



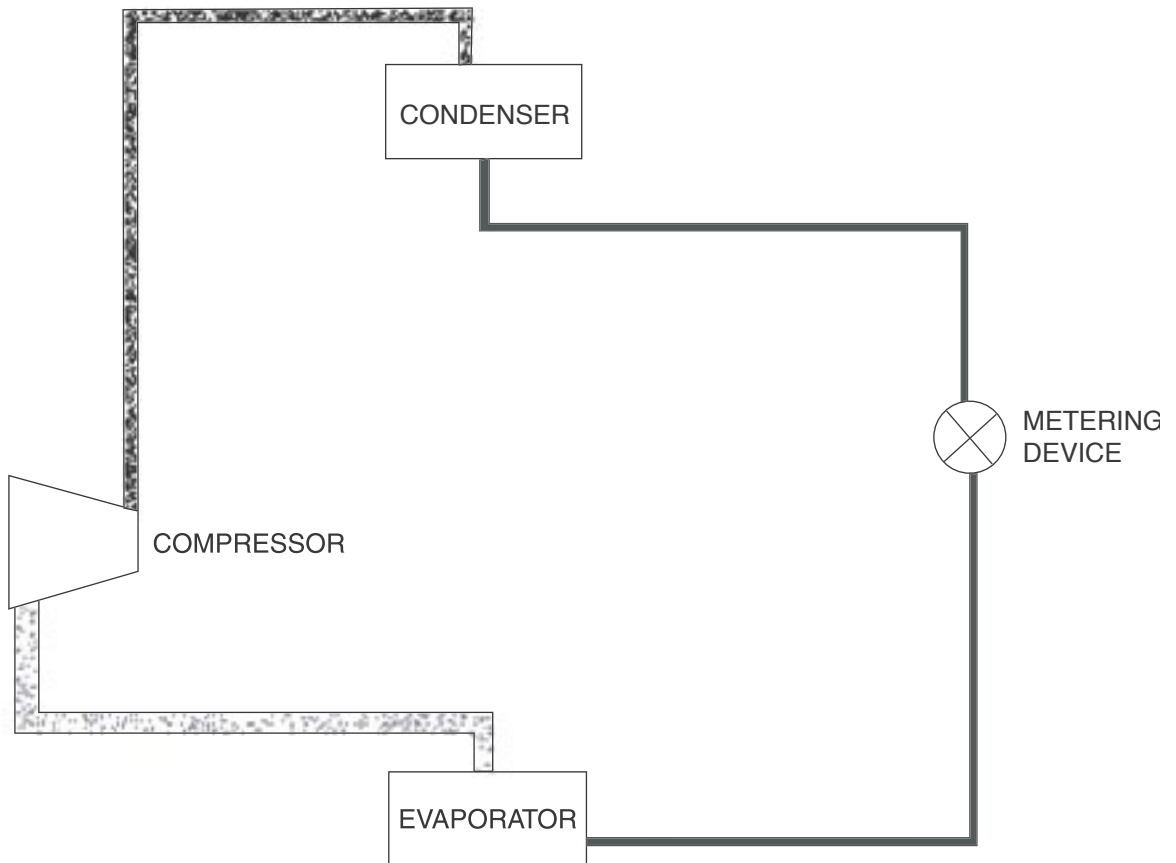
DX PIPING GUIDE

ENGINEERING SUPPLEMENT

New Release

Form No. 050.40-ES2 (102)

DX PIPING GUIDE



LD07429

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1.0 OVERVIEW

The current best practices for DX system piping are outlined in this document for systems using R-22, R-134a and R-407C refrigerants.

The objectives that influence the design of piping systems for refrigeration systems are to:

- Ensure proper refrigerant feed to evaporators
- Provide economical pipe sizes without excessive pressure drop
- Ensure lubricating oil return to the compressor by preventing excessive amounts of oil from being trapped in the system
- Minimize the loss of oil from the compressors
- Prevent liquid refrigerant or excessive amounts of oil from entering the compressor either during operation or during idle time

Improper design and sizing of refrigerant piping may result in loss of system efficiency and/or eventual failure of the system. Factors that must be considered in a piping design are the inter-relationships between velocity, pressure, friction, as well as, economics. Economics favor the use of the smallest possible line sizes. However, high suction and discharge line pressure drops will cause loss in capacity and increased power consumption. Another important design criterion is oil return to the compressor. The refrigerant line velocities have to be sufficiently high to carry oil up suction or hot gas risers at all operating capacities.

Sections 2.0 and 3.0 deal with refrigerant line sizing from a frictional pressure drop and oil return point of view. Section 5.0 discusses hot gas bypass arrangements. Losses in valves and fittings are discussed in Section 6.0. Multiple compressor piping recommendations are covered in Section 7.0. Tables are provided with factors for proper refrigerant line sizing.

2.0 REFRIGERANT LINE SIZING

The pressure drops (line losses) are typically presented as a given change in the corresponding saturation temperature. The effect of line losses on the capacity and energy consumption (kW/ton) is illustrated in Figure 1. Line sizing is a balance between pressure drop (reflected in system performance) and oil return (for system reliability).

Pressure drop issues are addressed in Section 3.1 while oil return considerations are dealt with in Section 4.0.

3.0 PRESSURE DROP CONSIDERATIONS

As mentioned previously, pressure drop calculations are determined as pressure changes associated with a change in saturation temperature of the refrigerant. Systems are typically sized for pressure losses of 2°F or less for the discharge, suction and liquid lines. This is the conventional method for sizing and is accepted practice throughout the industry (ASHRAE).

Tables 1 through 6 show capacities for R-22, R-134a and R-407C at specified pressure drops for the various refrigerant lines.

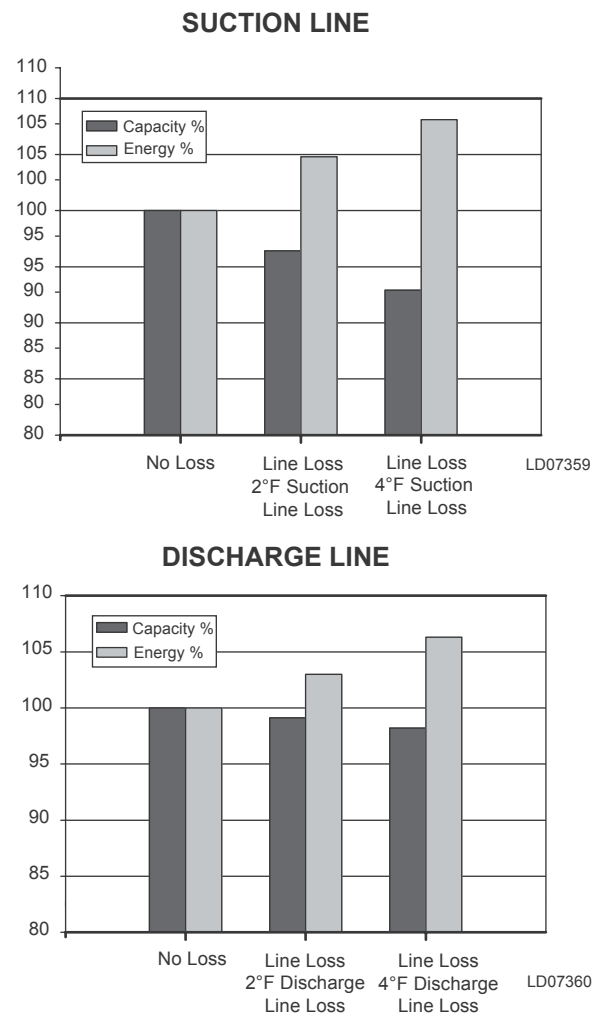


FIG. 1 – EFFECT OF SUCTION AND DISCHARGE LINE PRESSURE DROP ON CAPACITY AND POWER (ASHRAE). (R-22 system operating at 100°F saturated condensing and 40°F saturated evaporating temperature. Energy percentage is rated at kW/ton.)

Section 3.1 will address suction line sizing while discharge and liquid line sizing will be discussed in Sections 3.2 and 3.3, respectively.

3.1 SUCTION LINES

Pressure drop in the suction line reduces a system's capacity by forcing the compressor to operate at a lower suction pressure in order to maintain the desired temperature in the evaporator. The suction line is normally designed to have pressure drop no more than 2°F change in saturation temperature due to friction. Tables 1, 3 and 5 contain suction line sizing data for refrigerants 22, 134a and 407C, respectively. Capacities for Type L copper and steel tubes at three different pressure drops of 2, 1 and 0.5°F at various saturated suction temperatures and condensing temperature of 105°F are provided. Correction factors for capacity for condensing temperatures other than 105°F is given in the notes at the bottom of Tables 2, 4 and 6 for the above mentioned refrigerants.

For low pressure drops, the suction riser must be properly sized to ensure oil entrainment up the riser thereby assuring oil is always returned to the compressor. This is dealt with in detail in Section 4.0. When pipe size needs to be reduced to ensure sufficient velocities for oil return at partial loads, the pressure drops may get excessive at full load. This can be compensated for by over sizing the horizontal and down run lines.

3.2 DISCHARGE LINES

Pressure loss in the discharge line increases the power to capacity ratio (kW/ton) of the system and decreases the compressor capacity. This is illustrated in Figure 1. ASHRAE recommends a saturation temperature change of 1°F based on frictional pressure drop for discharge line sizing. Tables 2, 4 and 6 contain discharge line sizing data for refrigerants 22, 134a and 407C, respectively. Capacities are for a condensing temperature of 105°F and correction factors are given at the bottom of the tables for other condensing temperatures.

3.3 LIQUID LINES

Pressure drop in liquid lines causes flashing of the refrigerant and reduction in pressure at the liquid feed device. ASHRAE recommends that systems be designed so that the pressure drop be no more than 1 to 2°F change in saturation temperature. Tables 2, 4 and 6 contain liquid line sizing data for refrigerants 22, 134a and 407C, respectively, based on frictional pressure drop causing a 1°F change in saturation temperature. Liquid subcooling is the only means of over-

coming the liquid line pressure loss to ensure liquid entering the expansion device. Insufficient subcooling may lead to flashing of liquid refrigerant in the liquid line, which may result in degradation of system performance. Liquid line risers are an additional source of pressure loss. The loss in the risers is approximately 0.5 psi per foot of liquid lift (ASHRAE). Other losses include those caused by accessories like solenoid valves, filter driers, hand valves, etc.

Liquid lines from the condensers to the receivers should be sized for a refrigerant velocity of 100 fpm or less to ensure positive gravity flow without backup of liquid flow (ASHRAE). Sizing data is provided in Tables 2, 4 and 6.

4.0 OIL CIRCULATION CONSIDERATIONS

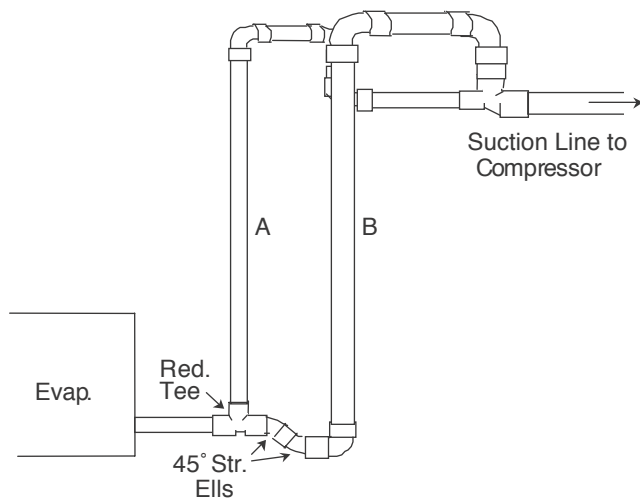
All compressors inevitably lose some oil during normal operation. The oil leaving the condenser is dissolved in the liquid refrigerant and oil return through the liquid line to the evaporator usually poses no problem. The oil separated in the evaporator returns to the compressor by gravity or by shear forces induced by the suction vapor. Some systems utilize oil separators, but they are not 100% effective, and hence the oil that finds its way into the system needs to be returned to the compressor. In systems where the capacity can be modulated, the system piping needs to be designed to return oil at the lowest load condition, while not imposing excessive pressure drop at full load.

Section 4.1 describes the methodology used for sizing of single and multiple suction risers as well as their construction. Suction piping at multiple evaporator coils is illustrated in Figures 3 and 4 of Section 4.2. Hot gas riser sizing details are addressed in Section 4.3.

