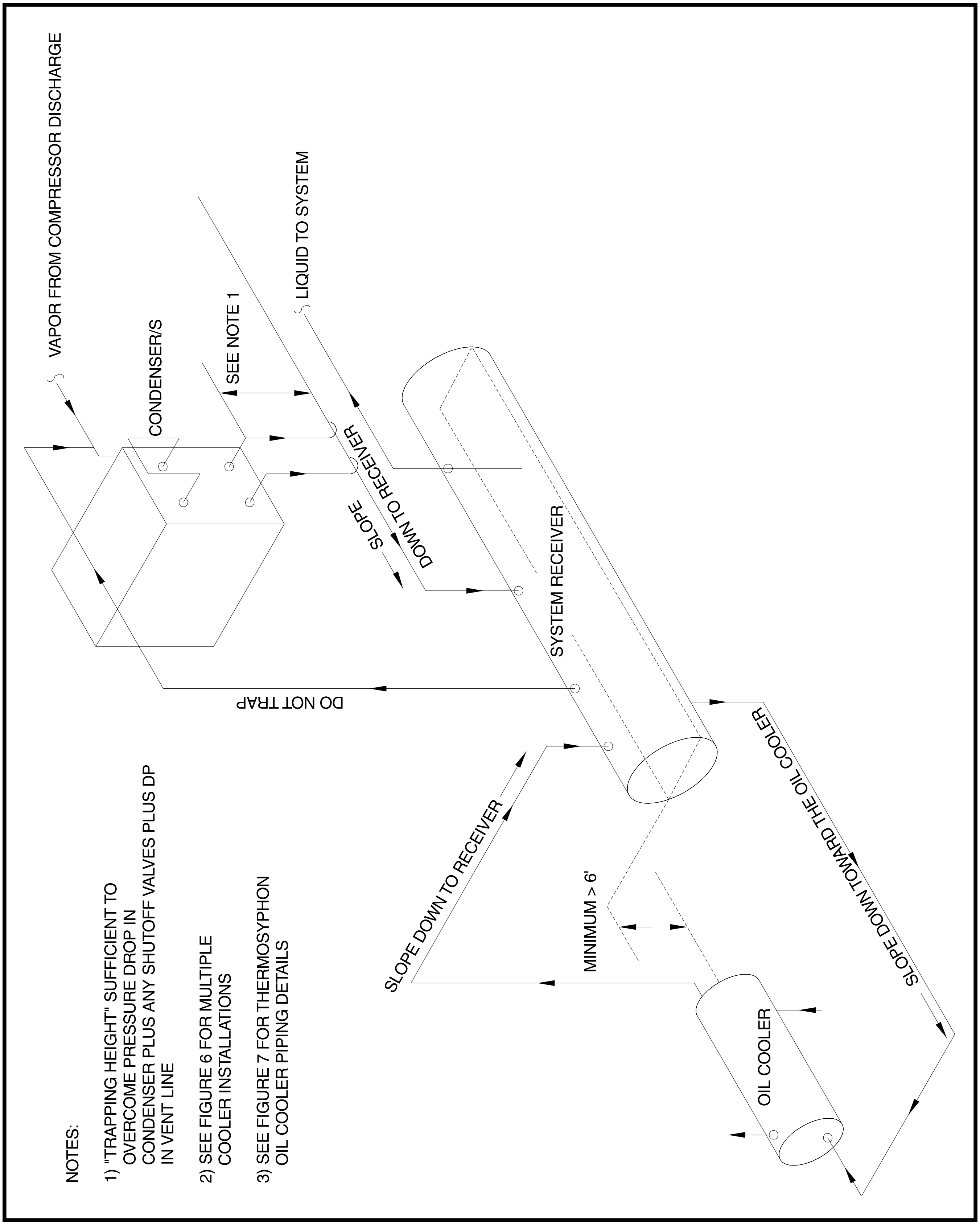


Figure 4. Elevated System Receiver

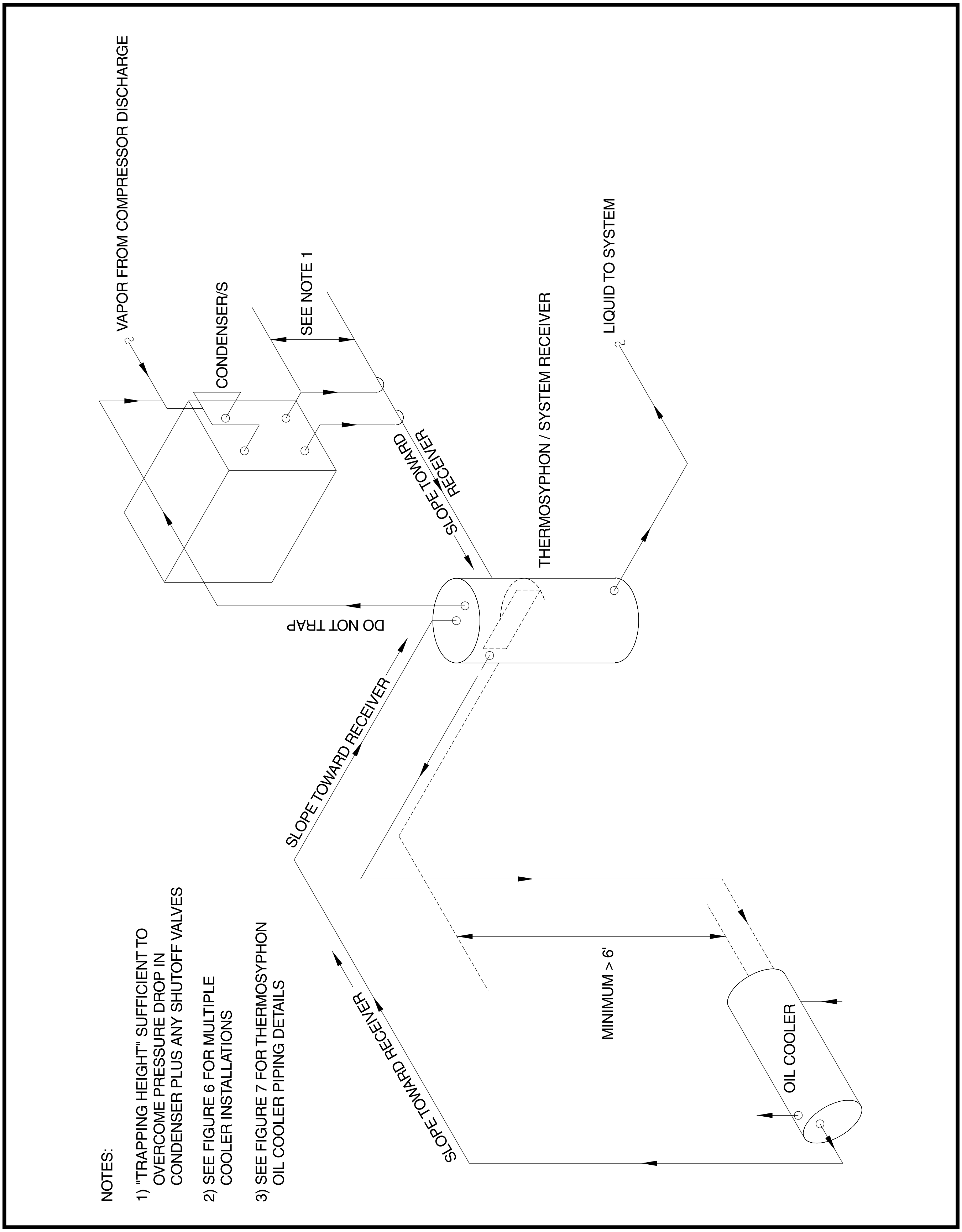


Figure 5. Combination Thermosyphon/System Receiver

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. The outlet connections on this vessel must be located 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 minute supply of refrigerant liquid to the oil cooler regardless of system demands. This can be accomplished by taking refrigerant for oil cooling off the bottom of the receiver and using a dip tube for the main system liquid line. Note that the nozzle on the receiver for the thermosyphon liquid supply, should project up into the receiver to prevent dirt being drawn into the supply line and eventually into the oil cooler.

With this approach, the system receiver must be located such that the minimum liquid level in the receiver vessel is at an elevation greater than 6 feet (1.8 meters) above that of the oil cooler. The operation of the thermosyphon oil cooler is the same as described previously.

If the system receiver can not be elevated 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/s 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 is used to feed the rest of the system. For this case, the liquid level in the weir dam must be elevated greater than 6 feet (1.8 meters) above that of the oil cooler.

