CHAPTER 2

SYSTEM PRACTICES FOR HALOCARBON REFRIGERANTS

REFRIGERATION is the process of moving heat from one location to another by use of refrigerant in a closed cycle. Oil management; gas and liquid separation; subcooling, superheating, and piping of refrigerant liquid and gas; and two-phase flow are all part of refrigeration. Applications include air conditioning, commercial refrigeration, and industrial refrigeration.

Desired characteristics of a refrigeration system may include

- Year-round operation, regardless of outdoor ambient conditions
- Possible wide load variations (0 to 100% capacity) during short periods without serious disruption of the required temperature levels
- Frost control for continuous-performance applications
- Oil management for different refrigerants under varying load and temperature conditions
- A wide choice of heat exchange methods (e.g., dry expansion, liquid overfeed, or flooded feed of the refrigerants) and use of secondary coolants such as salt brine, alcohol, and glycol
- System efficiency, maintainability, and operating simplicity
- Operating pressures and pressure ratios that might require multistaging, cascading, and so forth

A successful refrigeration system depends on good piping design and an understanding of the required accessories. This chapter covers the fundamentals of piping and accessories in halocarbon refrigerant systems. Hydrocarbon refrigerant pipe friction data can be found in petroleum industry handbooks. Use the refrigerant properties and information in Chapters 2, 19, and 20 of the 2005 *ASHRAE Handbook—Fundamentals* to calculate friction losses.

For information on refrigeration load, see Chapter 12. For R-502 information, refer to the 1998 *ASHRAE Handbook—Refrigeration*.

Piping Basic Principles

The design and operation of refrigerant piping systems should (1) ensure proper refrigerant feed to evaporators; (2) provide practical refrigerant line sizes without excessive pressure drop; (3) prevent excessive amounts of lubricating oil from being trapped in any part of the system; (4) protect the compressor at all times from loss of lubricating oil; (5) prevent liquid refrigerant or oil slugs from entering the compressor during operating and idle time; and (6) maintain a clean and dry system.

REFRIGERANT FLOW

Refrigerant Line Velocities

Economics, pressure drop, noise, and oil entrainment establish feasible design velocities in refrigerant lines (Table 1).

Table 1 Recommended Gas Line Velocities

Suction line	900 to 4000 fpm
Discharge line	2000 to 3500 fpm

Higher gas velocities are sometimes found in relatively short suction lines on comfort air-conditioning or other applications where the operating time is only 2000 to 4000 h per year and where low initial cost of the system may be more significant than low operating cost. Industrial or commercial refrigeration applications, where equipment runs almost continuously, should be designed with low refrigerant velocities for most efficient compressor performance and low equipment operating costs. An owning and operating cost analysis will reveal the best choice of line sizes. (See Chapter 36 of the 2003 *ASHRAE Handbook—HVAC Applications* for information on owning and operating costs.) Liquid lines from condensers to receivers should be sized for 100 fpm or less to ensure positive gravity flow without incurring backup of liquid flow. Liquid lines from receiver to evaporator should be sized to maintain velocities below 300 fpm, thus minimizing or preventing liquid hammer when solenoids or other electrically operated valves are used.

Refrigerant Flow Rates

Refrigerant flow rates for R-22 and R-134a are indicated in Figures 1 and 2. To obtain total system flow rate, select the proper rate value and multiply by system capacity. Enter curves using saturated refrigerant temperature at the evaporator outlet and actual liquid temperature entering the liquid feed device (including subcooling in condensers and liquid-suction interchanger, if used).

Because Figures 1 and 2 are based on a saturated evaporator temperature, they may indicate slightly higher refrigerant flow rates than are actually in effect when suction vapor is superheated in excess of the conditions mentioned. Refrigerant flow rates may be reduced approximately 3% for each 10°F increase in superheat in the evaporator.

Suction-line superheating downstream of the evaporator from line heat gain from external sources should not be used to reduce evaluated mass flow, because it increases volumetric flow rate and line velocity per unit of evaporator capacity, but not mass flow rate. It should be considered when evaluating suction-line size for satisfactory oil return up risers.

Suction gas superheating from use of a liquid-suction heat exchanger has an effect on oil return similar to that of suction-line superheating. The liquid cooling that results from the heat exchange reduces mass flow rate per ton of refrigeration. This can be seen in Figures 1 and 2 because the reduced temperature of the liquid supplied to the evaporator feed valve has been taken into account.

Superheat caused by heat in a space not intended to be cooled is always detrimental because the volumetric flow rate increases with no compensating gain in refrigerating effect.

The preparation of this chapter is assigned to TC 10.3, Refrigerant Piping.

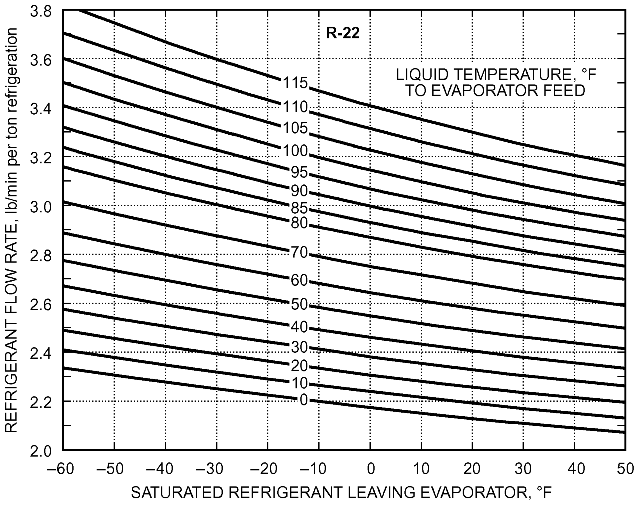


Fig. 1 Flow Rate per Ton of Refrigeration for Refrigerant 22

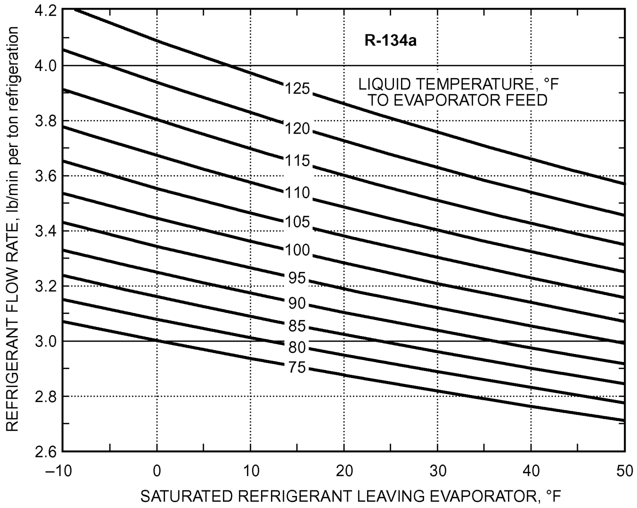


Fig. 2 Flow Rate per Ton of Refrigeration for Refrigerant 134a

REFRIGERANT LINE SIZING

In sizing refrigerant lines, cost considerations favor minimizing line sizes. However, suction and discharge line pressure drops cause loss of compressor capacity and increased power usage. Excessive liquid line pressure drops can cause liquid refrigerant to flash, resulting in faulty expansion valve operation. Refrigeration systems are designed so that friction pressure losses do not exceed a pressure differential equivalent to a corresponding change in the saturation boiling temperature. The primary measure for determining pressure drops is a given change in saturation temperature.

Pressure Drop Considerations

Pressure drop in refrigerant lines reduces system efficiency. Correct sizing must be based on minimizing cost and maximizing efficiency. Table 2 indicates the approximate effect of refrigerant pressure drop on an R-22 system operating at a 40°F saturated evaporator temperature with a 100°F saturated condensing temperature.

Pressure drop calculations are determined as normal pressure loss associated with a change in saturation temperature of the refrigerant. Typically, the refrigeration system is sized for pressure losses of 2°F or less for each segment of the discharge, suction, and liquid lines.

Table 2 Approximate Effect of Gas Line Pressure Drops on R-22 Compressor Capacity and Power[a]

Line Loss, °F	Capacity, %	Energy, %[b]
Suction Line		
0	100	100
2	96.4	104.8
4	92.9	108.1
Discharge Line		
0	100	100
2	99.1	103.0
4	98.2	106.3

[a] For system operating at 40°F saturated evaporator temperature and 100°F saturated condensing temperature.
[b] Energy percentage rated at hp/ton.

Liquid Lines. Pressure drop should not be so large as to cause gas formation in the liquid line, insufficient liquid pressure at the liquid feed device, or both. Systems are normally designed so that pressure drop in the liquid line from friction is not greater than that corresponding to about a 1 to 2°F change in saturation temperature. See Tables 3 to 9 for liquid-line sizing information.

Liquid subcooling is the only method of overcoming liquid line pressure loss to guarantee liquid at the expansion device in the evaporator. If subcooling is insufficient, flashing occurs in the liquid line and degrades system efficiency.

Friction pressure drops in the liquid line are caused by accessories such as solenoid valves, filter-driers, and hand valves, as well as by the actual pipe and fittings between the receiver outlet and the refrigerant feed device at the evaporator.