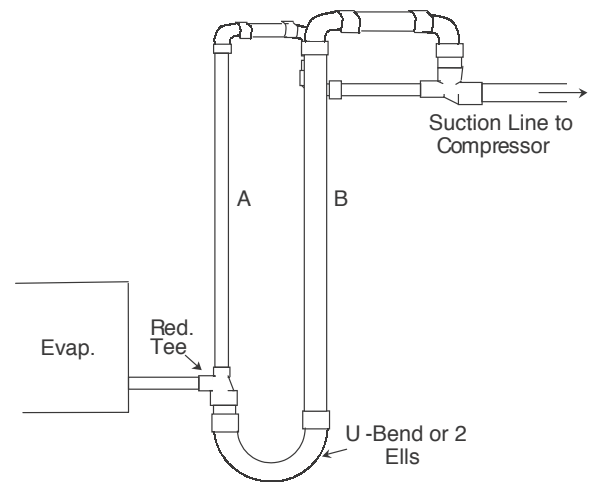
4.1 SUCTION LINE SIZING

Many systems contain a suction riser because the evaporator is located at a lower level than the compressor. Oil circulating in the system returns up the suction riser with the returning gas. The principal factors that govern the transport of oil by refrigerant vapor are the vapor velocity, vapor density, and inside pipe diameter.

Suction risers must be sized for the minimum capacity. Table 7 lists minimum refrigeration capacity for various pipe sizes at various saturated suction and suction gas temperatures for refrigerants 22, 134a and 407C.



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Method A

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Method B**FIG. 2A – DOUBLE SUCTION RISER DESIGN****FIG. 2B – DOUBLE SUCTION RISER DESIGN**

When a suction riser is sized to allow oil return at minimum load condition, the pressure drop in this line may be too high when operating at full load. If a correctly sized suction riser causes excessive pressure drop at full load, a double riser should be used. Figures 2A and 2B shows two methods of riser construction. The operation of the double riser is as follows:

Riser A is designed for the minimum load possible.

Riser B is sized for satisfactory pressure drop through both risers at full load. Riser B is sized such that the combined cross sectional area of A and B is equal to or slightly greater than that of a single pipe sized for an acceptable pressure drop at full load without any consideration for oil return at minimum load. The combined cross sectional area, however, should not exceed that of a single pipe that would return oil in an upflow riser under maximum load conditions.

During minimum operation, the gas velocity is not sufficient to carry oil up both the risers. Oil tends to accumulate in the trap between the two risers until riser B is completely sealed off. The gas velocity is now sufficient to carry oil along riser A.

The trap capacity should be maintained to a minimum by close coupling fittings otherwise the oil hold-up could lower the oil level in the compressor crankcase,

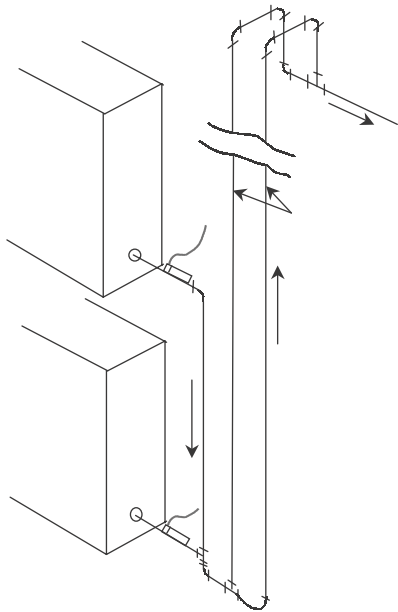
thereby impairing compressor operation. The two risers form an inverted loop and enter the horizontal line. This prevents oil drainage into risers that may be idle during part load operation. Another situation that warrants the use of double suction risers is when multiple compressors are used in a circuit. In this case, one continues to operate while the others may shut down, and the ratio of maximum to minimum capacity becomes large.

4.2 MULTIPLE EVAPORATOR COILS

Suction lines should be designed such that oil from the active evaporator does not drain into an idle one. Figures 3A through 3D show the recommended piping construction for multiple evaporators and their relative positions with respect to the compressor. Figure 4 shows the typical piping for evaporators above and below a common suction line. All horizontal runs should be level or pitched approximately 1/4" per linear foot toward the compressor to ensure oil return.

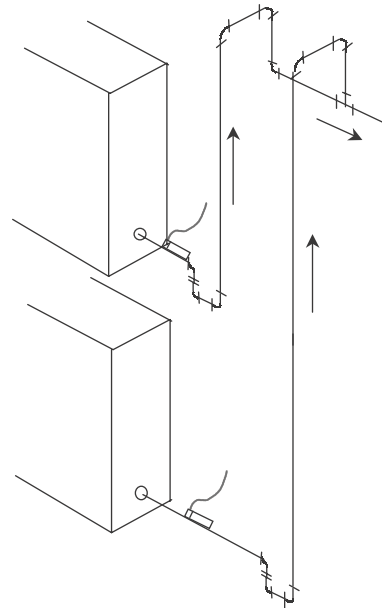
The traps after the evaporator suction outlets are recommended to prevent erratic functioning of the thermal expansion valve. The expansion valve bulbs are located on the suction lines between the evaporator and these traps. The traps serve as drains for refrigerant and helps prevent accumulation of liquid under the valve bulbs during compressor off cycles.

Multiple Evaporators Stacked on Same Level – Compressor Above



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**Multiple Evaporators Stacked on Same Level – Compressor Above
Arrangement “A” Preferred**

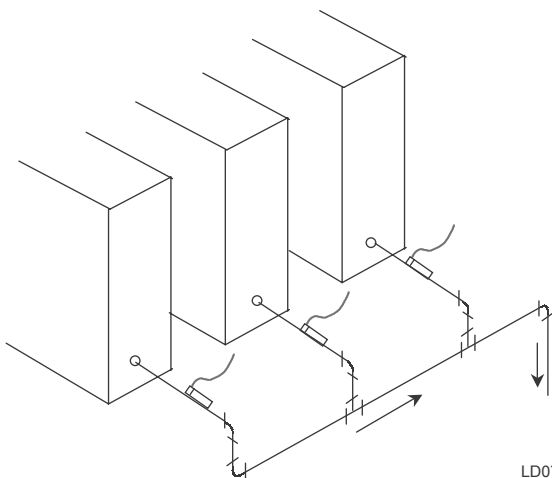


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FIG. 3A – SUCTION LINE PIPING ON MULTIPLE EVAPORATOR

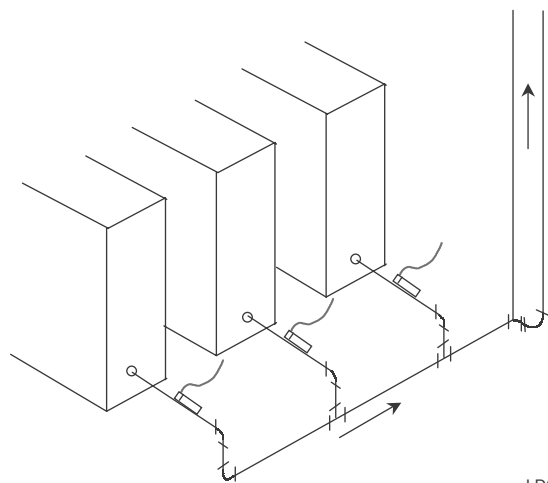
FIG. 3B – SUCTION LINE PIPING ON MULTIPLE EVAPORATOR

Multiple Evaporators on Same Elevation – Compressor Below



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Multiple Evaporators on Same Level – Compressor Above



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FIG. 3C – SUCTION LINE PIPING ON MULTIPLE EVAPORATOR COILS

FIG. 3D – SUCTION LINE PIPING ON MULTIPLE EVAPORATOR COILS

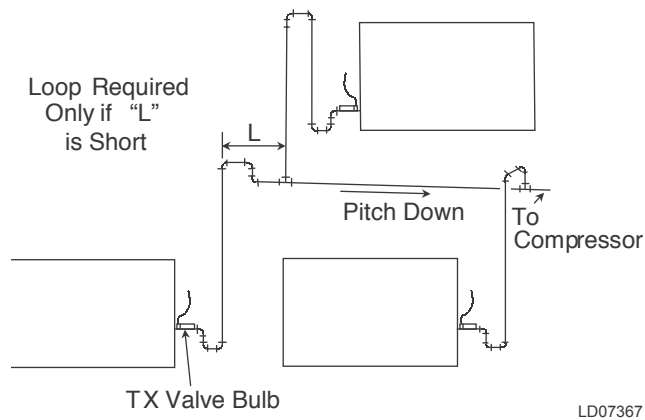


FIG. 4 – TYPICAL PIPING FOR EVAPORATORS LOCATED ABOVE AND BELOW SUCTION LINE

4.3 HOT GAS LINE SIZING

Low pressure drops are desired in hot gas lines. However, oversized lines can reduce gas velocities to the extent that oil is not transported by the refrigerant. Therefore, when using multiple compressors with capacity control, a hot gas riser should be appropriately sized to transport oil at all possible load conditions. Table 8 lists the minimum capacity in tons for oil transport up hot gas risers of different sizes for refrigerants 22, 134a and 407C. The capacities are given for various saturated condensing temperatures and discharge gas temperatures.

In installations with multiple compressors and with capacity control, a vertical hot gas line designed for oil transport at minimum load may cause excessive pressure drop when operating at full load. In such cases, either a double riser or a single riser with an oil separator should be used. A double hot gas riser can be applied similar to the case of the suction line. Figure 5 shows a schematic of a double hot gas riser design. Section 4.1 should be referred to for the operating principle and sizing of the risers.

As an alternative to double risers, a single riser and an oil separator can be used. The separator is located just before the riser so that any oil draining back down the riser accumulates in the oil separator. The oil is then returned to the compressor by feed mechanisms such as a pump, etc. Horizontal lines should be level or pitched approximately 1/4" per linear foot in the direction of gas flow to facilitate transport of oil through the system and back to the compressor. Mufflers are recommended to dampen discharge gas pulsations. Mufflers should

be installed horizontally or in the downflow portion of the hot gas line immediately after the compressor. The gas velocity through the muffler being lower than that through the line, there is a tendency of oil to accumulate within the muffler. The muffler should be installed such that it prevents the accumulation of oil.

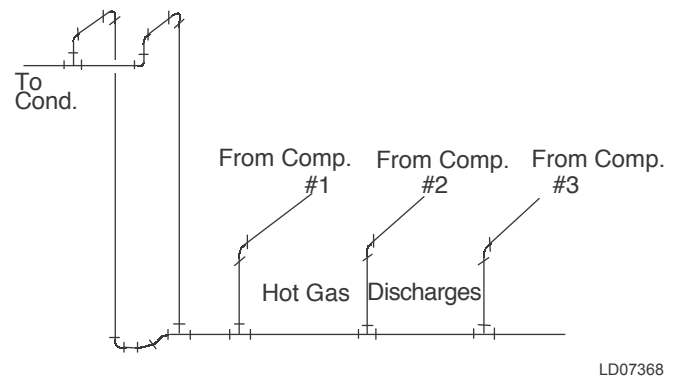


FIG. 5 – DOUBLE HOT GAS RISERS

5.0 HOT GAS BYPASS ARRANGEMENTS

Most compressors are equipped with unloaders that help the compressor start with a low starting torque and permit capacity control without stopping the compressor. Reduction of starting torque can also be accomplished by using a manual or automatic valve between the compressor discharge and suction lines. To avoid overheating the compressor, the valve is opened only during the start of the compressor and closed when the compressor attains full speed.

Figures 6(a) through 6(d) show various recommended bypass arrangements (ASHRAE). Figure 6(a) shows the simplest configuration. The compressor will overheat when used for long periods of time. The arrangement in Figure 6(b) shows the use of hot gas bypass to the exit of the evaporator. It is recommended that the expansion valve bulb be placed at least 5 feet downstream from the bypass point entrance. In Figure 6(c), the hot gas bypass enters after the thermostatic expansion valve bulb. Another expansion valve provides liquid to the bypass line for desuperheating purposes. The preferable arrangement is shown in Figure 6(d). Here, the bypass is connected into the low side between the expansion valve and the entrance to the evaporator. If a distributor is used, the bypass gas enters between the expansion valve and the distributor. The hot gas bypass line should be sized such that its pressure loss is only a small fraction of the pressure drop across the valve.

6.0 LOSSES IN FITTINGS AND VALVES

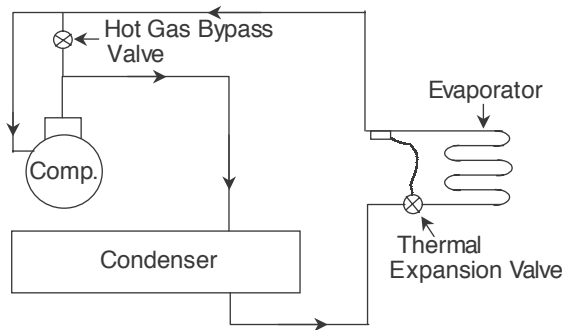
The refrigerant line capacity tables (Tables 1 through 6) are based on pressure drop per 100 equivalent feet of straight pipe, or combination of straight pipe and fittings and valves with friction drop equivalent to 100 feet of straight pipe. Pressure drop through fittings and valves is determined by establishing the equivalent straight length of pipe of the same size. Tables 9, 10 and 11 give the equivalent lengths of straight pipe for various fittings and valves based on nominal pipe sizes (ASHRAE).