Note that the liquid feed to the thermosyphon oil cooler should be taken off the receiver at the bottom of the weir dam. The volume of the weir dam should be sized to provide at least 5 minutes 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" seems to occur with these multiple-compressor systems. The operating principles for a thermosyphon system with multiple oil coolers however, 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 will be greatest at full load, while at other operating conditions heat rejection will be 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 will be added to the system in the future, common parts of the oil cooling system must be sized 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 recommended piping of a multiple compressor system using common supply and return headers. A word of caution: Consideration should 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, this will be an issue for a particular system, then separate liquid feed lines can be run as needed to avoid nuisance compressor shut downs. Alternatively, the compressor/s that run the most can be located 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 will form in this header as the saturated liquid refrigerant it contains absorbs heat from the warm engine room and vaporizes. These bubbles must be vented 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 ¼″ per foot in the direction of flow** (moving away from the supply vessel). If the "tail end" of the liquid header is then connected to the return header, any refrigerant vaporized can easily return to the supply vessel. The downcomers carrying liquid refrigerant to the individual oil coolers should exit the **bottom** of the supply header. Note that the end of the supply header can not exceed the elevation of the supply vessel.

The horizontal return header carrying the two-phase refrigerant mixture back to the supply vessel should also be pitched. In this case **the header should be pitched so it falls ¼″ per foot or more in the direction of flow** (moving towards the supply vessel). The risers carrying refrigerant liquid and vapor from the individual oil coolers should enter 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 riser in the middle. However, in this situation **the return header in the engine room must be sloped upward ¼″ per foot or more in the direction of flow** toward the vertical riser. This allows vapor bubbles in the riser in the engine room to return to the supply vessel. The part of the return header that is above the supply vessel should be sloped down toward the supply vessel as before.

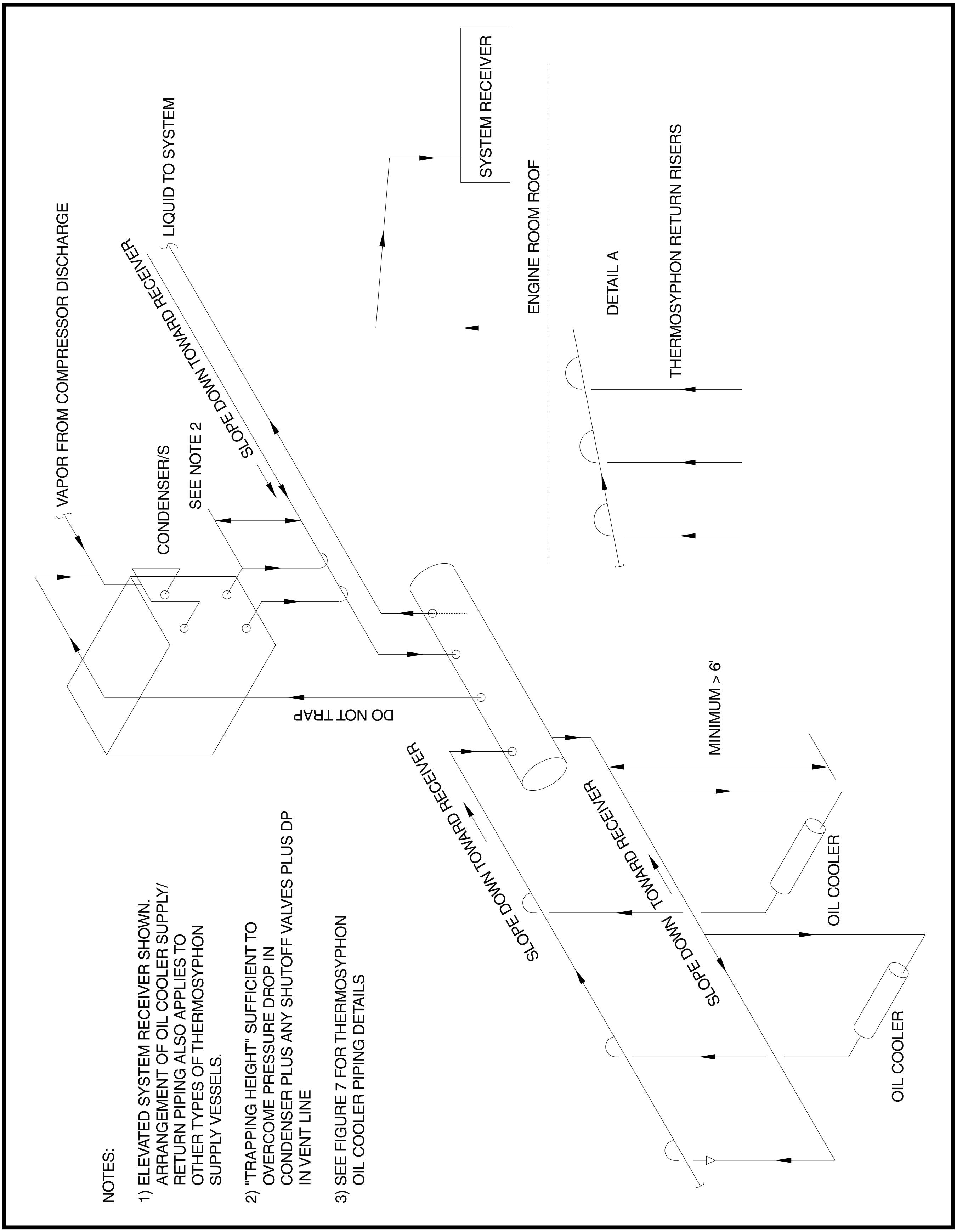


Figure 6. Multiple Thermosyphon Oil Coolers

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 recommended for installations where multiple thermosyphon coolers are connected to a common supply vessel. These valves should be generously sized and of a low pressure drop design. Globe valves should not be used. In addition a sight glass should be installed in the vertical leg of the refrigerant supply and return lines of every oil cooler to facilitate trouble shooting the system if necessary.

The supply vessel and the refrigerant side of the thermosyphon oil cooler(s) should be provided with oil drain valves at the low points. Over time, oil will accumulate in these locations and interfere with the efficient operation of the thermosyphon system. Periodic removal of oil is required.

Pressure vessel codes generally require that a pressure safety valve be installed 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 will generally be about 15 - 35°F (8.3 - 19.4°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 will simply "float" with the condensing temperature. If a thermosyphon oil cooling system without temperature control is contemplated for non-FRICK compressors, the compressor manufacturer should be contacted 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.

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. The recommended method, and by far the most common, 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 - above say 95°F (35°C) - the bypass remains closed and the oil temperature will "float" with the condensing temperature. At low condensing temperatures, some hot oil is bypassed around the cooler and mixed with cooled oil to maintain a constant oil supply temperature of approximately 130°F (54.4°C) to the compressor.

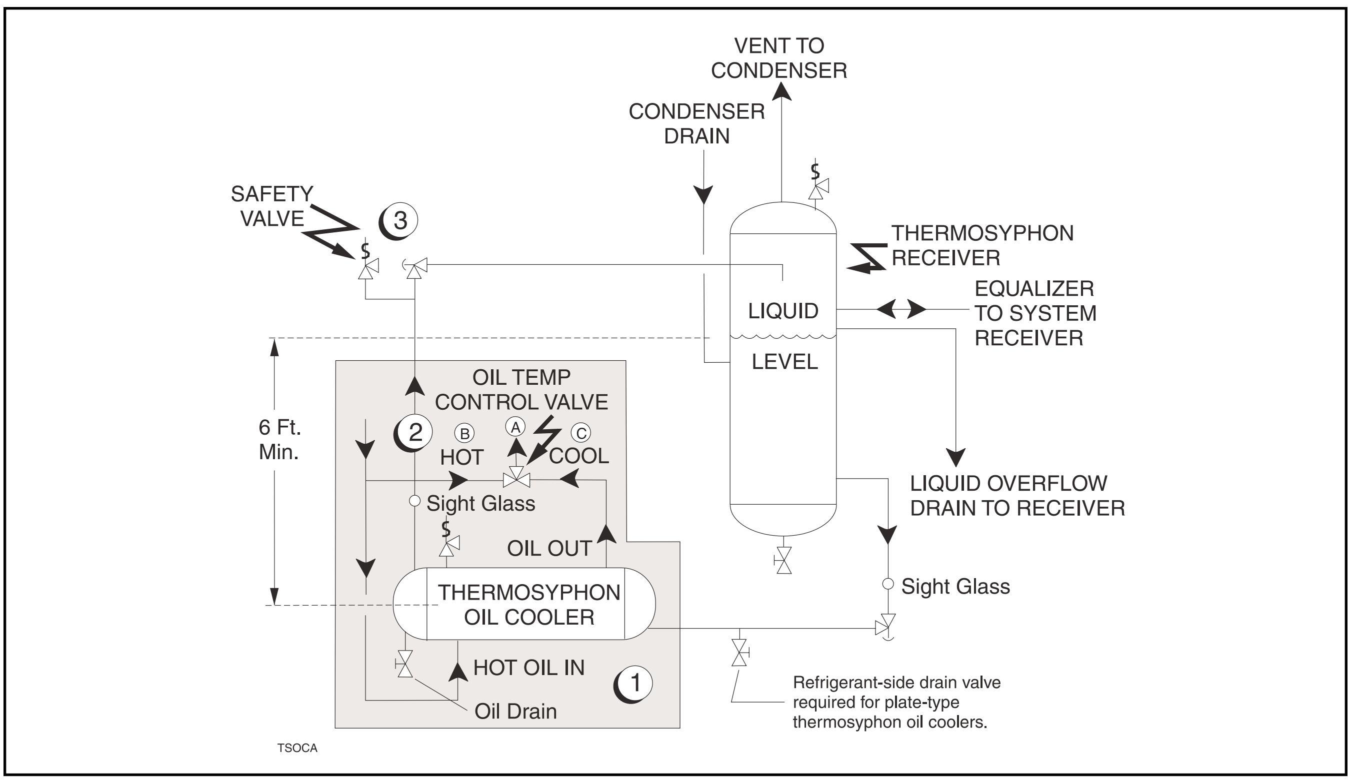


Figure 7. Thermosyphon Oil Cooler Piping

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 etc.

