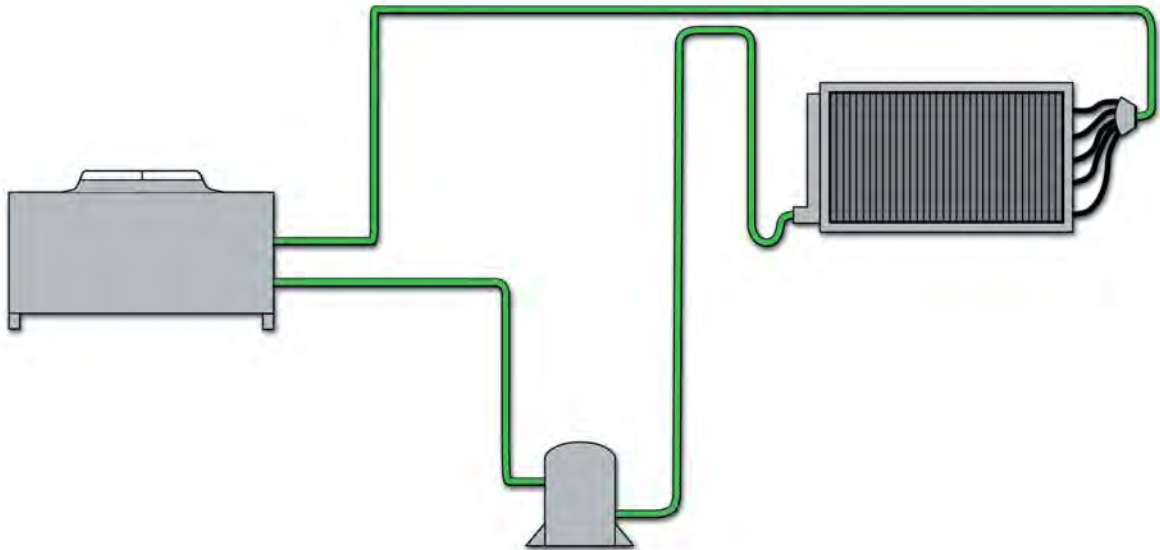




# Air Conditioning Clinic

## Refrigerant Piping

One of the Fundamental Series







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# **Refrigerant Piping**

**One of the Fundamental Series**

**A publication of Trane**

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# Preface

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## Refrigerant Piping

A Trane Air Conditioning Clinic

Figure 1

Trane believes that it is incumbent on manufacturers to serve the industry by regularly disseminating information gathered through laboratory research, testing programs, and field experience.

The Trane Air Conditioning Clinic series is one means of knowledge sharing. It is intended to acquaint a technical audience with various fundamental aspects of heating, ventilating, and air conditioning (HVAC). We have taken special care to make the clinic as uncommercial and straightforward as possible.

Illustrations of Trane products only appear in cases where they help convey the message contained in the accompanying text.

This particular clinic introduces the reader to **refrigerant piping**.

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period one

# Refrigerant Piping Requirements

## notes

### Refrigerant Piping

*period one*

### Refrigerant Piping Requirements

Figure 2

The focus of this clinic is on the design and installation of the interconnecting piping for vapor-compression refrigeration systems. Reviewing the physical changes that the refrigerant undergoes within the refrigeration cycle will help demonstrate certain demands that the piping design must meet.

This clinic focuses on systems that use Refrigerant-22 (R-22). While the general requirements are the same for systems that use other refrigerants, velocities and pressure drops will differ.

### Vapor-Compression Refrigeration

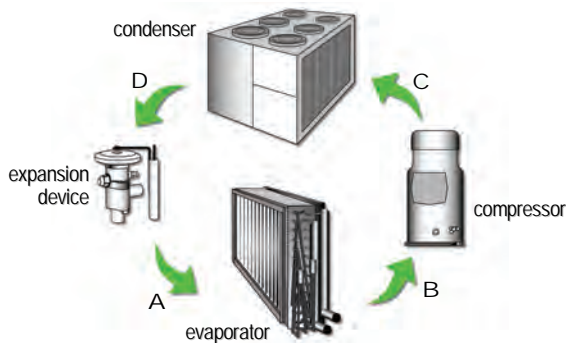


Figure 3

Figure 3 illustrates a basic vapor-compression refrigeration cycle. Refrigerant enters the evaporator in the form of a cool, low-pressure mixture of liquid and vapor (A). Heat is transferred to the refrigerant from the relatively warm air that is being cooled, causing the liquid refrigerant to boil. The resulting refrigerant vapor (B) is then pumped from the evaporator by the compressor, which increases the pressure and temperature of the vapor.

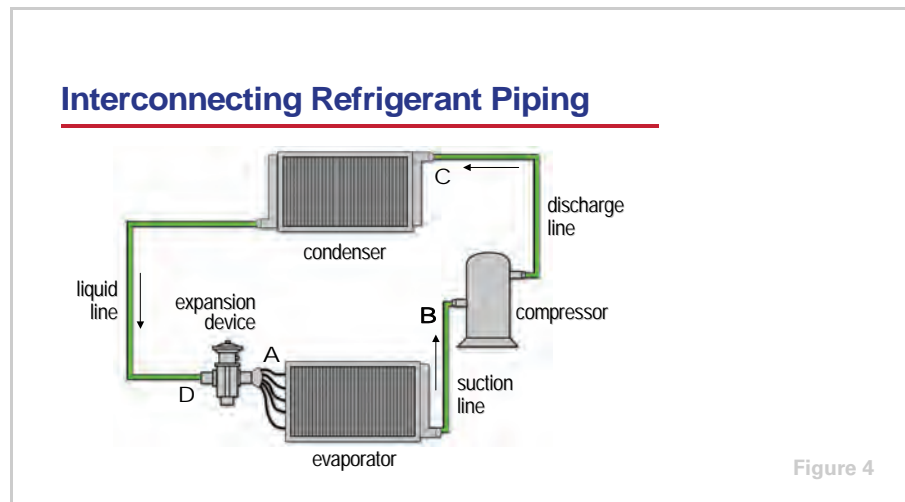
## period one

# Refrigerant Piping Requirements

### notes

The resulting hot, high-pressure refrigerant vapor (C) enters the condenser where heat is transferred to ambient air, which is at a lower temperature than the refrigerant. Inside the condenser, the refrigerant vapor condenses into a liquid and is subcooled. This liquid refrigerant (D) then flows from the condenser to the expansion device. This device creates a pressure drop that reduces the pressure of the refrigerant to that of the evaporator. At this low pressure, a small portion of the refrigerant boils (or flashes), cooling the remaining liquid refrigerant to the desired evaporator temperature. The cool mixture of liquid and vapor refrigerant (A) enters the evaporator to repeat the cycle.

The vapor-compression refrigeration cycle, and the four major components of the refrigeration system (evaporator, compressor, condenser, and expansion device), are discussed in more detail in separate clinics. Refer to the list of references at the end of Period Six.



These individual components are connected by refrigerant piping. The suction line connects the evaporator to the compressor, the discharge line connects the compressor to the condenser, and the liquid line connects the condenser to the expansion device. The expansion device is typically located at the end of the liquid line, at the inlet to the evaporator.

There is more to the design of refrigerant piping than moving refrigerant from one component to another. Regardless of the care exercised in selection and application of the components of the refrigeration system, operational problems may be encountered if the interconnecting piping is improperly designed or installed.



period one

# Refrigerant Piping Requirements

## notes

### Refrigerant Piping Requirements

- ▲ Return oil to compressor
- ▲ Ensure that only liquid refrigerant enters the expansion device
- ▲ Minimize system capacity loss
- ▲ Minimize refrigerant charge

Figure 5

When a refrigeration system includes field-assembled refrigerant piping to connect two or more of the components, the primary design goals are generally to maximize system reliability and minimize installed cost. To accomplish these two goals, the design of the interconnecting refrigerant piping must meet the following requirements:

- Return oil to the compressor at the proper rate, at all operating conditions
- Ensure that only liquid refrigerant (no vapor) enters the expansion device
- Minimize system capacity loss that is caused by pressure drop through the piping and accessories
- Minimize the total refrigerant charge in the system to improve reliability and minimize installed cost

### Scroll Compressor

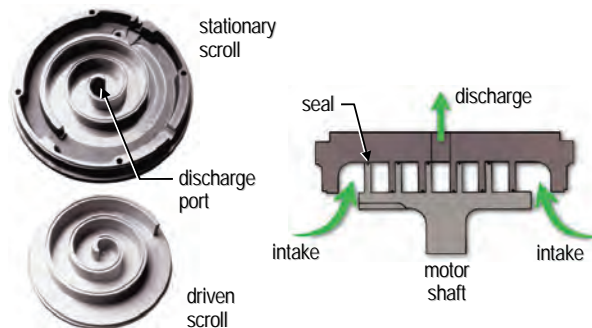


Figure 6

The first requirement is to ensure that oil is returned to the compressor at all

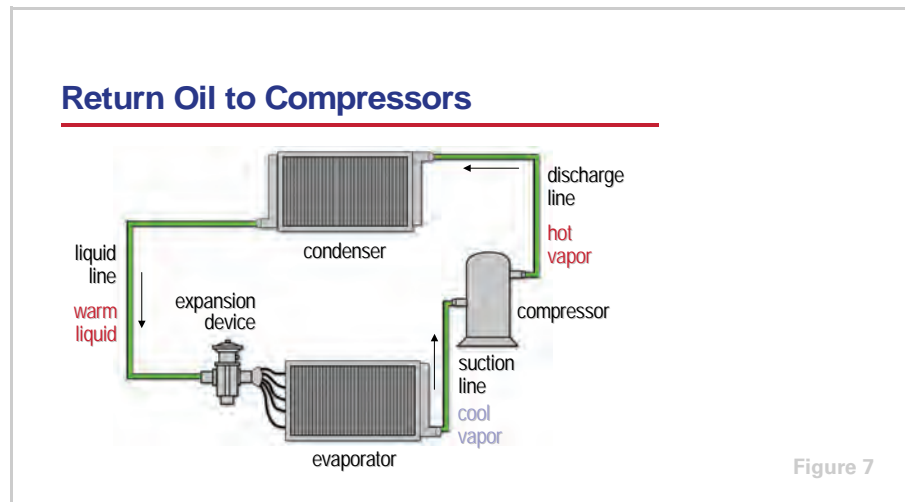
## period one

# Refrigerant Piping Requirements

## notes

operating conditions. Oil is used to lubricate and seal the moving parts of a compressor. For example, the scroll compressor shown in Figure 6 on page 3 uses two scroll configurations, mated face-to-face, to compress the refrigerant vapor. The tips of these scrolls are fitted with seals that, along with a thin layer of oil, prevent the compressed refrigerant vapor from escaping through the mating surfaces. Similarly, other types of compressors also rely on oil for lubrication and for providing a seal when compressing the refrigerant vapor.

Characteristically, some of this lubricating oil is pumped along with the refrigerant throughout the rest of the system. While this oil has no function anywhere else in the system, the refrigerant piping must be designed and installed so that this oil returns to the compressor at the proper rate, at all operating conditions.



Returning to the system schematic, droplets of oil are pumped out of the compressor along with the hot, high-pressure refrigerant vapor. The velocity of the refrigerant inside the discharge line must be high enough to carry the small oil droplets through the pipe to the condenser.

Inside the condenser, the refrigerant vapor condenses into a liquid. Liquid refrigerant and oil have an affinity for each other, so the oil easily moves along with the liquid refrigerant. From the condenser, this mixture of liquid refrigerant and oil flows through the liquid line to the expansion device.

Next, the refrigerant–oil mixture is metered through the expansion device into the evaporator, where the liquid refrigerant absorbs heat and vaporizes. Again, the velocity of the refrigerant vapor inside the suction line must be high enough to carry the droplets of oil through the pipe back to the compressor.

Without adequate velocity and proper pipe installation, oil may be trapped out in the system. If this condition is severe enough, the reduced oil level in the compressor could cause lubrication problems and, potentially, mechanical failure.

period one

## Refrigerant Piping Requirements

### notes

#### Thermostatic Expansion Valve (TXV)

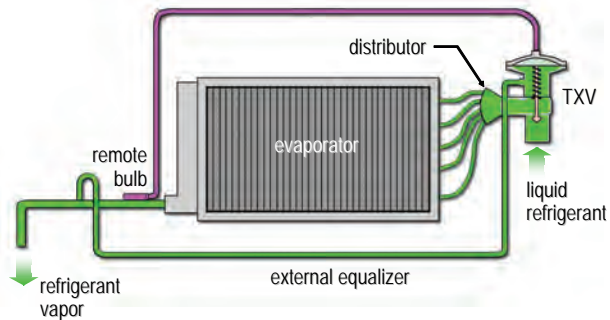


Figure 8

The second requirement of the refrigerant piping design is to ensure that only liquid refrigerant enters the expansion device. There are several types of expansion devices, including expansion valves (thermostatic or electronic), capillary tubes, and orifices.

In addition to maintaining the pressure difference between the high-pressure (condenser) and low-pressure (evaporator) sides of the system, a **thermostatic expansion valve (TXV)** also controls the quantity of liquid refrigerant that enters the evaporator. This ensures that the refrigerant will be completely vaporized within the evaporator, and maintains the proper amount of superheat in the system.

#### Subcooling

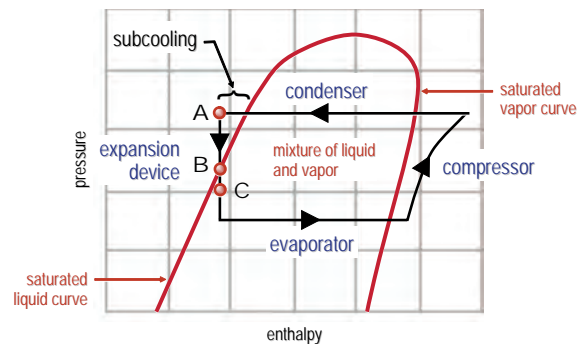


Figure 9

Inside the condenser, after all of the refrigerant vapor has condensed into liquid, the refrigerant is subcooled to further lower its temperature. This subcooled liquid refrigerant leaves the condenser (**A**) and experiences a pressure drop as it flows through the liquid line and accessories, such as a filter

## period one

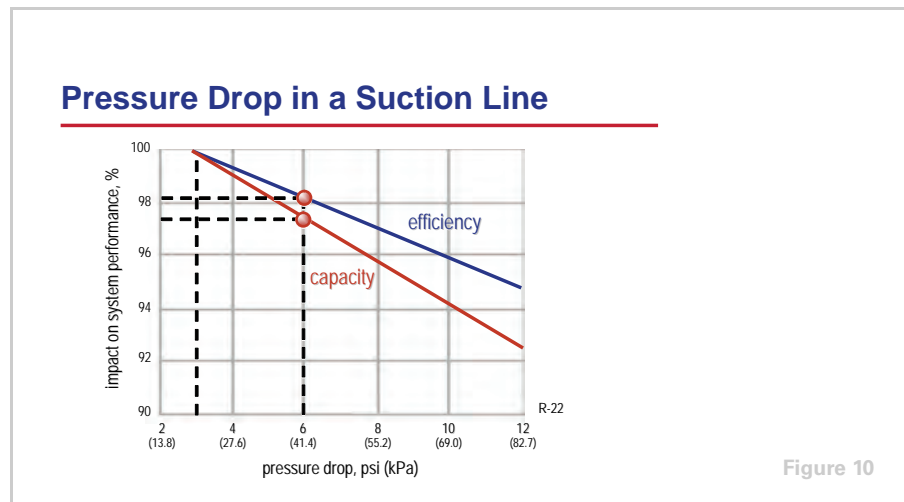
# Refrigerant Piping Requirements

### notes

drier and solenoid valve, installed upstream of the TXV. On the pressure-enthalpy chart, Figure 9 on page 5, this moves the condition of the refrigerant toward the saturated liquid curve (**B**). If this pressure drop is high enough, or if the refrigerant has not been subcooled enough by the condenser, a small portion of the refrigerant may boil (or flash), resulting in a mixture of liquid and vapor (**C**) entering the expansion device.

The presence of refrigerant vapor upstream of the expansion device is very undesirable. Bubbles of vapor displace liquid in the port of the TXV, reducing the flow rate of liquid through the valve, therefore substantially reducing the capacity of the evaporator. This results in erratic valve operation.

The design of the piping system must ensure that only liquid refrigerant (no vapor) enters the expansion device. This requires that the condenser provide adequate subcooling at all system operating conditions, and that the pressure drop through the liquid line and accessories not be high enough to cause flashing. Subcooling allows the liquid refrigerant to experience some pressure drop as it flows through the liquid line, without the risk of flashing.



The third requirement of the refrigerant piping design is to minimize system capacity loss. To achieve the maximum capacity from the system, the refrigerant must circulate through the system as efficiently as possible. This involves minimizing any pressure drop through the piping and other system components.

Whenever a fluid flows inside a pipe, a characteristic pressure drop is experienced. Pressure drop is caused by friction between the moving liquid (or vapor) and the inner walls of the pipe. The total pressure drop depends on the pipe diameter and length, the number and type of fittings and accessories installed in the line, and the mass flow rate, density, and viscosity of the refrigerant.

As an example, the chart in Figure 10 demonstrates the impact of pressure drop, through the suction line, on the capacity and efficiency of the system. For this example system operating with Refrigerant-22, increasing the total

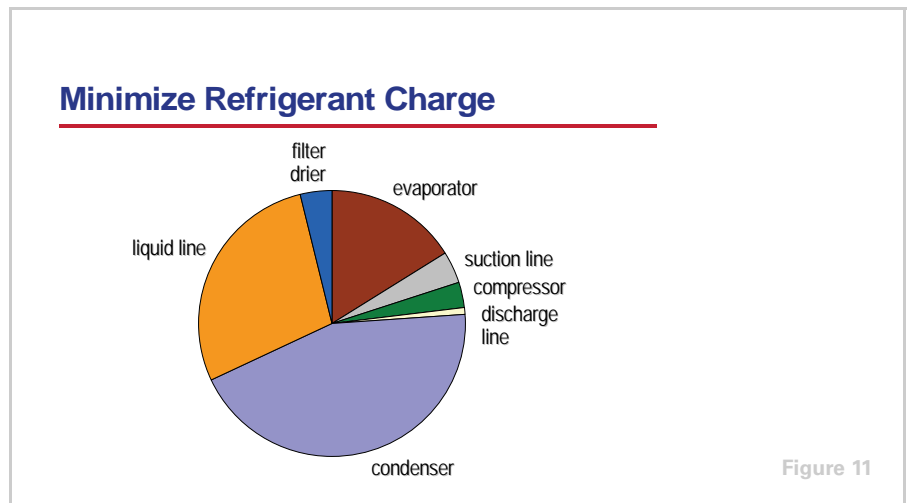
## period one

# Refrigerant Piping Requirements

## notes

pressure drop in the suction line from 3 psi (20.7 kPa) to 6 psi (41.4 kPa) decreases system capacity by about 2.5 percent and decreases system efficiency by about 2 percent.

This reveals a compromise that the system designer must deal with. The diameter of the suction line must be small enough that the resulting refrigerant velocity is sufficiently high to carry oil droplets through the pipe. However, the pipe diameter must not be so small that it creates an excessive pressure drop, reducing system capacity too much.



The first three requirements have remained unchanged for many years. However, years of observation and troubleshooting has revealed that the lower the system refrigerant charge, the more reliably the system performs. Therefore, a fourth requirement has been added for the design of refrigerant piping: minimize the total amount of refrigerant in the system. To begin with, this involves laying out the shortest, simplest, and most-direct pipe routing. It also involves using the smallest pipe diameter possible, particularly for the liquid line because, of the three lines, it impacts refrigerant charge the most. The chart in Figure 11 shows that the liquid line is second only to the condenser in the amount of refrigerant it contains.

This reveals another compromise for the system designer. The diameter of the liquid line must be as small as possible to minimize the total refrigerant charge. However, the pipe diameter cannot be small enough to create an excessive pressure drop that results in flashing before the liquid refrigerant reaches the expansion device.

period one

# Refrigerant Piping Requirements

## notes

### Involve the Manufacturer

- ▲ If provided, use refrigerant line sizes recommended by manufacturer



Figure 12

This clinic discusses the processes for sizing the interconnecting piping in an air-conditioning system. Some of the information required for selecting the optimal line sizes is best known by the manufacturer. Therefore, if the manufacturer of the refrigeration equipment provides recommended line sizes, or tools for selecting the optimal line sizes, we recommend that you use those line sizes.

If, however, line sizes are not provided by the manufacturer, the processes outlined within this clinic could be used for selecting the sizes.