7.0 MULTIPLE COMPRESSOR PIPING

Piping design for field-erected multiple compressor systems involves the need for uniform distribution of oil and refrigerant among the compressors. The design must also prevent oil and liquid refrigerant draining back to the heads of idle compressors. Suction, discharge and crankcase equalizing piping arrangements are discussed in Sections 7.1, 7.2 and 7.3, respectively.

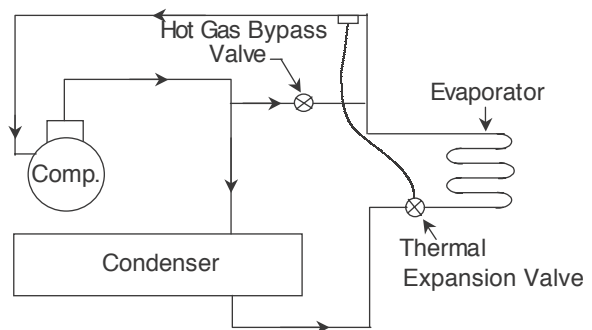
7.1 SUCTION PIPING

Suction piping in multiple compressors operating in parallel should be designed to maintain the same suction pressure and ensure that oil is returned in equal proportions. All suction lines should be brought to a common suction header. The suction header is a means of distributing oil and suction gas as uniformly as possible between the compressors. The header should run above the level of the compressor suction inlets so that oil can return to the compressors by gravity. Figure 7 shows a pyramidal or yoke-type suction header to maximize pressure and flow equalization for three compressors piped in parallel (ASHRAE). In case of two compressors in parallel, a single feed between two compressor take offs is an acceptable configuration. The suction header should be sized such that the suction gas and oil separate. The suction flow for the compressors should be tapped off at the top of the header and devices to feed the oil back to the compressor need to be utilized.



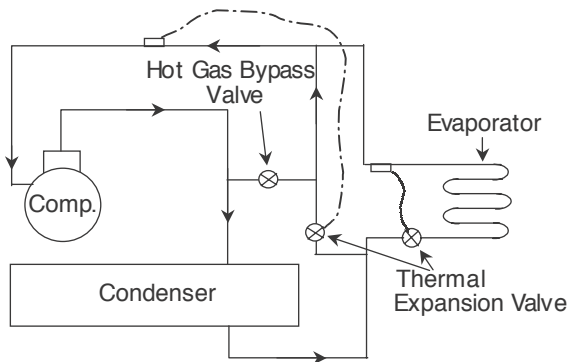
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FIG. 6A – HOT GAS BYPASS ARRANGEMENTS



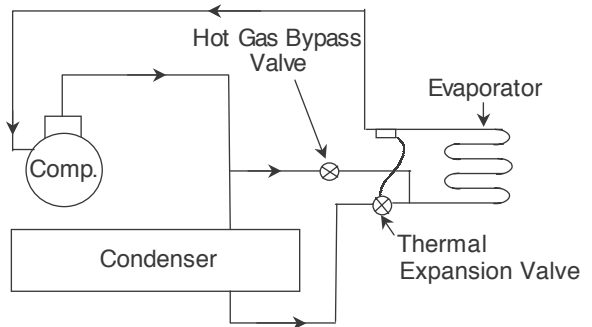
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FIG. 6B – HOT GAS BYPASS ARRANGEMENTS



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FIG. 6C – HOT GAS BYPASS ARRANGEMENTS



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FIG. 6D – HOT GAS BYPASS ARRANGEMENTS

7.2 DISCHARGE PIPING

Figure 7 also shows the piping arrangement for the discharge piping. The arrangement is such that liquid refrigerant and oil are prevented from draining back to the heads of idle compressors. A check valve is recommended in the discharge line to prevent refrigerant and oil from migrating to the compressor heads. The piping should be routed to a lower elevation so that a trap is formed for the drainback. An oil separator, if used, may serve as the trap.

7.3 CRANKCASE EQUALIZERS

When two or more compressors are connected, the crankcases should be equalized. The compressors should be placed on foundations and all equalizer tapping locations must be maintained level. An oil equalization line should connect all crankcases to maintain uniform oil levels (refer to Figure 7).

In order to allow the oil equalizer to perform satisfactorily, a gas equalizer should be installed above the oil level (refer Figure 7). It should be piped such that oil or liquid refrigerant will not be trapped.

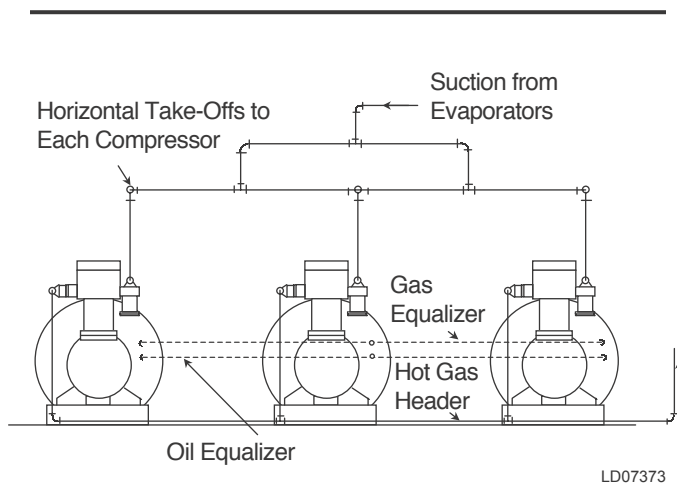


FIG. 7 – CHILLERS WITH REMOTELY INSTALLED DX COOLERS

8.0 CHILLERS WITH REMOTELY INSTALLED DX COOLERS

Outdoor chillers with remote DX water cooler applications have grown in popularity. It is prudent practice

to design these systems so that the remote, indoor DX cooler is as close to the outdoor section as possible. This assures optimum performance, reduces piping pressure penalties, and promotes reliability.

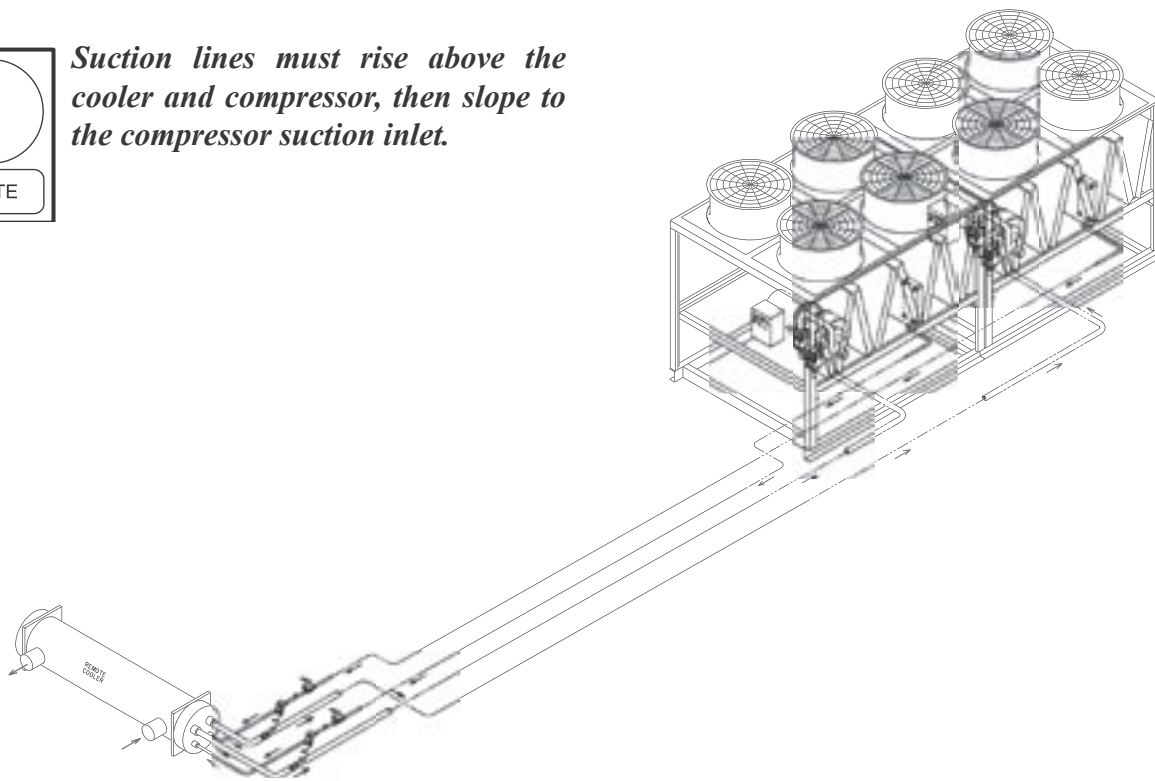
To assure these, the following recommendations should be followed:

- It is suggested that the linear feet of piping be 200 feet or less.
- The total equivalent feet of piping (which includes tees, elbows, fittings, etc.) be 300 feet or less.
- The DX cooler should be no lower than 100 feet below the outdoor section.
- The DX cooler should be no more than 12 to 15 feet above the outdoor section. [The combined friction and static refrigerant liquid column penalty should be no more than 25 psi.]

The design must be based on refrigerant lines (suction and liquid) which will meet full load and provide for proper oil return at the minimum system load condition. These should follow the practices contained in this catalog. See Figure 8A, which illustrates the DX cooler at the same level as the outdoor section. Figure B. illustrates the DX cooler below the outdoor section. Figure 8C illustrates the DX cooler being above the outdoor section. Regarding Figure 8C, the suction lines should loop up 1 to 2 feet above the DX cooler suction refrigerant connections before they proceed down vertically. This will allow any suction gas which condenses to drain back into cooler. A long radius elbow should be used at the bottom of the vertical drop to transition to the horizontal suction line (pitched approximately 1/4" per linear foot toward the compressor), which should be routed directly to the compressor suction valve connection, without any traps. Thus, any small amount of refrigerant which may condense in the suction line will drain into the compressor where it will be vaporized by the compressor crankcase heater, when the compressor is not operating. It is extremely important that the compressor heater is sized to handle the amount of condensed liquid that may occur in long suction lines and that the heater is allowed to remain energized during the compressor off cycle. Start-up logic should prevent the compressor from restarting if the crankcase temperature has not risen above the saturation temperature corresponding to the pressure of the compressor crankcase. Typically, the crankcase temperature is measured against the ambient and restart is prevented if the crankcase temperature is not at least 25°F above ambient.



Suction lines must rise above the cooler and compressor, then slope to the compressor suction inlet.



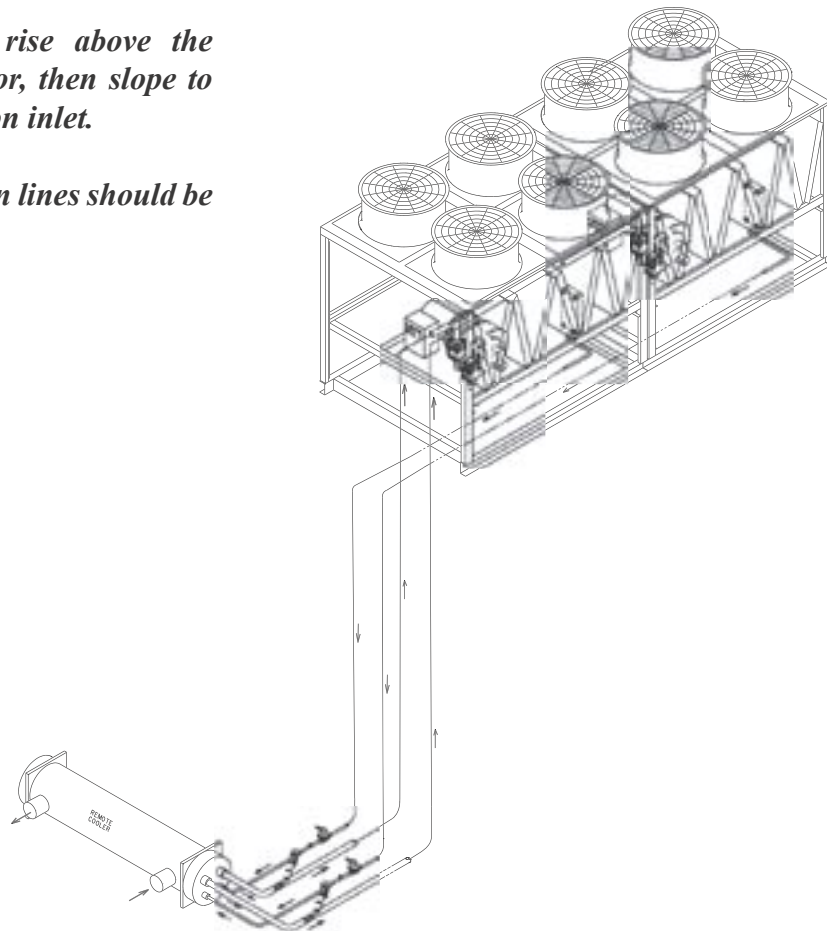
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FIG. 8A – REMOTE COOLER LOCATED ON THE SAME LEVEL AS THE CONDENSING UNIT