The system dynamics most likely to affect the operation of thermosyphon oil cooling systems, are those that cause intermittent interruptions in system refrigerant flows and/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 will permit a system to “ride through” brief interruptions in system refrigerant flow.

Rapid decreases in system condensing pressure however, 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 (i.e. defrosting of air units, harvesting ice makers etc.) although it can also be caused by a large, step increase in condenser capacity (i.e. 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 will be at a temperature higher than the equivalent boiling pressure. This will cause 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 downcomers and the risers.

Following this evacuation of liquid refrigerant from the heat exchanger, the liquid will return by gravity and normal thermosyphon operation will resume. 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”. Also note that 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, note that increasing the elevation of the thermosyphon receiver and oversizing the downcomers supplying liquid to the oil coolers will tend 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 condensing and a 4 to 1 recirculation ratio for both R-717 and R-22. For higher condensing temperatures it is recommended that all sizing be increased to assure that proper refrigerant flow is maintained. 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. Therefore, 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 presented 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/HR on an ammonia system condensing at 95°F.

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″ x 6 Ft. 500,000 BTU/HR
653,000 BTU/HR - - - - - - -ExampleHeat Load
16″ x 6 Ft. 875,000 BTU/HR - - - - - - -Select

STEP 2 - Sizing the Liquid Feed Line to the Oil Cooler and the Return Line to the Thermosyphon Receiver (See Graph 1, 2, or 3, for R-717 or Graph 4, 5, or 6, for R-22 depending on refrigerant and heat load).

A) Liquid Feed Line to Oil Cooler

Refer to Graph 1 (for R-717 heat loads to 1000 KBTU/HR). 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 (Graphs 1, 2 and 3)
0.50 PSID/100 Ft., R-22 (Graphs 4, 5 and 6)

Example: 653,000 BTU/HR

2″ - above 0.10 PSID/100 Ft. - - - - - - -Too Small
2-1/2″ - 0.058 PSID/100 Ft. - - - - - - -Select
3″ - 0.019 PSID/100 Ft. - - - - - - -Larger Than Required

B) Return Line from oil cooler

Return to Graph 1. Follow the same procedure as Step 2A but do not exceed:

0.04 PSID/100 Ft., R-717 (Graphs 1, 2 and 3)
0.20 PSID/100 Ft., R-22 (Graphs 4, 5 and 6)

Example: 653,000 BTU/HR

2-1/2″ - 0.058 PSID/100 Ft. - - - - - - Too Small
3″ - 0.019 PSID/100 Ft. - - - - - - - - - -Select

STEP 3 - Sizing the Return Vent Line from the Thermosyphon Receiver to the Condenser (See Graph 7 or 8 depending upon refrigerant).

Calculate the mass flow rate (LB/MIN) as follows:

A. Convert the oil cooler heat load in BTU/HR 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.

$$\frac{\text{BTU/MIN}}{\text{H}} = \text{LBS of Refrigerant / MIN}$$

H = 483.2 BTU/LB (R-717 @95°F)

H = 488.5 BTU/LB (R-717 @90°F)

H = 74.2 BTU/LB (R-22 @95°F)

H = 75.5 BTU/LB (R-22 @90°F)

Refer to Graph 7. 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/HR

A) $\frac{\text{653,000 BTU/HR}}{60} = \text{10,883.33 BTU/MIN}$

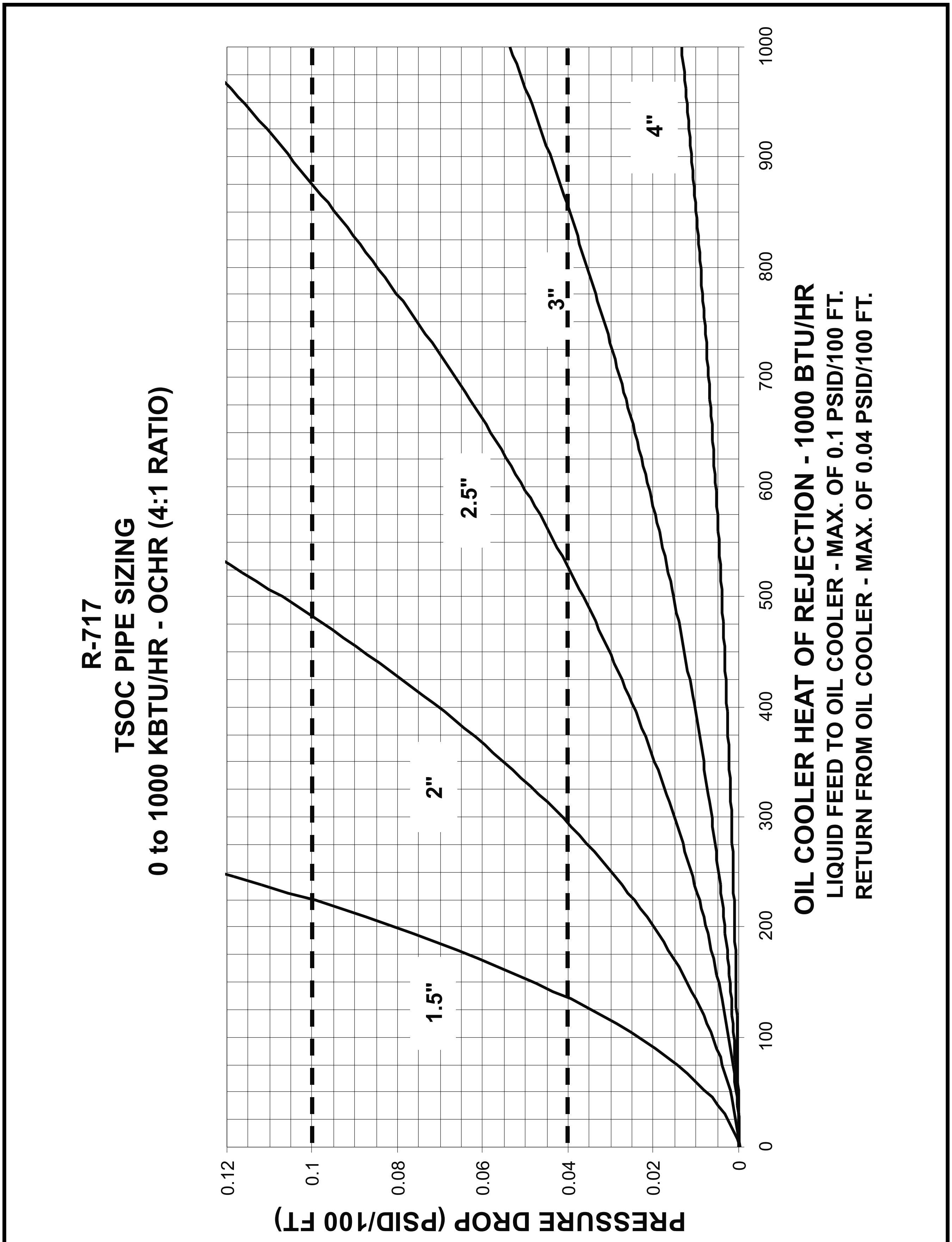
B) $\frac{\text{10,883 BTU/MIN}}{\text{483.2 BTU/LB}} = \text{22.52 LB/MIN}$