Liquid-line risers are a source of pressure loss and add to the total loss of the liquid line. Loss caused by risers is approximately 0.5 psi per foot of liquid lift. Total loss is the sum of all friction losses plus pressure loss from liquid risers.

Example 1 illustrates the process of determining liquid-line size and checking for total subcooling required.

Example 1. An R-22 refrigeration system using copper pipe operates at 40°F evaporator and 105°F condensing. Capacity is 5 tons, and the liquid line is 100 ft equivalent length with a riser of 20 ft. Determine the liquid-line size and total required subcooling.

Solution: From Table 3, the size of the liquid line at 1°F drop is 5/8 in. OD. Use the equation in Note 3 of Table 3 to compute actual temperature drop. At 5 tons,

Actual temperature drop = $1.0(5.0/6.7)^{1.8}$ = 0.59°F
Estimated friction loss = 0.59 × 3.05 = 1.8 psi
Loss for the riser = 20 × 0.5 = 10 psi
Total pressure losses = 10.0 + 1.8 = 11.8 psi
R-22 saturation pressure at 105°F condensing (see R-22 properties in Chapter 20, 2005 *ASHRAE Handbook—Fundamentals*) 210.8 psig
Initial pressure at beginning of liquid line 210.8 psig
Total liquid line losses – 11.8 psi
Net pressure at expansion device = 199 psig
The saturation temperature at 199 psig is 101.1°F.
Required subcooling to overcome the liquid losses = (105.0 – 101.1) or 3.9°F

Refrigeration systems that have no liquid risers and have the evaporator below the condenser/receiver benefit from a gain in pressure caused by liquid weight and can tolerate larger friction losses without flashing. Regardless of the liquid-line routing when flashing occurs, overall efficiency is reduced, and the system may malfunction.

The velocity of liquid leaving a partially filled vessel (such as a receiver or shell-and-tube condenser) is limited by the height of the liquid above the point at which the liquid line leaves the vessel, whether or not the liquid at the surface is subcooled. Because liquid

Table 3 Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 22 (Single- or High-Stage Applications)

Line Size		Suction Lines (Δt = 2°F)					Discharge Lines (Δt = 1°F, Δp = 3.05 psi)		Line Size		Liquid Lines	
		Saturated Suction Temperature, °F									See notes a and b	
Type L Copper, OD		−40	−20	0	20	40	Saturated Suction Temperature, °F		Type L Copper, OD		Vel. = 100 fpm	Δt = 1°F Δp = 3.05
		Corresponding Δp, psi/100 ft										
		0.79	1.15	1.6	2.22	2.91	−40	40				
1/2		—	—	—	0.40	0.6	0.75	0.85	1/2		2.3	3.6
5/8		—	0.32	0.51	0.76	1.1	1.4	1.6	5/8		3.7	6.7
7/8		0.52	0.86	1.3	2.0	2.9	3.7	4.2	7/8		7.8	18.2
1 1/8		1.1	1.7	2.7	4.0	5.8	7.5	8.5	1 1/8		13.2	37.0
1 3/8		1.9	3.1	4.7	7.0	10.1	13.1	14.8	1 3/8		20.2	64.7
1 5/8		3.0	4.8	7.5	11.1	16.0	20.7	23.4	1 5/8		28.5	102.5
2 1/8		6.2	10.0	15.6	23.1	33.1	42.8	48.5	2 1/8		49.6	213.0
2 5/8		10.9	17.8	27.5	40.8	58.3	75.4	85.4	2 5/8		76.5	376.9
3 1/8		17.5	28.4	44.0	65.0	92.9	120.2	136.2	3 1/8		109.2	601.5
3 5/8		26.0	42.3	65.4	96.6	137.8	178.4	202.1	3 5/8		147.8	895.7
4 1/8		36.8	59.6	92.2	136.3	194.3	251.1	284.4	4 1/8		192.1	1263.2
Steel									Steel			
IPS	SCH								IPS	SCH		
1/2	40	—	0.38	0.58	0.85	1.2	1.5	1.7	1/2	80	3.8	5.7
3/4	40	0.50	0.8	1.2	1.8	2.5	3.3	3.7	3/4	80	6.9	12.8
1	40	0.95	1.5	2.3	3.4	4.8	6.1	6.9	1	80	11.5	25.2
1 1/4	40	2.0	3.2	4.8	7.0	9.9	12.6	14.3	1 1/4	80	20.6	54.1
1 1/2	40	3.0	4.7	7.2	10.5	14.8	19.0	21.5	1 1/2	80	28.3	82.6
2	40	5.7	9.1	13.9	20.2	28.5	36.6	41.4	2	40	53.8	192.0
2 1/2	40	9.2	14.6	22.1	32.2	45.4	58.1	65.9	2 1/2	40	76.7	305.8
3	40	16.2	25.7	39.0	56.8	80.1	102.8	116.4	3	40	118.5	540.3
4	40	33.1	52.5	79.5	115.9	163.2	209.5	237.3	4	40	204.2	1101.2

Notes:

1. Table capacities are in tons of refrigeration.
   Δp = pressure drop from line friction, psi per 100 ft of equivalent line length
   Δt = corresponding change in saturation temperature, °F per 100 ft
2. Line capacity for other saturation temperatures Δt and equivalent lengths L_e

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

3. Saturation temperature Δt for other capacities and equivalent lengths L_e

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

4. Values based on 105°F condensing temperature. Multiply table capacities by the following factors for other condensing temperatures.

Condensing Temperature, °F	Suction Line	Discharge Line
80	1.11	0.79
90	1.07	0.88
100	1.03	0.95
110	0.97	1.04
120	0.90	1.10
130	0.86	1.18
140	0.80	1.26

[a] Sizing shown is recommended where any gas generated in receiver must return up condensate line to condenser without restricting condensate flow. Water-cooled condensers, where receiver ambient temperature may be higher than refrigerant condensing temperature, fall into this category.

[b] Line pressure drop Δp is conservative; if subcooling is substantial or line is short, a smaller size line may be used. Applications with very little subcooling or very long lines may require a larger line.

Table 4 Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 22 (Intermediate- or Low-Stage Duty)

Line Size	Suction Lines (Δt = 2°F)*							Discharge Lines (Δt = 2°F)*	Liquid Lines
Type L Copper, OD	Saturated Suction Temperature, °F								
	−90	−80	−70	−60	−50	−40	−30		
5/8								0.7	See Table 3
7/8	0.18	0.25	0.34	0.46	0.61	0.79	1.0	1.9	
1 1/8	0.36	0.51	0.70	0.94	1.2	1.6	2.1	3.8	
1 3/8	0.6	0.9	1.2	1.6	2.2	2.8	3.6	6.6	
1 5/8	1.0	1.4	1.9	2.6	3.4	4.5	5.7	10.5	
2 1/8	2.1	3.0	4.1	5.5	7.2	9.3	11.9	21.7	
2 5/8	3.8	5.3	7.2	9.7	12.7	16.5	21.1	38.4	
3 1/8	6.1	8.5	11.6	15.5	20.4	26.4	33.8	61.4	
3 5/8	9.1	12.7	17.3	23.1	30.4	39.4	50.2	91.2	
4 1/8	12.9	18.0	24.5	32.7	43.0	55.6	70.9	128.6	
5 1/8	23.2	32.3	43.9	58.7	77.1	99.8	126.9	229.5	
6 1/8	37.5	52.1	71.0	94.6	124.2	160.5	204.2	369.4	

Notes:

1. Table capacities are in tons of refrigeration.
   Δp = pressure drop from line friction, psi per 100 ft of equivalent line length
   Δt = corresponding change in saturation temperature, °F per 100 ft
2. Line capacity for other saturation temperatures Δt and equivalent lengths L_e

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

3. Saturation temperature Δt for other capacities and equivalent lengths L_e

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

4. Refer to refrigerant thermodynamic property tables (Chapter 20 of the 2005 *ASHRAE Handbook—Fundamentals*) for pressure drop corresponding to Δt.
5. Values based on 0°F condensing temperature. Multiply table capacities by the following factors for other condensing temperatures. Flow rates for discharge lines are based on –50°F evaporating temperature.

Condensing Temperature, °F	Suction Line	Discharge Line
−30	1.09	0.58
−20	1.06	0.71
−10	1.03	0.85
0	1.00	1.00
10	0.97	1.20
20	0.94	1.45
30	0.90	1.80

*See section on Pressure Drop Considerations.

in the vessel has a very low (or zero) velocity, the velocity V in the liquid line (usually at the vena contracta) is $V^2 = 2gh$, where h is the liquid height in the vessel. Gas pressure does not add to the velocity unless gas is flowing in the same direction. As a result, both gas and liquid flow through the line, limiting the rate of liquid flow. If this factor is not considered, excess operating charges in receivers and flooding of shell-and-tube condensers may result.

No specific data are available to precisely size a line leaving a vessel. If the height of liquid above the vena contracta produces the desired velocity, liquid leaves the vessel at the expected rate. Thus, if the level in the vessel falls to one pipe diameter above the bottom of the vessel from which the liquid line leaves, the capacity of copper lines for R-22 at 3 lb/min per ton of refrigeration is approximately as follows:

OD, in.	Tons
1 1/8	14
1 3/8	25
1 5/8	40
2 1/8	80
2 5/8	130
3 1/8	195
4 1/8	410

The whole liquid line need not be as large as the leaving connection. After the vena contracta, the velocity is about 40% less. If the line continues down from the receiver, the value of h increases. For a 200 ton capacity with R-22, the line from the bottom of the receiver should be about 3 1/8 in. After a drop of 1 ft, a reduction to 2 5/8 in. is satisfactory.