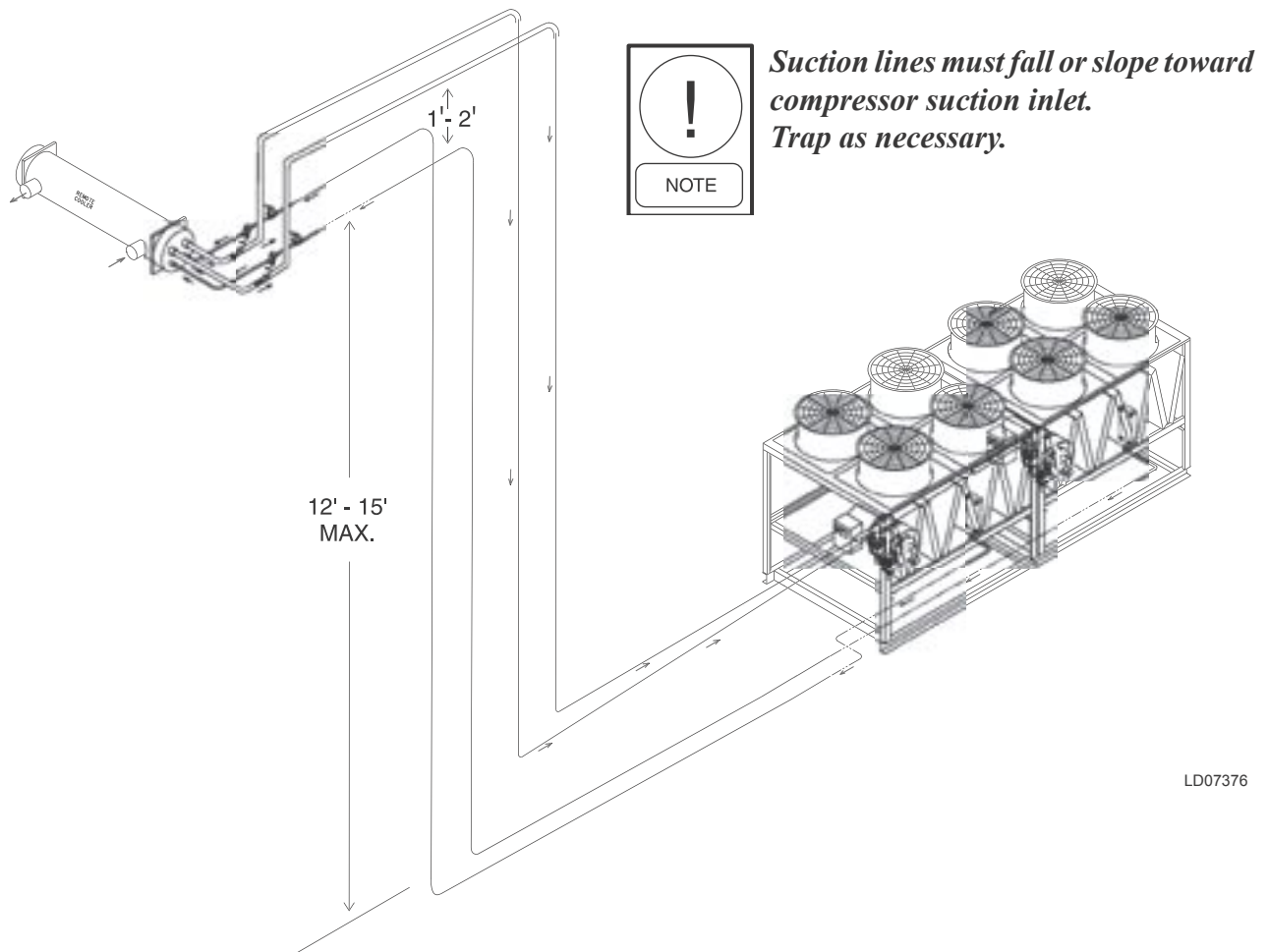
Suction lines must rise above the cooler and compressor, then slope to the compressor suction inlet.

The horizontal suction lines should be as short as possible.



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FIG. 8B – REMOTE COOLER LOCATED BELOW THE CONDENSING UNIT



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FIG. 8C – REMOTE COOLER LOCATED ABOVE THE CONDENSING UNIT

9.0 OTHER PIPING SUGGESTIONS

When vertical suction or discharge lines are greater than 25 feet, an extra oil trap is recommended for each 25 feet of vertical rise.

Suction and hot gas bypass lines must be insulated to help maintain optimum system performance.

Provisions should be made for contraction and expansion of 3/4" per 100 feet of copper piping.

Installing suction and discharge lines underground is not recommended. These have the potential to become liquid traps, which will damage compressors and reduce reliability.

Suction line accumulators may be required in certain instances where large volumes of liquid can periodically return to the compressors.

If there are reliability concerns regarding a proposed piping design, consult YORK Application Engineering personnel for advice.

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10.0 SYSTEM CHARGE REQUIREMENTS

Equipment literature should be reviewed to determine the necessary system charge requirements. There are additional requirements for the suction, discharge and liquid lines. Tables 12, 13, and 14 are included to help the designer determine these piping requirements, which need to be added to the equipment requirements to determine the total operating charge requirements. However, the charge will need to be trimmed during the commissioning process by a certified technician checking both the subcooling and superheat under design conditions.

11.0 EXPANSION VALVE SENSING BULB PLACEMENT

TXV sensing bulb mounting is of extreme importance. The purpose of the bulb is to sense the temperature of the refrigerant flowing through the suction line. Anything that hampers the bulb from doing this accurately will adversely affect the long-term reliability of the system.

The following points must be stressed when considering how and where to mount the TXV bulb (a more detailed explanation for each point mentioned is provided below the list).

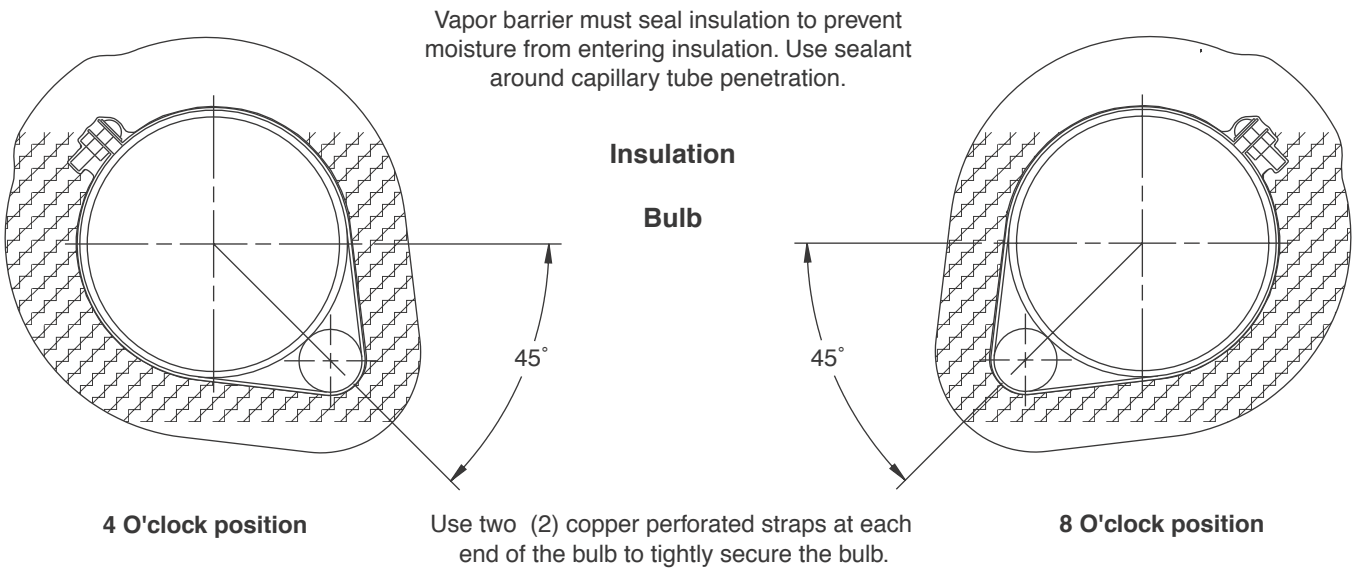
You can think of the steps as **Location, Location, Strap-it and Wrap-it!**

1. **Location:** The bulb should be mounted on a horizontal run of piping at the outlet of the evaporator.
2. **Location:** The bulb should be mounted radially on the suction line so that liquid oil returning from the evaporator does not influence the sensing bulb.
3. **Strap-it:** The bulb must be mounted to the suction line using the pair of metal straps normally supplied by the TXV manufacturer in order to provide good thermal contact with the suction line.
4. **Wrap-it:** The sensing bulb and the suction line must be thermally insulated with a vapor barrier¹ so that ambient conditions do not affect the bulb.

A general rule should be to mount the TXV sensing bulb on a horizontal run of suction line close to the outlet of the evaporator for which it is supplying liquid. In cases of multiple evaporators, especially stacked, multiple evaporators, consideration must be given to

the avoidance of bulb locations subject to liquid draining from higher evaporators. Avoid mounting the bulb next to large massive items such as valves or flanges that may act as a heat sink and influence the sensing bulb. Likewise, do not mount the bulb after a suction/liquid line heat exchanger since the suction gas is going to be artificially heated at that point.

The proper radial mounting location on the suction line will vary depending on the diameter of the suction line. Since the objective is to mount the bulb where it will quickly sense liquid refrigerant traveling down the suction line, but not be falsely affected by normal amounts of liquid oil returning from the evaporator, it stands to reason that the bulb should be mounted as close to the bottom of the line as possible, but not so close to the bottom to be affected by any normal oil return through the suction line. Smaller diameter suction lines will require the bulb to be mounted more closely to the top of the line, while large suction lines should have the bulb mounted nearer to the bottom of the suction line. On suction line sizes over 1.5 inches, YORK recommends the bulb to be mounted at 4 or 8 o'clock. Mounting the bulb on sweeping inside or outside bends may also result in false sensing of liquid oil, depending on the piping geometry and consequently those locations should be avoided. The oil will travel in the area of the



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FIG. 9 – THERMAL EXPANSION VALVE BULB MOUNTED RADIALLY ON THE SUCTION PIPING AT THE 4 AND 8 O’CLOCK POSITIONS

pipe that gravity and velocity dictate. On long stretches of horizontal runs, the oil will be in the bottom of the pipe. On bends in the pipe, the oil may be forced along the outside of the piping and it may even tend to swirl around the piping under certain circumstances.

Since the sensing bulb is round and the suction line is normally round, the contact area between the bulb and the pipe is at best one single line of contact for surface mounted sensing bulbs. With that in mind, the method used to position the bulb and clamp it in place is of utmost importance. Good heat transfer between the bulb and outer surface of the suction line is critical to TXV responding quickly to low and high suction gas superheat conditions. On surface-mounted bulbs (not inserted in a thermal well), the bulb must be mounted tightly using the manufacturer's supplied mounting hardware. Normally the mounting straps are made of copper. This is not because copper straps are soft and easy to bend. It is because copper straps provide an additional path for heat transfer to occur between the suction line and the bulb. The straps should be installed tightly but not so tight as to deform the bulb or piping. In addition, if desired, heat conductive compound could be added to further increase the heat transfer. Be careful to install the bulb on a round, straight uninterrupted, section of suction line. Do not straddle braze joints and other irregularities in the piping that would prevent that single line of contact over the length of the bulb-to-pipe mating surface.

Regardless of how well the bulb is mounted, adequate insulation must cover the bulb and surrounding suction line to prevent the ambient conditions from influencing the sensing bulb. Insulation must not only be thick enough to prevent the bulb from sensing ambient temperatures, but also include a vapor barrier sufficient to prevent any moisture from condensing on or around the bulb mounting area. If the vapor barrier is compromised, moisture may condense and form a pocket under the insulation that eventually will damage the insulation as well as cause erratic TXV operation. In low ambient applications, the moisture may freeze which will further hamper TXV operation and may cause eventual damage to the piping and/or insulation system. Since the capillary line attached to the bulb will penetrate the

vapor barrier, some method of sealing, such as a waterproof caulking compound, around the capillary tube should be used to keep the vapor barrier intact.