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

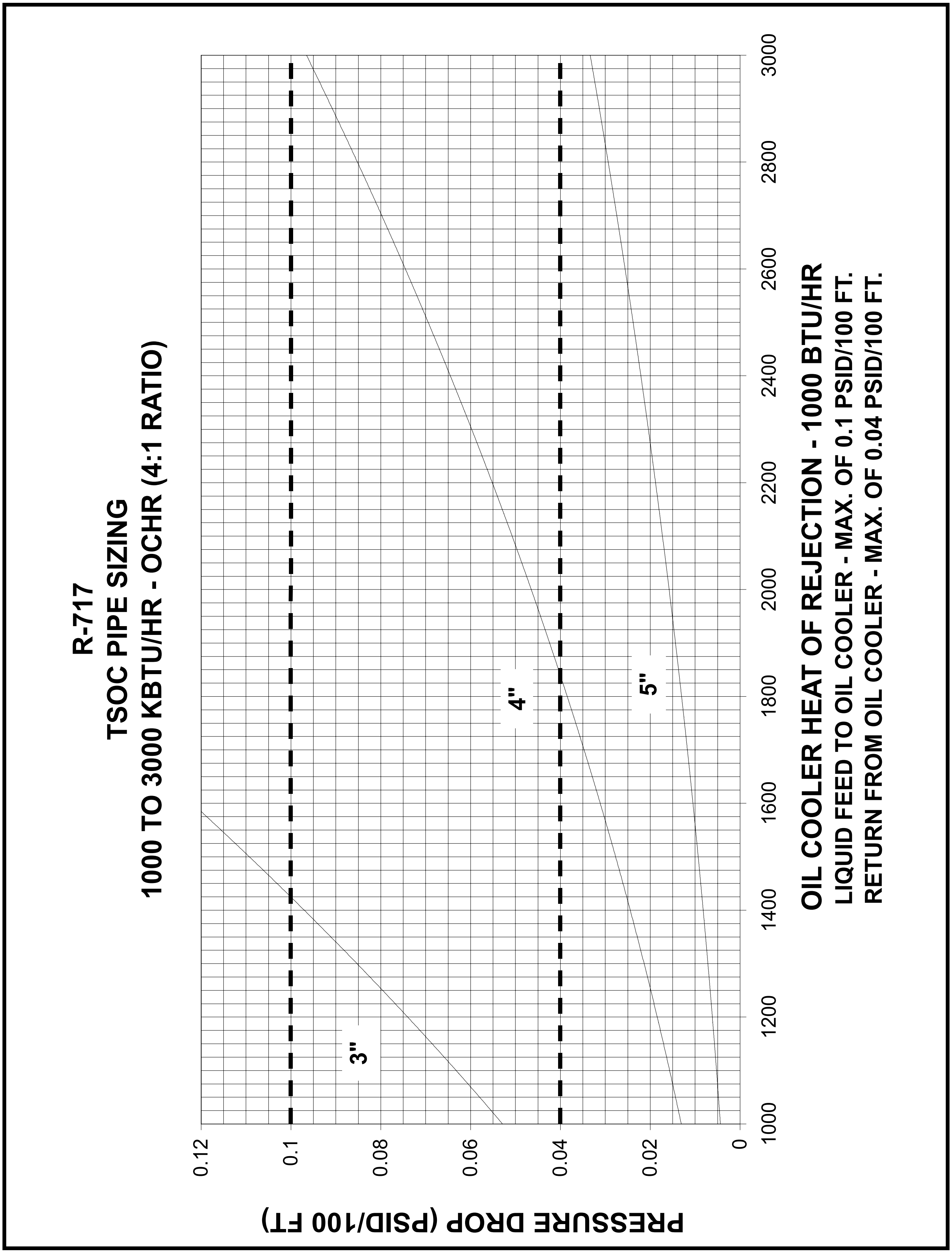
TABLE 1. THERMOSYPHON RECEIVER SIZING

(Based on 5 Min. Liquid Supply, 95°F Condensing)

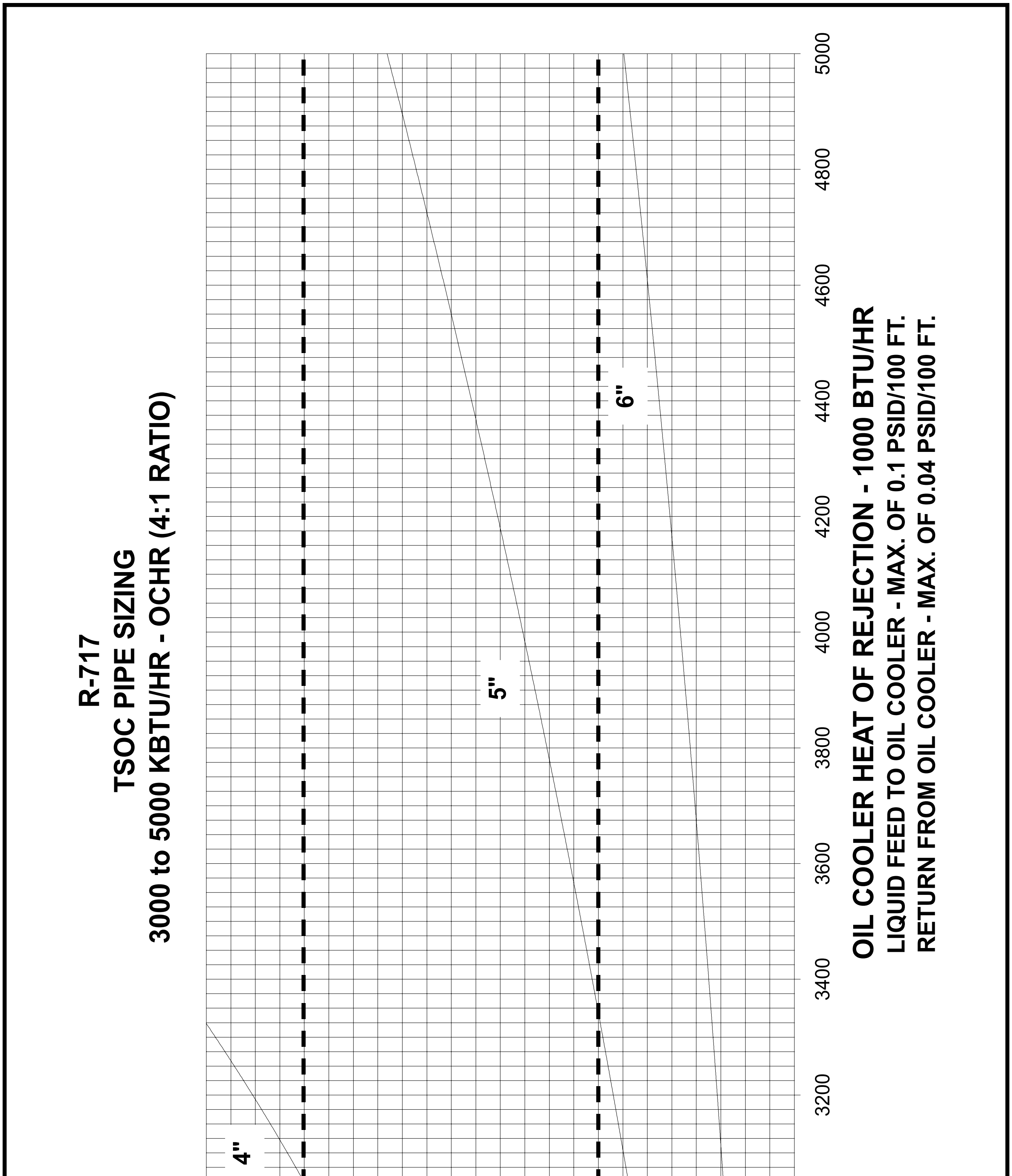
Shell Size	R-717 OCHR (BTU/HR) MAX.	R-22 OCHR (BTU/HR) MAX.
8 In x 6 Ft (Vert)	210,000	65,000
12 In x 6 Ft (Vert)	500,000	150,000
16 In x 6 Ft (Vert)	875,000	270,000
20 In x 6 Ft (Horiz or Vert)	1,400,000	420,000
24 In x 5 Ft (Horiz or Vert)	2,000,000	600,000
30 In x 5 Ft (Horiz or Vert)	2,600,000	720,000