Suction Lines. Suction lines are more critical than liquid and discharge lines from a design and construction standpoint. Refrigerant lines should be sized to (1) provide a minimum pressure drop at full load, (2) return oil from the evaporator to the compressor under minimum load conditions, and (3) prevent oil from draining from an active evaporator into an idle one. A pressure drop in the suction line reduces a system's capacity because it forces the compressor to operate at a lower suction pressure to maintain a desired evaporating temperature in the coil. The suction line is normally sized to have a pressure drop from friction no greater than the equivalent of about a 2°F change in saturation temperature. See Tables 3 to 15 for suction line sizing information.

At suction temperatures lower than 40°F, the pressure drop equivalent to a given temperature change decreases. For example, at –40°F suction with R-22, the pressure drop equivalent to a 2°F change in saturation temperature is about 0.8 psi. Therefore,

Table 5 Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 134a (Single- or High-Stage Applications)

Line Size		Suction Lines (Δt = 2°F)					Discharge Lines (Δt = 1°F, Δp = 2.2 psi/100 ft)			Line Size		Liquid Lines	
		Saturated Suction Temperature, °F										See notes a and b	
Type L Copper, OD		0	10	20	30	40	Saturated Suction Temperature, °F			Type L Copper, OD		Velocity = 100 fpm	Δt = 1°F
		Corresponding Δp, psi/100 ft											
		1.00	1.19	1.41	1.66	1.93	0	20	40				Δp = 2.2
1/2		0.14	0.18	0.23	0.29	0.35	0.54	0.57	0.59	1/2		2.13	2.79
5/8		0.27	0.34	0.43	0.54	0.66	1.01	1.07	1.12	5/8		3.42	5.27
7/8		0.71	0.91	1.14	1.42	1.75	2.67	2.81	2.94	7/8		7.09	14.00
1 1/8		1.45	1.84	2.32	2.88	3.54	5.40	5.68	5.95	1 1/8		12.10	28.40
1 3/8		2.53	3.22	4.04	5.02	6.17	9.42	9.91	10.40	1 3/8		18.40	50.00
1 5/8		4.02	5.10	6.39	7.94	9.77	14.90	15.70	16.40	1 5/8		26.10	78.60
2 1/8		8.34	10.60	13.30	16.50	20.20	30.80	32.40	34.00	2 1/8		45.30	163.00
2 5/8		14.80	18.80	23.50	29.10	35.80	54.40	57.20	59.90	2 5/8		69.90	290.00
3 1/8		23.70	30.00	37.50	46.40	57.10	86.70	91.20	95.50	3 1/8		100.00	462.00
3 5/8		35.10	44.60	55.80	69.10	84.80	129.00	135.00	142.00	3 5/8		135.00	688.00
4 1/8		49.60	62.90	78.70	97.40	119.43	181.00	191.00	200.00	4 1/8		175.00	971.00
5 1/8		88.90	113.00	141.00	174.00	213.00	323.00	340.00	356.00	—		—	—
6 1/8		143.00	181.00	226.00	280.00	342.00	518.00	545.00	571.00	—		—	—
Steel										**Steel**			
IPS	**SCH**									**IPS**	**SCH**		
1/2	80	0.22	0.28	0.35	0.43	0.53	0.79	0.84	0.88	1/2	80	3.43	4.38
3/4	80	0.51	0.64	0.79	0.98	1.19	1.79	1.88	1.97	3/4	80	6.34	9.91
1	80	1.00	1.25	1.56	1.92	2.33	3.51	3.69	3.86	1	80	10.50	19.50
1 1/4	40	2.62	3.30	4.09	5.03	6.12	9.20	9.68	10.10	1 1/4	80	18.80	41.80
1 1/2	40	3.94	4.95	6.14	7.54	9.18	13.80	14.50	15.20	1 1/2	80	25.90	63.70
2	40	7.60	9.56	11.90	14.60	17.70	26.60	28.00	29.30	2	40	49.20	148.00
2 1/2	40	12.10	15.20	18.90	23.10	28.20	42.40	44.60	46.70	2 1/2	40	70.10	236.00
3	40	21.40	26.90	33.40	41.00	49.80	75.00	78.80	82.50	3	40	108.00	419.00
4	40	43.80	54.90	68.00	83.50	101.60	153.00	160.00	168.00	4	40	187.00	853.00

Notes:

1. Table capacities are in tons of refrigeration.

   Δp = pressure drop from line friction, psi per 100 ft of equivalent line length

   Δt = corresponding change in saturation temperature, °F per 100 ft

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

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

3. Saturation temperature Δt for other capacities and equivalent lengths L_e

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

4. Values based on 105°F condensing temperature. Multiply table capacities by the following factors for other condensing temperatures.

Condensing Temperature, °F	Suction Line	Discharge Line
80	1.158	0.804
90	1.095	0.882
100	1.032	0.961
110	0.968	1.026
120	0.902	1.078
130	0.834	1.156

[a] Sizing shown is recommended where any gas generated in receiver must return up condensate line to the condenser without restricting condensate flow. Water-cooled condensers, where receiver ambient temperature may be higher than refrigerant condensing temperature, fall into this category.

[b] Line pressure drop Δp is conservative; if subcooling is substantial or line is short, a smaller size line may be used. Applications with very little subcooling or very long lines may require a larger line.

low-temperature lines must be sized for a very low pressure drop, or higher equivalent temperature losses, with resultant loss in equipment capacity, must be accepted. For very low pressure drops, any suction or hot-gas risers must be sized properly to ensure oil entrainment up the riser so that oil is always returned to the compressor.

Where pipe size must be reduced to provide sufficient gas velocity to entrain oil up vertical risers at partial loads, greater pressure drops are imposed at full load. These can usually be compensated for by oversizing the horizontal and down run lines and components.

Discharge Lines. Pressure loss in hot-gas lines increases the required compressor power per unit of refrigeration and decreases compressor capacity. Table 2 illustrates power losses for an R-22 system at 40°F evaporator and 100°F condensing temperature. Pressure drop is minimized by generously sizing lines for low friction losses, but still maintaining refrigerant line velocities to entrain and carry oil along at all loading conditions. Pressure drop is normally designed not to exceed the equivalent of a 2°F change in saturation temperature. Recommended sizing tables are based on a 1°F change in saturation temperature.

Location and Arrangement of Piping

Refrigerant lines should be as short and direct as possible to minimize tubing and refrigerant requirements and pressure drops. Plan piping for a minimum number of joints using as few elbows and other fittings as possible, but provide sufficient flexibility to absorb compressor vibration and stresses caused by thermal expansion and contraction.

Arrange refrigerant piping so that normal inspection and servicing of the compressor and other equipment is not hindered. Do not obstruct the view of the oil-level sight glass or run piping so that it interferes with removing compressor cylinder heads, end bells, access plates, or any internal parts. Suction-line piping to the compressor should be arranged so that it will not interfere with removal of the compressor for servicing.

Provide adequate clearance between pipe and adjacent walls and hangers or between pipes for insulation installation. Use sleeves that are sized to permit installation of both pipe and insulation through floors, walls, or ceilings. Set these sleeves prior to pouring of concrete or erection of brickwork.

Run piping so that it does not interfere with passages or obstruct headroom, windows, and doors. Refer to ASHRAE *Standard* 15 and other governing local codes for restrictions that may apply.

Protection Against Damage to Piping

Protection against damage is necessary, particularly for small lines, which have a false appearance of strength. Where traffic is heavy, provide protection against impact from carelessly handled hand trucks, overhanging loads, ladders, and fork trucks.

Piping Insulation

All piping joints and fittings should be thoroughly leak-tested before insulation is sealed. Suction lines should be insulated to prevent sweating and heat gain. Insulation covering lines on which moisture can condense or lines subjected to outside conditions must be vapor-sealed to prevent any moisture travel through the insulation or condensation in the insulation. Many commercially available types are provided with an integral waterproof jacket for this purpose. Although the liquid line ordinarily does not require insulation, suction and liquid lines can be insulated as a unit on installations where the two lines are clamped together. When it passes through a warmer area, the liquid line should be insulated to minimize heat gain. Hot-gas discharge lines usually are not insulated; however, they should be insulated if the heat dissipated is objectionable or to prevent injury from high-temperature surfaces. In the latter case, it is not essential to provide insulation with a tight vapor seal because moisture condensation is not a problem unless the line is located outside. Hot-gas defrost lines are customarily insulated to minimize heat loss and condensation of gas inside the piping.

All joints and fittings should be covered, but it is not advisable to do so until the system has been thoroughly leak-tested. See Chapter 33 for additional information.

Vibration and Noise in Piping

Vibration transmitted through or generated in refrigerant piping and the resulting objectionable noise can be eliminated or minimized by proper piping design and support.