### General Piping Requirements

- ▲ Use clean Type L copper tubing
  - ◆ Copper-to-copper joints: BCuP-6 without flux
  - ◆ Copper-to-steel (or brass) joints: BAg-28, non-acid flux
- ▲ Properly support piping to account for expansion, vibration, and weight
- ▲ Avoid installing piping underground
- ▲ Test entire refrigerant circuit for leaks

Figure 13

Before discussing the design and installation of the suction, discharge, and liquid lines, there are some general requirements that apply to all of these lines.

First, copper tubing is typically used for refrigerant piping in air-conditioning systems. This tubing is available in various standard diameters and wall thicknesses. The nominal diameter of the tubing is expressed in terms of its

## period one

# Refrigerant Piping Requirements

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### notes

outside diameter. This tubing must be completely free from dirt, scale, and oxide. New Type L or Type ACR tubing that has been cleaned by the manufacturer and capped at both ends is recommended for air-conditioning applications.

The piping system is constructed by brazing copper tubes and fittings together. When brazing copper-to-copper joints, use BCuP-6\* without flux. For copper-to-steel or copper-to-brass joints, use BAg-28\* with a non-acid flux.

The refrigerant piping must be properly supported to account for expansion, vibration, and the total weight of the piping. When a pipe experiences a temperature change, it is subject to a certain amount of expansion and contraction. Because the refrigerant piping is connected to the compressor, vibration forces are transmitted to the piping itself. Finally, the weight of the refrigerant-filled pipe and fittings must be supported to prevent the pipes from sagging, bending, or breaking.

Avoid installing refrigerant piping underground. It is very difficult to maintain cleanliness during installation or to test for leaks. If underground installation is unavoidable, each line must be insulated separately, and then the lines must be waterproofed and protected with a hard casing (such as PVC).

After the piping has been installed, the entire refrigeration circuit must be tested for leaks before it can be charged with refrigerant. This process typically involves pressurizing the entire piping system with dry nitrogen to examine each brazed joint for leaks.

Each of these issues is discussed in greater detail in the Trane Reciprocating Refrigeration Manual.

\* Based on the American Welding Society's (AWS) *Specification for Filler Metals for Brazing and Braze Welding*, publication A5.8–1992

## period two

# Suction Line

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## notes

### Refrigerant Piping

*period two*

Suction Line

Figure 14

The first line to be considered is the suction line. Again, this pipe conducts low-pressure refrigerant vapor from the evaporator to the compressor.

suction line

### Requirements for Sizing and Routing

- ▲ Ensure adequate velocity to return oil to compressor at all steps of unloading
- ▲ Avoid excessive noise
- ▲ Minimize system capacity and efficiency loss

Figure 15

### **Requirements for Sizing and Routing**

The diameter of the suction line must be small enough that the resulting refrigerant velocity is sufficiently high to carry oil droplets, at all steps of compressor unloading. If the velocity in the pipe is too high, however, objectionable noise may result. Also, the pipe diameter should be as large as possible to minimize pressure drop and thereby maximize system capacity and efficiency.



## period two

# Suction Line

## notes

suction line Sizing “Rules”	
old rules for R-22	new rules for R-22
<b>minimum velocity</b>	
1,000 fpm (5 m/s) for vertical risers	based on diameter of riser
500 fpm (2.5 m/s) for horizontal sections	percentage of minimum riser velocity
<b>maximum velocity</b>	
4,000 fpm (20 m/s)	4,000 fpm (20 m/s)
<b>maximum pressure drop</b>	
3 psi (20 kPa)	based on specific system requirements

Figure 16

It may be helpful to compare the old “rules” for selecting the diameter of the suction line with the newer rules that result from changes in compressor technology, recent research, and the additional requirement to minimize system refrigerant charge.

In the past, many suction lines for systems operating with Refrigerant-22 were sized to ensure that the minimum velocity in a vertical suction riser was more than 1,000 fpm (5 m/s), and the minimum velocity in a horizontal section was more than 500 fpm (2.5 m/s). Actually, the minimum allowable velocity in a suction riser depends on the diameter of the pipe.

The minimum velocity required to carry oil droplets up a vertical riser is higher for a larger diameter pipe than it is for a smaller diameter pipe. This is due to the velocity profile of the refrigerant flowing inside the pipe. In a smaller diameter pipe, the higher-velocity refrigerant is closer to the inner walls of the pipe than it is in a larger-diameter pipe. For instance, while the minimum allowable velocity in a 2 1/8 in. (54 mm)-diameter suction riser is approximately 1,000 fpm (5 m/s), the minimum velocity in a 1 1/8 in. (28 mm)-diameter riser is only 700 fpm (3.6 m/s). While the old minimum-velocity limits were easy to remember, they may lead to the unnecessary use of double suction risers.

The recommended maximum-velocity limit of 4,000 fpm (20 m/s) has not changed. A higher velocity inside the suction line may cause objectionable noise for those nearby.

Another common rule, in a system operating with R-22, was to limit the pressure drop through the suction line to 3 psi (20 kPa). Although this was often thought to be a maximum limit, this value was originally intended to be only a recommendation, or guideline, for minimizing capacity and efficiency loss. Today, architects and HVAC system design engineers are placing the components of the refrigeration system farther apart, and this 3 psi (20 kPa) limit is often overly restrictive. Longer line lengths and the associated higher pressure drop can be tolerated, assuming that the loss of system capacity and efficiency is acceptable for the given application. Of course, it is still good

## period two

# Suction Line

### notes

practice to minimize pressure drop, but an arbitrary limit places an unnecessary restriction on the system designer.

#### suction line

#### Process for Sizing

- 1 Determine total length of piping
- 2 Calculate refrigerant velocity at maximum and minimum capacities
- 3 Select largest pipe diameter that results in acceptable velocity at both maximum and minimum capacities
- 4 Calculate total equivalent length of straight pipe and fittings
- 5 Determine pressure drop due to pipe and fittings
- 6 Add pressure drop due to accessories

Figure 17

#### Sizing Process

Following are the steps to follow when selecting the proper diameter of the suction line:

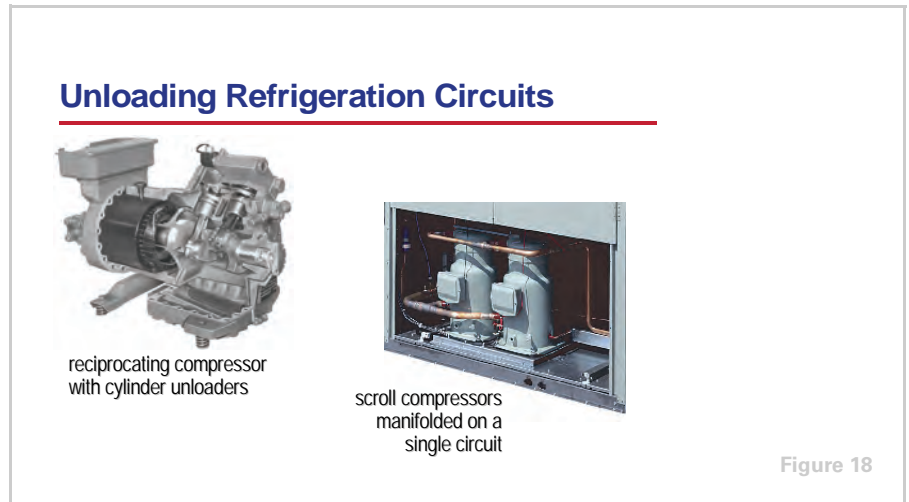
- 1) Determine the total length of suction-line piping.
- 2) Calculate the refrigerant velocity at both maximum and minimum system capacities.
- 3) Select the largest pipe diameter that will result in acceptable refrigerant velocity at both maximum and minimum capacities.
- 4) Calculate the total “equivalent” length of piping by adding the actual length of straight pipe to the equivalent length of any fittings to be installed in the suction line.
- 5) Determine the pressure drop (based on the total equivalent length) due to the straight pipe and fittings.
- 6) Add the pressure drop due to any accessories installed in the suction line.

To begin with, the refrigerant piping should be routed in the shortest and simplest manner possible, minimizing the total length of piping. From the initial layout, the total measured length of the suction line can be estimated.

## period two

# Suction Line

## notes



If the system contains more than one independent refrigerant circuit, each circuit requires its own set of refrigerant lines. Therefore, the capacity of each individual circuit must be considered separately.

Some refrigeration circuits include only one compressor that cycles on and off. This is very common in residential and light-commercial air-conditioning systems. In this case, the refrigeration circuit only operates at one capacity—fully on—so only the maximum system capacity needs to be considered.

However, if the circuit contains a compressor that is capable of unloading, such as a single reciprocating compressor with cylinder unloaders, or if more than one compressor is manifolded together on a single circuit, such as multiple scroll compressors, then the minimum capacity of the circuit must also be determined. When the circuit unloads, less refrigerant flows through the system and the refrigerant velocity inside the piping is reduced.

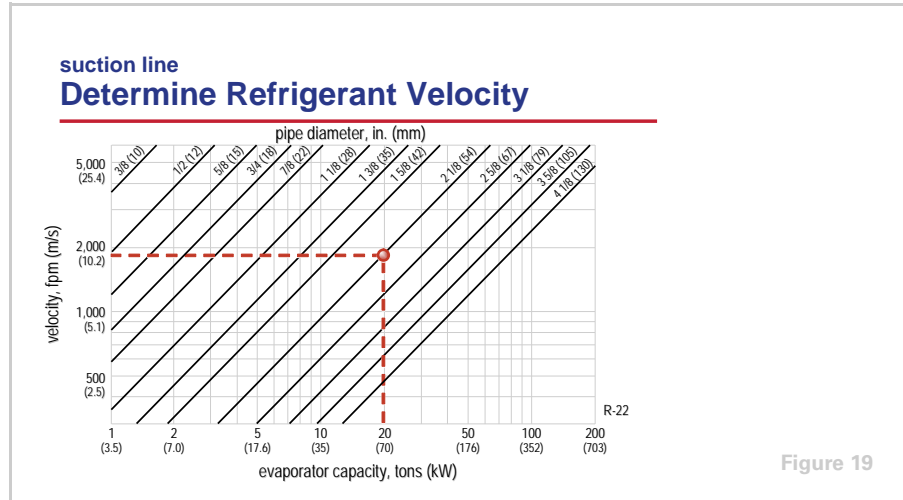
Recall that the diameter of the suction line must provide adequate velocity at both maximum and minimum capacities. At maximum capacity, the refrigerant velocity through the suction line will be the highest. Therefore, maximum capacity is important to ensure that the refrigerant velocity is below the upper limit of 4,000 fpm (20 m/s). At minimum capacity, the refrigerant velocity will be the lowest. Therefore, the minimum capacity of the refrigeration circuit is critical for ensuring that the refrigerant velocity is high enough to properly return oil to the compressor.

It is important to note that the diameter of a vertical riser does not necessarily need to be the same as the diameter of the horizontal or vertical drop sections of pipe. The horizontal or vertical drop sections can often be selected one diameter larger than a vertical riser, reducing the overall pressure drop due to the suction line. This will be demonstrated later in this period.

## period two

# Suction Line

## notes



The refrigerant velocity inside a pipe depends on the mass flow rate and density of the refrigerant, and on the inside diameter of the pipe. The chart in Figure 19 shows the velocity of R-22 inside pipes of various diameters at one particular operating condition—40°F (4.4°C) saturated suction temperature, 125°F (51.7°C) saturated condensing temperature, 12°F (6.7°C) of superheat, 15°F (8.3°C) of subcooling, and 70°F (38.9°C) of compressor superheat. For an example system with an evaporator capacity of 20 tons (70.3 kW), the refrigerant velocity inside a 2 1/8 in. (54 mm)-diameter pipe at this condition is about 1,850 fpm (9.4 m/s).

The easiest and most accurate method for determining refrigerant velocity is to use a computer program that can calculate the velocity for various pipe sizes based on actual conditions. However, if you do not have access to such a program, a chart like this may be useful.

**suction line**  
**Determine Refrigerant Velocity**

pipe diameter, in. (mm)	velocity, fpm (m/s)	
	20 tons (70.3 kW)	10 tons (35.2 kW)
1 1/8 (28)	7,000 (35.6)	3,500 (17.8)
1 3/8 (35)	4,600 (23.4)	2,300 (11.7)
1 5/8 (42)	3,250 (16.5)	1,625 (8.3)
2 1/8 (54)	1,850 (9.4)	925 (4.7)
2 5/8 (67)	1,200 (6.1)	600 (3.1)
3 1/8 (79)	850 (4.3)	425 (2.2)

Figure 20

Assume that this example 20-ton (70.3-kW) system contains one refrigeration

## period two

# Suction Line

## notes

circuit with two steps of capacity. Maximum system (evaporator) capacity is 20 tons (70.3 kW) and the circuit can unload to 10 tons (35.2 kW) of capacity. Using the chart in Figure 19 on page 14, the refrigerant velocity at both maximum and minimum capacities is determined for several pipe diameters.

After these velocities have been determined, the largest acceptable pipe diameter is selected to minimize the overall pressure drop due to the suction line.

When this system operates at maximum capacity, use of either the 1 1/8 in. (28 mm)- or the 1 3/8 in. (35 mm)-diameter pipes results in a refrigerant velocity that is greater than the recommended upper limit of 4,000 fpm (20 m/s). Again, these high velocities may cause objectionable noise, so these pipe sizes should probably not be considered.

### suction line

### Minimum Allowable Velocities

pipe diameter, in. (mm)	minimum velocity, fpm (m/s)	
	riser	horiz/drop
3/8 (10)	370 (1.9)	275 (1.4)
1/2 (12)	460 (2.3)	350 (1.8)
5/8 (15)	520 (2.6)	390 (2.0)
3/4 (18)	560 (2.8)	420 (2.1)
7/8 (22)	600 (3.1)	450 (2.3)
1 1/8 (28)	700 (3.6)	525 (2.7)
1 3/8 (35)	780 (4.0)	585 (3.0)
1 5/8 (42)	840 (4.3)	630 (3.2)
2 1/8 (54)	980 (5.0)	735 (3.7)
2 5/8 (67)	1,080 (5.5)	810 (4.1)
3 1/8 (79)	1,180 (6.0)	885 (4.5)
3 5/8 (105)	1,270 (6.5)	950 (4.8)
4 1/8 (130)	1,360 (6.9)	1,020 (5.2)

Figure 21

Figure 21 shows the minimum allowable refrigerant velocity, for both a vertical suction riser and a horizontal (or vertical drop) section of suction line, for each standard pipe diameter. As mentioned earlier in this period, the minimum allowable velocity in a suction riser depends on the diameter of the pipe. The minimum velocity for a horizontal, or vertical drop, section is 75 percent of the minimum allowable velocity for a vertical riser of the same diameter.

The minimum velocities listed in this table assume a worst-case operating condition of 20°F (-6.7°C) saturated suction temperature. This provides a safety factor, because a system will probably operate at this type of condition at some time in its life.

## period two

# Suction Line

## notes

### Select Suction Line Size

circuit capacity, tons (kW)	velocity inside 2 1/8 in. (54 mm) pipe, fpm (m/s)	minimum velocity for 2 1/8 in. (54 mm) pipe, fpm (m/s)	
		riser	horiz/drop
20 (70.3)	1,850 (9.4)	980 (5.0)	735 (3.7)
10 (35.2)	925 (4.7)		

circuit capacity, tons (kW)	velocity inside 1 5/8 in. (42 mm) pipe, fpm (m/s)	minimum velocity for 1 5/8 in. (42 mm) pipe, fpm (m/s)	
		riser	horiz/drop
20 (70.3)	3,250 (16.5)	840 (4.3)	630 (3.2)
10 (35.2)	1,625 (8.3)		

Figure 22

Based on the minimum allowable velocities listed in Figure 21 on page 15, a 2 1/8 in. (54 mm) pipe will result in acceptable velocity, at both maximum and minimum capacities, for the horizontal and vertical drop sections of this example suction line. One size larger would result in a velocity that is too low when the system operates at minimum capacity.

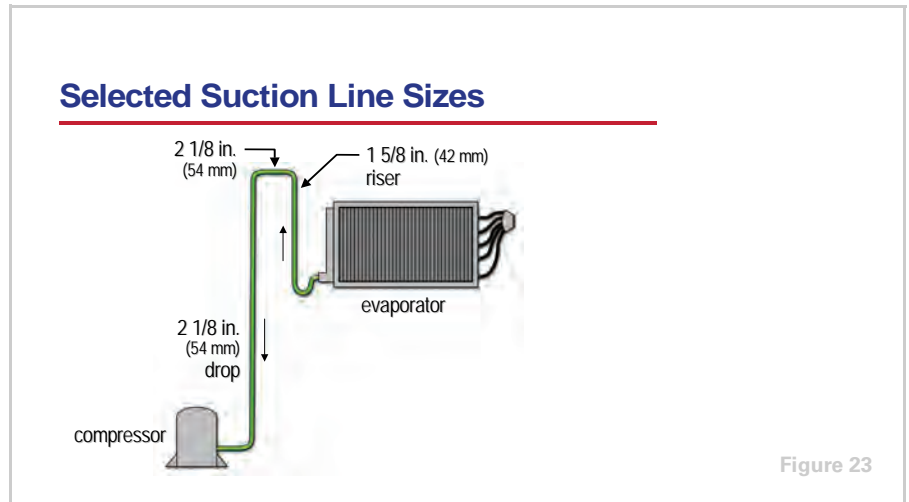
When the circuit unloads to 10 tons (35.2 kW), however, the velocity inside a 2 1/8 in. (54 mm) pipe—925 fpm (4.7 m/s)—will drop below the minimum allowable velocity for a vertical riser of this diameter—980 fpm (5.0 m/s).

Downsizing all vertical suction risers to 1 5/8 in. (42 mm)-diameter pipe will result in acceptable velocity at both maximum and minimum capacities. At minimum capacity, the velocity—1,625 fpm (8.3 m/s)—is above the minimum allowable velocity for a riser of this diameter—840 fpm (4.3 m/s). At maximum capacity, the velocity—3,250 fpm (16.5 m/s)—is below the 4,000 fpm (20 m/s) upper limit.

## period two

# Suction Line

## notes



For this example system, a 2 1/8 in. (54 mm)-diameter pipe can be used for the horizontal and vertical drop sections of the suction line, and a 1 5/8 in. (42 mm)-diameter pipe should be used for the vertical riser. This combination of pipe sizes will ensure proper oil movement at both maximum and minimum system capacities, and minimize the overall pressure drop due to the suction line.

This process suggests selecting the largest-possible pipe diameter to minimize pressure drop. For some applications, however, using a smaller pipe diameter may be possible and can reduce the installed cost of the system. Of course, the smaller pipe diameter will result in a higher pressure drop and must meet the minimum and maximum velocity requirements.

For this example system, for the vertical riser, a single pipe diameter was selected that provides adequate velocity at minimum capacity without exceeding the upper velocity limit at maximum capacity. This is generally possible when the minimum capacity of the refrigeration circuit is not less than 30 to 37 percent of maximum capacity.

## period two

# Suction Line

### notes

#### Double Suction Riser

circuit capacity, tons (kW)	velocity inside 1 5/8 in. (42 mm) pipe, fpm (m/s)	minimum velocity for 1 5/8 in. (42 mm) riser, fpm (m/s)
20 (70.3)	3,250 (16.5)	840 (4.3)
5 (17.6)	810 (4.1)	
circuit capacity, tons (kW)	velocity inside 1 3/8 in. (35 mm) pipe, fpm (m/s)	minimum velocity for 1 3/8 in. (35 mm) riser, fpm (m/s)
20 (70.3)	4,600 (23.4)	780 (4.0)
5 (17.6)	1,150 (5.9)	

Figure 24

If the refrigeration circuit unloads to less than 30 to 37 percent of maximum capacity, a **double suction riser** may be required.

Using the same example 20-ton (70.3-kW) system, assume that the refrigeration circuit can, instead, unload to 5 tons (17.6 kW). At this minimum capacity, the velocity inside the 1 5/8 in. (42 mm) pipe—810 fpm (4.1 m/s)—will drop below the minimum allowable velocity for a riser of this diameter—840 fpm (4.3 m/s).

Selecting the smaller 1 3/8 in. (35 mm)-diameter pipe for the vertical riser would result in a velocity that is too high—4,600 fpm (23.4 m/s)—at maximum capacity.