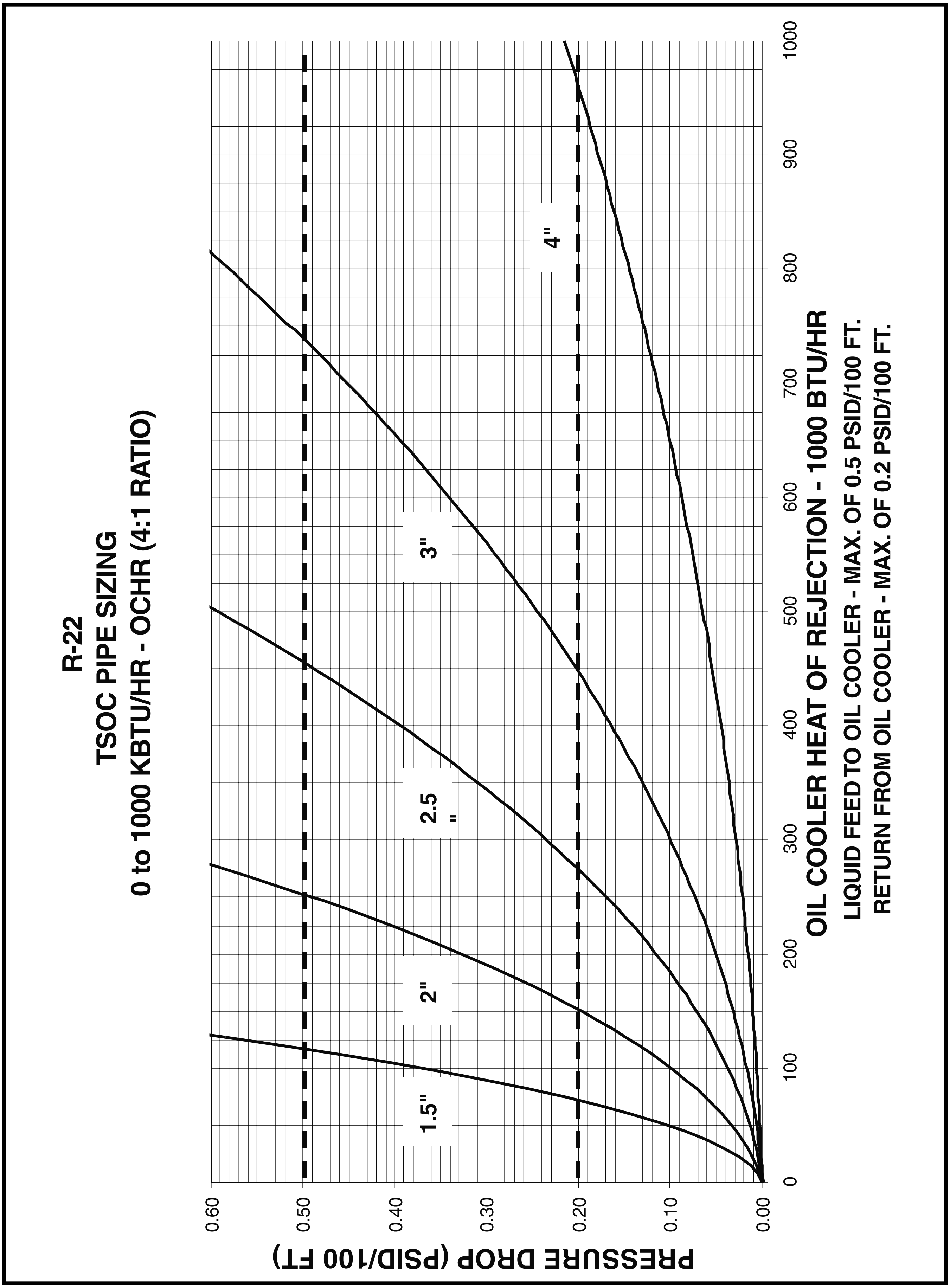
Graph 1. R-717 TSOC Pipe Sizing, 0 to 1000 KBTU/HR - OCHR



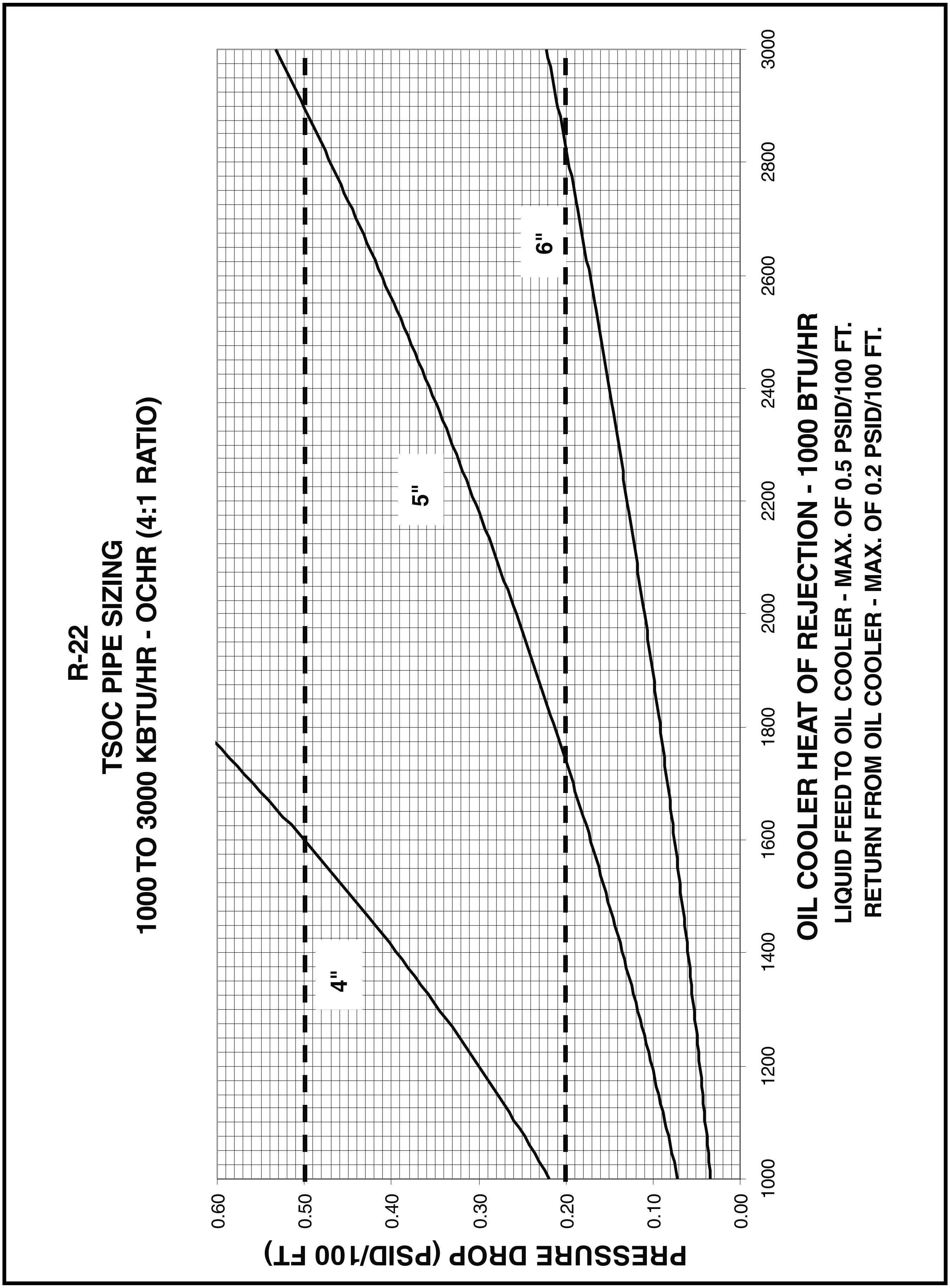
Graph 2. R-717 TSOC Pipe Sizing, 1000 to 3000 KBTU/HR - OCHR



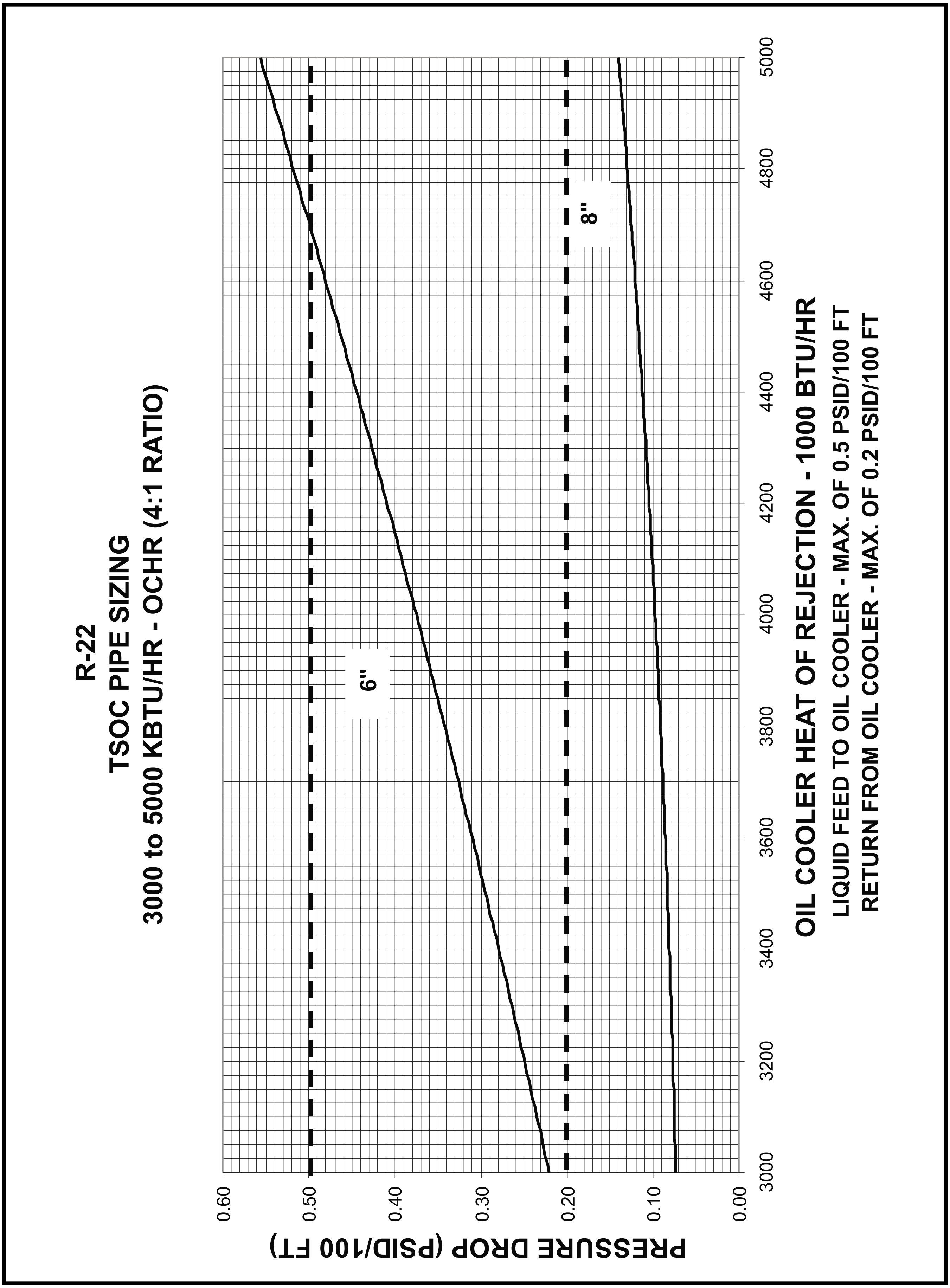
Graph 3. R-717 TSOC Pipe Sizing, 3000 to 5000 KBTU/HR - OCHR