Two undesirable effects of vibration of refrigerant piping are (1) physical damage to the piping, which can break brazed joints and, consequently, lose charge; and (2) transmission of noise through the piping itself and through building construction that may come into direct contact with the piping.

In refrigeration applications, piping vibration can be caused by rigid connection of the refrigerant piping to a reciprocating compressor. Vibration effects are evident in all lines directly connected to the compressor or condensing unit. It is thus impossible to eliminate vibration in piping; it is only possible to mitigate its effects.

Flexible metal hose is sometimes used to absorb vibration transmission along smaller pipe sizes. For maximum effectiveness, it should be installed parallel to the crankshaft. In some cases, two isolators may be required, one in the horizontal line and the other in the vertical line at the compressor. A rigid brace on the end of the flexible hose away from the compressor is required to prevent vibration of the hot-gas line beyond the hose.

Flexible metal hose is not as efficient in absorbing vibration on larger pipes because it is not actually flexible unless the ratio of length to diameter is relatively great. In practice, the length is often limited, so flexibility is reduced in larger sizes. This problem is best solved by using flexible piping and isolation hangers where the piping is secured to the structure.

When piping passes through walls, through floors, or inside furring, it must not touch any part of the building and must be supported only by the hangers (provided to avoid transmitting vibration to the building); this eliminates the possibility of walls or ceilings acting as sounding boards or diaphragms. When piping is erected where access is difficult after installation, it should be supported by isolation hangers.

Vibration and noise from a piping system can also be caused by gas pulsations from the compressor operation or from turbulence in the gas, which increases at high velocities. It is usually more apparent in the discharge line than in other parts of the system.

When gas pulsations caused by the compressor create vibration and noise, they have a characteristic frequency that is a function of the number of gas discharges by the compressor on each revolution. This frequency is not necessarily equal to the number of cylinders, because on some compressors two pistons operate together. It is also varied by the angular displacement of the cylinders, such as in V-type compressors. Noise resulting from gas pulsations is usually objectionable only when the piping system amplifies the pulsation by resonance. On single-compressor systems, resonance can be reduced by changing the size or length of the resonating line or by installing a properly sized hot-gas muffler in the discharge line immediately after the compressor discharge valve. On a paralleled compressor system, a harmonic frequency from the different speeds of multiple compressors may be apparent. This noise can sometimes be reduced by installing mufflers.

When noise is caused by turbulence and isolating the line is not effective enough, installing a larger-diameter pipe to reduce gas velocity is sometimes helpful. Also, changing to a line of heavier wall or from copper to steel to change the pipe natural frequency may help.

Table 6 Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 404a (Single- or High-Stage Applications)

Line Size		Suction Lines (Δt = 2°F)						Discharge Lines (Δt = 1°F, Δp = 3.55 psi)						Liquid Lines		
		Saturated Suction Temperature, °F						Saturated Suction Temperature, °F						See note a		
Type L Copper, OD		–60	–40	–20	0	20	40	–60	–40	–20	0	20	40	Velocity = 100 fpm	Δt = 1°F Drop Δp = 3.6	Δt = 5°F Drop Δp = 17.4
		Corresponding Δp, psi/100 ft						Corresponding Δp, psi/100 ft								
		0.64	0.97	1.41	1.96	2.62	3.44	3.55	3.55	3.55	3.55	3.55	3.55			
1/2		0.05	0.09	0.15	0.24	0.36	0.53	0.56	0.61	0.65	0.70	0.75	0.79	1.3	2.6	6.09
5/8		0.09	0.16	0.28	0.44	0.68	1.00	1.04	1.14	1.23	1.31	1.40	1.48	2.1	4.9	11.39
3/4		0.15	0.28	0.47	0.76	1.15	1.70	1.77	1.93	2.09	2.23	2.38	2.51	3.1	8.1	18.87
7/8		0.24	0.43	0.73	1.17	1.78	2.63	2.73	2.98	3.22	3.44	3.66	3.87	4.4	12.8	29.81
1 1/8		0.49	0.88	1.49	2.37	3.61	5.31	5.52	6.01	6.49	6.96	7.40	7.81	7.5	25.9	60.17
1 3/8		0.86	1.54	2.59	4.13	6.28	9.23	9.60	10.46	11.29	12.10	12.87	13.58	11.4	45.2	104.41
1 5/8		1.36	2.44	4.10	6.53	9.92	14.57	15.14	16.49	17.80	19.07	20.28	21.41	16.1	71.4	164.68
2 1/8		2.83	5.07	8.52	13.53	20.51	30.06	31.29	34.08	36.80	39.43	41.93	44.26	28.0	147.9	339.46
2 5/8		5.03	8.97	15.07	23.88	36.16	52.96	55.04	59.95	64.74	69.36	73.76	77.85	43.2	261.2	597.42
3 1/8		8.05	14.34	24.02	38.05	57.56	84.33	87.66	95.48	103.11	110.47	117.48	124.00	61.7	416.2	950.09
3 5/8		11.98	21.31	35.73	56.53	85.39	125.18	129.88	141.46	152.76	163.67	174.05	183.71	83.5	618.4	1407.96
4 1/8		16.93	30.09	50.32	79.66	120.39	176.20	182.83	199.13	215.05	230.40	245.01	258.61	108.5	871.6	1982.40
5 1/8		30.35	53.85	89.97	142.32	214.82	313.91	325.75	354.81	383.16	410.51	436.55	460.78	169.1	1554.2	3525.99
6 1/8		48.89	86.74	144.47	228.50	344.70	502.77	521.74	568.28	613.69	657.49	699.20	738.00	243.1	2497.7	5648.67
8 1/8		101.60	179.88	299.39	472.46	710.75	1037.34	1076.62	1172.66	1266.36	1356.75	1442.81	1522.89	424.6	5159.7	11660.71
Steel																
IPS	SCH															
3/8	80	0.04	0.07	0.11	0.18	0.27	0.39	0.40	0.44	0.47	0.51	0.54	0.57	1.3	1.9	4.3
1/2	80	0.08	0.14	0.22	0.35	0.53	0.76	0.79	0.86	0.93	0.99	1.06	1.12	2.1	3.8	8.5
3/4	80	0.18	0.31	0.51	0.79	1.18	1.71	1.78	1.93	2.09	2.24	2.38	2.51	3.9	8.6	19.2
1	80	0.35	0.60	0.99	1.55	2.32	3.36	3.48	3.79	4.09	4.38	4.66	4.92	6.5	16.9	37.5
1 1/4	80	0.75	1.30	2.13	3.33	4.97	7.20	7.45	8.12	8.77	9.39	9.99	10.54	11.6	36.3	80.3
1 1/2	80	1.14	1.98	3.26	5.08	7.57	10.96	11.35	12.37	13.35	14.31	15.21	16.06	16.0	55.3	122.3
2	40	2.65	4.61	7.55	11.78	17.57	25.45	26.36	28.71	31.01	33.22	35.33	37.29	30.4	128.4	283.5
2 1/2	40	4.23	7.34	12.04	18.74	27.94	40.49	41.93	45.67	49.32	52.84	56.19	59.31	43.3	204.7	450.9
3	40	7.48	12.98	21.26	33.11	49.37	71.55	74.10	80.71	87.16	93.38	99.31	104.82	66.9	361.6	796.8
4	40	15.30	26.47	43.34	67.50	100.66	145.57	150.75	164.20	177.32	189.98	202.03	213.24	115.3	735.6	1623.0
5	40	27.58	47.78	78.24	121.87	181.32	262.52	272.21	296.49	320.19	343.04	364.80	385.05	181.1	1328.2	2927.2
6	40	44.58	77.26	126.52	197.09	293.24	424.04	439.72	478.94	517.21	554.13	589.28	621.99	261.7	2148.0	4728.3
8	40	91.40	158.09	258.81	402.66	599.91	867.50	898.42	978.56	1056.75	1132.18	1203.99	1270.82	453.2	4394.4	9674.1
10	40	165.52	286.19	468.14	728.40	1083.73	1569.40	1625.34	1770.31	1911.78	2048.23	2178.15	2299.05	714.4	7938.5	17,477.4
12	ID[b]	264.36	457.37	748.94	1163.62	1733.87	2507.30	2600.54	2832.50	3058.84	3277.16	3485.04	3678.47	1024.6	12,681.8	27,963.7
14	30	342.81	592.13	968.21	1506.59	2244.98	3246.34	3362.07	3661.96	3954.59	4236.83	4505.59	4755.67	1249.2	16,419.6	36,152.5
16	30	493.87	852.84	1395.24	2171.13	3230.27	4678.48	4845.26	5277.44	5699.16	6105.92	6493.24	6853.65	1654.7	23,662.2	52,101.2

[a] Sizing shown is recommended where any gas generated in receiver must return up condensate line to condenser without restricting condensate flow. Water-cooled condensers, where receiver ambient temperature may be higher than refrigerant condensing temperature, fall into this category.

[b] Pipe inside diameter is same as nominal pipe size.