#### Double Suction Riser

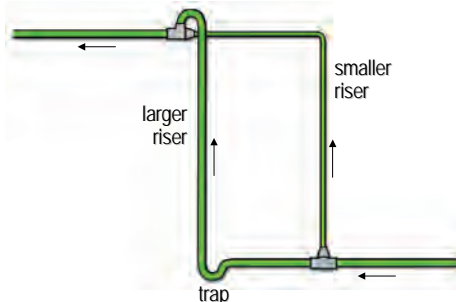


Figure 25

A double suction riser is constructed of a larger-diameter riser with a trap at the base and a smaller-diameter riser in parallel. At maximum capacity, the refrigerant vapor flows up both risers at velocities that are adequate to carry the oil droplets.



## period two

# Suction Line

### notes

At minimum capacity, the refrigerant velocity in the two risers becomes too low to carry the oil droplets. The oil from both risers therefore drains down, filling the trap at the base of the larger riser. When this trap becomes completely filled with oil, it prevents refrigerant vapor from flowing up the larger riser, and diverts all the refrigerant up the smaller riser. This smaller riser is constructed of a pipe with a diameter that is small enough to maintain adequate velocity at minimum capacity. When system capacity is increased again, the higher refrigerant velocity clears the trap of oil, and refrigerant vapor again flows up both risers.

Notice the inverted trap at the top of the larger riser. When the trap at the base of the larger riser is filled with oil, and refrigerant flows up only the smaller riser, this inverted trap prevents oil from draining back into the larger riser. This configuration minimizes the amount of oil trapped in the double riser under this condition, therefore maximizing the amount of oil that is returned to the compressor.

### Sizing a Double Suction Riser

#### ▲ Sizing the smaller riser

circuit capacity, tons (kW)	velocity inside 1 3/8 in. (35 mm) pipe, fpm (m/s)	minimum velocity for 1 3/8 in. (35 mm) riser, fpm (m/s)
5 (17.6)	1,150 (5.8)	780 (4.0)

#### ▲ Sizing the larger riser

circuit capacity, tons (kW)	velocity inside 1 5/8 in. (42 mm) pipe, fpm (m/s)	minimum velocity for 1 5/8 in. (42 mm) riser, fpm (m/s)
15 (52.8)	2,450 (12.4)	840 (4.3)

Figure 26

When sizing a double suction riser, the diameter of the smaller riser should be selected to provide the minimum allowable velocity at minimum capacity. The diameter of the larger riser should then be selected so that, at maximum capacity, the velocities in both risers are greater than the minimum allowable velocity and less than 4,000 fpm (20 m/s).

For this example system, the diameter of the smaller riser is 1 3/8 in. (35 mm). When the system operates at its minimum capacity of 5 tons (17.6 kW), the velocity inside the 1 3/8 in. (35 mm) riser—1,150 fpm (5.8 m/s)—is above the minimum allowable velocity for a riser of this diameter.

At the maximum system capacity of 20 tons (70.3 kW), the smaller riser is assumed to handle the same 5 tons (17.6 kW) of capacity, and the larger riser must handle the remaining capacity. At 15 tons (52.8 kW) of capacity, the refrigerant velocity in a 1 5/8 in. (42 mm) riser—2,450 fpm (12.4 m/s)—is between the maximum and minimum allowable velocities for a riser of this diameter.

## period two

# Suction Line

### notes

For this example system, a 2 1/8 in. (54 mm)-diameter pipe can be used for the horizontal or vertical drop sections, and the double suction riser should be constructed using 1 5/8 in. (42 mm)-diameter pipe for the larger riser and 1 3/8 in. (35 mm)-diameter pipe for the smaller riser.

Do not use a double suction riser unless it is absolutely necessary! The double riser stores a large amount of oil—oil that is preferably left in the crankcase of the compressor. If a double riser is required, check with the equipment manufacturer to determine if more oil must be added to the system.

### Calculate Total Equivalent Length

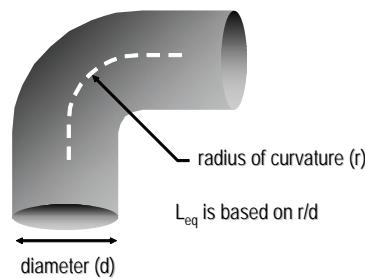


Figure 27

After the pipe diameters are selected, the overall pressure drop due to the suction line can be calculated. The next step is to calculate the total “equivalent” length of the suction line. The total equivalent length is the actual measured length of straight pipe plus the pressure drop through any fittings, such as elbows and tees. The pressure drop through the fittings is expressed in terms of the length of straight pipe that would produce the same pressure drop.

To determine the equivalent length of various copper fittings, Trane now uses the results from two decades of research conducted for the American Society of Mechanical Engineers (ASME). This research produced a method for predicting the pressure drop, in terms of equivalent length ( $L_{eq}$ ), through various types of copper fittings by using the radius of curvature ( $r$ ) of the fitting divided by its diameter ( $d$ ), or  $r/d$ . Table 1 on page 21 includes this data.

## period two

# Suction Line

## notes

<b>Table 1. Equivalent Lengths (<math>L_{eq}</math>) for Copper Pipe Fittings</b>						
nominal diameter*, in. (mm)	$L_{eq}$ , ft (m)					
	long radius elbows	short radius elbows	tee (thru flow)	tee (branch flow)	ball valve	sight glass
1/2 (12)	0.3 (0.09)	0.4 (0.12)	1.0 (0.30)	1.0 (0.30)	1.0 (0.30)	1.0 (0.30)
5/8 (15)	0.4 (0.12)	0.5 (0.15)	1.0 (0.30)	1.3 (0.40)	1.0 (0.30)	1.0 (0.30)
3/4 (18)	0.5 (0.15)	0.6 (0.18)	1.0 (0.30)	1.4 (0.43)	1.0 (0.30)	1.0 (0.30)
7/8 (22)	0.6 (0.18)	0.7 (0.21)	1.0 (0.30)	1.8 (0.55)	1.0 (0.30)	1.0 (0.30)
1 1/8 (28)	0.8 (0.24)	0.9 (0.27)	1.0 (0.30)	2.6 (0.79)	1.0 (0.30)	1.0 (0.30)
1 3/8 (35)	0.9 (0.27)	1.4 (0.43)	1.0 (0.30)	3.4 (1.04)	1.0 (0.30)	1.0 (0.30)
1 5/8 (42)	1.0 (0.30)	1.7 (0.52)	1.0 (0.30)	4.4 (1.34)	1.0 (0.30)	1.0 (0.30)
2 1/8 (54)	1.4 (0.43)	2.3 (0.70)	1.0 (0.30)	5.9 (1.80)	1.0 (0.30)	1.0 (0.30)
2 5/8 (67)	1.5 (0.46)	3.4 (1.04)	1.0 (0.30)	8.0 (2.44)	1.0 (0.30)	1.0 (0.30)
3 1/8 (79)	1.7 (0.52)	5.0 (1.52)	1.0 (0.30)	10.1 (3.08)	1.0 (0.30)	1.0 (0.30)
* For smaller diameters, use the 1/2 in. (12 mm) values						
<b>Source:</b> ASME Transactions, July 1950, page 685, table 4						

### $L_{eq}$ for Long-Radius Elbows

elbow diameter, in. (mm)	equivalent length, ft (m)	
	"old" data	"new" data
1 1/8 (28)	1.9 (0.6)	0.8 (0.24)
1 3/8 (35)	2.2 (0.7)	0.9 (0.27)
1 5/8 (42)	2.6 (0.8)	1.0 (0.30)
2 1/8 (54)	3.4 (1.0)	1.4 (0.43)
2 5/8 (67)	4.2 (1.3)	1.5 (0.46)
3 1/8 (79)	5.1 (1.6)	1.7 (0.52)

Figure 28

The research findings show that the equivalent lengths of copper fittings are much smaller than the data used in the past. The table in Figure 28 compares some of the equivalent-length data from this research to the values commonly used in the past. While this data is for long-radius elbows, the same method is used to predict the equivalent length of short-radius elbows and tees.

The suction line for this example 20-ton (70.3-kW) system, shown in Figure 23 on page 17, is 70 ft (21.4 m) long. The horizontal and vertical drop sections total 55 ft (16.8 m) in length, and are constructed of 2 1/8 in. (54 mm)-diameter pipe with three long-radius elbows and three short-radius elbows. The single vertical riser is 15 ft (4.6 m) long, and is constructed of 1 5/8 in. (42 mm)-diameter pipe with no fittings.

## period two

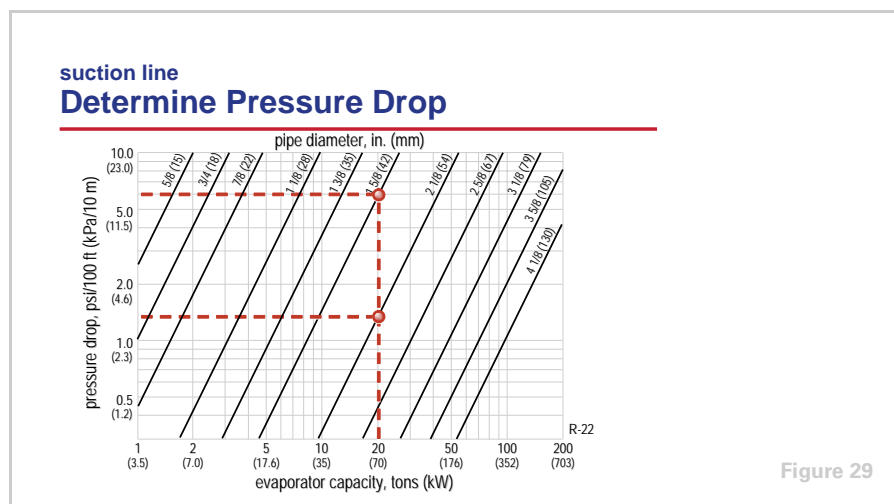
# Suction Line

## notes

Based on this “new” data, the equivalent length of each 2 1/8 in. (54 mm) long-radius elbow is 1.4 ft (0.43 m), and the equivalent length of each 2 1/8 in. (54 mm) short-radius elbow is 2.3 ft (0.70 m). The equivalent length of each fitting is then added to the actual length of straight pipe of the same diameter. The total equivalent length of suction line constructed of 2 1/8 in. (54 mm)-diameter pipe is calculated to be 66.1 ft (20.2 m), and the total equivalent length of suction line constructed of 1 5/8 in. (42 mm)-diameter pipe is 15 ft (4.6 m).

$$L_{eq} \text{ for } 2 \frac{1}{8} \text{ in. sections} = 55 \text{ ft} + (3 \times 1.4 \text{ ft}) + (3 \times 2.3 \text{ ft}) = 66.1 \text{ ft}$$

$$(L_{eq} \text{ for } 54 \text{ mm sections} = 16.8 \text{ m} + (3 \times 0.43 \text{ m}) + (3 \times 0.70 \text{ m}) = 20.2 \text{ m})$$



The next step is to determine the pressure drop due to the straight pipe and fittings, based on the calculated total equivalent length. The chart in Figure 29 shows the pressure drop, per unit of equivalent length, for different pipe diameters and various system capacities. At the maximum capacity of 20 tons (70.3 kW), the pressure drop due to the 2 1/8 in. (54 mm)-diameter sections of straight pipe and fittings is 1.4 psi per 100 ft of equivalent length (3.2 kPa/10 m), or 0.93 psi (6.4 kPa).

$$\Delta p_{2 \frac{1}{8} \text{ in.}} = 66.1 \text{ ft} \times 1.4 \text{ psi/100 ft} = 0.93 \text{ psi}$$

$$(\Delta p_{54 \text{ mm}} = 20.2 \text{ m} \times 3.2 \text{ kPa/10 m} = 6.4 \text{ kPa})$$

The pressure drop due to the 1 5/8 in. (42 mm)-diameter section of straight pipe is 6 psi per 100 ft of equivalent length (13.5 kPa/10 m), or 0.9 psi (6.2 kPa).

$$\Delta p_{1 \frac{5}{8} \text{ in.}} = 15 \text{ ft} \times 6 \text{ psi/100 ft} = 0.9 \text{ psi}$$

$$(\Delta p_{42 \text{ mm}} = 4.6 \text{ m} \times 13.5 \text{ kPa/10 m} = 6.2 \text{ kPa})$$

## period two

# Suction Line

## notes

Therefore, the total pressure drop through the straight pipe and fittings for this example suction line is 1.83 psi (12.6 kPa).

$$\Delta p = 0.93 \text{ psi} + 0.9 \text{ psi} = 1.83 \text{ psi}$$

$$(\Delta p = 6.4 \text{ kPa} + 6.2 \text{ kPa} = 12.6 \text{ kPa})$$

### suction line

### Add Pressure Drop of Accessories

- ▲ Suction filter = 2 psi (13.8 kPa)
- ▲ Angle valve = 1 psi (6.9 kPa)
- ▲ Ball valve = 1 equivalent ft (0.3 m)

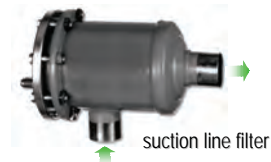


Figure 30

The final step toward calculating the total pressure drop is to add the pressure drop due to any accessories that are to be installed in the suction line. The most common accessories installed in a suction line are a suction filter and manual ball valves that can be used to isolate this filter when it needs to be replaced.

Assuming that this example system includes one suction filter and two manual ball valves installed in the 2 1/8 in. (54 mm) line, the total pressure drop due to the suction line is 3.86 psi (26.6 kPa).

$$\Delta p = 1.83 \text{ psi} + 2 \text{ psi} + (2 \times 1 \text{ ft} \times 1.4 \text{ psi}/100 \text{ ft}) = 3.86 \text{ psi}$$

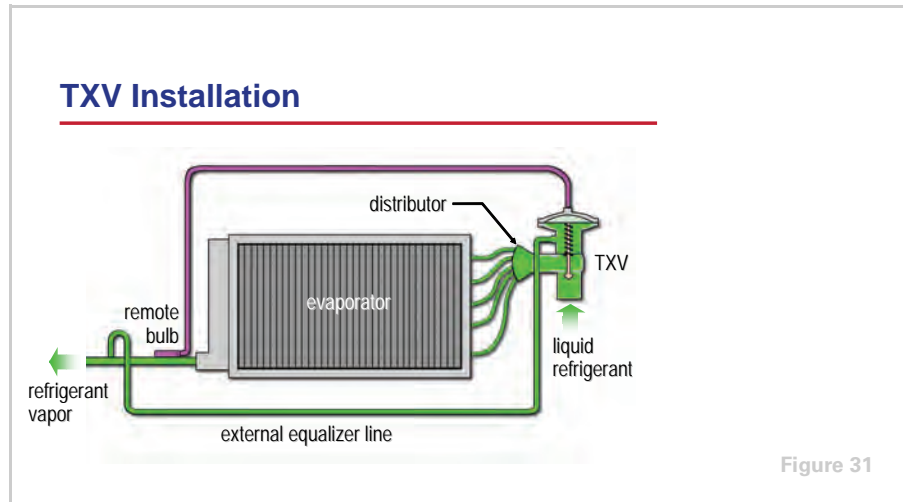
$$(\Delta p = 12.6 \text{ kPa} + 13.8 \text{ kPa} + (2 \times 0.3 \text{ m} \times 3.2 \text{ kPa}/10 \text{ m}) = 26.6 \text{ kPa})$$

As mentioned earlier, a common rule in the past was to limit the pressure drop through the suction line to 3 psi (20 kPa) for a system operating with R-22. However, long line lengths, and the associated higher pressure drops, can be tolerated assuming that the impact on system capacity and efficiency is acceptable for the given application.

## period two

# Suction Line

## notes



### Other Considerations

Certain precautions should be taken when routing the suction line. First, proper location and attachment of the **remote expansion-valve bulb** is very important. This remote bulb, which measures the temperature of the refrigerant leaving the evaporator, should be firmly attached to a straight, well-drained, horizontal section of the suction line. The **external equalizer line** should be inserted downstream of the remote bulb, in order to prevent influencing the temperature measured by the bulb due to any leakage of liquid refrigerant through the equalizer line.

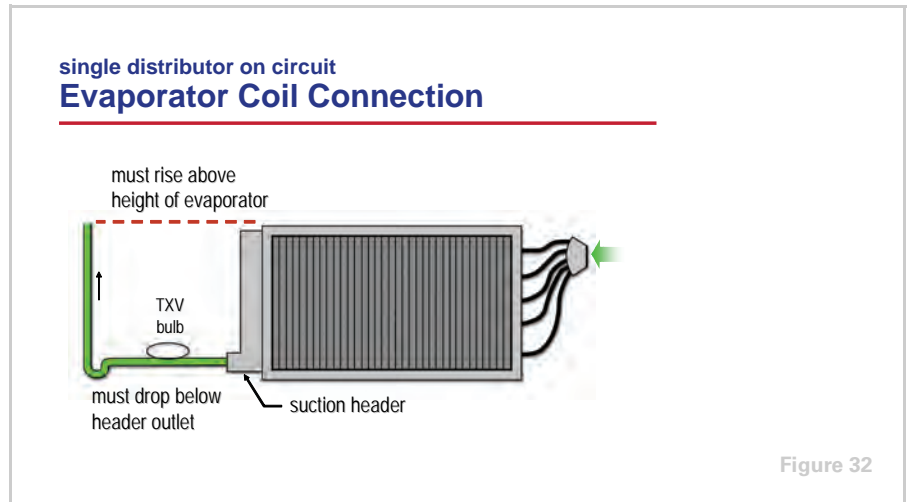
The section of pipe at the outlet of the evaporator must be long enough to permit the attachment of the remote bulb and the insertion of the external equalizer line. A good rule is to allow 12 in. (300 mm) of straight horizontal pipe for these two connections. Under no circumstances should the remote bulb be located on a section of pipe where oil or liquid refrigerant could be trapped. The presence of oil or liquid refrigerant can cause false temperature readings. Slightly pitch this short horizontal section of pipe downward from the suction header—1 in./10 ft (10 mm/3 m) in the direction of flow. For best temperature sensing, the remote bulb should be firmly attached to the top of this section of pipe and well insulated.

In addition, an **access port** should be located near the external equalizer line connection. This port provides a point for accurate pressure measurement when checking or adjusting the superheat setting of the TXV.

## period two

# Suction Line

## notes



After this short horizontal section, the suction line should drop vertically downward to allow the evaporator, and the section of pipe with the TXV bulb attached, to drain freely when the system is operating.

If the suction line leaves the evaporator and then must rise immediately, this can be accomplished by using a small trap at the end of the horizontal section of pipe, just before the suction line rises. The purpose of this trap is to provide free drainage from the evaporator and the section of pipe to which the TXV bulb is attached. This ensures that the TXV bulb is not the “low spot” in the piping where oil or liquid refrigerant could be trapped. The purpose of this trap is *not* to drain the suction riser.

The suction line must then rise above the height of the evaporator coil. This prevents refrigerant and oil inside the evaporator from free draining into the suction line, and toward the compressor, when the system is off.

The configuration in Figure 32 is typical for a system with an indoor air handler, where the refrigerant piping is routed along the ceiling and must drop down to the evaporator.

## period two

# Suction Line

## notes

### single distributor on circuit Evaporator Coil Connection

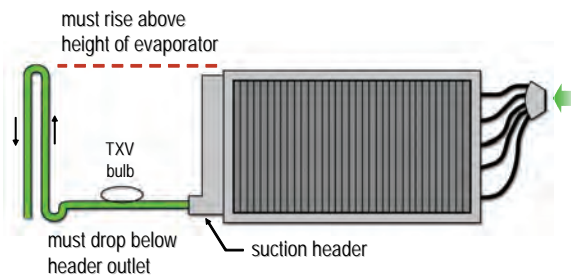


Figure 33

If the suction line leaves the evaporator and then must drop vertically, an inverted trap is used to prevent refrigerant and oil inside the evaporator from free draining into the suction line, and toward the compressor, when the system is off.

This involves piping the suction line to rise above the height of the evaporator coil, installing an inverted trap, and then allowing the line to drop back down below the evaporator.

### multiple distributors on circuit Evaporator Coil Connections

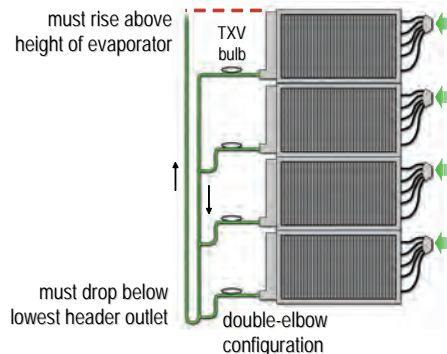


Figure 34

When an evaporator has more than one distributor connected to a single refrigeration circuit, there are some additional considerations for manifolding the suction headers into a single suction line.