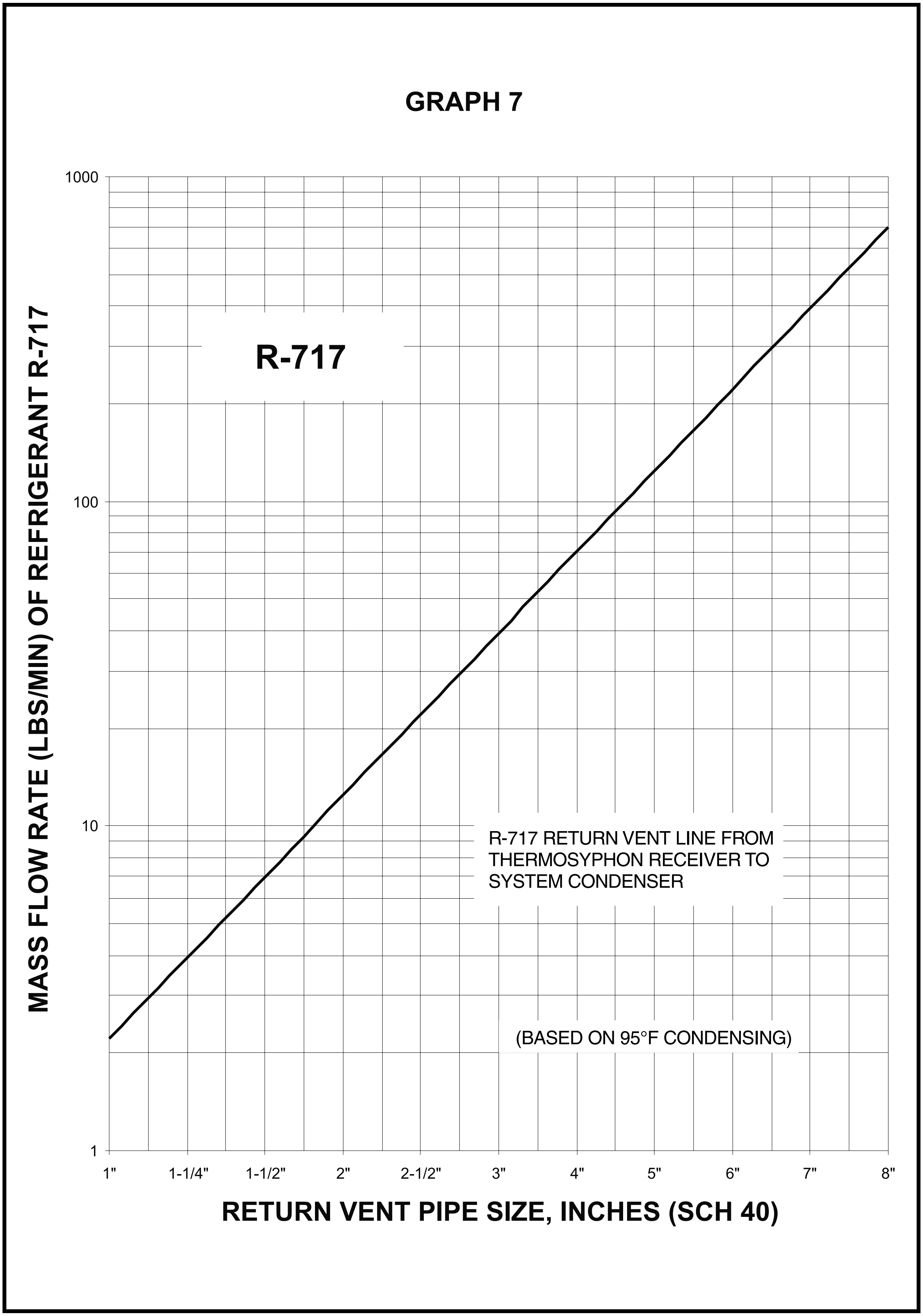
Graph 4. R-22 TSOC Pipe Sizing, 0 to 1000 KBTU/HR - OCHR



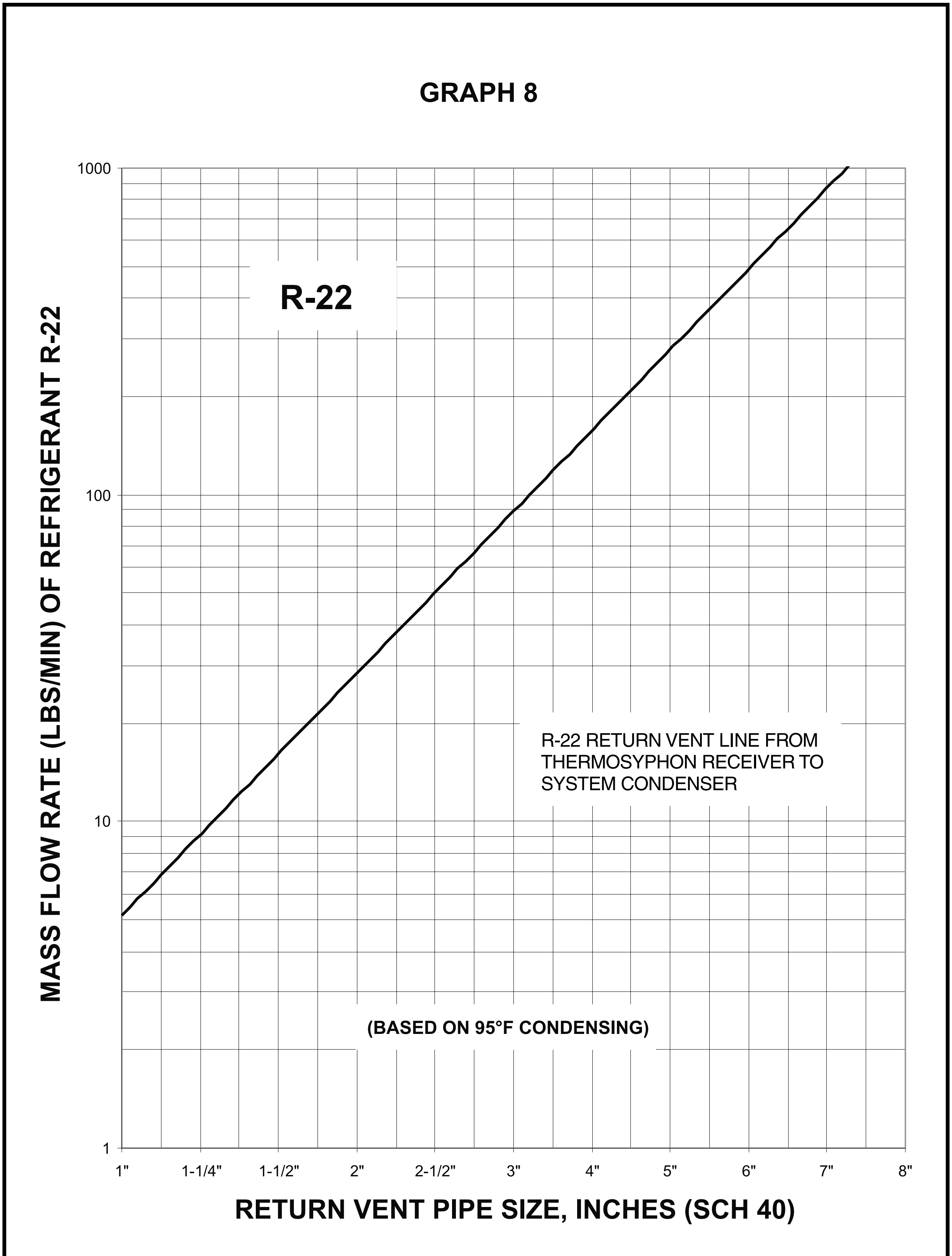
Graph 5. R-22 TSOC Pipe Sizing, 1000 to 3000 KBTU/HR - OCHR



Graph 6. R-22 TSOC Pipe Sizing, 3000 to 5000 KBTU/HR - OCHR



Graph 7. R-717 Return Vent Line Sizing



Graph 8. R-22 Return Vent Line Sizing

APPENDIX A - PRESSURE LOSS CALCULATION

The sizing example given previously will provide reasonably sized piping. However, if the graphs yield a size that seems questionable, a more rigorous calculation can be made as outlined below. In fact, once sizes have been selected from the graphs, we still recommend calculating pressure losses in the piping and comparing that to the available head.