Notes:

1. Table capacities are in tons of refrigeration.
   Δp = pressure drop from line friction, psi per 100 ft of equivalent line length
   Δt = corresponding change in saturation temperature, °F per 100 ft
2. Line capacity for other saturation temperatures Δt and equivalent lengths L_e

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

3. Saturation temperature Δt for other capacities and equivalent lengths L_e

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

4. Tons based on standard refrigerant cycle of 105°F liquid and saturated evaporator outlet temperature. Liquid tons based on 20°F evaporator temperature.
5. Thermophysical properties and viscosity data based on calculations from NIST REFPROP program Version 6.01.
6. For brazed Type L copper tubing larger than 1 1/8 in. OD for discharge or liquid service, see Safety Requirements section.
7. Values based on 105°F condensing temperature. Multiply table capacities by the following factors for other condensing temperatures.

Cond. Temp., °F	Suction Line	Discharge Line
80	1.246	0.870
90	1.150	0.922
100	1.051	0.974
110	0.948	1.009
120	0.840	1.026
130	0.723	1.043

Table 7 Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 507 (Single- or High-Stage Applications)

Line Size		Suction Lines (Δt = 2°F)						Discharge Lines (Δt = 1°F, Δp = 3.65 psi)						Liquid Lines		
		Saturated Suction Temperature, °F						Saturated Suction Temperature, °F						See note a		
Type L Copper, OD		–60	–40	–20	0	20	40	–60	–40	–20	0	20	40	Velocity = 100 fpm	Δt = 1°F Drop Δp = 3.65	Δt = 5°F Drop Δp = 17.8
		Corresponding Δp, psi/100 ft						Corresponding Δp, psi/100 ft								
		0.67	1.01	1.46	2.02	2.71	3.6	3.65	3.65	3.65	3.65	3.65	3.65			
1/2		0.05	0.09	0.15	0.24	0.37	0.55	0.55	0.60	0.65	0.70	0.75	0.79	1.3	2.5	5.96
5/8		0.09	0.17	0.28	0.45	0.69	1.02	1.04	1.13	1.22	1.31	1.40	1.48	2.0	4.7	11.13
3/4		0.16	0.28	0.48	0.77	1.17	1.74	1.76	1.92	2.08	2.24	2.38	2.52	3.0	7.9	18.45
7/8		0.25	0.44	0.74	1.18	1.81	2.68	2.72	2.97	3.22	3.45	3.68	3.89	4.2	12.5	29.14
1 1/8		0.50	0.90	1.51	2.40	3.66	5.41	5.48	5.99	6.49	6.96	7.41	7.84	7.2	25.2	58.74
1 3/8		0.88	1.57	2.63	4.18	6.35	9.41	9.54	10.42	11.28	12.11	12.90	13.63	11.0	44.0	102.09
1 5/8		1.39	2.48	4.17	6.61	10.04	14.84	15.04	16.43	17.79	19.09	20.34	21.50	15.6	69.5	161.04
2 1/8		2.91	5.17	8.65	13.70	20.76	30.66	31.03	33.90	36.70	39.40	41.96	44.36	27.1	144.0	331.97
2 5/8		5.15	9.14	15.27	24.19	36.62	54.04	54.69	59.74	64.68	69.43	73.96	78.18	41.8	254.3	584.28
3 1/8		8.24	14.61	24.40	38.55	58.29	85.90	86.95	94.98	102.84	110.39	117.58	124.29	59.6	405.2	929.27
3 5/8		12.27	21.75	36.22	57.15	86.47	127.52	129.07	140.99	152.66	163.87	174.54	184.50	80.6	601.0	1377.19
4 1/8		17.34	30.66	51.13	80.55	121.93	179.33	181.70	198.48	214.91	230.69	245.71	259.74	104.8	847.0	1935.27
5 1/8		31.09	54.88	91.25	143.93	217.14	319.89	323.48	353.35	382.60	410.70	437.44	462.40	163.3	1513.6	3449.44
6 1/8		49.99	88.20	146.87	230.77	348.36	512.29	518.62	566.52	613.40	658.45	701.32	741.34	234.8	2427.4	5526.55
8 1/8		103.91	182.97	303.62	477.80	720.09	1057.14	1070.49	1169.35	1266.13	1359.11	1447.60	1530.21	410.1	5019.4	11,383.18
Steel																
IPS	**SCH**															
3/8	80	0.04	0.07	0.12	0.18	0.27	0.39	0.40	0.43	0.47	0.51	0.54	0.57	1.2	1.9	4.2
1/2	80	0.08	0.14	0.23	0.35	0.53	0.77	0.78	0.86	0.93	0.99	1.06	1.12	2.1	3.7	8.3
3/4	80	0.18	0.31	0.51	0.80	1.20	1.74	1.76	1.93	2.09	2.24	2.39	2.52	3.8	8.4	18.7
1	80	0.35	0.61	1.01	1.57	2.34	3.41	3.45	3.77	4.08	4.38	4.67	4.94	6.3	16.4	36.6
1 1/4	80	0.76	1.32	2.16	3.36	5.02	7.32	7.39	8.08	8.74	9.39	10.00	10.57	11.2	35.2	78.4
1 1/2	80	1.16	2.01	3.29	5.12	7.65	11.15	11.26	12.30	13.32	14.30	15.23	16.10	15.5	53.8	119.4
2	40	2.70	4.68	7.65	11.89	17.76	25.88	26.15	28.56	30.93	33.20	35.36	37.38	29.4	124.8	276.7
2 1/2	40	4.31	7.45	12.18	18.93	28.24	41.17	41.59	45.43	49.19	52.80	56.24	59.45	41.9	198.9	440.6
3	40	7.63	13.19	21.54	33.45	49.90	72.75	73.50	80.29	86.93	93.32	99.39	105.06	64.6	351.5	777.9
4	40	15.57	26.88	43.92	68.12	101.75	148.00	149.53	163.33	176.85	189.84	202.20	213.74	111.4	714.9	1586.3
5	40	28.10	48.52	79.19	122.99	183.27	266.91	270.00	294.93	319.34	342.79	365.11	385.94	174.9	1290.8	2857.5
6	40	45.48	78.45	128.06	198.91	296.40	431.69	436.14	476.41	515.85	553.73	589.78	623.44	252.8	2087.5	4622.0
8	40	93.13	160.66	261.94	406.93	606.38	882.01	891.10	973.39	1053.96	1131.36	1205.02	1273.79	437.7	4270.8	9443.9
10	40	168.64	290.60	473.82	735.12	1095.44	1595.65	1612.10	1760.97	1906.72	2046.75	2180.00	2304.41	690.0	7715.1	17,086.7
12	ID[b]	269.75	464.87	758.01	1174.36	1752.56	2553.03	2579.36	2817.55	3050.75	3274.79	3488.00	3687.06	989.6	12,324.9	27,298.3
14	30	349.22	601.87	979.92	1520.49	2269.19	3300.65	3334.69	3642.64	3944.13	4233.77	4509.42	4766.76	1206.5	15,957.5	35,292.2
16	30	503.20	866.37	1414.32	2191.17	3265.09	4756.74	4805.79	5249.60	5684.09	6101.51	6498.76	6869.63	1598.2	22,996.2	50,861.5

[a] Sizing shown is recommended where any gas generated in receiver must return up condensate line to condenser without restricting condensate flow. Water-cooled condensers, where receiver ambient temperature may be higher than refrigerant condensing temperature, fall into this category.

[b] Pipe inside diameter is same as nominal pipe size.

Notes:

1. Table capacities are in tons of refrigeration.
   Δp = pressure drop from line friction, psi per 100 ft of equivalent line length
   Δt = corresponding change in saturation temperature, °F per 100 ft
2. Line capacity for other saturation temperatures Δt and equivalent lengths L_e

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

3. Saturation temperature Δt for other capacities and equivalent lengths L_e

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

4. Tons based on standard refrigerant cycle of 105°F liquid and saturated evaporator outlet temperature. Liquid tons based on 20°F evaporator temperature.
5. Thermophysical properties and viscosity data based on calculations from NIST REFPROP program Version 6.01.
6. For brazed Type L copper tubing larger than 1 1/8 in. OD for discharge or liquid service, see Safety Requirements section.
7. Values based on 105°F condensing temperature. Multiply table capacities by the following factors for other condensing temperatures.