12.0 SUMMARY

Design recommendations have been provided for the design and sizing of connecting piping in DX systems.

This information is compiled for refrigerants 22, 134a and 407C, for all sizes of piping and range of capacities that encompasses all of YORK's current DX products and are consistent with the industry practices of ASHRAE. Following these recommendations will result in optimal system performance and reliability.

13.0 REFERENCES

ASHRAE Refrigeration and Fundamentals Handbooks, American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta, GA.

York International, 1960, Refrigerant-12 Piping, Form 215.05-TM1 (R-12).

York International, 1960, Refrigerant-22 Piping, Form 215.05-TM1 (R-22).

York International, Refrigerant Piping – Compressors, Form 215.05-TM11.

York International, Refrigerant Piping – Suction Lines, Form 215.05-TM09.

York International, Refrigerant Piping – Suction Lines, Form 215.05-TM12.

¹ A vapor barrier consists of a moisture proof material such as aluminum foil, plastic tape, etc. that will prevent water vapor from passing through the insulation material around the suction line. The vapor barrier must be installed on the outside (warm side) of the insulation material and must be sealed so that water vapor cannot go around it.

TABLE 1 – SUCTION LINE CAPACITIES IN TONS FOR REFRIGERANT 22

LINE SIZE	SATURATED SUCTION TEMPERATURE, °F																			
	-40				-20				0				20				40			
	t = 2 F p = 0.77	t = 1 F p = 0.39	t = 0.5 F p = 0.19		t = 2 F p = 1.13	t = 1 F p = 0.57	t = 0.5 F p = 0.29		t = 2 F p = 1.60	t = 1 F p = 0.81	t = 0.5 F p = 0.40		t = 2 F p = 2.18	t = 1 F p = 1.1	t = 0.5 F p = 0.55		t = 2 F p = 2.87	t = 1 F p = 1.45	t = 0.5 F p = 0.73	
COPPER																				
O.D.																				
1/2	0.10	0.07	0.05	0.17	0.12	0.08	0.27	0.18	0.12	0.40	0.27	0.19	0.58	0.40	0.27	0.58	0.40	0.27		
5/8	0.19	0.13	0.09	0.32	0.22	0.15	0.50	0.34	0.23	0.75	0.52	0.35	1.10	0.74	0.51	1.10	0.74	0.51		
3/4	0.33	0.23	0.15	0.55	0.38	0.26	0.86	0.59	0.40	1.30	0.88	0.60	1.80	1.30	0.87	1.80	1.30	0.87		
7/8	0.52	0.35	0.24	0.85	0.58	0.40	1.30	0.91	0.62	2.00	1.40	0.93	2.90	2.00	1.30	2.90	2.00	1.30		
1-1/8	1.10	0.72	0.49	1.70	1.20	0.80	2.70	1.90	1.30	4.00	2.80	1.90	5.80	4.00	2.70	5.80	4.00	2.70		
1-3/8	1.80	1.30	0.90	3.00	2.10	1.40	4.70	3.20	2.20	7.00	4.80	3.30	10.00	6.90	4.80	10.00	6.90	4.80		
1-5/8	2.90	2.00	1.40	4.80	3.30	2.30	7.40	5.10	3.50	11.10	7.60	5.30	15.90	11.00	7.60	15.90	11.00	7.60		
2-1/8	6.10	4.20	2.90	10.00	6.90	4.70	15.40	10.70	7.30	22.90	15.90	10.90	32.80	22.80	15.70	32.80	22.80	15.70		
2-5/8	10.80	7.50	5.10	17.70	12.20	8.40	27.30	18.90	13.00	40.50	28.10	19.40	58.00	40.30	27.80	58.00	40.30	27.80		
3-1/8	17.30	12.00	8.20	28.20	19.50	13.40	43.60	30.20	20.80	64.60	44.80	31.00	92.30	64.20	44.40	92.30	64.20	44.40		
3-5/8	25.80	17.80	12.30	42.00	29.10	20.00	64.80	44.90	31.00	96.00	66.70	46.10	137.00	95.40	66.10	137.00	95.40	66.10		
4-1/8	36.50	25.20	17.40	59.30	41.10	28.30	91.40	63.40	43.80	135.20	94.00	65.10	193.00	134.40	93.20	193.00	134.40	93.20		
5-1/8	65.30	45.20	31.20	106.10	73.60	50.80	163.30	113.50	78.50	241.50	168.10	116.50	344.40	240.20	166.70	344.40	240.20	166.70		
6-1/8	105.20	72.93	50.30	170.70	118.50	81.90	262.50	182.60	126.40	387.90	270.30	187.50	552.70	385.80	268.10	552.70	385.80	268.10		
STEEL																				
1/2	0.24	0.16	0.11	0.38	0.27	0.19	0.58	0.41	0.29	0.84	0.60	0.42	1.20	0.84	0.59	1.20	0.84	0.59		
3/4	0.50	0.35	0.24	0.80	0.56	0.39	1.20	0.86	0.60	1.80	1.30	0.88	2.50	1.80	1.20	2.50	1.80	1.20		
1	0.95	0.66	0.46	1.50	1.10	0.75	2.30	1.60	1.10	3.40	2.40	1.70	4.80	3.40	2.40	4.80	3.40	2.40		
1-1/4	2.00	1.40	1.00	3.10	2.20	1.50	4.80	3.40	2.40	6.90	4.90	3.40	9.80	6.90	4.90	9.80	6.90	4.90		
1-1/2	2.90	2.10	1.40	4.70	3.30	2.30	7.10	5.00	3.60	10.40	7.40	5.20	14.70	10.40	7.30	14.70	10.40	7.30		
2	5.70	4.00	2.80	9.10	6.40	4.50	13.80	9.70	6.90	20.10	14.20	10.00	28.30	20.00	14.10	28.30	20.00	14.10		
2-1/2	9.10	6.40	4.50	14.50	10.20	7.20	22.00	15.50	10.90	32.00	22.60	16.00	45.10	31.90	22.50	45.10	31.90	22.50		
3	16.10	11.30	8.00	25.60	18.10	12.70	38.80	27.50	19.40	56.50	40.00	28.20	79.60	56.40	39.80	79.60	56.40	39.80		
3-1/2	23.50	16.60	11.70	37.40	26.50	18.60	56.80	40.20	28.30	82.70	58.50	41.30	116.50	82.50	58.30	116.50	82.50	58.30		
4	32.70	23.10	16.30	52.10	36.90	26.00	79.00	55.90	39.50	115.00	81.40	57.50	162.10	114.80	81.10	162.10	114.80	81.10		
5	59.20	41.80	29.40	94.10	66.60	47.00	142.60	101.00	71.30	207.60	147.00	103.80	292.60	207.20	146.40	292.60	207.20	146.40		
6	95.70	67.70	47.60	152.00	107.60	76.00	230.40	163.20	115.30	335.40	237.60	167.80	472.50	334.70	236.60	472.50	334.70	236.60		

NOTES:

Capacities are in tons of refrigeration.
 □ p = pressure drop due to line friction, psi per 100 feet equivalent length.
 □ t = change in saturation temperature corresponding to pressure drop, °F per 100 feet.
 All steel pipe sizes are nominal and are for schedule 40.
 See notes at the bottom of Table 2.

TABLE 2 – DISCHARGE AND LIQUID LINE CAPACITIES IN TONS FOR REFRIGERANT 22

LINE SIZE		DISCHARGE LINES ($\Delta t = 1^\circ\text{F}$, $\Delta p = 3.03 \text{ PSI}$)					LINE SIZE		LIQUID LINES	
TYPE L COPPER, O.D.		SATURATED SUCTION TEMPERATURE, $^\circ\text{F}$					TYPE L COPPER, O.D.		VEL. = 100 FPM	$\Delta t = 1^\circ\text{F}$ $\Delta p = 3.03$
		-40	-20	0	20	40				
1/2		0.75	0.78	0.8	0.83	0.85	1/2		2.4	3.7
5/8		1.4	1.5	1.5	1.6	1.6	5/8		3.8	7.0
3/4		2.4	2.5	2.6	2.6	2.7	3/4		5.7	12.0
7/8		3.7	3.8	4.0	4.1	4.2	7/8		8.0	18.6
1-1/8		7.5	7.8	8.0	8.3	8.5	1-1/8		13.6	37.8
1-3/8		13.1	13.5	14.0	14.4	14.8	1-3/8		20.7	66.1
1-5/8		20.6	21.4	22.1	22.8	23.4	1-5/8		29.3	104.7
2-1/8		42.7	44.2	45.7	47.1	48.4	2-1/8		51.0	217.5
2-5/8		75.3	78.0	80.6	83.1	85.3	2-5/8		78.7	385.0
3-1/8		119.9	124.3	128.4	132.3	135.9	3-1/8		112.3	615.0
3-5/8		177.9	184.4	190.6	196.3	201.6	3-5/8		151.8	914.6
4-1/8		250.6	259.7	268.4	276.5	283.9	4-1/8		197.4	1291.0
5-1/8		447.0	463.3	478.7	493.2	506.4	5-1/8		307.6	–
6-1/8		717.1	743.2	768.0	791.2	812.5	6-1/8		442.2	–
STEEL										
IPS	SCH						IPS	SCH		
1/2	40	1.5	1.6	1.7	1.7	1.8	1/2	80	3.9	5.8
3/4	40	3.3	3.4	3.5	3.6	3.7	3/4	80	7.1	13.1
1	40	6.1	6.4	6.6	6.8	7.0	1	80	11.9	25.8
1-1/4	40	12.7	13.1	13.6	14.0	14.3	1-1/4	80	21.1	55.4
1-1/2	40	19.0	19.7	20.3	21.0	21.5	1-1/2	80	29.1	84.5
2	40	36.6	37.9	39.2	40.4	41.5	2	40	55.3	196.5
2-1/2	40	58.3	60.4	62.5	64.3	66.1	2-1/2	40	78.9	313.4
3	40	103.0	106.7	110.3	113.6	116.7	3	40	121.8	554.0
4	40	209.6	217.3	224.5	231.3	237.5	4	40	209.8	1129.0
5	40	378.3	392.1	405.2	417.4	428.6	5	40	329.7	2039.0
6	40	611.1	633.3	654.4	674.1	692.3	6	40	476.2	3294.0

Capacities are in tons of refrigeration.

Δp = Pressure drop due to line friction, psi per 100 feet equivalent length.

Δt = Change in saturation temperature corresponding to pressure drop, $^\circ\text{F}$ per 100 feet.

Line capacity for other saturation temperatures Δt and equivalent lengths L.

$$\text{Line capacity} = \text{Table capacity} \left(\frac{\text{Table } L_e}{\text{Actual } L_e} \right) \times \left(\frac{\text{Actual } \Delta t}{\text{Table } \Delta t} \right)^{0.55}$$

Saturation temperature Δt for other capacities and equivalent lengths L_e .

$$\Delta t = \text{Table } \Delta t \left(\frac{\text{Actual } L_e}{\text{Table } L_e} \right) \left(\frac{\text{Actual Capacity}}{\text{Table Capacity}} \right)^{1.8}$$

The refrigerant cycle for determining capacity is based on saturated gas leaving the evaporator and no subcooling in the condenser. Discharge superheat is 105°F . The saturated suction temperature is 40°F for liquid line sizing.

Multiply table capacities by the following factors for condensing temperatures other than 105°F .