The basic calculation process is to first determine the initial pipe sizes from the sizing graphs. For a given oil heat rejection rate, the refrigerant flow rate through the piping can be determined. Once the flow rate is known, the pressure loss in the liquid supply line can be determined, using the Moody diagram. The pressure loss in the return line which is a mixture of gas and liquid can be determined from the charts in this appendix. The pressure losses in the thermosyphon supply and return lines, plus an estimate of the cooler pressure loss, is then added together to get the total loss in the piping. This number is compared to the available liquid head to ensure that the selected piping is adequate.

For this example let's assume that the thermosyphon oil cooling load is 425 MBH for an ammonia system at 95°F condensing. Further, initially assume that the liquid level in the thermosyphon receiver is elevated 6 feet above the center line of the thermosyphon oil cooler. From Graph 1, a 2″ liquid supply line to the oil cooler results in a pressure loss of 0.08 psi/100ft. Since this is below the limit of 0.10 psi/100ft, the 2″ pipe is acceptable. However, for the return line the 0.08 psi/100ft exceeds the limit of 0.04 psi/100ft. A 2.5″ 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 to 1 recirculation ratio from equation 1. Note that the graphs 1 to 6 are based on a 4 to 1 recirculation ratio. By definition the refrigerant flow rate is 4 times the evaporation rate which can be determined from the oil cooler heat rejection and the latent heat of vaporization. The latent heat of vaporization or h_{fg} can be determined from the FRICK Engineering Data and Tables pamphlet (E20-4G/J66) at 95°F condensing temperature.

$$\text{Refrigerant Flow Rate} = \frac{\text{4 x Oil Cooling Load}}{\text{Enthapy of Vaporization}} = \frac{\text{4 x 425,000 BTU/HR}}{\text{483.2 BTU/LB x 60 MIN/HR}} \qquad (1)$$

$$\text{Refrigerant Flow Rate} = \text{58.6 LB/MIN}$$

With the flow rate known the pressure loss in the liquid supply line can be calculated using the Moody diagram and the Darcy-Weisbach formula (see an undergraduate engineering text on fluid mechanics). To use the Moody diagram the velocity of the refrigerant in the liquid line and then the Reynolds number must first be determined. The velocity in the 2″ liquid supply pipe is defined by equation 2. The density of the liquid refrigerant and cross sectional area of a 2″, schedule 40 pipe can be obtained from the FRICK Engineering Data and Tables pamphlet (E20-4G/J66) at 95°F condensing temperature.

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

$$\text{Velocity} = \frac{\text{58.6 LB/MIN x 144 IN}^2\text{/FT}^2}{\text{36.67 LB/FT}^3\text{ x 3.36 IN}^2} = \text{68.5 FT/MIN} = \text{4110 FT/HR}$$

The next step is to calculate the Reynolds number from equation 3. The viscosity of liquid ammonia at 95°F can be obtained from CoolWare™ software. Note that the Reynolds number does not have units.

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

$$\text{Reynolds \#} = \frac{\text{36.67 LB/FT}^3\text{ x 4110 FT/HR x 0.1723 FT}}{\text{0.2985 LB/FT - HR}} = 87{,}000$$

The last piece of information needed to use a Moody diagram is to determine the relative roughness of the pipe given by equation 4. The absolute roughness for various materials along with a Moody diagram may be found in Mark's Standard Handbook for Mechanical Engineers.

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

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

Given the Reynolds number and relative roughness, the Moody diagram can be read to determine the friction factor is 0.023. The pressure drop in the pipe is calculated from the friction factor by equation 5. Note that 100 feet of pipe is assumed. The result is that 100 feet of 2″ schedule 40 pipe will have a pressure loss of 0.067 psi with 58.6 LB/MIN of liquid ammonia flowing through it at 95°F.

In our case we have assumed six feet of vertical pipe and there typically will also be elbows and at least 1 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 publication (E20-4G/J66). A table of equivalent lengths for valves and fittings is given.

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

$$\text{Pressure Drop} = \text{0.023 x 36.67 LB/FT}^3 \text{ x } \frac{\text{100 FT}}{\text{0.1723 FT}} \text{ x } \frac{(\text{4110 FT/HR})^2}{2} \text{ x}$$
$$\text{x} \left(\frac{1}{\text{32.2 FT - LB/LBF - SEC}^2}\right) \text{x} \left(\frac{\text{1 HR}}{\text{3600 SEC}}\right)^2 \text{ x } \frac{\text{1 FT}^2}{\text{144 IN}^2}$$

$$\text{Pressure Drop} = \text{0.069 psig / 100 feet of 2 inch pipe}$$

For our example assume that there are two, 2″ long radius elbows and one 2″ angle valve. Reading the table for ferrous fittings results in an equivalent length of 2.3 feet for a welded elbow and 25 feet for a flanged angle valve. The total equivalent length of straight pipe is given by equation 6.

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

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

The total pressure loss in the thermosyphon liquid supply line is given by equation 7.

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

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

The next step is to find the refrigerant side pressure drop through the thermosyphon oil cooler. This can be found on a heat exchanger rating sheet if available. If specific data is not available a reasonable estimate of the heat exchanger pressure drop is 0.25 psi for ammonia applications and 1 psi for R-22 applications.

It is now necessary to determine the pressure drop through the thermosyphon return line. It should be emphasized that the graphs 1 to 6 are based on a 4 to 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/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 thermosyphon return lines for ammonia systems are selected 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.

Personal computers now make it possible to estimate the actual pressure drop of the gas/refrigerant mixture in the thermosyphon return line. We have condensed a large amount of data into Graphs 9 and 10. 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 condensing temperatures.

Returning to our example, the refrigerant flow rate is 58.6 LB/MIN in a 2-1/2″ pipe. From graph 9, the two phase flow pressure drop is roughly 1.5 psig / 100 feet of pipe. As for the liquid line it is expected that there will be elbows and at least one valve in the line. There is little engineering data available for two phase flow pressure losses through fittings and valves. In lieu of better data, the recommended approach is to 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 2 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. The total equivalent length of pipe for the thermosyphon return line is given by equation 8. This equivalent length is then multiplied by the pressure loss for straight pipe. The total pressure loss is calculated by equation 9.

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

$$\text{Total equivalent length of pipe} = 39.4 \text{ feet of pipe}$$

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

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

The total pressure loss in the piping can now be determined 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. The total loss is given by equation 10.

$$\text{Total Piping Pressure Loss} = 0.025 \text{ psig} + 0.25 \text{ psig} + 0.59 \text{ psig} \quad (10)$$

$$\text{Total Piping Pressure Loss} = 0.87 \text{ psig}$$

The total piping pressure drop of 0.87 psig must be overcome by the available liquid head. To calculate the thermosyphon driving pressure, the density of the refrigerant in the return line must be determined. This can be calculated from equation 11. Note that the total flow of refrigerant is divided into 75% liquid and 25% gas for a 4 to 1 recirculation ratio.

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

$$\text{Refrigerant Density in Return Pipe} = \frac{58.6 \text{ LB/MIN}}{\frac{44 \text{ LB/MIN}}{36.67 \text{ LB/FT}^3} + \frac{14.6 \text{ LB/MIN}}{0.6517 \text{ LB/FT}^3}}$$

$$\text{Refrigerant Density in Return Pipe} = 2.5 \text{ LB/FT}^3$$

The driving pressure due to the difference in densities of the liquid in the supply and return pipe is given by equation 12.