Cond. Temp., °F	Suction Line	Discharge Line
80	1.267	0.873
90	1.163	0.924
100	1.055	0.975
110	0.944	1.005
120	0.826	1.014
130	0.701	1.024

Table 8 Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 410a (Single- or High-Stage Applications)

Line Size		Suction Lines (Δt = 2°F)						Discharge Lines (Δt = 1°F, Δp = 4.75 psi)						Liquid Lines		
		Saturated Suction Temperature, °F						Saturated Suction Temperature, °F						See note a		
Type L Copper, OD		–60	–40	–20	0	20	40	–60	–40	–20	0	20	40	Velocity = 100 fpm	Δt = 1°F Drop Δp = 4.75	Δt = 5°F Drop Δp = 23.3
		Corresponding Δp, psi/100 ft						Corresponding Δp, psi/100 ft								
		0.84	1.27	1.85	2.57	3.46	4.5	4.75	4.75	4.75	4.75	4.75	4.75			
1/2		0.10	0.17	0.27	0.42	0.62	0.89	1.13	1.17	1.22	1.26	1.30	1.33	2.0	4.6	10.81
5/8		0.18	0.31	0.51	0.79	1.17	1.67	2.11	2.20	2.29	2.36	2.43	2.49	3.2	8.6	20.24
3/4		0.31	0.53	0.87	1.35	2.00	2.84	3.59	3.74	3.88	4.02	4.14	4.23	4.7	14.3	33.53
7/8		0.48	0.83	1.35	2.08	3.08	4.39	5.53	5.76	5.99	6.19	6.38	6.52	6.7	22.6	52.92
1 1/8		0.98	1.69	2.74	4.22	6.23	8.86	11.16	11.64	12.09	12.50	12.88	13.17	11.4	45.8	106.59
1 3/8		1.72	2.95	4.78	7.34	10.85	15.41	19.39	20.21	21.00	21.72	22.37	22.88	17.4	79.7	185.04
1 5/8		2.73	4.67	7.56	11.61	17.14	24.28	30.63	31.92	33.16	34.30	35.33	36.14	24.6	125.9	291.48
2 1/8		5.69	9.71	15.71	24.05	35.45	50.19	63.20	65.88	68.44	70.78	72.90	74.57	42.8	260.7	601.13
2 5/8		10.09	17.17	27.74	42.45	62.53	88.43	111.20	115.90	120.41	124.53	128.25	131.20	66.0	459.7	1056.39
3 1/8		16.15	27.44	44.24	67.77	99.53	140.83	177.12	184.62	191.80	198.36	204.29	208.98	94.2	733.0	1680.52
3 5/8		24.06	40.84	65.81	100.50	147.66	208.65	262.44	273.54	284.19	293.90	302.70	309.64	127.4	1087.5	2491.00
4 1/8		33.98	57.58	92.66	141.61	208.22	293.70	369.45	385.08	400.07	413.75	426.13	435.90	165.7	1530.2	3500.91
5 1/8		60.95	103.03	165.73	253.05	370.82	523.21	658.32	686.18	712.88	737.26	759.31	776.72	258.2	2729.8	6228.40
6 1/8		98.05	166.00	266.14	405.75	594.85	839.82	1054.47	1099.10	1141.87	1180.91	1216.24	1244.13	371.1	4383.7	9980.43
8 1/8		203.77	344.31	551.73	840.04	1229.69	1733.02	2176.50	2268.62	2356.89	2437.49	2510.41	2567.98	648.3	9049.5	20,561.73
Steel																
IPS	SCH															
3/8	80	0.08	0.13	0.21	0.32	0.46	0.65	0.81	0.84	0.88	0.91	0.93	0.95	1.9	3.4	7.6
1/2	80	0.16	0.26	0.41	0.62	0.91	1.27	1.59	1.66	1.73	1.78	1.84	1.88	3.2	6.7	15.0
3/4	80	0.35	0.59	0.93	1.41	2.04	2.86	3.59	3.74	3.88	4.02	4.14	4.23	6.0	15.1	33.6
1	80	0.69	1.15	1.83	2.75	4.00	5.59	7.02	7.32	7.60	7.86	8.10	8.28	10.0	29.5	65.8
1 1/4	80	1.49	2.48	3.92	5.90	8.58	12.00	15.03	15.67	16.28	16.83	17.34	17.74	17.7	63.3	140.9
1 1/2	80	2.28	3.79	5.98	9.01	13.06	18.27	22.89	23.86	24.79	25.64	26.41	27.01	24.4	96.6	214.7
2	40	5.30	8.80	13.89	20.91	30.32	42.43	53.16	55.41	57.57	59.54	61.32	62.73	46.4	224.2	498.0
2 1/2	40	8.46	14.02	22.13	33.29	48.23	67.48	84.56	88.14	91.57	94.70	97.53	99.77	66.2	356.5	793.0
3	40	14.98	24.81	39.10	58.81	85.22	119.26	149.44	155.76	161.82	167.36	172.37	176.32	102.2	630.0	1398.4
4	40	30.58	50.56	79.68	119.77	173.76	242.63	304.02	316.88	329.21	340.47	350.66	358.70	176.1	1284.6	2851.7
5	40	55.19	91.27	143.84	216.23	312.97	437.56	548.97	572.20	594.46	614.79	633.19	647.71	276.5	2313.7	5137.0
6	40	89.34	147.57	232.61	349.71	506.16	707.69	886.76	924.29	960.25	993.09	1022.80	1046.26	399.6	3741.9	8308.9
8	40	182.90	301.82	475.80	715.45	1035.51	1445.92	1811.80	1888.48	1961.96	2029.05	2089.76	2137.68	692.0	7655.3	16,977.6
10	40	331.22	546.64	860.67	1292.44	1870.67	2615.83	3277.74	3416.46	3549.40	3670.77	3780.59	3867.29	1090.7	13,829.2	30,716.4
12	ID[b]	529.89	873.19	1376.89	2064.68	2992.85	4185.32	5244.38	5466.33	5679.03	5873.23	6048.94	6187.65	1564.3	22,125.4	49,074.9
14	30	685.86	1130.48	1779.99	2673.23	3875.08	5410.92	6780.14	7067.08	7342.06	7593.13	7820.29	7999.63	1907.2	28,647.5	63,445.8
16	30	988.28	1628.96	2569.05	3852.37	5575.79	7797.98	9771.20	10,184.73	10,581.02	10,942.85	11,270.23	11,528.68	2526.4	41,220.5	91,435.1

[a] Sizing shown is recommended where any gas generated in receiver must return up condensate line to condenser without restricting condensate flow. Water-cooled condensers, where receiver ambient temperature may be higher than refrigerant condensing temperature, fall into this category.

[b] Pipe inside diameter is same as nominal pipe size.

Notes:

1. Table capacities are in tons of refrigeration.
   Δp = pressure drop from line friction, psi per 100 ft of equivalent line length
   Δt = corresponding change in saturation temperature, °F per 100 ft
2. Line capacity for other saturation temperatures Δt and equivalent lengths L_e

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

3. Saturation temperature Δt for other capacities and equivalent lengths L_e

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

4. Tons based on standard refrigerant cycle of 105°F liquid and saturated evaporator outlet temperature. Liquid tons based on 20°F evaporator temperature.
5. Thermophysical properties and viscosity data based on calculations from NIST REFPROP program Version 6.01.
6. For brazed Type L copper tubing larger than 5/8 in. OD for discharge or liquid service, see Safety Requirements section.
7. Values based on 105°F condensing temperature. Multiply table capacities by the following factors for other condensing temperatures.

Cond. Temp., °F	Suction Line	Discharge Line
80	1.170	0.815
90	1.104	0.889
100	1.035	0.963
110	0.964	1.032
120	0.889	1.096
130	0.808	1.160

Table 9 Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 407c (Single- or High-Stage Applications)