CONDENSING TEMPERATURE, $^\circ\text{F}$	SUCTION LINE	DISCHARGE LINE
80	1.12	0.82
90	1.07	0.89
100	1.03	0.96
110	0.97	1.03
120	0.92	1.10
130	0.87	1.16
140	0.82	1.22

TABLE 3 – SUCTION LINE CAPACITIES IN TONS FOR REFRIGERANT 134a

LINE SIZE	SATURATED SUCTION TEMPERATURE, °F																			
	-40				-20				0				20				40			
	t = 2 F p = 0.98	t = 1 F p = 0.49	t = 0.5 F p = 0.25	t = 2 F p = 1.17	t = 1 F p = 0.59	t = 0.5 F p = 0.30	t = 2 F p = 1.39	t = 1 F p = 0.70	t = 0.5 F p = 0.35	t = 2 F p = 1.63	t = 1 F p = 0.82	t = 0.5 F p = 0.41	t = 2 F p = 1.90	t = 1 F p = 0.96	t = 0.5 F p = 0.48					
1/2	0.14	0.10	0.07	0.18	0.12	0.08	0.23	0.16	0.11	0.28	0.19	0.13	0.35	0.24	0.16					
5/8	0.27	0.18	0.12	0.34	0.23	0.16	0.43	0.29	0.20	0.53	0.37	0.25	0.66	0.45	0.31					
3/4	0.46	0.31	0.21	0.58	0.40	0.27	0.73	0.50	0.34	0.91	0.62	0.43	1.1	0.77	0.53					
7/8	0.71	0.48	0.33	0.90	0.62	0.42	1.1	0.78	0.53	1.4	1.0	0.66	1.7	1.2	0.82					
1-1/8	1.4	1.0	0.67	1.8	1.3	0.86	2.3	1.6	1.1	2.9	2.0	1.3	3.5	2.4	1.7					
1-3/8	2.5	1.7	1.2	3.2	2.2	1.5	4.0	2.8	1.9	5.0	3.4	2.4	6.1	4.2	2.9					
1-5/8	4.0	2.7	1.9	5.1	3.5	2.4	6.4	4.4	3.0	7.9	5.4	3.7	9.7	6.7	4.6					
2-1/8	8.3	5.7	3.9	10.5	7.3	5.0	13.2	9.1	6.3	16.4	11.3	7.8	20.1	13.9	9.6					
2-5/8	14.7	10.1	7.0	18.6	12.9	8.8	23.3	16.1	11.1	28.9	20.0	13.8	35.5	24.6	17.0					
3-1/8	23.4	16.2	11.2	29.7	20.6	14.2	37.2	25.8	17.8	46.1	32.0	22.1	56.6	39.3	27.1					
3-5/8	34.9	24.1	16.6	44.2	30.6	21.1	55.3	38.4	26.5	68.6	47.6	32.9	84.1	58.4	40.4					
4-1/8	49.2	34.1	23.5	62.4	43.2	29.8	78.1	54.2	37.4	96.7	67.1	46.4	118.5	82.4	57.0					
5-1/8	88.2	61.1	42.2	111.6	77.4	53.5	139.6	97.0	67.1	172.7	120.1	83.1	211.6	147.3	102.1					
6-1/8	141.8	98.5	68.0	179.4	124.7	86.2	224.4	156.0	108.0	277.5	193.2	133.9	339.9	236.8	164.2					

STEEL

IPS SCH	1/2	3/4	1	1-1/4	1-1/2	2	2-1/2	3	3-1/2	4	4	5	6
80	0.22	0.50	1.0	2.6	3.9	7.6	12.1	21.3	31.2	43.5	78.5	126.9	208.4
80	0.16	0.35	0.70	1.8	2.8	5.3	8.5	15.1	22.1	30.7	55.5	89.8	147.2
80	0.25	0.55	1.1	3.3	5.1	10.2	16.3	28.8	42.2	58.7	100.9	164.2	271.0
40	0.28	0.63	1.2	3.3	4.9	9.5	15.1	26.7	39.1	54.5	98.4	159.0	268.4
40	0.20	0.45	0.88	2.3	3.5	6.7	10.7	18.9	27.7	38.5	69.6	112.6	191.0
40	0.14	0.31	0.61	1.6	2.4	4.7	7.5	13.3	19.5	27.2	49.1	79.4	133.9
40	0.17	0.39	0.76	2.0	3.0	5.9	9.3	16.5	24.2	33.7	60.9	98.5	164.2
40	0.43	0.97	1.90	5.0	7.5	14.5	23.1	40.7	59.6	82.9	149.7	241.9	408.4
40	0.30	0.68	1.3	3.5	5.3	10.2	16.3	28.8	42.2	58.7	106.0	171.3	294.2
40	0.21	0.48	0.94	2.5	3.7	7.2	11.5	20.3	29.8	41.4	74.9	121.0	208.4
40	0.52	1.2	2.3	6.1	9.1	17.6	28.0	49.6	72.5	100.9	182.1	294.2	508.4
40	0.37	0.83	1.6	4.3	6.4	12.4	19.8	35.1	51.3	71.4	129.0	208.4	368.4
40	0.26	0.59	1.1	3.0	4.5	8.8	14.0	24.7	36.2	50.4	91.1	147.2	258.4

NOTES:

Capacities are in tons of refrigeration.
 □ p = pressure drop due to line friction, psi per 100 feet equivalent length.
 □ t = change in saturation temperature corresponding to pressure drop, °F per 100 feet.
 All steel pipe sizes are nominal and are for schedule 40.
 See notes at the bottom of Table 4.

TABLE 4 – DISCHARGE AND LIQUID LINE CAPACITIES IN TONS FOR REFRIGERANT 134a

LINE SIZE		DISCHARGE LINES ($\Delta t = 1^\circ\text{F}$, $\Delta p = 2.2 \text{ PSI}$)			LINE SIZE		LIQUID LINES	
TYPE L		SATURATED SUCTION TEMPERATURE, $^\circ\text{F}$			TYPE L		VEL. = 100	$\Delta t = 1^\circ\text{F}$
COPPER, O.D.		0	20	40	COPPER, O.D.		FPM	$\Delta p = 2.2$
1/2		0.54	0.57	0.59	1/2		2.4	3.7
5/8		1.0	1.1	1.1	5/8		3.8	7.0
3/4		1.7	1.8	1.9	3/4		5.7	12.0
7/8		2.7	2.8	2.9	7/8		8.0	18.6
1-1/8		5.4	5.7	5.9	1-1/8		13.6	37.8
1-3/8		9.4	9.9	10.4	1-3/8		20.7	66.1
1-5/8		14.9	15.6	6.4	1-5/8		29.3	104.7
2-1/8		30.8	32.4	33.9	2-1/8		51.0	217.5
2-5/8		54.4	57.1	59.8	2-5/8		78.7	385.0
3-1/8		86.7	91.0	95.2	3-1/8		112.3	615.0
3-5/8		128.7	135.1	141.4	3-5/8		151.8	914.6
4-1/8		181.3	190.4	199.1	4-1/8		197.4	–
5-1/8		323.5	339.7	355.4	5-1/8		307.6	–
6-1/8		519.2	545.2	570.3	6-1/8		442.2	–
STEEL STEEL								
IPS	SCH				IPS	SCH		
1/2	80	0.80	0.83	0.87	1/2	80	3.2	5.6
3/4	80	1.8	1.9	2.0	3/4	80	7.1	16.0
1	80	3.5	3.7	3.9	1	80	11.9	31.5
1-1/4	40	9.2	9.7	10.1	1-1/4	80	21.1	67.9
1-1/2	40	13.8	14.5	5.2	1-1/2	80	29.1	103.7
2	40	26.6	28.0	29.2	2	40	55.3	242.0
2-1/2	40	42.4	44.6	46.6	2-1/2	40	78.9	386.7
3	40	75.0	78.7	82.3	3	40	121.8	684.8
4	40	152.6	160.2	167.6	4	40	209.8	–
5	40	275.4	289.2	02.5	5	40	329.7	–
6	40	444.8	467.1	488.5	6	40	476.2	–

Capacities are in tons of refrigeration.

Δp = Pressure drop due to line friction, psi per 100 feet equivalent length.

Δt = Change in saturation temperature corresponding to pressure drop, $^\circ\text{F}$ per 100 feet.

Line capacity for other saturation temperatures Δt and equivalent lengths L_e .

$$\text{Line capacity} = \text{Table capacity} \left(\frac{\text{Table } L_e}{\text{Actual } L_e} \right) \times \left(\frac{\text{Actual } \Delta t}{\text{Table } \Delta t} \right)^{0.55}$$

Saturation temperature Δt for other capacities and equivalent lengths L_e .

$$\Delta t = \text{Table } \Delta t \left(\frac{\text{Actual } L_e}{\text{Table } L_e} \right) \left(\frac{\text{Actual Capacity}}{\text{Table Capacity}} \right)^{1.8}$$

The refrigerant cycle for determining capacity is based on saturated gas leaving the evaporator and no subcooling in the condenser. Discharge superheat is 105°F . The saturated suction temperature is 40°F for liquid line sizing.

Multiply table capacities by the following factors for condensing temperatures other than 105°F .

CONDENSING TEMPERATURE, $^\circ\text{F}$	SUCTION LINE	DISCHARGE LINE
80	1.14	0.81
90	1.09	0.89
100	1.03	0.96
110	0.97	1.03
120	0.91	1.10
130	0.84	1.16
140	0.78	1.20

TABLE 6 – DISCHARGE AND LIQUID LINE CAPACITIES IN TONS FOR REFRIGERANT 407C

LINE SIZE		DISCHARGE LINES ($\Delta t = 1^\circ\text{F}$, $\Delta p = 3.28 \text{ PSI}$)					LINE SIZE		LIQUID LINES	
TYPE L		SATURATED SUCTION TEMPERATURE, $^\circ\text{F}$					TYPE L		VEL. = 100	$\Delta t = 1^\circ\text{F}$
COPPER, O.D.		-40	-20	0	20	40	COPPER, O.D.		FPM	$\Delta p = 3.43$
1/2		0.71	0.75	0.78	0.82	0.86	1/2		2.2	4.0
5/8		1.3	1.4	1.5	1.5	1.6	5/8		3.6	7.5
3/4		2.3	2.4	2.5	2.6	2.7	3/4		5.4	12.9
7/8		3.5	3.7	3.9	4.0	4.2	7/8		7.5	19.9
1-1/8		7.0	7.4	7.8	8.2	8.5	1-1/8		12.7	40.3
1-3/8		12.3	12.9	13.6	14.3	14.9	1-3/8		19.4	70.3
1-5/8		19.3	20.4	21.5	22.5	23.5	1-5/8		27.5	111.2
2-1/8		40.0	42.2	44.4	46.5	48.6	2-1/8		47.8	230.5
2-5/8		70.5	74.5	78.3	82.1	85.6	2-5/8		73.7	407.3
3-1/8		112.3	118.6	124.8	130.7	136.4	3-1/8		105.2	649.6
3-5/8		166.6	176.0	185.1	193.9	202.3	3-5/8		142.3	965.0
4-1/8		234.7	247.8	260.7	273.1	284.9	4-1/8		185.0	1360.0
5-1/8		418.5	441.9	464.9	487.0	508.1	5-1/8		288.3	–
6-1/8		671.3	708.9	745.7	781.2	815.0	6-1/8		414.4	–

STEEL												
IPS							IPS		SCH			
1/2	40	1.4	1.5	1.6	1.7	1.8	1/2	80	4.7	8.5		
3/4	40	3.0	3.2	3.4	3.5	3.7	3/4	80	8.2	17.9		
1	40	5.7	6.1	6.4	6.7	7.0	1	80	13.4	34.0		
1-1/4	40	11.8	12.5	3.1	13.8	14.4	1-1/4	80	23.1	70.0		
1-1/2	40	17.7	18.7	19.7	20.6	21.5	1-1/2	80	31.4	105.0		
2	40	34.2	36.1	38.0	39.8	41.5	2	40	51.8	202.4		
2-1/2	40	54.5	57.5	60.5	63.4	66.1	2-1/2	40	73.9	322.7		
3	40	96.2	101.6	106.9	112.0	116.8	3	40	114.2	570.2		
4	40	195.8	206.8	217.5	227.9	237.8	4	40	196.6	1161.0		
5	40	353.4	373.2	392.5	411.3	429.0	5	40	309.0	–		
6	40	570.8	602.7	634.0	664.2	692.9	6	40	446.2	–		

Capacities are in tons of refrigeration.

Δp = Pressure drop due to line friction, psi per 100 feet equivalent length.

Δt = Change in saturation temperature corresponding to pressure drop, $^\circ\text{F}$ per 100 feet.

Line capacity for other saturation temperatures Δt and equivalent lengths L.

$$\text{Line capacity} = \text{Table capacity} \left(\frac{\text{Table } L_e}{\text{Actual } L_e} \right) \times \left(\frac{\text{Actual } \Delta t}{\text{Table } \Delta t} \right)^{0.55}$$

Saturation temperature Δt for other capacities and equivalent lengths L_e .

$$\Delta t = \text{Table } \Delta t \left(\frac{\text{Actual } L_e}{\text{Table } L_e} \right) \left(\frac{\text{Actual Capacity}}{\text{Table Capacity}} \right)^{1.8}$$

The refrigerant cycle for determining capacity is based on saturated gas leaving the evaporator and no subcooling in the condenser. Discharge superheat is 105°F . The saturated suction temperature is 40°F for liquid line sizing.

Multiply table capacities by the following factors for condensing temperatures other than 105°F .