First, arrange the suction line so that the refrigerant vapor leaving the top suction header flows downward, connecting to the pipes leaving the other suction headers. This manifold pipe should drop below the lowest suction



## period two

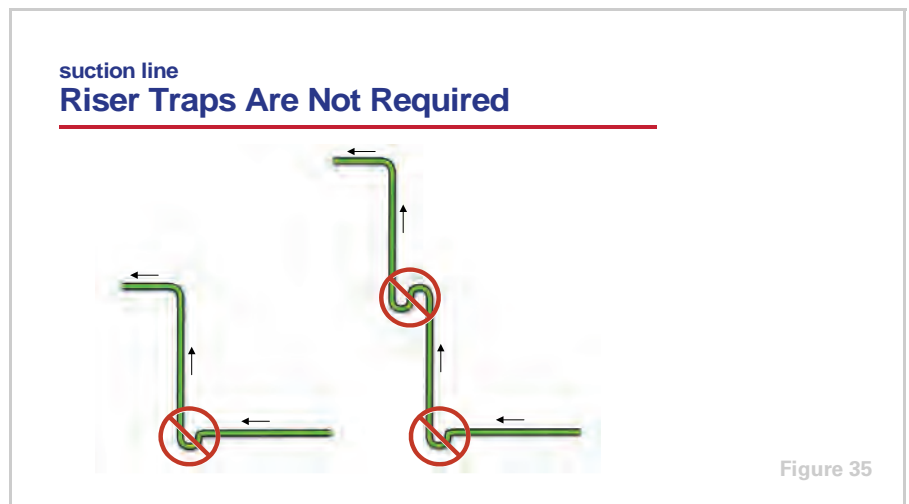
# Suction Line

### notes

header outlet before being allowed to turn upward. This allows all of the oil to drain out of the evaporator and congregate before it must be carried up a vertical riser.

Second, the remote TXV bulbs and external equalizing line connections must be located so that the conditions affecting one valve cannot influence the other valves. The double-elbow configuration allows the corresponding section of the evaporator coil to drain freely while also isolating the remote bulb and equalizer line from the suction pressure and temperature of the above coil sections.

The single suction line then rises above the height of the evaporator coil, preventing refrigerant and oil inside the evaporator from free draining into the suction line, and toward the compressor, when the system is off.



As mentioned earlier, the trap shown in Figure 32 on page 25, and Figure 33 on page 26, is used to provide free drainage from the evaporator and section of pipe to which the TXV bulb is attached. Its purpose is *not* to drain the suction riser.

A common practice in the past was to install a trap at the base of any vertical suction riser, and often part of the way up the riser. It was believed that these traps would cause turbulence at the base of the riser section so that oil accumulating in the trap would be more easily “broken up” into a mist, which could then be carried to the top of the riser with the refrigerant vapor.

Today, we understand that oil droplets are moved inside a pipe by the force of mass flow, not by turbulence. It is possible to size a vertical suction riser, which will allow the refrigerant to carry oil droplets at minimum capacity, without the need for a trap. If a suction riser is sized properly, oil will return to the compressor regardless of whether a trap is present at the base of the riser. If a suction riser is oversized, however, adding such a trap will not restore proper oil movement.

## period two

# Suction Line

### notes

#### suction line

#### Other Considerations

- ▲ **Do not use suction riser traps: they are not required**
- ▲ **Pitch horizontal sections to drain toward evaporator**
- ▲ **Insulate entire suction line**
  - ◆ Prevents condensation
  - ◆ Minimizes loss of capacity due to heat gain
- ▲ **Install suction-line filter close to compressor**
  - ◆ Manual shutoff valves allow isolation for replacement
- ▲ **Install access ports to measure suction pressure and superheat**

Figure 36

Horizontal sections of the suction line should be pitched slightly—1 in./10 ft (10 mm/3 m)—so that the refrigerant drains back toward the evaporator. This prevents any refrigerant that condenses in the suction line from flowing into the compressor when the circuit is off.

The refrigerant temperature inside the suction line is generally cooler than the surrounding air. Therefore, it is always good practice to insulate the entire suction line to prevent condensation and loss of capacity due to heat gain. Without insulation, moisture may condense on the outside surface of the pipe and drip onto the floor or suspended ceiling below. Also, any heat gained by the suction line places an additional load on the system that reduces cooling capacity, and also may result in improper motor cooling with some compressor designs.

The **suction-line filter** should be located as close as possible to the compressor—its purpose is to remove foreign matter from the system and protect the compressor. This filter should be accompanied by manual shutoff valves on both sides, allowing it to be isolated when the filter core needs to be replaced.

Additionally, the suction line typically includes two access ports. One is installed near the compressor and is used to measure suction pressure. The other is located near the external equalizer line connection for the TXV, and is used to measure superheat when checking or adjusting the TXV.

## period two

# Suction Line

## notes

### Suction-Line Accumulator

- ▲ Check with equipment manufacturer to determine if required, recommended, or discouraged



photo provided by Henry Technologies

Figure 37

The compressor is designed to compress refrigerant vapor only. A **suction-line accumulator** is a device that attempts to prevent a slug of liquid refrigerant or oil from causing damage to the compressor. The accumulator allows liquid refrigerant and oil to separate from the refrigerant vapor, and then be drawn into the compressor at a rate that will not cause damage.

Oversizing the accumulator, however, can cause inadequate oil return due to low refrigerant velocity through the accumulator. An accumulator also increases the refrigerant charge of the system and increases the pressure drop.

Some refrigeration systems that use a flooded evaporator may require a suction-line accumulator for freeze protection or other functions. Check with the equipment manufacturer to determine if a suction-line accumulator is required, recommended, or discouraged.

## period two

# Suction Line

## notes

### suction line Reciprocating Compressors

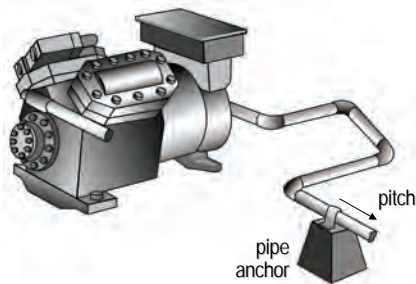


Figure 38

Finally, when the system includes a reciprocating compressor, an anchored, 45-degree canted loop should be constructed very close to the location at which the suction line connects to the compressor. This loop absorbs vibration generated by the reciprocating motion of the compressor, minimizing the amount of vibration transmitted to the piping system.

The loop is constructed of four straight sections of pipe, each one at least ten pipe diameters in length, and four elbows. The pipe is anchored a short distance downstream from the loop, allowing the loop to absorb vibration in both the vertical and horizontal planes. Note that the loop is pitched downward to provide free drainage away from the compressor.

## period three

# Discharge Line

## notes

### Refrigerant Piping

*period three*

Discharge Line

Figure 39

The next line to be considered is the discharge line. This section of pipe conducts hot, high-pressure refrigerant vapor from the compressor to the condenser.

The design of the discharge line is probably less critical than that of the suction line because the refrigerant vapor is at a higher temperature, allowing the oil to be carried along more easily than in the cooler suction line. Even so, if the discharge line is not properly sized and installed, reliability or performance problems can result.

*discharge line*

### Requirements for Sizing and Routing

- ▲ Ensure adequate velocity to return oil to compressor at all steps of unloading
- ▲ Avoid excessive noise
- ▲ Minimize efficiency loss

Figure 40

### Requirements for Sizing and Routing

Similar to the suction line, the diameter of the discharge line must be small enough that the resulting refrigerant velocity is sufficiently high to carry oil droplets, at all steps of compressor unloading. However, if the velocity in the pipe is too high, it may cause objectionable noise.

## period three

# Discharge Line

### notes

Finally, anything that causes the compressor discharge pressure to rise causes the compressor to work harder. Therefore, the pipe diameter should be as large as possible to minimize pressure drop and maximize compressor efficiency.

#### discharge line Sizing “Rules”

##### old rules for R-22

###### minimum velocity

1,000 fpm (5 m/s)  
for risers

500 fpm (2.5 m/s)  
for horizontal sections

###### maximum velocity

3,500 fpm (17.5 m/s)

###### maximum pressure drop

6 psi (41 kPa)

##### new rules for R-22

based on diameter of riser

percentage of minimum  
riser velocity

3,500 fpm (17.5 m/s)

based on specific system  
requirements

Figure 41

As with the suction line, it may be helpful to compare the old “rules” for selecting the diameter of the discharge line with the newer rules that result from changes in compressor technology, recent research, and the additional requirement to minimize system refrigerant charge.

In the past, many discharge lines for systems operating with R-22 were sized using the same minimum-velocity values that were used for sizing suction lines. As explained in Period Two, the minimum allowable velocity at a specific operating condition depends on the diameter of the pipe. The minimum allowable velocity values for discharge lines are different than for suction lines, because the refrigerant vapor is at a higher temperature, and oil moves more easily than in the cooler suction line.

The recommended maximum velocity limit for discharge lines is 3,500 fpm (17.5 m/s). Again, a higher velocity inside the pipe may cause objectionable noise.

Another “old” rule, for a system operating with R-22, was to limit the pressure drop through the discharge line to 6 psi (41 kPa). While it is still good practice to minimize pressure drop, higher pressure drops can be tolerated, assuming that the impact on system efficiency is acceptable for the given application.

## period three

# Discharge Line

## notes

discharge line

### Process for Sizing

- 1 Determine total length of piping
- 2 Calculate refrigerant velocity at maximum and minimum capacities
- 3 Select largest pipe diameter that results in acceptable velocity at both maximum and minimum capacity
- 4 Calculate total equivalent length of straight pipe and fittings
- 5 Determine pressure drop due to pipe and fittings
- 6 Add pressure drop due to accessories

Figure 42

### Sizing Process

The steps for selecting the proper diameter of the discharge line are identical to the steps for sizing the suction line:

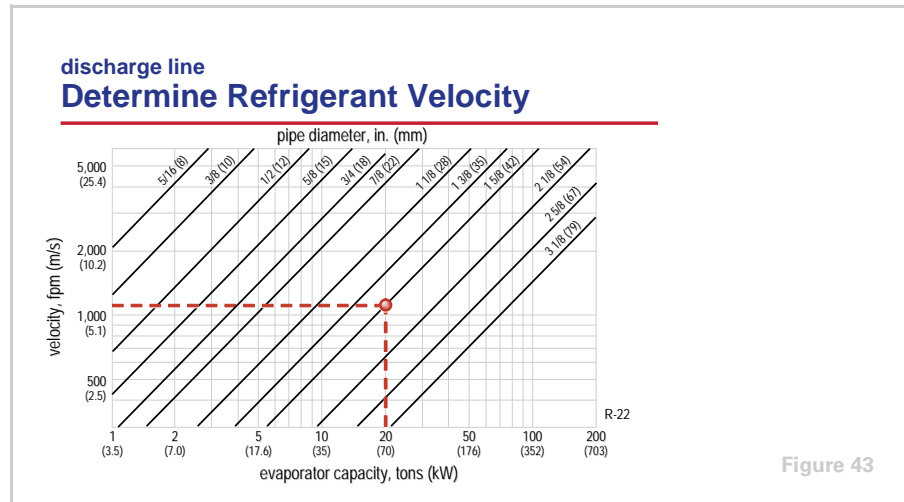
- 1) Determine the total length of discharge-line piping.
- 2) Calculate the refrigerant velocity at both maximum and minimum system capacities.
- 3) Select the largest pipe diameter that will result in acceptable refrigerant velocity at both maximum and minimum capacities.
- 4) Calculate the total "equivalent" length of piping by adding the actual length of straight pipe to the equivalent length of any fittings to be installed in the discharge line.
- 5) Determine the pressure drop (based on the total equivalent length) due to the straight pipe and fittings.
- 6) Add the pressure drop due to any accessories installed in the discharge line.

Again, the total measured length of the discharge line can be estimated from the initial layout.

## period three

# Discharge Line

### notes



The chart in Figure 43 shows the velocity of R-22 inside discharge pipes of various diameters at one particular operating condition—40°F (4.4°C) saturated suction temperature, 125°F (51.7°C) saturated condensing temperature, 12°F (6.7°C) of superheat, 15°F (8.3°C) of subcooling, and 70°F (38.9°C) of compressor superheat.

For the same example 20-ton (70.3-kW) system, the refrigerant velocity inside a 1 5/8 in. (42 mm)-diameter pipe is about 1,100 fpm (5.6 m/s) at this condition.

Again, a computer program can be used to calculate the velocity based on actual conditions, but without a program, a chart like this may be useful.

**discharge line**  
**Determine Refrigerant Velocity**

pipe diameter, in. (mm)	velocity, fpm (m/s)	
	20 tons (70.3 kW)	10 tons (35.2 kW)
7/8 (22)	4,050 (20.6)	2,025 (10.3)
1 1/8 (28)	2,400 (12.2)	1,200 ( 6.1)
1 3/8 (35)	1,550 ( 7.9)	775 ( 3.9)
1 5/8 (42)	1,100 ( 5.6)	550 ( 2.8)
2 1/8 (54)	650 ( 3.3)	325 ( 1.7)

Figure 44

This example 20-ton (70.3-kW) system contains one refrigeration circuit with two steps of capacity. Maximum capacity is 20 tons (70.3 kW) and minimum capacity is 10 tons (35.2 kW). Using the chart in Figure 43, the refrigerant



## period three

# Discharge Line

### notes

velocity at both maximum and minimum capacities is determined for several pipe diameters.

After these velocities have been determined, the largest acceptable pipe diameter is selected to minimize the overall pressure drop due to the discharge line.

When this system operates at maximum capacity, use of the 7/8 in. (22 mm)-diameter pipe results in a refrigerant velocity that is greater than the recommended upper limit of 3,500 fpm (17.5 m/s) for discharge lines. This high velocity may cause objectionable noise, so this pipe size should probably not be considered.

#### discharge line

### Minimum Allowable Velocities

pipe diameter, in. (mm)	minimum velocity, fpm (m/s)	
	riser	horiz/drop
5/16 (8)	220 (1.1)	165 (0.8)
3/8 (10)	250 (1.3)	185 (0.9)
1/2 (12)	285 (1.4)	215 (1.1)
5/8 (15)	315 (1.6)	235 (1.2)
3/4 (18)	345 (1.8)	260 (1.3)
7/8 (22)	375 (1.9)	285 (1.4)
1 1/8 (28)	430 (2.2)	325 (1.7)
1 3/8 (35)	480 (2.4)	360 (1.8)
1 5/8 (42)	520 (2.6)	390 (2.0)
2 1/8 (54)	600 (3.0)	450 (2.3)
2 5/8 (67)	665 (3.4)	500 (2.5)
3 1/8 (79)	730 (3.7)	550 (2.8)

Figure 45

The table in Figure 45 shows the minimum allowable refrigerant velocity, for both a vertical riser and a horizontal (or vertical drop) section of discharge line, for each standard pipe diameter. As with suction lines, the minimum velocity for a horizontal, or vertical drop, section of discharge line is 75 percent of the minimum allowable velocity for a vertical riser of the same diameter.

The minimum velocities listed in this table assume a worst-case operating condition of 80°F (26.7°C) saturated condensing temperature.

## period three

# Discharge Line

## notes

### Select Discharge Line Size

circuit capacity, tons (kW)	velocity inside 1 5/8 in. (42 mm) pipe, fpm (m/s)	minimum velocity for 1 5/8 in. (42 mm) pipe, fpm (m/s)	
		riser	horiz/drop
20 (70.3)	1,100 (5.6)	520 (2.6)	390 (2.0)
10 (35.2)	550 (2.8)		

Figure 46

Based on the minimum allowable velocities listed in Figure 45 on page 35, 1 5/8 in. (42 mm) is the largest diameter of pipe that will result in acceptable velocity, at both maximum and minimum capacities, within horizontal and vertical drop sections.

When the circuit unloads to 10 tons (35.2 kW), the velocity inside a 1 5/8 in. (42 mm) pipe—550 fpm (2.8 m/s)—remains above the minimum allowable velocity for a riser of this diameter—520 fpm (2.6 m/s). One size larger (2 1/8 in. [54 mm]) would result in a velocity that is too low when the system operates at minimum capacity.

### Selected Discharge Line Size

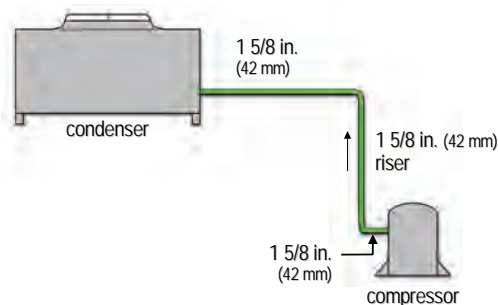


Figure 47

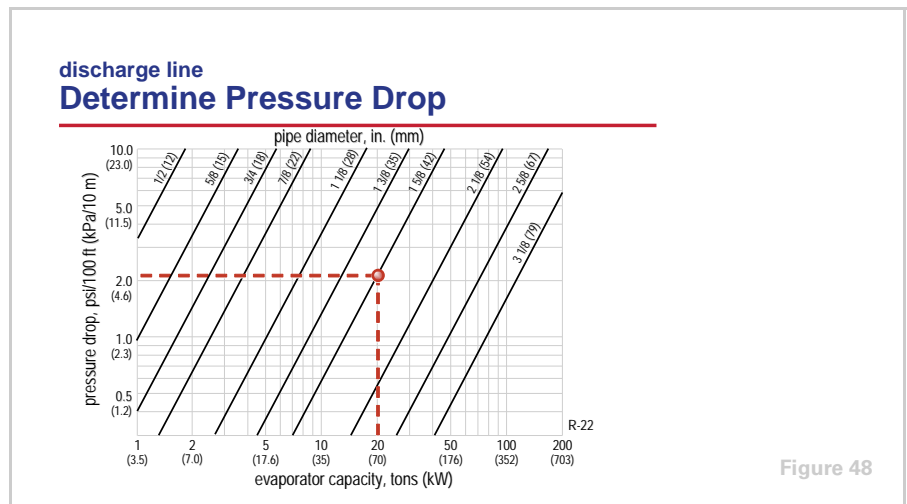
For this example system, a 1 5/8 in. (42 mm)-diameter pipe can be used for all horizontal and vertical sections of the discharge line, including vertical risers. This pipe size will ensure proper oil movement at both maximum and minimum system capacities, and minimize the overall pressure drop due to the discharge line.

## period three

# Discharge Line

## notes

For this example, a single pipe diameter can be selected that provides adequate velocity at minimum capacity, without exceeding the maximum velocity limit at maximum capacity. If the maximum and minimum velocity requirements cannot be achieved with a single pipe diameter, a double riser may be required.



After the pipe diameter is selected, the overall pressure drop due to the discharge line can be calculated. The next step is to calculate the total "equivalent" length of the straight pipe and fittings. The equivalent length data for copper fittings used in the discharge line is the same as for the suction line.

The discharge line for this example 20-ton (70.3-kW) system, shown in Figure 47 on page 36, is a total of 50 ft (15.2 m) long. Both the horizontal sections and the vertical riser are constructed of 1 5/8 in. (42 mm)-diameter pipe and contain a total of two long-radius elbows. Based on the "new" data from Table 1 on page 21, the equivalent length of each long-radius elbow of this diameter is 1.0 ft (0.30 m), so the total equivalent length of the discharge line is 52 ft (15.8 m).

$$L_{eq} = 50 \text{ ft} + (2 \times 1.0 \text{ ft}) = 52 \text{ ft}$$

$$(L_{eq} = 15.2 \text{ m} + (2 \times 0.30 \text{ m}) = 15.8 \text{ m})$$

The next step is to determine the pressure drop due to the straight pipe and fittings, based on this calculated total equivalent length. The chart in Figure 48 shows the pressure drop, per unit of equivalent length, for different pipe diameters and various capacities. At the maximum capacity of 20 tons (70.3 kW), the pressure drop due to the straight, 1 5/8 in. (42 mm)-diameter pipe and fittings is 2.1 psi per 100 ft of equivalent length (4.8 kPa/10 m), or 1.1 psi (7.6 kPa).

$$\Delta p = 52 \text{ ft} \times 2.1 \text{ psi/100 ft} = 1.1 \text{ psi}$$

$$(\Delta p = 15.8 \text{ m} \times 4.8 \text{ kPa/10 m} = 7.6 \text{ kPa})$$

## period three

# Discharge Line

### notes

discharge line

#### **Add Pressure Drop of Accessories**

- ▲ Angle valve = 1 psi (6.9 kPa)
- ▲ Ball valve = 1 equivalent ft (0.3 m)

Figure 49

The final step is to add the pressure drop due to any accessories that will be installed in the discharge line, and calculate the total pressure drop. The most common accessories installed in a discharge line are manual ball valves that are used to isolate portions of the system during maintenance.