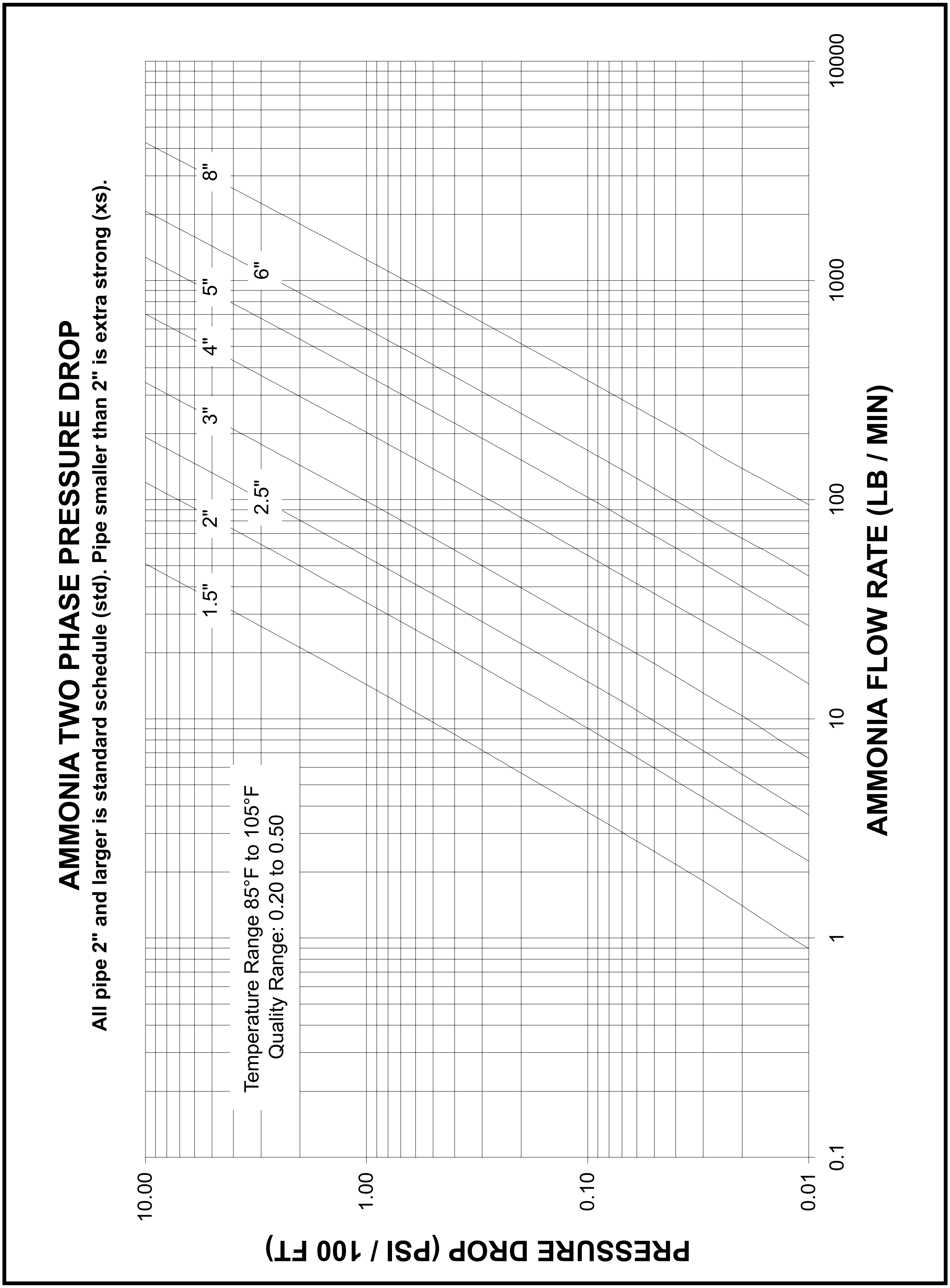
$$\text{Driving Pressure} = \text{Density Difference x Gravity Constant x Height} \quad (12)$$

$$\text{Driving Pressure} = \frac{(36.67 \text{ LB/FT}^3 - 2.5 \text{ LB/FT}^3) \text{x } 32.2 \text{ FT/SEC}^2 \text{ x } 6 \text{ FT}}{144 \text{ IN}^2/\text{FT}^2 \text{ x } 32.2 \text{ FT - LB/LBF - SEC}^2}$$

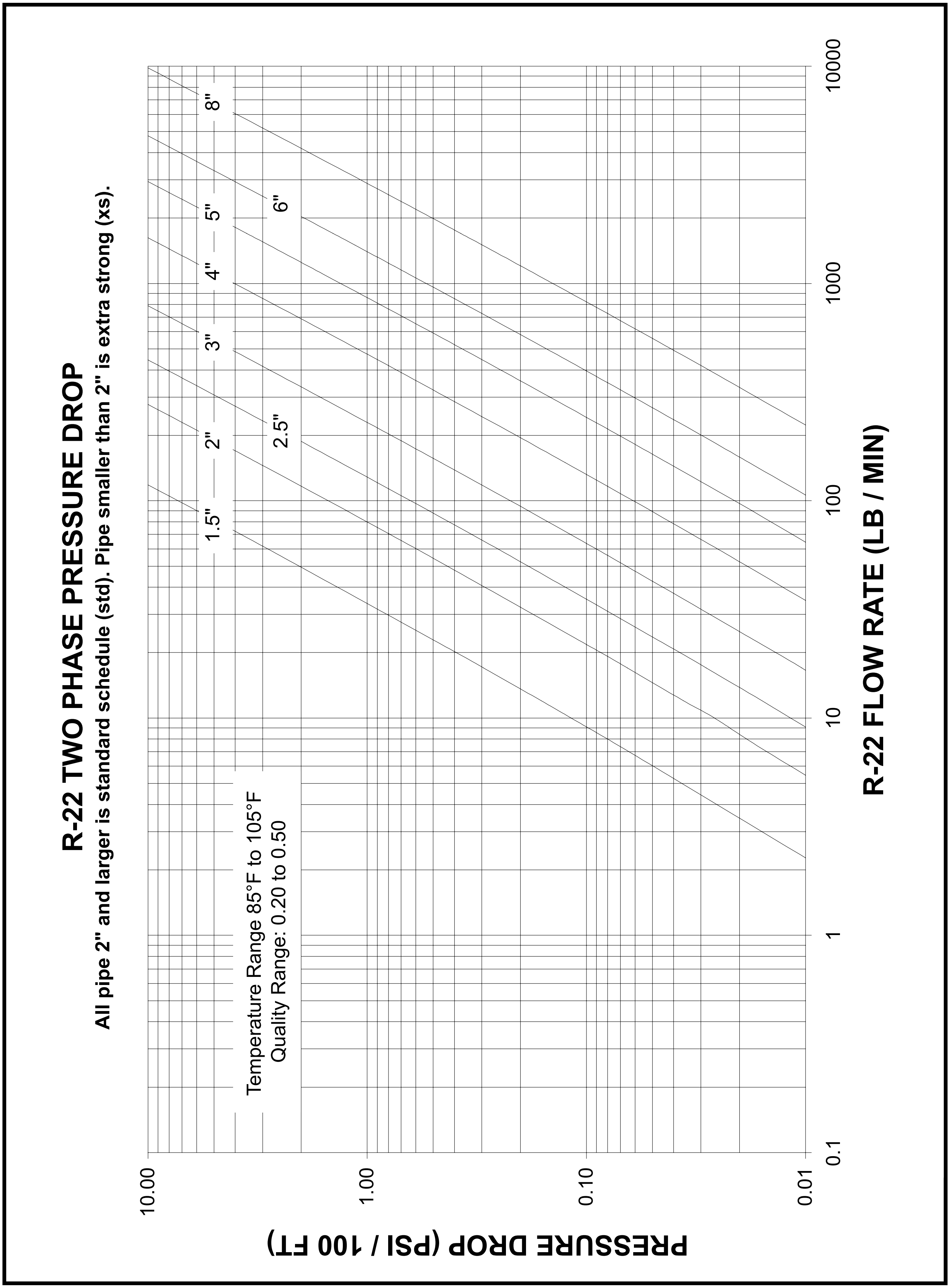
$$\text{Driving Pressure} = 1.4 \text{ psig}$$

Therefore, the driving pressure to push the refrigerant through the piping and oil cooler is 1.4 psi. The frictional losses are lower at 0.87 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 will be somewhat higher that initially assumed. This design is acceptable.

The calculation presented has been simplified for ease of presentation. A real system will likely have more complicated piping with many more fittings, valves, etc. However, the techniques presented here can be extended to any system. As mentioned in the beginning of this paper, these systems must be designed by qualified individuals familiar with industrial refrigeration piping practice, and who can determine the adequacy of the techniques presented.



Graph 9. R-717 Two-Phase Pressure Drop



Graph 10. R-22 Two-Phase Pressure Drop

This page left blank intentionaly.

Form E70-900E (AUG 2006)
Supersedes: E70-900E JUL 2004
Subject to change without notice
Printed in USA • GUI 500 805 1.08

A JOHNSON CONTROLS COMPANY

Frick®
100 CV Avenue • P.O. Box 997
Waynesboro, PA USA 17268-0997
Phone: 717-762-2121 • FAX: 717-762-8624
www.frickcold.com