CONDENSING TEMPERATURE, $^\circ\text{F}$	SUCTION LINE	DISCHARGE LINE
80	1.16	0.81
90	1.09	0.89
100	1.03	0.96
110	0.97	1.03
120	0.90	1.10
130	0.83	1.16
140	0.76	1.19

TABLE 7 – MINIMUM REFRIGERATION CAPACITY IN TONS FOR OIL ENTRAINMENT UP SUCTION RISERS (TYPE L COPPER TUBING)

REFRIGERANT	SATURATION SUCTION TEMP., °F	SUCTION GAS TEMP., °F	PIPE O.D., IN.												
			1/2	5/8	3/4	7/8	1-1/8	1-3/8	1-5/8	2-1/8	2-5/8	3-1/8	3-5/8	4-1/8	
			AREA, IN ²												
22	-40	-30	0.07	0.12	0.20	0.30	0.484	0.825	1.256	1.780	3.094	4.770	6.812	9.213	11.97
		-10	0.07	0.12	0.20	0.30	0.30	0.58	0.97	1.51	3.00	5.16	8.06	11.75	16.30
		10	0.07	0.12	0.20	0.30	0.30	0.58	0.98	1.52	3.02	5.19	8.11	11.82	16.40
	-20	-10	0.09	0.16	0.26	0.39	0.39	0.76	1.28	1.97	3.94	6.77	10.57	15.41	21.38
		10	0.09	0.15	0.25	0.38	0.38	0.74	1.26	1.95	3.89	6.68	10.42	15.20	21.09
		30	0.09	0.15	0.26	0.39	0.39	0.75	1.27	1.96	3.92	6.73	10.51	15.33	21.26
	0	10	0.11	0.20	0.33	0.49	0.49	0.96	1.63	2.52	5.03	8.63	13.48	19.65	27.26
		30	0.11	0.19	0.32	0.49	0.49	0.94	1.60	2.47	4.93	8.47	13.22	19.29	26.75
		50	0.11	0.20	0.32	0.49	0.49	0.95	1.61	2.49	4.97	8.54	13.33	19.44	26.96
	20	30	0.14	0.25	0.41	0.62	0.62	1.20	2.03	3.15	6.28	10.78	16.84	24.56	34.06
		50	0.14	0.24	0.40	0.61	0.61	1.19	2.01	3.11	6.21	10.66	16.65	24.28	33.68
		70	0.14	0.24	0.40	0.61	0.61	1.19	2.01	3.11	6.20	10.65	16.63	24.25	33.64
40	50	0.17	0.30	0.50	0.76	0.76	1.48	2.49	3.86	7.70	13.22	20.64	30.11	41.76	
	70	0.17	0.30	0.49	0.74	0.74	1.45	2.45	3.78	7.55	12.96	20.24	29.52	40.94	
	90	0.17	0.30	0.49	0.74	0.74	1.45	2.45	3.78	7.55	12.97	20.26	29.54	40.98	
134a	0	10	0.09	0.16	0.26	0.39	0.39	0.76	1.29	2.00	3.99	6.86	10.71	15.61	21.66
		30	0.09	0.15	0.25	0.38	0.38	0.75	1.26	1.95	3.90	6.70	10.46	15.26	21.16
		50	0.09	0.16	0.26	0.39	0.39	0.76	1.28	1.98	3.95	6.79	10.60	15.46	21.45
	10	20	0.10	0.17	0.28	0.43	0.43	0.84	1.41	2.18	4.36	7.49	11.69	17.05	23.65
		40	0.10	0.17	0.29	0.43	0.43	0.85	1.43	2.21	4.41	7.58	11.84	17.26	23.95
		60	0.10	0.18	0.29	0.44	0.44	0.86	1.45	2.24	4.47	7.68	11.99	17.49	24.26
	20	30	0.11	0.19	0.32	0.49	0.49	0.95	1.60	2.48	4.94	8.49	13.25	19.32	26.80
		50	0.11	0.20	0.33	0.49	0.49	0.96	1.62	2.51	5.01	8.61	13.44	19.60	27.19
		70	0.11	0.20	0.33	0.50	0.50	0.97	1.64	2.54	5.06	8.70	13.58	19.81	27.48
	30	40	0.12	0.22	0.37	0.55	0.55	1.08	1.82	2.82	5.63	9.67	15.10	22.02	30.55
		60	0.12	0.22	0.37	0.55	0.55	1.08	1.82	2.82	5.62	9.66	15.08	21.99	30.51
		80	0.13	0.23	0.37	0.56	0.56	1.10	1.86	2.87	5.73	9.85	15.38	22.43	31.11
40	50	0.14	0.25	0.41	0.62	0.62	1.20	2.03	3.14	6.26	10.75	16.79	24.49	33.97	
	70	0.14	0.25	0.41	0.62	0.62	1.20	2.03	3.14	6.28	10.78	16.83	24.55	34.05	
	90	0.14	0.25	0.42	0.63	0.63	1.23	2.08	3.22	6.43	11.04	17.24	25.15	34.88	

TABLE 7 – MINIMUM REFRIGERATION CAPACITY IN TONS FOR OIL ENTRAINMENT UP SUCTION RISERS (TYPE L COPPER TUBING) (CONTINUED)

REFRIGERANT	SATURATION SUCTION TEMP., °F	SUCTION GAS TEMP., °F	PIPE O.D., IN.												
			1/2	5/8	3/4	7/8	1-1/8	1-3/8	1-5/8	2-1/8	2-5/8	3-1/8	3-5/8	4-1/8	
			AREA, IN ²												
407C	-40	-30	0.06	.10	0.17	0.25	0.49	0.825	1.256	1.780	3.094	4.770	6.812	9.213	11.97
		-10	0.06	0.10	0.17	0.26	0.51	0.86	1.33	2.65	4.55	7.10	10.35	14.36	14.36
		10	0.06	0.11	0.18	0.27	0.52	0.88	1.37	2.73	4.69	7.32	10.67	14.81	14.81
	-20	-10	0.08	0.14	0.23	0.34	0.67	1.13	1.75	3.49	6.00	9.36	13.65	18.93	18.93
		10	0.08	0.14	0.23	0.35	0.68	1.15	1.78	3.56	6.11	9.54	13.92	19.30	19.30
		30	0.08	0.14	0.24	0.36	0.70	1.18	1.83	3.65	6.28	.80	14.29	19.82	19.82
	0	10	0.10	0.18	0.29	0.45	0.87	1.47	2.27	4.53	7.78	12.14	17.71	24.56	24.56
		0	0.10	0.18	0.30	0.46	0.89	1.51	2.33	4.65	7.98	12.46	18.17	25.20	25.20
		50	0.10	0.19	0.31	0.47	0.91	1.54	2.38	4.76	8.17	12.76	18.61	25.82	25.82
	20	30	0.13	0.23	0.38	0.58	1.12	1.90	2.94	5.86	10.06	15.71	22.92	31.79	31.79
		50	0.13	0.23	0.39	0.59	1.14	1.93	2.98	5.95	10.22	15.95	23.27	32.28	32.28
		70	0.13	0.24	0.39	0.60	1.16	1.96	3.03	6.05	10.40	16.24	23.68	32.85	32.85
40	50	0.16	0.29	0.47	0.71	1.39	2.35	3.64	7.26	2.47	19.46	28.39	39.38	39.38	
	70	0.16	0.29	0.48	0.73	1.41	2.39	3.70	7.38	12.68	19.79	28.87	40.05	40.05	
	90	0.17	0.30	0.49	0.74	1.44	2.44	3.77	7.52	12.92	20.17	29.42	40.82	40.82	

Refrigeration capacity in tons is based on 90°F liquid temperature and superheat as indicated by the temperature in the table. The saturated condensing and suction conditions are referenced to the dewpoint for R-407C. For other liquid line temperatures, use correction factors to the capacity given in the table below.

REFRIGERANT	LIQUID TEMPERATURE, °F													
	50	60	70	80	100	110	120	130	140					
22	1.16	1.12	1.08	1.04	0.96	0.91	0.87	0.82	0.78					
134a	1.19	1.15	1.10	1.05	0.95	0.90	0.84	0.79	0.73					
407C	1.21	1.16	1.11	1.05	0.94	0.89	0.83	0.77	0.70					

TABLE 8 – MINIMUM REFRIGERATION CAPACITY IN TONS FOR OIL ENTRAINMENT UP HOT GAS RISERS (TYPE L COPPER TUBING)

REFRIGERANT	SATURATION SUCTION TEMP., °F	SUCTION GAS TEMP., °F	PIPE O.D., IN.											
			1/2	5/8	3/4	7/8	1-1/8	1-3/8	1-5/8	2-1/8	2-5/8	3-1/8	3-5/8	4-1/8
			AREA, IN ²											
22	80	110	1.146	0.233	0.348	0.484	0.825	1.256	1.780	3.094	4.770	6.812	9.213	11.97
		140	0.23	0.42	0.69	1.04	2.02	3.42	5.29	10.56	18.15	28.33	41.32	57.31
		170	0.22	0.39	0.65	0.98	1.91	3.23	5.00	9.98	17.15	26.77	39.04	54.15
	90	120	0.21	0.38	0.62	0.94	1.84	3.11	4.80	9.58	16.46	25.70	37.49	52.00
		150	0.24	0.43	0.71	1.07	2.09	3.53	5.46	10.90	18.72	29.22	42.62	59.12
		180	0.23	0.40	0.67	1.01	1.96	3.32	5.13	10.25	17.60	27.48	40.08	55.60
	100	130	0.22	0.39	0.64	0.96	1.88	3.17	4.90	9.78	16.81	26.24	38.27	53.09
		160	0.25	0.44	0.73	1.11	2.16	3.65	5.64	11.26	19.34	30.20	44.05	61.10
		190	0.23	0.41	0.68	1.03	2.01	3.41	5.27	10.51	18.05	28.18	41.11	57.02
	110	140	0.22	0.40	0.65	0.99	1.92	3.25	5.02	10.02	17.21	26.86	39.18	54.34
		170	0.25	0.45	0.75	1.13	2.21	3.73	5.77	11.52	19.79	30.89	45.06	62.50
		200	0.24	0.42	0.70	1.06	2.06	3.48	5.38	10.73	18.44	28.79	41.99	58.24
120	150	0.22	0.40	0.66	0.99	1.93	3.27	5.06	10.09	17.33	27.06	39.46	54.74	
	180	0.26	0.46	0.76	1.15	2.25	3.80	5.87	11.72	20.14	31.44	45.85	63.60	
	210	0.24	0.43	0.70	1.06	2.07	3.50	5.41	10.81	18.56	28.98	42.27	58.63	
134a	80	110	0.23	0.41	0.67	1.01	1.97	3.34	5.16	10.30	17.70	27.63	40.30	55.90
		140	0.20	0.36	0.59	0.89	1.73	2.93	4.52	9.03	15.51	24.22	35.32	48.99
		170	0.19	0.33	0.55	0.83	1.62	2.74	4.24	8.47	14.55	22.71	33.13	45.95
	90	20	0.18	0.32	0.53	0.80	1.55	2.63	4.06	8.10	13.92	21.73	31.70	43.97
		150	0.20	0.37	0.61	0.92	1.79	3.02	4.67	9.32	16.01	24.99	36.45	50.56
		180	0.19	0.34	0.56	0.85	1.66	2.80	4.33	8.64	14.84	23.17	33.79	46.87
	100	130	0.18	0.33	0.54	0.82	1.59	2.69	4.16	8.30	14.26	22.27	32.48	45.05
		160	0.21	0.37	0.62	0.93	1.81	3.07	4.74	9.47	16.26	25.39	37.03	51.36
		190	0.20	0.35	0.58	0.88	1.71	2.90	4.48	8.95	15.37	23.99	34.99	48.54
	110	140	0.19	0.34	0.55	0.84	1.63	2.76	4.26	8.50	14.61	22.80	33.26	46.13
		170	0.21	0.38	0.63	0.96	1.86	3.15	4.88	9.73	16.71	26.09	38.06	52.79
		200	0.20	0.36	0.59	0.90	1.75	2.95	4.57	9.11	15.66	24.44	35.65	49.45
120	150	0.19	0.34	0.56	0.85	1.65	2.80	4.32	8.63	14.83	23.15	33.76	46.83	
	180	0.21	0.38	0.63	0.96	1.87	3.16	4.88	9.74	16.73	26.12	38.10	52.85	
	210	0.20	0.36	0.60	0.90	1.76	2.97	4.60	9.18	15.76	24.61	35.89	49.79	







TABLE 8 – MINIMUM REFRIGERATION CAPACITY IN TONS FOR OIL ENTRAINMENT UP HOT GAS RISERS (TYPE L COPPER TUBING) (CONTINUED)