Assuming that this example system includes one manual angle valve, the total pressure drop due to the discharge line is 2.1 psi (14.5 kPa).

$$\Delta p = 1.1 \text{ psi} + 1 \text{ psi} = 2.1 \text{ psi}$$

$$(\Delta p = 7.6 \text{ kPa} + 6.9 \text{ kPa} = 14.5 \text{ kPa})$$

As mentioned earlier, in the past, a common rule was to limit the pressure drop through the discharge line to 6 psi (41 kPa) for a system operating with R-22. However, long line lengths, and the associated higher pressure drops, can be tolerated assuming that the impact on system efficiency is acceptable for the given application.

## period three

# Discharge Line

## notes

discharge line

### Other Considerations

- ▲ Do not use discharge riser traps: they are not required
- ▲ Pitch horizontal sections to drain toward condenser
- ▲ Insulate only to prevent accidental burns or when system must operate in cold ambient conditions

Figure 50

### Other Considerations

As with the suction line, riser traps are not required in a discharge line, assuming that the riser is properly sized to return oil at minimum capacity.

Horizontal sections of the discharge line should be pitched slightly—1 in./10 ft (10 mm/3 m)—so that the refrigerant drains toward the condenser. This prevents any refrigerant that may condense in the discharge line from flowing back into the compressor when the system is off.

In most cases, the discharge line does not need to be insulated. The refrigerant temperature inside the line is generally warmer than the surrounding air. Any heat loss reduces the heat-rejection load of the condenser, improving system efficiency. However, the outer surface of the discharge line can be hot. Some sections of the discharge line may be insulated to prevent injury to someone who may come in contact with the piping. Also, if the system has long refrigerant lines and is expected to operate when it is cold outdoors, the discharge line should be insulated to prevent refrigerant from condensing inside the line before it reaches the condenser.

period three

## Discharge Line

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### notes

discharge line

#### Reciprocating Compressors

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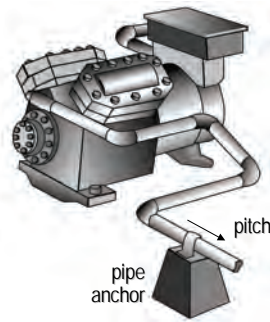


Figure 51

Finally, when the system includes a reciprocating compressor, an anchored, 45-degree canted loop should be constructed very close to the location at which the discharge line connects to the compressor. This loop absorbs vibration generated by the reciprocating motion of the compressor. Note that the loop is pitched downward to provide free drainage away from the compressor.

## period four

# Liquid Line

### notes

#### Refrigerant Piping

*period four*

Liquid Line

Figure 52

The third line to be considered is the liquid line. This section of pipe conducts warm, high-pressure liquid refrigerant from the condenser to the expansion device and evaporator.

*liquid line*

#### Requirements for Sizing and Routing

- ▲ Ensure that only liquid refrigerant enters expansion device
- ▲ Minimize refrigerant charge
- ▲ Avoid excessive noise and pipe erosion

Figure 53

#### Requirements for Sizing and Routing

The liquid line must be designed and installed to ensure that only liquid refrigerant (no vapor) enters the expansion device. The presence of refrigerant vapor upstream of a TXV can result in erratic valve operation and reduced system capacity. In order to meet this requirement, the condenser must provide adequate subcooling, and the pressure drop through the liquid line and accessories must not be high enough to cause flashing upstream of the expansion device. Subcooling allows the liquid refrigerant to experience some pressure drop as it flows through the liquid line, without the risk of flashing.

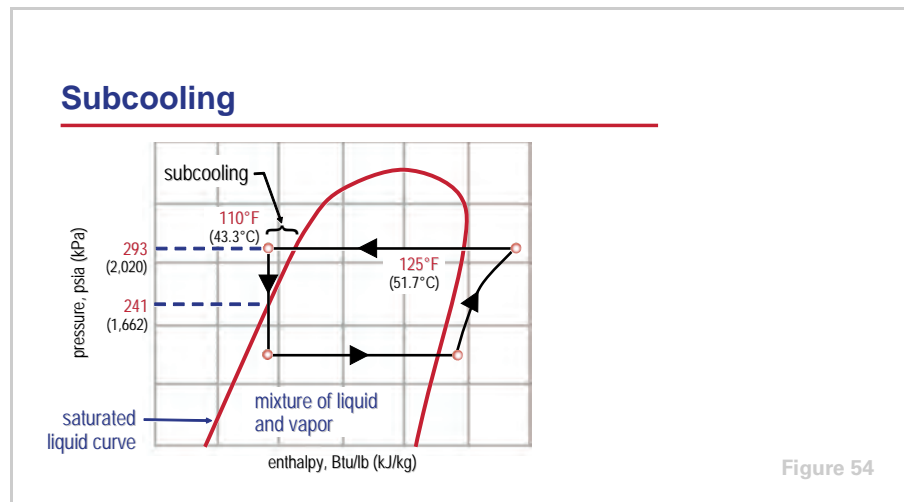
## period four

# Liquid Line

### notes

Oil and liquid refrigerant mix readily, so oil movement within the liquid line is not a concern. However, the design of the liquid line is the most critical when it comes to minimizing the system refrigerant charge. This is because, of the three lines, it has the greatest impact on the quantity of refrigerant required to charge the system. The diameter of the liquid line must be as small as possible to minimize the refrigerant charge, therefore improving reliability and minimizing installed cost. However, if the pipe is too small, the increased pressure drop may cause flashing upstream of the expansion device.

The final requirement is to limit the refrigerant velocity in order to prevent objectionable noise or erosion of the inner surfaces of the piping.



In this example system, the refrigerant (R-22) is condensed at a temperature of 125°F (51.7°C), which corresponds to a pressure of 293 psia (2,020 kPa). Inside the subcooling tubes, the temperature of the liquid refrigerant is reduced by 15°F (8.3°C) to 110°F (43.3°C). Ignoring the pressure drop due to the condenser and subcooler tubes, the pressure of this subcooled liquid refrigerant is the same, 293 psia (2,020 kPa).

At 110°F (43.3°C), the pressure of the refrigerant can be reduced to 241 psia (1,662 kPa) before it reaches the saturated liquid condition, where flashing occurs. In other words, the liquid refrigerant could experience a pressure drop of up to 52 psi (358 kPa) before flashing occurs.

In years past, the liquid line was considered an independent component of the system and was sized based solely on the physics of fluid flow through the pipe. It was believed that increasing the diameter of the liquid line, which would reduce the pressure drop through the piping, would increase the amount of subcooling remaining at the inlet of the expansion device.

The internal volume of the liquid line, however, has a significant impact on the ability of the condenser to provide subcooling at part-load operating conditions. When this interaction between the liquid line and condenser is considered, we find that the amount of subcooling available at the expansion device is as dependent on the ability of the condenser to provide subcooling as it is on the



## period four

# Liquid Line

### notes

pressure drop due to the liquid-line piping. In fact, if the liquid line is too large, it will actually reduce the subcooling at the expansion device under cooler ambient conditions.

Therefore, proper sizing of the liquid line requires an understanding of the entire refrigeration system, not just the liquid-line piping.

#### Impact of Refrigerant Density Change

	liquid line diameter, in. (mm)	outdoor temperature	available subcooling, °F (°C)		
			leaving subcooler	loss due to piping	remaining at TXV
A	7/8 (22)	high	18.0 (10)	- 5.7 (3.2)	= 12.3 (6.8)
B	1 1/8 (28)	high	18.0 (10)	- 3.8 (2.1)	= 14.2 (7.9)
A	7/8 (22)	low	12.9 (7.2)	- 7.4 (4.1)	= 5.5 (3.1)
B	1 1/8 (28)	low	7.2 (4.0)	- 5.2 (2.9)	= 2.0 (1.1)

Figure 55

Take a look at an example 25-ton (87.9-kW) system with 150 ft (45.7 m) of liquid-line piping to illustrate this point. System **A** uses a 7/8 in. (22 mm)-diameter pipe for the liquid line, while system **B** uses a larger 1 1/8 in. (28 mm)-diameter pipe. Both systems are charged so that the condenser provides 18°F (10°C) of subcooling at the same high outdoor temperature. Even though the rest of the system components are identical, system **B** requires 25 lb (11.3 kg) more refrigerant to charge the system because of the larger liquid line.

As expected, the pressure drop due to the piping is less for system **B** with the larger diameter pipe. When the outdoor temperature is high, this reduced pressure drop results in more subcooling remaining at the expansion device—14.2°F (7.9°C) compared to 12.3°F (6.8°C) for system **A**.

However, one should examine what happens when the system operates at a cooler outdoor temperature. As the outdoor temperature decreases, the temperature of the refrigerant inside the condenser and liquid line also decreases. This lower temperature causes the density of the refrigerant to increase, requiring more mass of refrigerant in order to fill up the internal volume of the liquid line. The result is less refrigerant remaining inside the condenser, and therefore less subcooling provided by the condenser.

This change in density has a more severe impact on system **B** with the larger-diameter liquid line, because the internal volume of the liquid line is greater. For these two example systems, when the outdoor temperature is cooler, the amount of subcooling provided by the same condenser is 12.9°F (7.2°C) for system **A**, but only 7.2°F (4°C) for system **B**.

Even though the pressure drop is still less through the larger-diameter line—5.2°F (2.9°C) loss in subcooling due to pressure drop in system **B** versus 7.4°F

## period four

# Liquid Line

### notes

(4.1°C) in system **A**—the impact of the change in refrigerant density has a greater impact on the overall system. At this cooler outdoor condition, only 2.0°F (1.1°C) of subcooling is available at the TXV in system **B**, while 5.5°F (3.1°C) of subcooling is available at the TXV in system **A**.

Again, proper sizing of the liquid line requires an understanding of the entire refrigeration system, not just the liquid-line piping. The diameter of the liquid line should be as small as possible, to minimize the impact of the change in refrigerant density. If the liquid line is too large, it may result in too little subcooling at some part-load operating conditions.

#### liquid line Sizing “Rules”

##### old rules for R-22

##### maximum pressure drop

6 psi (41 kPa)

##### maximum velocity

360 fpm (1.8 m/s)

##### new rules for R-22

5°F (2.8°C) subcooling  
remaining at inlet to expansion  
device at all operating conditions

600 fpm (3 m/s)

Figure 56

Figure 56 compares the old “rules” for selecting the diameter of the liquid line with the newer rules that result from this understanding of the impact of refrigerant density change, recent research, and the additional requirement to minimize system refrigerant charge.

In the past, many liquid lines in systems operating with R-22 were sized to limit the pressure drop through the piping and accessories to 6 psi (41 kPa). However, as we just explained, limiting the pressure drop through the liquid line does not necessarily ensure that only liquid refrigerant (no vapor) enters the expansion device. This rule has been rewritten to specify that adequate subcooling must be available at the inlet of the expansion device *at all operating conditions*, acknowledging this interaction between the liquid line and condenser. The condenser must provide adequate subcooling, and the pressure drop through the liquid line and accessories must not be high enough to cause flashing upstream of the expansion device. A margin of safety, such as 5°F (2.8°C), may be used to account for unexpected variations.

In the past, the recommended maximum velocity limit for a liquid line was 360 fpm (1.8 m/s). Velocities in excess of this limit caused concern for liquid-hammer effect when the flow of refrigerant was suddenly stopped, such as when a solenoid valve closed. However, most solenoid valves used in air-conditioning systems today are not quick-closing valves. That is, they do not close rapidly enough to cause a serious hammer effect.

## period four

# Liquid Line

### notes

The new maximum-velocity limit of 600 fpm (3 m/s) is based on preventing erosion of the inner surfaces of the pipe.

#### liquid line

#### Process for Sizing

- 1 Determine total length of piping
- 2 Obtain subcooling provided by condenser
- 3 Determine refrigerant velocity
- 4 Select smallest pipe diameter that results in acceptable velocity at maximum capacity
- 5 Calculate total equivalent length of pipe and fittings
- 6 Determine pressure drop due to pipe and fittings
- 7 Add pressure drop due to accessories
- 8 Calculate loss of subcooling due to pressure drop and elevation change

Figure 57

#### Sizing Process

Following are the steps to follow when selecting the proper diameter of the liquid line:

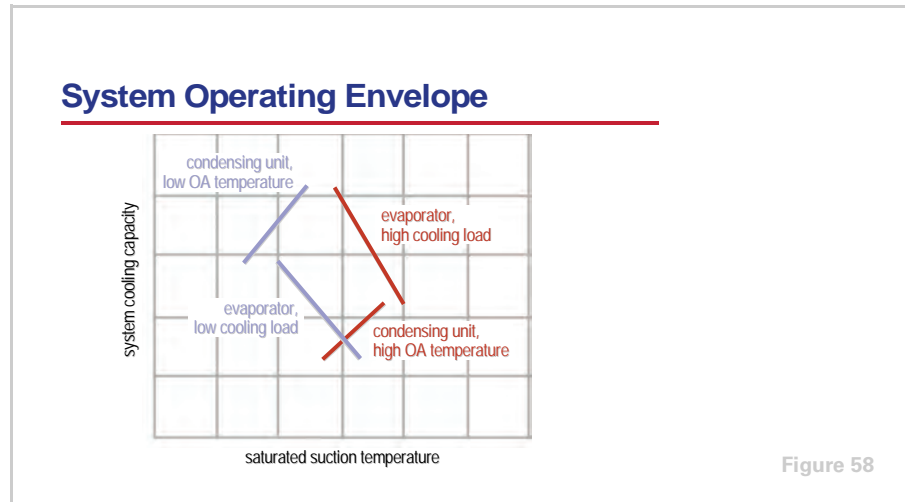
- 1) Determine the total length of liquid-line piping.
- 2) Obtain the amount of subcooling provided by the condenser from the manufacturer.
- 3) Determine the refrigerant velocity at maximum system capacity.
- 4) Select the smallest pipe diameter that will result in acceptable refrigerant velocity at maximum system capacity.
- 5) Calculate the total equivalent length of piping by adding the actual measured length of straight pipe to the equivalent length of any fittings to be installed in the liquid line.
- 6) Determine the pressure drop (based on the total equivalent length) due to the straight pipe and fittings.
- 7) Add the pressure drop due to any accessories installed in the liquid line.
- 8) Calculate the loss of subcooling due to pressure drop and to any change in elevation, to verify adequate subcooling at the inlet to the expansion device.

From the initial layout, the total measured length of the liquid line can be estimated. Again, the refrigerant piping should be routed in the shortest and simplest manner possible, minimizing the total length of piping.

## period four

# Liquid Line

### notes



The next step is to determine how much subcooling is provided by the condenser. This is difficult because the amount of subcooling provided by an air-cooled condenser depends on many factors, including the total amount of heat that must be rejected, the outdoor dry-bulb temperature, the internal volume of the liquid line, and the system refrigerant charge.

To demonstrate, look at an example operating envelope for a system comprised of an air-cooled condensing unit (compressors and condenser packaged together) and a cooling coil (evaporator). The operating envelope is the entire range of cooling loads for the system over the entire range of outdoor temperatures at which the system will be operating. Identifying the extreme corners of this system operating envelope helps us to understand how the amount of subcooling provided by the condenser varies.

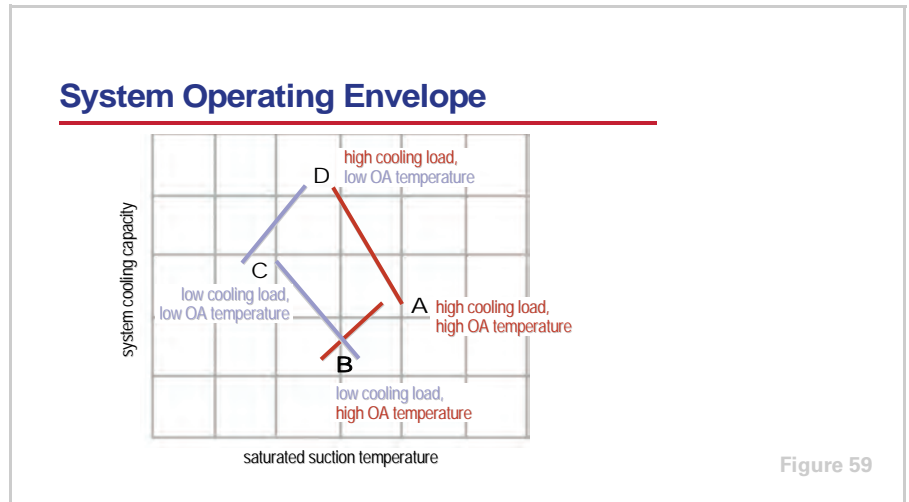
In Figure 58, two of the lines depict the performance of the condensing unit operating over a range of cooling loads. One line assumes that the unit operates at a constant high outdoor-air (outdoor or OA) temperature, while the other line assumes that the same unit operates at a constant low outdoor temperature.

The other two lines depict the performance of the evaporator coil operating over a range of outdoor-air temperatures. One line assumes that the coil operates with a constant high cooling load, while the other line assumes that the same coil operates with a constant low cooling load.

## period four

# Liquid Line

## notes



This operating envelope defines four extreme corners. Condition **A** depicts the system operating with a high cooling load at a high outdoor temperature, and condition **B** depicts a low cooling load at the same high outdoor temperature. Condition **C** depicts the system operating with a low cooling load at a low outdoor temperature, and condition **D** depicts a high cooling load at the same low outdoor temperature.

**effect of cooling load and outdoor temperature**

**Subcooling Leaving Condenser**

	cooling load	outdoor temperature	subcooling leaving condenser*, °F (°C)
A	high	high	18 (10.0)
B	low	high	21 (11.7)
C	low	low	13 (7.2)
D	high	low	5 (2.8)

\* This example system uses a TXV for the expansion device.

Figure 60

A common belief is that the amount of subcooling provided by the condenser increases when the outdoor temperature is cooler. But this is not necessarily true in systems that use a TXV for the expansion device. Figure 60 shows how subcooling changes throughout the operating envelope of an example system. When the system operates with a high cooling load and a high outdoor temperature (**A**), the condenser provides 18°F (10°C) of subcooling. When operating with the low cooling load at the same high outdoor temperature (**B**), less heat must be rejected by the condenser, allowing it to provide more subcooling—21°F (11.7°C).

## period four

# Liquid Line

### notes

When the system operates with the low cooling load at a low outdoor temperature (**C**), the impact of the refrigerant density change reduces the subcooling provided by the condenser to 13°F (7.2°C). Finally, when the system operates with the high cooling load at the same low outdoor temperature (**D**), subcooling drops to 5°F (2.8°C). At condition **D**, the mass flow rate of refrigerant is high due to the high cooling load, and the refrigerant density is high due to the low outdoor temperature. This results in very little liquid refrigerant remaining in the condenser for subcooling.

The key points here are that the amount of subcooling provided by the condenser changes within the operating envelope, and this subcooling is actually reduced at cooler outdoor temperatures due to the change in refrigerant density. This example demonstrates that the condenser provides the smallest amount of subcooling when the system is operating with a high evaporator load at a low outdoor temperature.

#### Involve the Manufacturer

- ▲ If provided, use refrigerant line sizes recommended by manufacturer



Figure 61

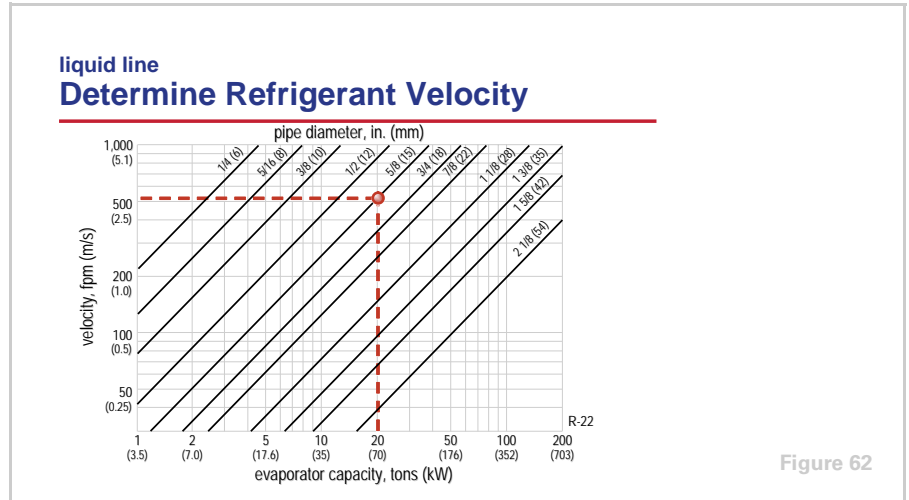
Defining the operating envelope for a particular system is no easy task, and therefore is generally not done for each project. Determining the amount of subcooling provided by the condenser throughout an operating envelope is also quite complicated. It requires a model of the entire refrigeration system and help from the manufacturer of the condenser. Therefore, if the manufacturer of the refrigeration equipment provides recommended line sizes, or tools for selecting the optimal line sizes, we suggest that you use them.