Line Size		Suction Lines (Δt = 2°F)						Discharge Lines (Δt = 1°F, Δp = 3.3 psi)						Liquid Lines		
		Saturated Suction Temperature, °F						Saturated Suction Temperature, °F						See note a		
Type L Copper, OD		–60	–40	–20	0	20	40	–60	–40	–20	0	20	40	Velocity = 100 fpm	Δt = 1°F Drop Δp = 3.5	Δt = 5°F Drop Δp = 16.9
		Corresponding Δp, psi/100 ft						Corresponding Δp, psi/100 ft								
		0.435	0.7	1.06	1.55	2.16	2.92	3.3	3.3	3.3	3.3	3.3	3.3			
1/2		0.04	0.08	0.14	0.23	0.36	0.54	0.71	0.75	0.78	0.82	0.86	0.89	2.1	3.8	8.90
5/8		0.08	0.15	0.26	0.43	0.68	1.02	1.33	1.40	1.47	1.54	1.61	1.67	3.4	7.1	16.68
3/4		0.14	0.26	0.45	0.74	1.16	1.74	2.26	2.38	2.50	2.62	2.73	2.84	4.9	11.8	27.66
7/8		0.21	0.40	0.70	1.15	1.79	2.68	3.48	3.67	3.86	4.05	4.22	4.38	6.9	18.7	43.73
1 1/8		0.44	0.82	1.42	2.33	3.63	5.42	7.05	7.43	7.82	8.19	8.53	8.86	11.8	37.9	88.21
1 3/8		0.77	1.43	2.48	4.07	6.33	9.45	12.25	12.92	13.59	14.23	14.83	15.40	18.0	66.2	153.45
1 5/8		1.23	2.27	3.93	6.44	10.00	14.93	19.33	20.39	21.44	22.46	23.40	24.30	25.5	104.7	241.93
2 1/8		2.56	4.74	8.18	13.37	20.72	30.90	39.99	42.17	44.35	46.45	48.40	50.27	44.4	217.1	499.23
2 5/8		4.55	8.42	14.49	23.64	36.62	54.50	70.56	74.41	78.25	81.96	85.40	88.70	68.5	383.7	879.85
3 1/8		7.30	13.47	23.15	37.76	58.34	86.88	112.34	118.47	124.59	130.50	135.97	141.22	97.7	611.3	1401.50
3 5/8		10.90	20.08	34.44	56.15	86.64	128.89	166.39	175.47	184.54	193.29	201.39	209.17	132.2	907.9	2076.59
4 1/8		15.42	28.37	48.62	79.21	122.10	181.34	234.63	247.42	260.22	272.56	283.98	294.95	171.8	1281.5	2923.40
5 1/8		27.70	50.85	86.97	141.60	218.05	323.50	417.91	440.69	463.48	485.46	505.80	525.33	267.8	2288.8	5209.13
6 1/8		44.70	81.91	140.04	227.86	350.42	519.62	670.58	707.15	743.71	778.97	811.62	842.96	385.0	3676.9	8344.10
8 1/8		92.98	170.14	290.93	471.55	725.11	1072.54	1383.29	1458.72	1534.15	1606.88	1674.23	1738.88	672.4	7599.4	17,220.64
Steel																
IPS	**SCH**															
3/8	80	0.04	0.07	0.11	0.18	0.27	0.40	0.52	0.55	0.57	0.60	0.63	0.65	2.0	2.9	6.4
1/2	80	0.07	0.13	0.22	0.35	0.54	0.79	1.02	1.07	1.13	1.18	1.23	1.28	3.4	5.7	12.6
3/4	80	0.16	0.30	0.50	0.80	1.22	1.79	2.29	2.42	2.54	2.66	2.78	2.88	6.2	12.8	28.4
1	80	0.32	0.58	0.98	1.57	2.38	3.50	4.50	4.74	4.99	5.22	5.44	5.65	10.3	25.1	55.6
1 1/4	80	0.69	1.25	2.10	3.37	5.12	7.50	9.63	10.15	10.68	11.18	11.65	12.10	18.4	53.7	118.9
1 1/2	80	1.06	1.91	3.21	5.13	7.79	11.44	14.66	15.46	16.26	17.03	17.74	18.43	25.4	82.0	181.1
2	40	2.49	4.46	7.47	11.93	18.13	26.57	34.04	35.89	37.75	39.54	41.20	42.79	48.1	190.3	420.6
2 1/2	40	3.97	7.11	11.90	19.01	28.83	42.25	54.25	57.21	60.16	63.02	65.66	68.19	68.6	303.2	669.0
3	40	7.04	12.59	21.05	33.59	50.94	74.66	95.76	100.99	106.21	111.24	115.90	120.38	106.0	535.7	1182.3
4	40	14.38	25.70	42.97	68.47	103.84	152.24	195.04	205.68	216.31	226.57	236.06	245.18	182.6	1092.0	2405.3
5	40	26.00	46.36	77.55	123.61	187.25	274.21	351.31	370.46	389.62	408.09	425.19	441.61	286.8	1969.0	4343.2
6	40	42.13	75.15	125.49	199.88	302.82	443.47	568.16	599.14	630.12	659.99	687.65	714.21	414.5	3184.3	7015.7
8	40	86.32	153.84	256.66	408.86	619.47	907.26	1162.36	1225.74	1289.12	1350.24	1406.83	1461.15	717.7	6514.5	14,334.3
10	40	156.54	278.57	464.86	739.58	1120.60	1638.95	2102.83	2217.49	2332.15	2442.72	2545.10	2643.38	1131.3	11,784.6	25,932.3
12	ID[b]	250.23	445.65	742.54	1183.19	1790.17	2622.17	3359.45	3542.64	3725.82	3902.46	4066.02	4223.03	1622.5	18,826.0	41,491.5
14	30	324.38	576.93	961.33	1529.58	2317.81	3395.13	4349.77	4586.95	4824.14	5052.85	5264.62	5467.92	1978.2	24,374.8	53,641.7
16	30	468.29	831.27	1385.24	2204.17	3340.17	4885.19	6258.81	6600.09	6941.37	7270.46	7575.17	7867.69	2620.4	35,126.4	77,305.8

[a] Sizing shown is recommended where any gas generated in receiver must return up condensate line to condenser without restricting condensate flow. Water-cooled condensers, where receiver ambient temperature may be higher than refrigerant condensing temperature, fall into this category.

[b] Pipe inside diameter is same as nominal pipe size.

Notes:

1. Table capacities are in tons of refrigeration.
   Δp = pressure drop from line friction, psi per 100 ft of equivalent line length
   Δt = corresponding change in saturation temperature, °F per 100 ft
2. Line capacity for other saturation temperatures Δt and equivalent lengths L_e

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

3. Saturation temperature Δt for other capacities and equivalent lengths L_e

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

4. Tons based on standard refrigerant cycle of 105°F liquid and saturated evaporator outlet temperature. Liquid tons based on 20°F evaporator temperature.
5. Thermophysical properties and viscosity data based on calculations from NIST REFPROP program Version 6.01.
6. For brazed Type L copper tubing larger than 2 1/8 in. OD for discharge or liquid service, see Safety Requirements section.
7. Values based on 105°F condensing temperature. Multiply table capacities by the following factors for other condensing temperatures.

Cond. Temp., °F	Suction Line	Discharge Line
80	1.163	0.787
90	1.099	0.872
100	1.033	0.957
110	0.966	1.036
120	0.896	1.109
130	0.824	1.182

Refrigerant Line Capacity Tables

Tables 3 to 9 show line capacities in tons of refrigeration for R-22, R-134a, R-404a, R-507, R-410a, and R-407c. Capacities in the tables are based on the refrigerant flow that develops a friction loss, per 100 ft of equivalent pipe length, corresponding to a 2°F change in the saturation temperature (Δt) in the suction line, and a 1°F change in the discharge line. The capacities shown for liquid lines are for pressure losses corresponding to 1 and 5°F change in saturation temperature and also for velocity corresponding to 100 fpm. Tables 10 to 15 show capacities for the same refrigerants based on reduced suction line pressure loss corresponding to 1.0 and 0.5°F per 100 ft equivalent length of pipe. These tables may be used when designing system piping to minimize suction line pressure drop.

The refrigerant line sizing capacity tables are based on the Darcy-Weisbach relation and friction factors as computed by the Colebrook function (Colebrook 1938, 1939). Tubing roughness height is 0.000005 ft for copper and 0.00015 ft for steel pipe. Viscosity extrapolations and adjustments for pressures other than 1 atm were based on correlation techniques as presented by Keating and Matula (1969). Discharge gas superheat was 80°F for R-134a and 105°F for R-22.

The refrigerant cycle for determining capacity is based on saturated gas leaving the evaporator. The calculations neglect the presence of oil and assume nonpulsating flow.

For additional charts and discussion of line sizing refer to Atwood (1990), Timm (1991), and Wile (1977).

Equivalent Lengths of Valves and Fittings

Refrigerant line capacity tables are based on unit pressure drop per 100 ft length of straight pipe, or per combination of straight pipe, fittings, and valves with friction drop equivalent to a 100 ft length of straight pipe.

Generally, pressure drop through valves and fittings is determined by establishing the equivalent straight length of pipe of the same size with the same friction drop. Line sizing tables can then be used directly. Tables 16 to 18 give equivalent lengths of straight pipe for various fittings and valves, based on nominal pipe sizes.

The following example illustrates the use of various tables and charts to size refrigerant lines.

Example 2. Determine the line size and pressure drop equivalent (in degrees) for the suction line of a 30 ton R-22 system, operating at 40°F suction and 100°F condensing temperatures. Suction line is copper tubing, with 50 ft of straight pipe and six long-radius elbows.

Solution: Add 50% to the straight length of pipe to establish a trial equivalent length. Trial equivalent length is 50 × 1.5 = 75 ft. From Table 3 (for 40°F suction, 105°F condensing), 33.1 tons capacity in 2 1/8 in. OD results in a 2°F loss per 100 ft equivalent length. Referring to Note 4, Table 3, capacity at 40°F evaporator and 100°F condensing temperature is 1.03 × 33.1 = 34.1 ton. This trial size is used to evaluate actual equivalent length.

Straight pipe length	=	50.0 ft
Six 2 in. long-radius elbows at 3 ft each (Table 16)	=	19.8 ft
Total equivalent length	=	69.8 ft

$$\Delta t = 2(69.8/100)(30/34.1)^{1.8} = 1.1\text{°F or } 1.6 \text{ psi}$$

Oil Management in Refrigerant Lines

Oil Circulation. All compressors lose some lubricating oil during normal operation. Because oil inevitably leaves the compressor with the discharge gas, systems using halocarbon refrigerants must return this oil at the same rate at which it leaves (Cooper 1971).

Oil that leaves the compressor or oil separator reaches the condenser and dissolves in the liquid refrigerant, enabling it to pass readily through the liquid line to the evaporator. In the evaporator, the refrigerant evaporates, and the liquid phase becomes enriched in oil. The concentration of refrigerant in the oil depends on the evaporator temperature and types of refrigerant and oil used. The viscosity of the oil/refrigerant solution is determined by the system parameters. Oil separated in the evaporator is returned to the compressor by gravity or by drag forces of the returning gas. Oil's effect on pressure drop is large, increasing the pressure drop by as much as a factor of 10 (Alofs et al. 1990).

One of the most difficult problems in low-temperature refrigeration systems using halocarbon refrigerants is returning lubrication oil from the evaporator to the compressors. Except for most centrifugal compressors and rarely used nonlubricated compressors, refrigerant continuously carries oil into the discharge line from the compressor. Most of this oil can be removed from the stream by an oil separator and returned to the compressor. Coalescing oil separators are far better than separators using only mist pads or baffles; however, they are not 100% effective. Oil that finds its way into the system must be managed.