REFRIGERANT	SATURATION SUCTION TEMP., °F	SUCTION GAS TEMP., °F	PIPE O.D., IN.											
			1/2	5/8	3/4	7/8	1-1/8	1-3/8	1-5/8	2-1/8	2-5/8	3-1/8	3-5/8	4-1/8
			AREA, IN ²											
407C	80	110	0.146	0.233	0.348	0.484	0.825	1.256	1.780	3.094	4.770	6.812	9.213	11.97
		140	0.23	0.42	0.69	1.05	2.04	3.46	5.34	10.66	18.32	28.60	41.72	57.87
		170	0.22	0.40	0.66	0.99	1.93	3.27	5.05	10.08	17.31	27.02	39.42	54.67
	90	120	0.21	0.38	0.63	0.95	1.85	3.13	4.84	9.66	16.60	25.91	37.79	52.42
		150	0.24	0.43	0.71	1.07	2.07	3.51	5.42	10.83	18.60	29.03	42.35	58.74
		180	0.23	0.41	0.67	1.02	1.98	3.35	5.19	10.35	17.78	27.76	40.49	56.17
	100	130	0.22	0.39	0.65	0.98	1.90	3.22	4.97	9.93	17.06	26.63	38.83	53.87
		160	0.24	0.44	0.72	1.09	2.13	3.60	5.56	11.10	19.07	29.77	43.42	60.23
		190	0.23	0.42	0.69	1.05	2.04	3.44	5.32	10.62	18.25	28.49	41.55	57.63
	110	140	0.22	0.40	0.66	1.00	1.94	3.28	5.07	10.12	17.39	27.15	39.60	54.93
		170	0.25	0.45	0.73	1.11	2.16	3.66	5.65	11.28	19.38	30.25	44.12	61.21
		200	0.24	0.42	0.70	1.06	2.06	3.49	5.39	10.76	18.49	28.86	42.10	58.39
120	150	0.23	0.41	0.67	1.01	1.98	3.34	5.17	10.31	17.71	27.65	40.32	55.93	
	180	0.25	0.45	0.75	1.13	2.20	3.71	5.74	11.45	19.68	30.72	44.80	62.15	
	210	0.24	0.43	0.71	1.07	2.09	3.53	5.46	10.91	18.73	29.25	42.66	59.17	
			0.23	0.41	0.68	1.03	2.01	3.40	5.26	10.49	18.03	28.14	41.04	56.93

Refrigeration capacity in tons is based on a saturated suction temperature of 20°F with 15°F superheat at the indicated saturated condensing temperature with 15°F subcooling. The saturated condensing and suction conditions are referenced to the dewpoint for R-407C. For other saturated suction temperatures with 15°F superheat, use correction factors to the capacity given in the table below.

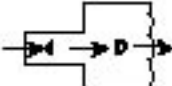
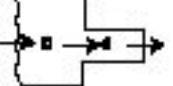




REFRIGERANT	SATURATED SUCTION TEMPERATURE, °F		
	-40	-20	0
22	0.91	0.94	0.97
134a	-	-	0.96
407C	0.88	0.92	0.96

TABLE 9 – FITTING LOSSES IN EQUIVALENT FEET OF PIPE (SCREWED, WELDED, FLANGED AND BRAZED CONNECTIONS)

Nominal Pipe or Tube Size, in.	Smooth Bend Elbows							Smooth Bend Tee		
	90° Std. ^a	90° Long Radius ^b	90° Street ^a	45° Std. ^a	45° Std. ^a	180° Std. ^a	Flow Through Branch	Straight-Through Flow		
								No Reduction	Reduced 1/4	Reduced 1/2
3/8	1.4	0.9	2.5	0.7	1.1	2.5	2.7	0.9	1.2	1.4
1/2	1.5	1.0	2.5	0.8	1.3	2.5	3.0	1.0	1.4	1.5
3/4	2.0	1.4	3.2	0.9	1.6	3.2	4.0	1.4	1.9	2.0
1	2.5	1.7	4.1	1.3	2.1	4.1	6.0	1.7	2.2	2.5
1-1/4	3.5	2.5	5.5	1.7	3.0	5.5	7.0	2.5	3.1	3.5
1-1/2	4.0	2.6	6.5	2.1	3.4	6.5	8.0	2.6	3.7	4.0
2	5.0	3.5	8.2	2.6	4.5	8.2	10.0	3.5	4.7	5.0
2-1/2	6.0	4.1	10.0	3.2	6.2	10.0	12.0	4.1	5.6	6.0
3	7.5	5.0	12.0	4.0	6.4	12.0	15.0	5.0	7.0	7.5
3-1/2	8.0	5.9	12.0	4.7	7.3	16.0	18.0	5.9	8.0	8.0
4	10.0	6.7	10.0	6.2	8.5	17.0	21.0	6.7	9.0	10.0
5	13.0	8.2	12.0	6.5	11.0	21.0	25.0	8.2	12.0	13.0
6	15.0	10.0	15.0	7.9	13.0	25.0	30.0	10.0	14.0	15.0

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TABLE 10 – SPECIAL FITTING LOSSES IN EQUIVALENT FEET OF PIPE (ASHRAE)

Nominal Pipe or Tube Size, in.	Sudden Enlargement, d/D			Sudden Contraction, d/D			Sharp Edge		Pipe Projection	
	1/4	1/2	3/4	1/4	1/2	3/4	Entrance	Exit	Entrance	Exit
										
3/8	1.4	0.8	0.3	0.7	0.5	0.3	1.5	0.8	1.5	1.1
1/2	1.8	1.1	0.4	0.8	0.7	0.4	1.8	1.0	1.8	1.5
3/4	2.5	1.5	0.5	1.2	1.0	0.5	2.5	1.4	2.5	2.2
1	3.2	2.0	0.7	1.6	1.2	0.7	3.7	1.8	3.7	2.7
1-1/4	4.7	3.0	1.0	2.3	1.8	1.0	5.3	2.6	5.3	4.2
1-1/2	6.5	3.8	1.2	2.8	2.2	1.2	6.5	3.3	6.5	5.0
2	8.0	4.5	1.6	4.0	3.0	1.6	8.0	4.4	8.0	6.5
2-1/2	10.0	6.1	2.0	5.0	3.8	2.0	12.0	5.8	12.0	8.7
3	13.0	8.0	2.6	6.5	4.9	2.6	14.0	7.2	14.0	11.0
3-1/2	15.0	9.2	3.0	7.7	6.0	3.0	17.0	8.5	17.0	13.0
4	17.0	11.0	3.8	9.0	6.8	3.8	20.0	10.0	20.0	16.0
5	24.0	15.0	5.0	12.0	9.0	5.0	27.0	14.0	27.0	20.0
6	29.0	22.0	6.0	15.0	11.0	6.0	32.0	19.0	32.0	25.0

LD07401

TABLE 11 – VALVE LOSSES IN EQUIVALENT FEET OF PIPE (ASHRAE)

Nominal Pipe or Tube Size, in.	Globe ^a	60° Wye	45° Wye	Angle ^a	Gate ^b	Swing Check ^c	Lift Check
3/8	17	8	6	6	0.6	5	Globe and vertical lift same as globe valve ^d
1/2	18	9	7	7	0.7	6	
3/4	22	11	9	9	0.9	8	
1	29	15	12	12	1.0	10	
1-1/4	38	20	15	15	1.5	14	
1-1/2	43	24	18	18	1.8	16	
2	55	30	24	24	2.3	20	
2-1/2	69	35	29	29	2.8	25	Angle lift same as angle valve
3	84	43	35	35	3.2	30	
3-1/2	100	50	41	41	4.0	35	
4	12	58	47	47	4.5	40	
5	140	71	58	58	6.0	50	
6	170	88	70	70	7.0	60	

NOTE:

Losses are for valves in fully open position and with screwed, welded, flanged, or flared connections.

^a These losses do not apply to valves with needlepoint seats.

^b Regular and short pattern plug cock valves, when fully open, have the same loss as gate valve.

^c Losses also apply to the in-line, ball-type check valve.

^d For Y pattern globe lift check valve with seat approximately equal to the nominal pipe diameter, use values of 60° Wye valves for loss.

TABLE 12 – REFRIGERANT CHARGE IN POUNDS PER 100 FEET OF SUCTION LINE

Saturated Suction Temperature, °F									
Line Size, O.D.	R-22			R-134a			R-407C		
	-40	0	40	0	20	40	-40	0	40
1/2	0.03	0.07	0.15	0.05	0.07	0.11	0.02	0.06	0.14
5/8	0.05	0.12	0.25	0.08	0.12	0.17	0.04	0.10	0.22
3/4	0.07	0.18	0.37	0.11	0.17	0.26	0.06	0.15	0.33
7/8	0.10	0.24	0.51	0.16	0.24	0.36	0.08	0.21	0.46
1-1/8	0.17	0.42	0.87	0.27	0.41	0.61	0.14	0.36	0.79
1-3/8	0.26	0.64	1.33	0.41	0.62	0.92	0.21	0.55	1.20
1-5/8	0.37	0.90	1.88	0.58	0.88	1.31	0.30	0.78	1.70
2-1/8	0.65	1.57	3.27	1.00	1.54	2.27	0.53	1.35	2.95
2-5/8	1.01	2.42	5.04	1.55	2.37	3.50	0.81	2.09	4.55
3-1/8	1.44	3.45	7.19	2.21	3.38	5.00	1.16	2.98	6.50
3-5/8	1.94	4.66	9.73	2.99	4.57	6.77	1.57	4.03	8.79
4-1/8	2.52	6.06	12.65	3.88	5.94	8.79	2.04	5.24	11.42
5-1/8	3.93	9.45	19.71	6.05	9.26	13.70	3.17	8.17	17.80
6-1/8	5.65	13.58	28.34	8.69	13.31	19.70	4.56	11.74	25.59

TABLE 13 – REFRIGERANT CHARGE IN POUNDS PER 100 FEET OF DISCHARGE LINE

Saturated Suction Temperature, °F									
Line Size, O.D.	R-22			R-134a			R-407C		
	-40	0	40	0	20	40	-40	0	40
1/2	0.22	0.33	0.48	0.17	0.27	0.40	0.21	0.33	0.50
5/8	0.35	0.53	0.77	0.27	0.43	0.65	0.34	0.53	0.80
3/4	0.53	0.79	1.15	0.41	0.64	0.97	0.50	0.78	1.19
7/8	0.73	1.10	1.60	0.57	0.89	1.34	0.70	1.09	1.65
1-1/8	1.25	1.87	2.72	0.97	1.52	2.29	1.19	1.86	2.81
1-3/8	1.90	2.85	4.14	1.48	2.31	3.49	1.81	2.83	4.28
1-5/8	2.69	4.04	5.86	2.09	3.28	4.94	2.57	4.01	6.06
2-1/8	4.67	7.02	10.20	3.63	5.70	8.59	4.47	6.97	10.55
2-5/8	7.20	10.83	15.72	5.60	8.79	13.25	6.89	10.74	16.27
3-1/8	10.28	15.45	22.45	8.00	12.54	18.92	9.83	15.34	23.22
3-5/8	13.91	20.90	30.36	10.82	16.96	25.59	13.30	20.74	31.40
4-1/8	18.08	27.17	39.46	14.06	22.05	33.26	17.29	26.96	40.82
5-1/8	28.17	42.34	61.50	21.91	34.37	51.84	26.94	42.02	63.62
6-1/8	40.50	60.87	88.41	31.50	49.40	74.52	38.73	60.41	91.46

Discharge superheat is 105°F for R-22 and R-407C, and 85°F for R-134a.

TABLE 14 – REFRIGERANT CHARGE IN POUNDS PER 100 FEET OF LIQUID LINE

Line Size, O.D.	Saturated Suction Temperature, °F								
	R-22			R-134a			R-407C		
	-40	0	40	0	20	40	-40	0	40
1/2	7.47	7.03	6.50	7.56	7.13	6.64	7.11	6.64	6.11
5/8	12.01	11.29	10.44	12.15	11.46	10.66	1.42	10.67	9.81
3/4	17.93	16.86	15.59	18.14	17.11	15.92	17.05	15.93	14.65
7/8	24.91	23.42	21.66	25.20	23.78	22.11	23.69	22.14	20.35
1-1/8	42.47	39.93	36.93	42.96	40.54	37.70	40.39	37.74	34.70
1-3/8	64.69	60.82	56.24	65.44	61.74	57.42	61.52	57.49	52.85
1-5/8	91.56	86.09	79.61	92.62	87.39	81.28	87.08	81.37	74.81
2-1/8	159.30	149.80	138.50	161.10	152.00	141.40	151.50	141.60	130.10
2-5/8	245.60	231.00	213.60	248.50	234.40	218.00	233.60	218.30	200.70
3-1/8	350.60	329.70	304.80	354.70	334.60	311.20	333.40	311.60	286.40
3-5/8	474.20	445.90	412.30	479.70	452.60	420.90	451.00	421.40	387.40
4-1/8	616.40	579.60	536.00	623.60	588.40	547.20	586.20	547.80	503.60
5-1/8	960.70	903.30	835.30	971.80	917.00	852.80	913.70	853.80	784.90
6-1/8	1381.00	1299.00	1201.00	1397.00	1318.00	1226.00	1313.00	1227.00	1128.00

Discharge superheat is 105°F for R-22 and R-407C, and 85°F for R-134a.