If recommended line sizes are not provided, however, consult the manufacturer to determine the amount of subcooling provided by the condenser. For this example, we will assume 15°F (8.3°C) of subcooling is provided by the condenser.

## period four

# Liquid Line

## notes



The chart in Figure 62 shows the velocity of liquid R-22 inside liquid lines of various diameters at one particular operating condition—40°F (4.4°C) saturated suction temperature, 125°F (51.7°C) saturated condensing temperature, 12°F (6.7°C) of superheat, 15°F (8.3°C) of subcooling, and 70°F (38.9°C) of compressor superheat.

For the same example 20-ton (70.3-kW) system, the refrigerant velocity inside a 5/8 in. (15 mm)-diameter pipe is about 520 fpm (2.6 m/s) at this condition.

Again, a computer program can be used to calculate the velocity based on actual conditions, but without a program, a chart like this may be useful.

**liquid line**  
**Determine Refrigerant Velocity**

pipe diameter, in. (mm)	velocity, fpm (m/s)
	20 tons (70.3 kW)
1/2 (12)	850 (4.3)
5/8 (15)	525 (2.7)
3/4 (18)	350 (1.8)
7/8 (22)	250 (1.3)
1 1/8 (28)	150 (0.8)
1 3/8 (35)	100 (0.5)

Figure 63

Using the chart in Figure 62, the refrigerant velocity at maximum system capacity—20 tons (70.3 kW), for this example—is determined for various pipe diameters. Use of the 1/2 in. (12 mm)-diameter pipe results in a refrigerant

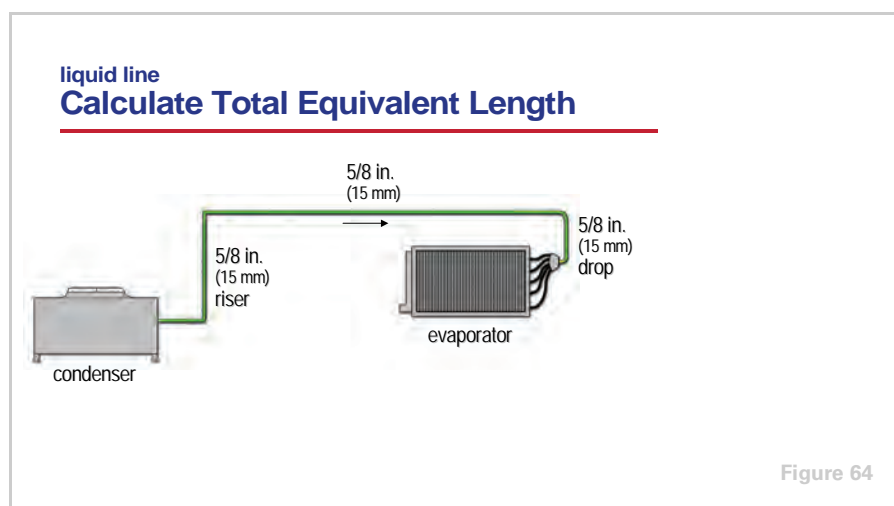
## period four

# Liquid Line

### notes

velocity that is greater than the recommended upper limit of 600 fpm (3 m/s) for liquid lines, so it should probably not be considered.

After the velocity inside pipes of various diameters has been determined, the smallest acceptable pipe diameter is selected to minimize the system refrigerant charge. The table in Figure 63 on page 49 shows that 5/8 in. (15 mm) is the smallest diameter of pipe that will result in a velocity below the recommended upper limit of 600 fpm (3 m/s).



The next step is to calculate the total equivalent length by adding the actual measured length of straight pipe to the equivalent length of any fittings to be installed in the liquid line. The equivalent length data for copper fittings used in the liquid line is the same as for the suction and discharge lines.

The liquid line for this example 20-ton (70.3-kW) system, shown in Figure 64, is 100 ft (30.5 m) long. It is constructed entirely of 5/8 in. (15 mm)-diameter pipe and contains a total of four long-radius elbows. Based on the “new” equivalent length data from Table 1 on page 21, the equivalent length of each elbow of this diameter is 0.4 ft (0.12 m). The total equivalent length of this example liquid line is calculated to be 101.6 ft (31.0 m).

$$L_{eq} = 100 \text{ ft} + (4 \times 0.4 \text{ ft}) = 101.6 \text{ ft}$$

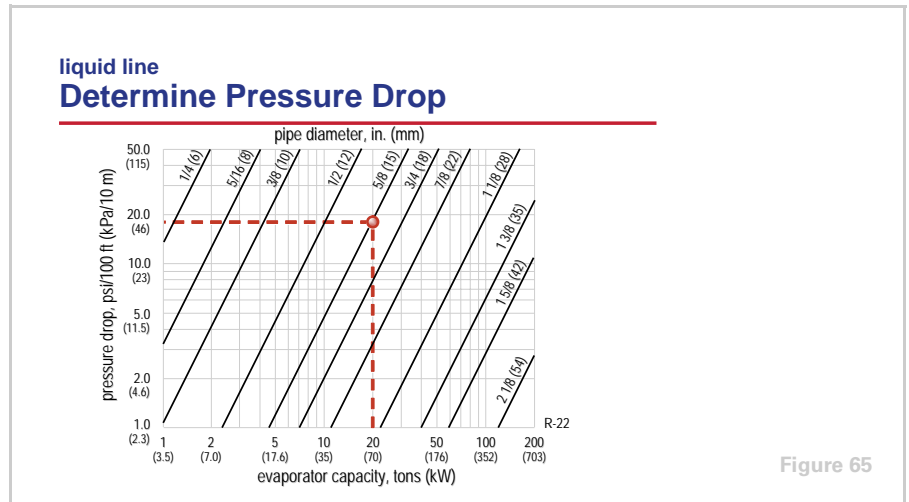
$$(L_{eq} = 30.5 \text{ m} + (4 \times 0.12 \text{ m}) = 31.0 \text{ m})$$



## period four

# Liquid Line

## notes



The next step is to determine the pressure drop due to the straight pipe and fittings, based on the calculated total equivalent length. The chart in Figure 65 shows the pressure drop, per unit of equivalent length, for different pipe diameters and capacities. This chart is based on one specific operating condition, but is likely to be representative of the condition when the system operates at maximum capacity. Alternatively, a computer program can be used to determine pressure drop at specific conditions.

At the maximum capacity of 20 tons (70.3 kW), the pressure drop due to the straight pipe and fittings of the liquid line is 17 psi per 100 ft of equivalent length (72.2 kPa/10 m), or 17.3 psi (119 kPa).

$$\Delta p = 101.6 \text{ ft} \times 17 \text{ psi/100 ft} = 17.3 \text{ psi}$$

$$(\Delta p = 31.0 \text{ m} \times 38.4 \text{ kPa/10 m} = 119 \text{ kPa})$$

## period four

# Liquid Line

## notes

### liquid line

### Add Pressure Drop of Accessories

- ▲ Filter-drier = 6 psi (41.4 kPa)
- ▲ Solenoid valve = 4 psi (27.6 kPa)
- ▲ Sight glass = 1 equivalent ft (0.3 m)
- ▲ Angle valve = 1 psi (6.9 kPa)
- ▲ Ball valve = 1 equivalent ft (0.3 m)



Figure 66

The next step is to add the pressure drop due to any accessories that will be installed in the liquid line, and calculate the total pressure drop. The most common accessories installed in a liquid line are a filter-drier, a sight glass, a solenoid valve, and manual ball valves that are used to isolate portions of the system during maintenance, such as when replacing the core of the filter-drier. The pressure drop for the filter-drier, shown in Figure 66, assumes that it is properly sized but dirty.

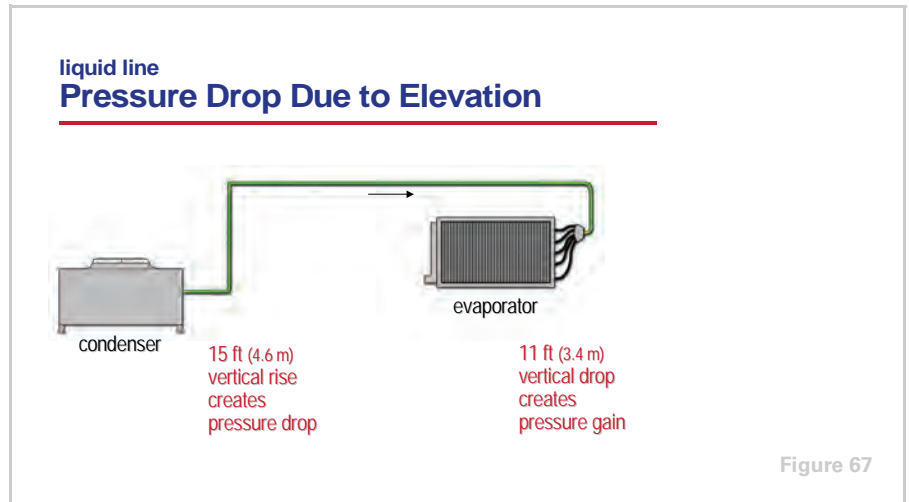
Assuming that this example system includes a filter-drier, solenoid valve, sight glass, and two ball valves, the total pressure drop due to the liquid line is 27.8 psi (191 kPa).

$$\begin{aligned}
 \Delta p &= 17.3 \text{ psi} + 6 \text{ psi} + 4 \text{ psi} + (1 \text{ ft} \times 17 \text{ psi}/100 \text{ ft}) + (2 \times 1 \text{ ft} \times 17 \text{ psi}/100 \text{ ft}) \\
 &= 27.8 \text{ psi} \\
 (\Delta p &= 119 \text{ kPa} + 41.4 \text{ kPa} + 27.6 \text{ kPa} + (0.3 \text{ m} \times 38.4 \text{ kPa}/10 \text{ m}) \\
 &\quad + (2 \times 0.3 \text{ m} \times 38.4 \text{ kPa}/10 \text{ m})) \\
 &= 191 \text{ kPa}
 \end{aligned}$$

## period four

# Liquid Line

## notes



The next step is to calculate the loss of subcooling due to the liquid-line pressure drop and any elevation change.

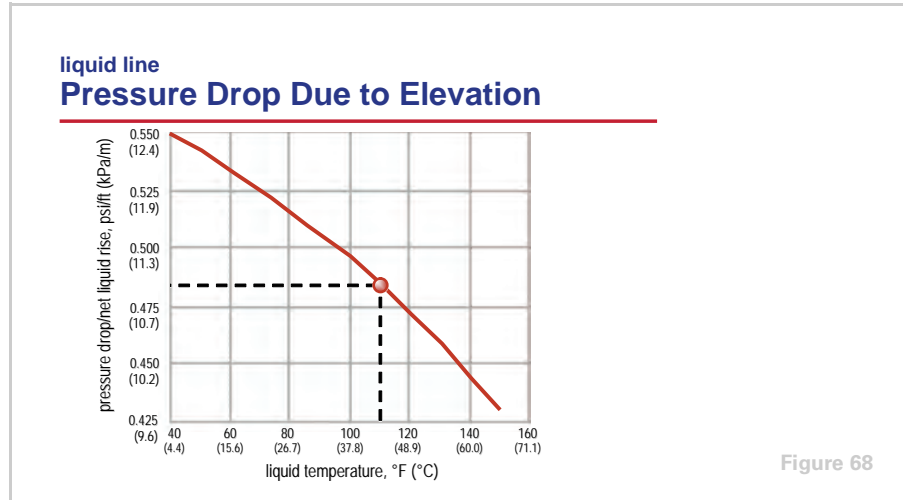
When the condenser is at a *lower* elevation than the evaporator, the refrigerant must overcome the pressure due to a column of liquid refrigerant when flowing up to the evaporator. This liquid column causes a pressure drop that results in a decrease in the amount of subcooling available at the expansion device. When the condenser is at a *higher* elevation than the evaporator, however, the liquid column causes a pressure gain that results in an increase in subcooling.

Most systems will include some vertical risers and some vertical drops in the liquid line, but the critical distance is *net* elevation difference. For example, the liquid line in this example system (Figure 67) rises 15 ft (4.6 m) and then later drops 11 ft (3.4 m). The vertical rise will cause a pressure drop and the vertical drop will cause a pressure gain, but the *net* effect of these two vertical sections will be 4 ft (1.2 m) of rise.

## period four

# Liquid Line

## notes



The chart in Figure 68 shows that the magnitude of the associated pressure drop (or gain) depends primarily on the temperature (density) of the liquid refrigerant. The greatest pressure drop (or gain) due to this vertical column of liquid occurs when the temperature of the liquid refrigerant is the lowest.

Using the example liquid line in Figure 67 on page 53, and assuming a liquid-refrigerant temperature of 110°F (43.3°C), the pressure drop due to the 4 ft (1.2 m) net liquid-line rise is 1.9 psi (13.4 kPa).

$$\Delta p = 4 \text{ ft} \times 0.485 \text{ psi/ft} = 1.9 \text{ psi}$$

$$(\Delta p = 1.2 \text{ m} \times 11.1 \text{ kPa/m} = 13.4 \text{ kPa})$$

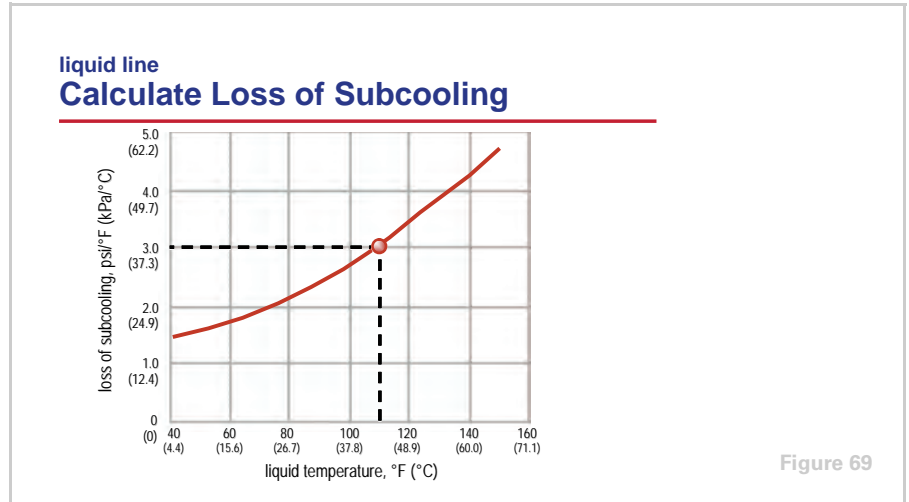
As you can see, the effect of this liquid column of refrigerant can dramatically affect the amount of subcooling available at the expansion device, especially at cooler ambient conditions when the condenser is already providing less subcooling than it could under design conditions. In fact, it may even use up all of the subcooling provided by the condenser, resulting in no subcooling remaining at the expansion device.

Historically, the rule-of-thumb has been to assume a pressure drop of 0.5 psi for each foot (11.3 kPa/m) of net liquid-line rise.

## period four

# Liquid Line

## notes



Next, the pressure drop due to the elevation change is added to the frictional pressure drop due to the straight pipe, fittings, and accessories. For our example 20-ton (70.3-kW) system, the total pressure drop due to all of these effects is 29.7 psi (204.4 kPa).

$$\Delta p = 27.8 \text{ psi} + 1.9 \text{ psi} = 29.7 \text{ psi}$$

$$(\Delta p = 191 \text{ kPa} + 13.4 \text{ kPa} = 204.4 \text{ kPa})$$

Then, using the chart in Figure 69, this total pressure drop is converted to an equivalent loss in subcooling. Assuming a liquid refrigerant temperature of 110°F (43.3°C), the loss of subcooling is 3 psi/°F (37.3 kPa/°C), or 9.9°F (5.5°C).

$$\text{loss of subcooling} = \frac{29.7 \text{ psi}}{3 \text{ psi/°F}} = 9.9^\circ\text{F}$$

$$\left( \text{loss of subcooling} = \frac{204.4 \text{ kPa}}{37.3 \text{ kPa/°C}} = 5.5^\circ\text{C} \right)$$

## period four

# Liquid Line

### notes

#### Subcooling Remaining at Expansion Device

subcooling, °F (°C)		
leaving condenser	loss due to liquid line	remaining at expansion device
15 (8.3)	- 9.9 (5.5)	= 5.1 (2.8)

Figure 70

The last step is to verify that adequate subcooling is available at the inlet to the expansion device. This involves subtracting the impact of pressure drop and elevation difference from the amount of subcooling provided by the condenser.

$$\text{subcooling remaining at expansion device} = 15^{\circ}\text{F} - 9.9^{\circ}\text{F} = 5.1^{\circ}\text{F}$$

$$(\text{subcooling remaining at expansion device} = 8.3^{\circ}\text{C} - 5.5^{\circ}\text{C} = 2.8^{\circ}\text{C})$$

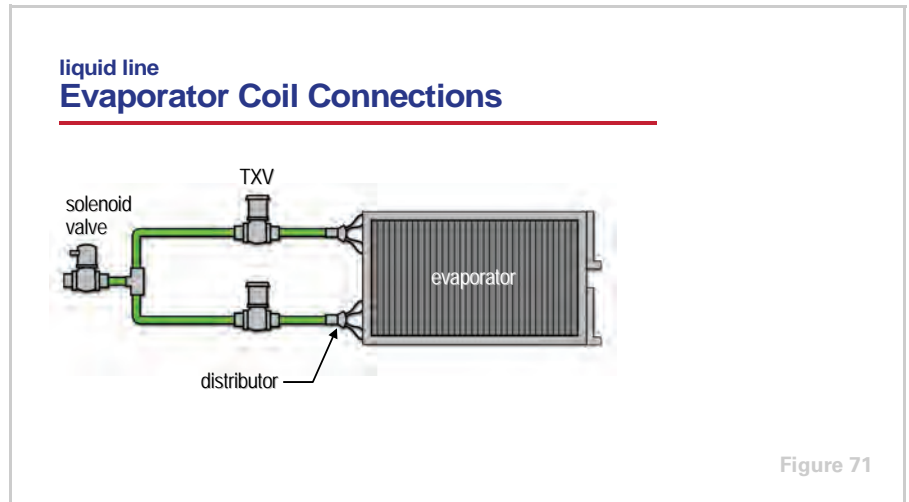
At this condition, 5.1°F (2.8°C) of subcooling is remaining at the inlet to the expansion device. This is above the minimum limit of 5°F (2.8°C).

As mentioned earlier, in the past, a common rule was to limit the pressure drop through the liquid line to 6 psi (41 kPa) for a system operating with R-22. However, as this example demonstrates, in many cases a much higher pressure drop can be tolerated and allows the liquid line to be smaller. This reduces the refrigerant charge, which improves system reliability and lowers the installed cost.

## period four

# Liquid Line

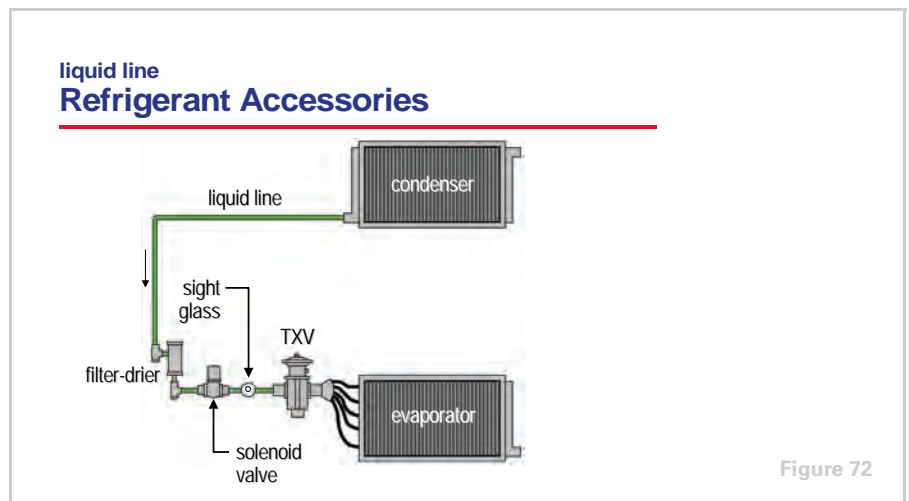
## notes



### Other Considerations

When an evaporator has more than one distributor connected to a single refrigeration circuit, the liquid line should be designed and installed to provide equal inlet pressures to the distributors.

In this example, the liquid line is split and the same sizes and lengths of piping are used to make the connections to the expansion devices and distributors. This equalizes the pressure drops of the two connections and provides the most-balanced expansion-device performance.



For proper refrigerant distribution into the evaporator coil, one TXV is required for each distributor.

A **sight glass** is installed as close as practical, upstream of the expansion device. This is a fitting that contains a glass window through which the refrigerant entering the expansion device may be viewed and its state

## period four

# Liquid Line

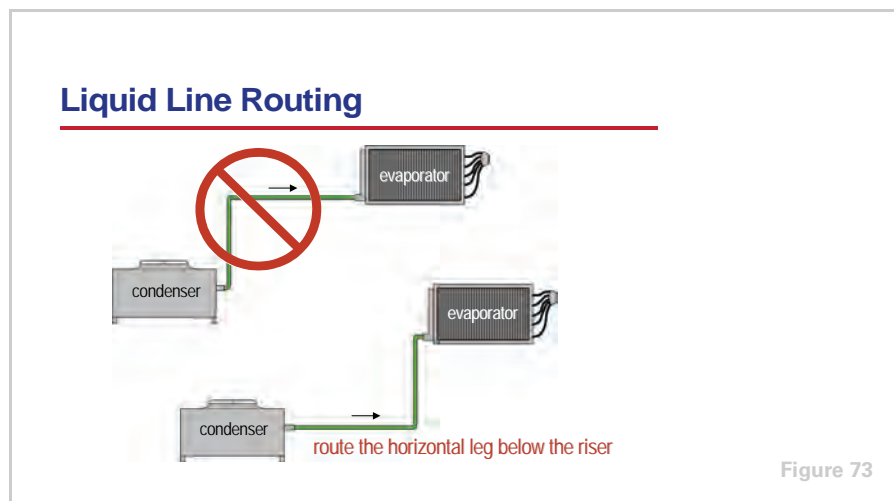
### notes

determined. As mentioned earlier, flashing of refrigerant upstream of the expansion device is undesirable. A sight glass can be used to indicate that this problem exists.

A **solenoid-operated shutoff valve** is typically installed in the liquid line to facilitate system pumpdown or to prevent refrigerant migration when the system is off. Solenoid valves may also be used to control the flow of liquid refrigerant to multiple sections of the evaporator. In this application, a valve would be installed upstream of the TXV for each individually controlled section of the evaporator coil.

A **liquid line filter-drier** should be installed upstream of, and as close as possible to, the solenoid valve and TXV. The filter-drier prevents moisture and foreign matter, introduced during the installation process, from damaging the solenoid valve or TXV. Manual ball valves should be installed for isolating the filter-drier when the core needs to be replaced. It is important to realize that the filter-drier is not a cure-all for poor installation practice. The piping must be clean, and any air or moisture must be thoroughly evacuated from the system.

Finally, an **access port** is typically installed in the liquid line in a convenient location and is used to charge the system with liquid refrigerant. It is also used to measure the amount of subcooling in the system.



How the liquid line is routed can be important, particularly when the condenser is located below the evaporator. As an example, the upper system in Figure 73 routes the liquid-line riser *before* a long horizontal run. The lower system routes the riser *after* a long horizontal run.

In both cases, when the refrigeration circuit shuts off, the liquid refrigerant inside the vertical riser will drain back into the condenser. In the upper system, however, some of the liquid refrigerant from inside the horizontal run will also drain back into the condenser. If this quantity of refrigerant is large enough, it will fill the condenser and overflow into the discharge line and then into the compressor. This can result in compressor lubrication problems, system shutdown due to activation of a protective safety device, or mechanical failure.



## period four

# Liquid Line

### notes

Alternatively, a check valve can be installed in the liquid line, near the outlet of the condenser, whenever the liquid line rises above the height of the condenser. This check valve will prevent liquid refrigerant from draining back into the condenser when the system is off.

A pressure-relief valve is required if both a check valve and a solenoid valve are installed in the same liquid line. When the refrigeration circuit is off, these two valves trap liquid refrigerant between them. The pressure-relief valve is required to relieve the increased pressure that can result when the temperature of this trapped refrigerant increases.

#### liquid line

#### Other Considerations

- ▲ Pitch horizontal sections to run alongside suction or discharge line
- ▲ Insulate lines that are routed through very warm spaces

Figure 74

Generally, horizontal sections of the liquid line should be pitched to run alongside of the accompanying suction or discharge line. This simplifies the installation of the line set. Sloping the liquid line in one direction will not always prevent the flow of refrigerant in the opposite direction because it may move due to the siphon effect.

The refrigerant temperature inside the liquid line is generally higher than that of the surrounding air. Therefore, insulation is not typically required. In fact, heat loss may be desirable because it provides additional subcooling. If the liquid line is routed through areas that are significantly warmer than the liquid refrigerant (such as unconditioned attic spaces or boiler rooms), however, the sections of the liquid line that are exposed to the warmer temperatures should be insulated to prevent loss of subcooling.

period five

## Hot-Gas Bypass Line

---

### notes

#### Refrigerant Piping

*period five*

Hot-Gas Bypass Line

Figure 75

The last refrigerant line to be considered is the hot-gas bypass line. The hot-gas bypass line diverts hot, high-pressure refrigerant vapor from the discharge line to the low-pressure side of the refrigeration system. This adds a “false load” to the system that can help stabilize the suction pressure and temperature.

Hot-gas bypass is sometimes used to help prevent evaporator frosting in direct-expansion (DX) applications, or in applications where the compressor cannot be allowed to cycle on and off, such as a process-cooling application.

In general, however, hot-gas bypass should be avoided for most comfort-cooling applications. It increases the refrigerant charge in the system, adds more paths for potential refrigerant leaks, and increases the likelihood of refrigerant-distribution and oil-return problems. It also increases the first cost of the system, and the operating cost, because it prevents compressors from cycling off with fluctuating loads.

period five

## Hot-Gas Bypass Line

### notes

hot-gas bypass line

#### **Requirements for Sizing and Routing**

- ▲ Ensure that oil and refrigerant are not trapped in piping
- ▲ Avoid excessive noise
- ▲ Proper routing is more critical than pipe diameter

Figure 76

Similar to the discharge line, the refrigerant flowing through the hot-gas bypass line is in the form of a hot, high-pressure vapor. However, unlike the discharge line, the minimum velocity is difficult to determine. A typical hot-gas bypass valve is a modulating valve, so the refrigerant velocity through the valve and piping can range all the way from zero flow to full flow. At times, the refrigerant velocity inside a given pipe diameter will fall below the minimum velocity required to carry oil droplets.

Therefore, it is not possible to select a pipe diameter that is small enough to move oil at all valve positions. The hot-gas bypass line should be sized as small as possible, without the maximum velocity being high enough to cause objectionable noise. Because the hot-gas bypass line is basically an extension of the discharge line, the recommended maximum velocity is 3,500 fpm (17.5 m/s). Selecting the smallest possible line size also minimizes the amount of oil and condensed refrigerant that could be trapped in the hot-gas bypass line.

Also, the hot-gas bypass line must be installed so that oil and condensed refrigerant can freely drain out of all sections of the line, at all operating conditions.

## period five

# Hot-Gas Bypass Line

## notes

### Hot-Gas Bypass to Evaporator Inlet

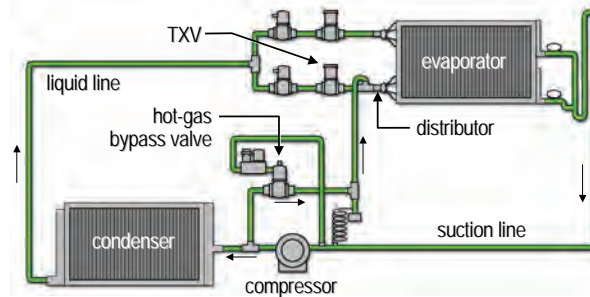


Figure 77

### Hot-Gas Bypass to Evaporator Inlet

In a DX application, there are two bypass methods that are commonly used. The first, and preferred, method bypasses refrigerant vapor from the compressor discharge line to the inlet of the evaporator. Sensing a decrease in suction pressure, the hot-gas bypass valve opens to bypass hot refrigerant vapor from the compressor discharge line to the inlet of the evaporator, between the TXV and the distributor. This adds a “false load” to the evaporator and increases the suction pressure (and temperature).

The principal advantage of hot-gas bypass to the evaporator inlet is that the refrigerant velocity inside the evaporator and the suction line is higher at low loads. This promotes a uniform movement of oil through the evaporator and the suction line.

### Hot-Gas Bypass Valve Installation

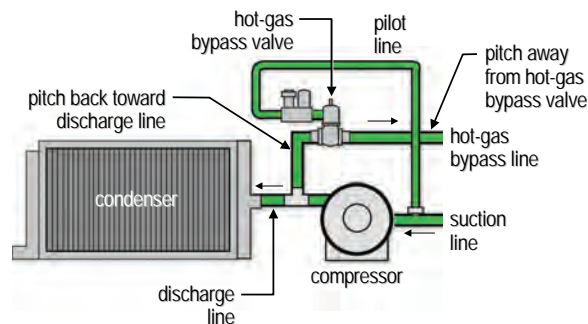


Figure 78

Due to the widely varying velocity in the hot-gas bypass line, the selected diameter of the pipe is not as critical as its routing. Install the **hot-gas bypass**

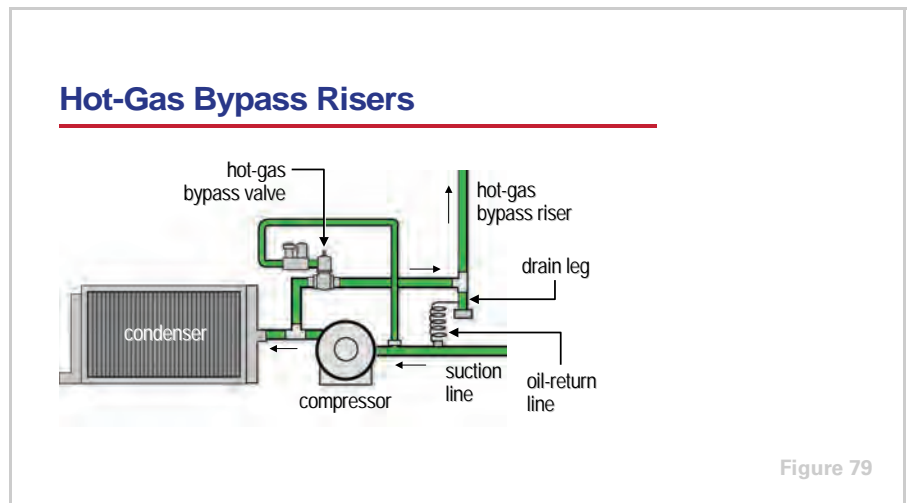
period five

## Hot-Gas Bypass Line

### notes

**valve** above the discharge line and close to the compressor. Pitch the section of pipe between the discharge line and the hot-gas bypass valve so oil will drain back into the discharge line.

When using a pressure-actuated hot-gas bypass valve, tap the pilot line for the valve close to the compressor inlet. This pilot line senses suction pressure, to which the valve responds by modulating open or closed. Pitch the piping downstream of the hot-gas bypass valve so that oil or condensed refrigerant will drain away from the valve, toward the evaporator.



If the hot-gas bypass line includes a vertical riser of *any* height, or if the line does not allow free drainage from the valve to the evaporator, the retention of oil or condensed refrigerant within the riser must be considered. As mentioned, because the refrigerant velocity within the hot-gas bypass line varies over a wide range, no pipe, regardless of its diameter, can ensure adequate velocity to carry oil up a riser of any height. Oil will collect at the base of the riser when the hot-gas bypass valve throttles to lower flow rates.

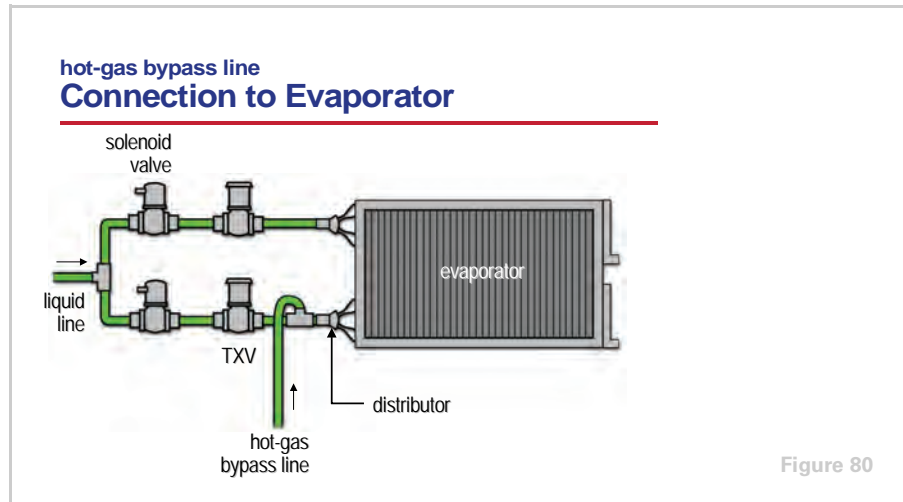
This problem is commonly addressed by installing a small **oil-return line** between the base of the riser and the suction line. First, a drain leg is provided at the base of the riser to collect the oil and condensed refrigerant. This short section of pipe should be about 12 in (300 mm) long and be the same diameter as the riser.

Next, the oil-return line is connected to the side of the drain leg to prevent any foreign material from plugging the tube. This oil-return line is constructed of 1/8 in. (3 mm)-diameter copper tubing and should be at least 5 ft (1.5 m) in length. This length is required to create the necessary pressure drop between the high-pressure and low-pressure sides of the system, yet this tubing is large enough to drain oil and liquid refrigerant from the drain leg. If the oil-return line must be longer than 5 ft (1.5 m), 1/4 in. (6 mm)-diameter tubing can be used for the rest of the distance.

## period five

# Hot-Gas Bypass Line

## notes



The hot-gas bypass line is then piped to the evaporator inlet. It must be connected to the top of the pipe, between the TXV and the distributor. Any hot-gas bypass piping upstream of this connection should be pitched so that oil or condensed refrigerant will drain toward the evaporator.

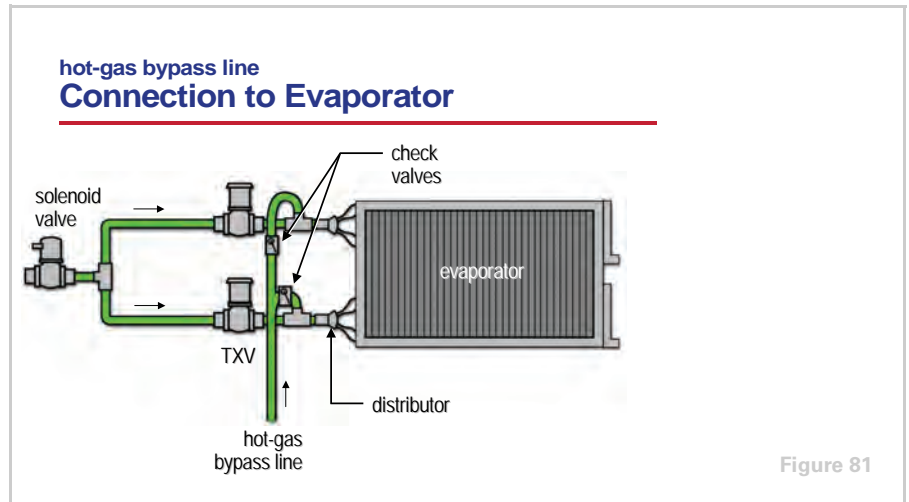
An orifice-type distributor is typically not able to permit adequate flow of refrigerant vapor through the orifice. If this type of distributor is used, the orifice must be relocated upstream of the point at which the hot-gas bypass line connects to the distributor.

When the evaporator contains multiple distributors on a single refrigeration circuit, the hot-gas bypass line should be piped to each distributor that is “active” when the hot-gas bypass valve opens. The evaporator shown in Figure 80 contains two distributors on the refrigeration circuit, each equipped with a solenoid valve. In this example, the top solenoid valve closes when the compressor unloads. Only the bottom distributor is “active” at minimum capacity when the hot-gas bypass valve opens. Therefore, the hot-gas bypass pipe only connects to the bottom distributor.

period five

## Hot-Gas Bypass Line

### notes



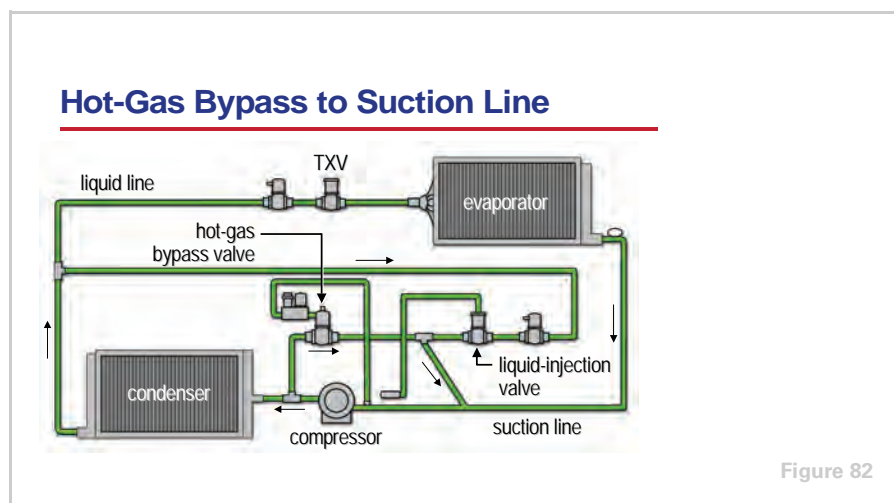
The evaporator shown in Figure 81 also contains two distributors on a single refrigeration circuit, but it only contains one solenoid valve. In this example, the solenoid valve closes only after the last compressor on the circuit shuts off. Both distributors are “active” at all times and the hot-gas bypass pipe must be connected to both distributors.

When the hot-gas bypass line is piped to more than one distributor, **check valves** should be installed in each section of pipe that feeds a distributor. Without these check valves, a path is created between coil circuits whenever the hot-gas bypass valve is closed. The coil circuits will become unequally loaded and refrigerant will travel from the outlet of one TXV into the section of coil that is served by a different TXV. This will result in the loss of superheat control and potential compressor damage.

## period five

# Hot-Gas Bypass Line

## notes



### Hot-Gas Bypass to the Suction Line

The second bypass method conducts refrigerant vapor from the compressor discharge line to the suction line. This method requires an additional expansion valve, called a **liquid-injection valve**. The remote bulb of this liquid-injection valve is attached to the suction line near the inlet to the compressor. When reduced suction pressure causes the hot-gas bypass valve to open, the liquid-injection valve senses the resulting rise in suction temperature and opens to mix liquid refrigerant with the bypassed refrigerant vapor. The hot refrigerant vapor causes the liquid refrigerant to vaporize, cooling the mixture and stabilizing the suction pressure.

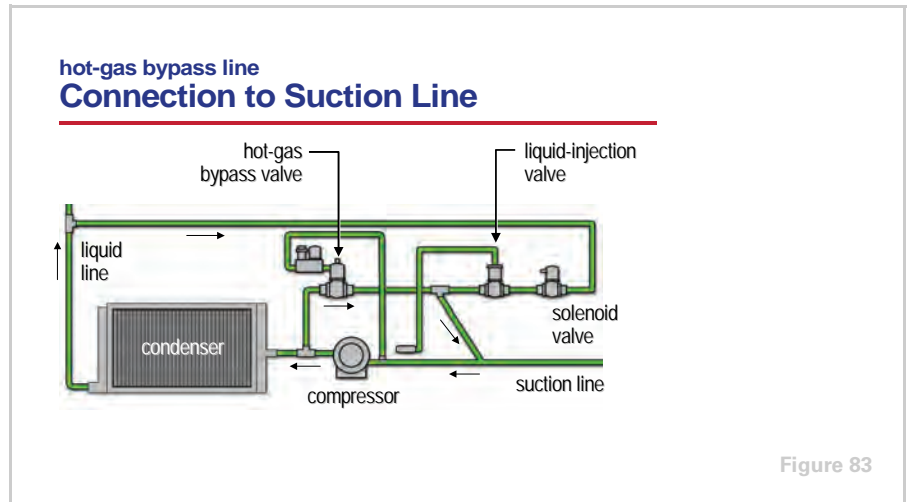
The advantage of this method is that it requires less actual piping than bypassing hot gas to the evaporator inlet. However, the disadvantage is that the refrigerant velocity inside the evaporator and suction line (upstream of the hot-gas bypass connection) can drop very low when the hot-gas bypass valve is open. This may result in oil being trapped in the evaporator and the suction line. For this reason, this method is only acceptable in applications where the evaporator and the suction line are able to freely drain to the point at which the hot-gas bypass line connects into the suction line. Therefore, hot-gas bypass to the evaporator inlet is the preferred method.



period five

## Hot-Gas Bypass Line

### notes



Again, routing of the hot-gas bypass piping is critical. Install the hot-gas bypass valve above the compressor discharge and close to the compressor. Pitch the section of pipe between the discharge line and the hot-gas bypass valve so that oil will drain back into the discharge line. Tap the pilot line for the hot-gas bypass valve close to the compressor inlet, and pitch the piping downstream of the hot-gas bypass valve so that oil will drain away from the valve, toward the suction line.

A separate pipe routes liquid refrigerant from the liquid line to mix with the hot, bypassed refrigerant vapor. This mixture is then piped into the suction line at an angle, at least 5 ft (1.5 m) upstream of the compressor inlet, and upstream of the pilot line tap for the hot-gas bypass valve. This angled connection promotes mixing, and the minimum distance helps ensure complete vaporization of the liquid refrigerant before it reaches the compressor.

The liquid-injection line contains a thermal expansion valve, called a liquid-injection valve. The remote bulb for the liquid-injection valve is attached to the suction line, downstream of the point at which the hot gas enters the suction line. This valve is adjusted to maintain between 30°F and 35°F (-1°C and 1.5°C) of superheat at the inlet to the compressor. A solenoid valve is installed upstream of the liquid-injection valve to shut off the flow of liquid refrigerant whenever the hot-gas bypass valve is closed.

## period five

# Hot-Gas Bypass Line

## notes

### Hot-Gas Bypass Line Sizing

pipe diameter, in. (mm)	velocity, fpm (m/s)
	10 tons (35.2 kW)
1/2 (12)	6,700 (34.0)
5/8 (15)	4,200 (21.3)
3/4 (18)	2,825 (14.4)
7/8 (22)	2,025 (10.3)
1 1/8 (28)	1,200 ( 6.1)
1 3/8 (35)	775 ( 3.9)

Figure 84

### Other Considerations

The hot-gas bypass valve, and the diameter of the hot-gas bypass line, are selected based on the maximum amount of refrigerant vapor that will be bypassed. This depends on the application and the purpose of the hot-gas bypass system.

For a traditional comfort-cooling application, the hot-gas bypass valve is used to modulate the amount of refrigerant vapor bypassed to maintain the suction temperature at a setpoint, such as 28°F (-2.2°C). This typically requires the valve to only open when the cooling load on the system drops below the minimum compressor capacity.

For example, consider the same 20-ton (70.3-kW), single-circuit system that can unload to a minimum capacity of 10 tons (35.2 kW). The hot-gas bypass valve and line would probably be sized for 10 tons (35.2 kW) of capacity or less. Assuming a maximum bypass capacity of 10 tons (35.2 kW), the refrigerant velocity is determined for various pipe diameters using the chart in Figure 43 on page 34.

As explained at the beginning of this period, the diameter of the hot-gas bypass line should be as small as possible while preventing the maximum velocity from exceeding the recommended upper limit of 3,500 fpm (17.5 m/s). For this example system, a 3/4 in. (18 mm)-diameter pipe is selected. If the overall length of the hot-gas bypass line exceeds 75 ft (23 m), special precautions, including increasing the line size, may be required.

period five

## Hot-Gas Bypass Line

### notes

hot-gas bypass line

#### Other Considerations

- ▲ Insulate entire line to prevent refrigerant vapor from condensing
- ▲ Minimize overall length of the line
- ▲ Install solenoid valve if system includes a pump-down cycle

Figure 85

The full length of the hot-gas bypass line must be insulated to prevent the hot refrigerant vapor from condensing before it reaches the evaporator.

As the length of the hot-gas bypass line increases, the risk of refrigerant vapor condensing inside the line increases, even if the line is insulated. This can cause a problem when the hot-gas bypass valve opens, because the refrigerant vapor could force a slug of liquid refrigerant into the evaporator. This over-feeding of the evaporator may cause liquid refrigerant to enter the suction line, possibly slugging the compressor. For this reason, the overall length of the hot-gas bypass line should be as short as possible.

If the refrigeration system includes a **pump-down cycle**, then a solenoid valve must be installed in the hot-gas bypass line. A pump-down cycle closes the liquid-line solenoid valve to stop the flow of refrigerant into the evaporator, and allows the compressor to run for a short period of time. This pumps refrigerant from the low-pressure side of the system (evaporator and suction line) to the high-pressure side of the system (discharge line, condenser, and liquid line). The solenoid valve in the hot-gas bypass line must also close when the system begins the pump-down cycle, to prevent the high-pressure refrigerant vapor from migrating to the low-pressure side of the system. Some hot-gas bypass valves are equipped with an integral solenoid that can be used to hold the valve closed for this purpose.

## period six Review

### notes

#### Refrigerant Piping

*period six*  
**Review**

Figure 86

We will now review the main concepts that were covered in this clinic regarding the design and installation of the interconnecting piping for vapor-compression air-conditioning systems.

#### Review—Period One

##### ▲ Refrigerant Piping Requirements

- ◆ Return oil to compressor
- ◆ Ensure that only liquid refrigerant (no vapor) enters the expansion device
- ◆ Minimize system capacity loss
- ◆ Minimize refrigerant charge

##### ▲ If provided, use line sizes recommended by the manufacturer

Figure 87

As discussed in Period One, when a refrigeration system includes field-assembled refrigerant piping, the primary design goals are generally to maximize system reliability and minimize installed cost. When the refrigerant piping is designed and installed, it is imperative that the following requirements are met:

- Return oil to the compressor at all operating conditions
- Ensure that only liquid refrigerant (no vapor) enters the expansion device
- Minimize system capacity loss that is caused by pressure drop through the piping

## period six Review

### notes

- Minimize the total refrigerant charge in the system to improve reliability and minimize installed cost

If the manufacturer of the refrigeration equipment provides recommended line sizes, or tools for selecting the proper line sizes, use those line sizes.

### Review—Period Two

#### ▲ Suction line

- ◆ Ensure adequate velocity to return oil to the compressor at all steps of unloading
- ◆ Avoid excessive noise
- ◆ Minimize system capacity and efficiency loss

Figure 88

Period Two reviewed the design of the suction line. The diameter of the suction line should be selected small enough so the refrigerant velocity is high enough to move oil, at all steps of compressor unloading. However, the pipe diameter should be as large as possible to minimize pressure drop, maximize system capacity and efficiency, and avoid causing objectionable noise.

The minimum allowable velocity in a suction line varies based on the diameter of the pipe. The smaller the pipe diameter, the less velocity is required to move the oil. Long line lengths and the associated higher pressure drop can be tolerated, assuming that the loss of system capacity and efficiency is acceptable for the given application. Of course, it is still good practice to minimize pressure drop.

## period six

# Review

## notes

### Review—Period Three

#### ▲ Discharge line

- ◆ Ensure adequate velocity to return oil to the compressor at all steps of unloading
- ◆ Avoid excessive noise
- ◆ Minimize efficiency loss

Figure 89

Period Three discussed the discharge line. Similar to the suction line, the diameter of the discharge line must be small enough that the refrigerant velocity is high enough to move oil, at all steps of compressor unloading. However, the pipe diameter should be as large as possible to minimize pressure drop, maximize system efficiency, and avoid causing objectionable noise.

### Review—Period Four

#### ▲ Liquid line

- ◆ Ensure that only liquid refrigerant enters the expansion device
- ◆ Minimize refrigerant charge
- ◆ Avoid excessive noise and pipe erosion

Figure 90

The liquid line was the focus of Period Four. The liquid line must be designed and installed to ensure that only liquid refrigerant (no vapor) enters the expansion device. This requires the condenser to provide adequate subcooling, and the pressure drop through the liquid line and accessories must not be high enough to cause flashing upstream of the expansion device.

An understanding of the entire refrigeration system, not just the piping, is required to select the optimal liquid line size. The diameter of the liquid line must be as small as possible to minimize the quantity of refrigerant in the system and minimize the impact of changes in refrigerant density on the ability

## period six

# Review

### notes

of the condenser to provide subcooling. This improves system reliability and minimizes the installed cost. However, if the pipe is too small, the increased pressure drop may cause flashing upstream of the expansion device, and the high velocity may result in objectionable noise or erosion to the inside surfaces of the piping.

### Review—Period Five

#### ▲ Hot-gas bypass line

- ◆ Ensure that oil and refrigerant are not trapped in the piping
- ◆ Avoid excessive noise
- ◆ Proper routing is more critical than pipe diameter

Figure 91

Period Five discussed the importance of proper routing when hot-gas bypass is to be used. The two most common methods are hot-gas bypass to the evaporator inlet and hot-gas bypass to the suction line. Hot-gas bypass to the evaporator inlet is preferred because the refrigerant velocity in the evaporator and the suction line is higher at low loads, resulting in improved oil return.

Because the hot-gas bypass valve modulates from full flow down to zero flow, at times the refrigerant velocity will fall below the minimum velocity required to carry oil droplets. Therefore, it is not possible to select a pipe diameter that is small enough to move oil at all valve positions. The hot-gas bypass line must be designed and installed so that oil and condensed refrigerant can freely drain out of all sections of the line, at all operating conditions.

When used, the overall length of the hot-gas bypass line should be as short as possible. In general, however, hot-gas bypass should be avoided for most comfort-cooling applications.

## period six **Review**

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### notes



Figure 92

For more information, refer to the following references:

- *Refrigeration Cycle* Air Conditioning Clinic (Trane literature order number TRG-TRC003-EN)
- *Refrigeration Compressors* Air Conditioning Clinic (TRG-TRC004-EN)
- *Refrigeration System Components* Air Conditioning Clinic (TRG-TRC005-EN)
- *Tube Size and Component Selection for RAUC Split Systems (20 - 120 tons)* application guide (SS-APG001-EN)
- *Trane Reciprocating Refrigeration Manual*
- *Hot Gas Bypass Control* Applications Engineering Manual (AM-CON10)
- "Split System Refrigerant Piping Design" *Engineers Newsletter Live* satellite broadcast (APP-APV009-EN)
- *Trane Refrigerant Line Sizing* software (part of the *Engineering Toolbox*)

For more information on these and other educational materials available from Trane, contact your local Trane office (request a copy of the Educational Materials catalog—Trane order number EM-ADV1) or visit our online bookstore at [www.trane.com/bookstore/](http://www.trane.com/bookstore/).

For information on Trane's *Engineering Toolbox* software, visit the Trane C.D.S. Web site at [www.tranecds.com](http://www.tranecds.com).





# Quiz

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## Questions for Period 1

- 1** What is the name of the refrigerant line that connects the compressor to the condenser?
  - a** suction line
  - b** discharge line
  - c** liquid line
  - d** hot-gas bypass line
- 2** Which of the following best describes the condition of the refrigerant flowing through the suction line?
  - a** hot, high-pressure vapor
  - b** warm, high-pressure liquid
  - c** cool, high-pressure vapor
  - d** cool, low-pressure vapor
- 3** Which of the following best describes the condition of the refrigerant flowing through the liquid line?
  - a** hot, high-pressure vapor
  - b** cool, high-pressure liquid
  - c** warm, high-pressure liquid
  - d** cool, low-pressure liquid

## Questions for Period 2

- 4** What is the minimum allowable velocity for a 3 1/8-in. (79 mm)-diameter vertical suction riser?
- 5** Using Figure 19 on page 14 and Figure 21 on page 15, what is the smallest pipe diameter that ensures adequate velocity in a vertical suction riser with a capacity of 10 tons (35.2 kW)?
- 6** Should a suction line typically be insulated? Why or why not?

## Questions for Period 3

- 7** A 75 ft (22.9 m) section of discharge line is constructed of 1 3/8-in. (35 mm)-diameter pipe, and contains five long-radius elbows and two manual ball valves. Using Figure 48 on page 37, what is the pressure drop through the entire discharge line when the system capacity is 10 tons (35.2 kW)?
- 8** Why is the discharge line pitched away from the compressor?

# Quiz

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## Questions for Period 4

- 9** Why is the diameter of the liquid line selected as small as possible?
- 10** Explain why minimum refrigerant velocity is not a concern when sizing a liquid line.
- 11** Why does a net liquid-line rise, such as when the condenser is at a lower elevation than the evaporator, cause concern when sizing the liquid line?
- 12** What is the purpose of the liquid-line filter-drier?

## Questions for Period 5

- 13** Why is it difficult to determine the minimum refrigerant velocity in a hot-gas bypass line?
- 14** When piping hot-gas bypass to the evaporator inlet, is the hot gas introduced upstream or downstream of the TXV?
- 15** Should a hot-gas bypass line be insulated? Why or why not?

## Answers

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- 1** b
- 2** d
- 3** c
- 4** 1,180 fpm (6.0 m/s) — see Figure 21 on page 15.
- 5** The velocity inside a 1 1/8-in. (35 mm)-diameter pipe is 3,500 fpm (17.8 m/s), which is below the recommended upper limit of 4,000 fpm (20 m/s) and above the minimum allowable velocity in a riser of this diameter— 700 fpm (3.6 m/s).
- 6** Yes, to prevent condensation on the outside surfaces of the pipe and to prevent loss of capacity due to heat gain.
- 7** 1.06 psi (7.4 kPa)  
$$L_{eq} = 75 \text{ ft} + 5 \times 0.9 \text{ ft} + 2 \times 1.0 \text{ ft} = 81.5 \text{ ft}$$
$$(L_{eq} = 22.9 \text{ m} + 5 \times 0.27 \text{ m} + 2 \times 0.30 \text{ m} = 24.85 \text{ m})$$
$$\Delta p = 81.5 \text{ ft} \times 1.3 \text{ psi}/100 \text{ ft} = 1.06 \text{ psi}$$
$$(\Delta p = 24.85 \text{ m} \times 3.0 \text{ kPa}/10 \text{ m} = 7.4 \text{ kPa})$$
- 8** To prevent any refrigerant that may condense inside the discharge line from draining back into the compressor when the system is off.
- 9** To minimize the quantity of refrigerant in the system.
- 10** The oil and liquid refrigerant mix readily, so oil moves along with the refrigerant at any velocity.
- 11** When the condenser is at a lower elevation than the evaporator, the refrigerant must overcome a column of liquid refrigerant when flowing up to the evaporator. This liquid column causes a pressure drop that results in a decrease in the amount of subcooling available at the expansion device.
- 12** To prevent moisture and foreign matter, introduced during the installation process, from entering the solenoid valve and TXV.
- 13** The hot-gas bypass valve is a modulating valve, so the refrigerant velocity through the valve and piping can range from zero flow to full flow.
- 14** Downstream of the TXV.
- 15** Yes, to prevent the hot refrigerant vapor from condensing before it reaches the evaporator.

# Glossary

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**access port** A device that allows a technician to gain access to the refrigeration system for charging or measurement.

**air-cooled condenser** A type of condenser in which refrigerant flows through the tubes and rejects heat to air that is drawn across the tubes.

**angle valve** A type of manual shutoff valve in which the inlet and outlet are at an angle, and consisting of a valve stem and seat that allow flow through the valve to be stopped.

**ASHRAE** American Society of Heating, Refrigerating and Air-Conditioning Engineers

**ASME** American Society of Mechanical Engineers

**ball valve** A type of manual shutoff valve that consists of a rotating ball with a hole through its center to allow flow to pass through.

**check valve** A type of valve that allows fluid to flow in one direction only.

**compressor** A mechanical device in the refrigeration system used to increase the pressure and temperature of the refrigerant vapor.

**compressor superheat** The difference between the actual temperature of the refrigerant vapor leaving the compressor and the saturated condensing temperature of the refrigerant at that pressure.

**condenser** A component of the refrigeration system in which refrigerant vapor is converted to liquid as it rejects heat to air, water, or some other fluid.

**condensing unit** A piece of air conditioning equipment that includes the condenser and compressors packaged together in a single unit.

**cycling** The practice of alternately turning a compressor on and off to match the system load.

**cylinder unloader** A device used to unload the capacity of a reciprocating compressor by closing the suction passage to the cylinder, holding open the suction valve of a piston-cylinder pair, or diverting the compressed refrigerant vapor back to the suction side of the compressor.

**direct expansion (DX) system** A type of air-conditioning system that uses the refrigerant directly as the cooling media. The refrigerant inside the evaporator absorbs heat directly from the air used for space conditioning.

**discharge line** A pipe that transports refrigerant vapor from the compressor to the condenser in a mechanical refrigeration system.

**distributor** A device used to ensure uniform refrigerant distribution through the multiple coil circuits of a finned-tube evaporator.

**enthalpy** A measure of heat quantity, both sensible and latent, per lb (kg) of refrigerant.

## Glossary

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**equivalent length** A method of expressing the pressure drop through a pipe fitting or accessory in terms of the length of straight pipe, of the same internal diameter, that would produce an equal pressure drop.

**evaporator** A component of the refrigeration system in which cool, liquid refrigerant absorbs heat from air, water, or some other fluid, causing the refrigerant to boil.

**expansion device** A component of the refrigeration system used to reduce the pressure and temperature of the refrigerant to the evaporator conditions.

**expansion valve** A type of expansion device that maintains the pressure difference between the high-pressure and low-pressure sides of the system, and maintains the proper amount of superheat in the system by metering the quantity of liquid refrigerant entering the evaporator, ensuring it will be completely vaporized within the evaporator.

**flash** The process of liquid refrigerant being vaporized by a sudden reduction of pressure.

**hot-gas bypass** The process of diverting hot, high-pressure refrigerant vapor from the discharge line to the low-pressure side of the refrigeration system, either to the inlet of the evaporator or to the suction line.

**liquid-hammer effect** A hammer-like noise caused by suddenly shutting off the flow of liquid inside a pipe.

**liquid-injection valve** An additional thermal expansion valve that is used when bypassing hot gas to the suction line. This valve injects liquid refrigerant to mix with the bypassed hot gas, cooling the mixture before introducing it into the suction line.

**liquid line** A pipe that transports refrigerant vapor from the condenser to the evaporator in a mechanical refrigeration system.

**liquid-line filter drier** A device installed in the liquid line to remove moisture and foreign matter, introduced during the installation process, from the refrigeration system.

**pressure–enthalpy chart** A graphical representation of the properties of a refrigerant, plotting refrigerant pressure versus enthalpy.

**pump-down cycle** A control sequence used in a refrigeration system to pump the refrigerant from the low-pressure side of the system to the high-pressure side of the system.

**reciprocating compressor** A type of compressor that uses a piston that travels up and down inside a cylinder to compress the refrigerant vapor.

**saturated condensing temperature** The temperature at which refrigerant vapor condenses into a liquid inside the condenser.

# Glossary

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**saturated suction temperature** The temperature at which liquid refrigerant boils inside the evaporator.

**scroll compressor** A type of compressor that uses two opposing scrolls to trap the refrigerant vapor and compress it by gradually shrinking the volume of the refrigerant.

**shutoff valve** A device used to isolate one part of the refrigeration system from the rest.

**sight glass** A device installed in the liquid line, upstream of the expansion device, used to allow visual detection of moisture in the system and determine if the liquid refrigerant has flashed into vapor before entering the expansion device.

**solenoid valve** A device used to automatically stop the flow of refrigerant within the refrigeration system.

**subcooling** The amount of heat removed from the liquid refrigerant after it has completely condensed inside the condenser.

**suction line** A pipe that transports refrigerant vapor from the evaporator to the compressor in a mechanical refrigeration system.

**suction-line accumulator** A device, installed in the suction line, that attempts to prevent a slug of liquid refrigerant or oil from causing damage to the compressor, by allowing liquid refrigerant and oil to separate from the refrigerant vapor and then be drawn into the compressor at a rate that will not cause damage.

**suction-line filter** A device installed in the suction line to remove foreign matter from the refrigeration system.

**superheat** The amount of heat added to the refrigerant vapor after it has completely vaporized within the evaporator.

**thermostatic expansion valve (TXV)** A type of expansion device that uses a thermally-actuated valve to sense and control the flow rate of liquid refrigerant to the evaporator.













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