Oil mixes well with halocarbon refrigerants at higher temperatures. As temperature decreases, miscibility is reduced, and some oil separates to form an oil-rich layer near the top of the liquid level in a flooded evaporator. If the temperature is very low, the oil becomes a gummy mass that prevents refrigerant controls from functioning, blocks flow passages, and fouls heat transfer surfaces. Proper oil management is often key to a properly functioning system.

In general, direct-expansion and liquid overfeed system evaporators have fewer oil return problems than do flooded system evaporators because refrigerant flows continuously at velocities high enough to sweep oil from the evaporator. Low-temperature systems using hot-gas defrost can also be designed to sweep oil out of the circuit each time the system defrosts. This reduces the possibility of oil coating the evaporator surface and hindering heat transfer.

Flooded evaporators can promote oil contamination of the evaporator charge because they may only return dry refrigerant vapor back to the system. Skimming systems must sample the oil-rich layer floating in the drum, a heat source must distill the refrigerant, and the oil must be returned to the compressor. Because flooded halocarbon systems can be elaborate, some designers avoid them.

System Capacity Reduction. Using automatic capacity control on compressors requires careful analysis and design. The compressor can load and unload as it modulates with system load requirements through a considerable range of capacity. A single compressor can unload down to 25% of full-load capacity, and multiple compressors connected in parallel can unload to a system capacity of 12.5% or lower. System piping must be designed to return oil at the lowest loading, yet not impose excessive pressure drops in the piping and equipment at full load.

Oil Return up Suction Risers. Many refrigeration piping systems contain a suction riser because the evaporator is at a lower level than the compressor. Oil circulating in the system can return up gas risers only by being transported by returning gas or by auxiliary means such as a trap and pump. The minimum conditions for oil transport correlate with buoyancy forces (i.e., density difference between liquid and vapor, and momentum flux of vapor) (Jacobs et al. 1976).

The principal criteria determining the transport of oil are gas velocity, gas density, and pipe inside diameter. Density of the oil/refrigerant mixture plays a somewhat lesser role because it is almost constant over a wide range. In addition, at temperatures somewhat lower than –40°F, oil viscosity may be significant. Greater gas velocities are required as temperature drops and the gas becomes less dense. Higher velocities are also necessary if the pipe diameter increases. Table 19 translates these criteria to minimum refrigeration capacity requirements for oil transport. Suction risers must be sized for minimum system capacity. Oil must be returned to the compressor at the operating condition corresponding to the minimum displacement and minimum suction temperature at which the

Table 10 Suction Line Capacities in Tons for Refrigerant 22 (Single- or High-Stage Applications)

Line Size		Saturated Suction Temperature, °F									
Type L Copper, OD		-40		-20		0		20		40	
		Δt = 1°F	Δt = 0.5°F	Δt = 1°F	Δt = 0.5°F	Δt = 1°F	Δt = 0.5°F	Δt = 1°F	Δt = 0.5°F	Δt = 1°F	Δt = 0.5°F
		Δp = 0.393	Δp = 0.197	Δp = 0.577	Δp = 0.289	Δp = 0.813	Δp = 0.406	Δp = 1.104	Δp = 0.552	Δp = 1.455	Δp = 0.727
1/2		0.07	0.05	0.12	0.08	0.18	0.12	0.27	0.19	0.40	0.27
5/8		0.13	0.09	0.22	0.15	0.34	0.23	0.52	0.35	0.75	0.51
3/4		0.22	0.15	0.37	0.25	0.58	0.39	0.86	0.59	1.24	0.85
7/8		0.35	0.24	0.58	0.40	0.91	0.62	1.37	0.93	1.97	1.35
1 1/8		0.72	0.49	1.19	0.81	1.86	1.27	2.77	1.90	3.99	2.74
1 3/8		1.27	0.86	2.09	1.42	3.25	2.22	4.84	3.32	6.96	4.78
1 5/8		2.02	1.38	3.31	2.26	5.16	3.53	7.67	5.26	11.00	7.57
2 1/8		4.21	2.88	6.90	4.73	10.71	7.35	15.92	10.96	22.81	15.73
2 5/8		7.48	5.13	12.23	8.39	18.97	13.04	28.19	19.40	40.38	27.84
3 1/8		11.99	8.22	19.55	13.43	30.31	20.85	44.93	31.00	64.30	44.44
3 5/8		17.89	12.26	29.13	20.00	45.09	31.03	66.81	46.11	95.68	66.09
4 1/8		25.29	17.36	41.17	28.26	63.71	43.85	94.25	65.12	134.81	93.22
Steel											
IPS	SCH										
3/8	80	0.06	0.04	0.10	0.07	0.15	0.10	0.21	0.15	0.30	0.21
1/2	80	0.12	0.08	0.19	0.13	0.29	0.20	0.42	0.30	0.60	0.42
3/4	80	0.27	0.18	0.43	0.30	0.65	0.46	0.95	0.67	1.35	0.95
1	80	0.52	0.36	0.84	0.59	1.28	0.89	1.87	1.31	2.64	1.86
1 1/4	40	1.38	0.96	2.21	1.55	3.37	2.36	4.91	3.45	6.93	4.88
1 1/2	40	2.08	1.45	3.32	2.33	5.05	3.55	7.38	5.19	10.42	7.33
2	40	4.03	2.81	6.41	4.51	9.74	6.85	14.22	10.01	20.07	14.14
2 1/2	40	6.43	4.49	10.23	7.19	15.56	10.93	22.65	15.95	31.99	22.53
3	40	11.38	7.97	18.11	12.74	27.47	19.34	40.10	28.23	56.52	39.79
4	40	23.24	16.30	36.98	26.02	56.12	39.49	81.73	57.53	115.24	81.21
5	40	42.04	29.50	66.73	47.05	101.16	71.27	147.36	103.82	207.59	146.38
6	40	68.04	47.86	108.14	76.15	163.77	115.21	238.29	168.07	335.71	236.70
8	40	139.48	98.06	221.17	155.78	334.94	236.21	488.05	344.19	686.71	484.74
10	40	252.38	177.75	400.53	282.05	606.74	427.75	881.59	622.51	1243.64	876.79
12	ID*	403.63	284.69	639.74	451.09	969.02	683.22	1410.30	995.80	1987.29	1402.63

Δp = pressure drop from line friction, psi per 100 ft equivalent line length

Δt = change in saturation temperature corresponding to pressure drop, °F per 100 ft

*Pipe inside diameter is same as nominal pipe size.

Table 11 Suction Line Capacities in Tons for Refrigerant 134a (Single- or High-Stage Applications)

Line Size		Saturated Suction Temperature, °F									
Type L Copper, OD		0		10		20		30		40	
		Δt = 1°F	Δt = 0.5°F	Δt = 1°F	Δt = 0.5°F	Δt = 1°F	Δt = 0.5°F	Δt = 1°F	Δt = 0.5°F	Δt = 1°F	Δt = 0.5°F
		Δp = 0.50	Δp = 0.25	Δp = 0.60	Δp = 0.30	Δp = 0.71	Δp = 0.35	Δp = 0.83	Δp = 0.42	Δp = 0.97	Δp = 0.48
1/2		0.10	0.07	0.12	0.08	0.16	0.11	0.19	0.13	0.24	0.16
5/8		0.18	0.12	0.23	0.16	0.29	0.20	0.37	0.25	0.45	0.31
7/8		0.48	0.33	0.62	0.42	0.78	0.53	0.97	0.66	1.20	0.82
1 1/8		0.99	0.67	1.26	0.86	1.59	1.08	1.97	1.35	2.43	1.66
1 3/8		1.73	1.18	2.21	1.51	2.77	1.89	3.45	2.36	4.25	2.91
1 5/8		2.75	1.88	3.50	2.40	4.40	3.01	5.46	3.75	6.72	4.61
2 1/8		5.73	3.92	7.29	5.00	9.14	6.27	11.40	7.79	14.00	9.59
2 5/8		10.20	6.97	12.90	8.87	16.20	11.10	20.00	13.80	24.70	17.00
3 1/8		16.20	11.10	20.60	14.20	25.90	17.80	32.10	22.10	39.40	27.20
3 5/8		24.20	16.60	30.80	21.20	38.50	26.50	47.70	32.90	58.70	40.40
4 1/8		34.20	23.50	43.40	29.90	54.30	37.40	67.30	46.50	82.60	57.10
5 1/8		61.30	42.20	77.70	53.60	97.20	67.10	121.00	83.20	148.00	102.00
6 1/8		98.80	68.00	125.00	86.30	157.00	108.00	194.00	134.00	237.00	165.00
Steel											
IPS	SCH										
1/2	80	0.16	0.11	0.20	0.14	0.25	0.17	0.30	0.21	0.37	0.26
3/4	80	0.36	0.25	0.45	0.31	0.56	0.39	0.69	0.48	0.84	0.59
1	80	0.70	0.49	0.88	0.61	1.09	0.77	1.34	0.94	1.64	1.15
1 1/4	40	1.84	1.29	2.31	1.62	2.87	2.02	3.54	2.48	4.31	3.03
1 1/2	40	2.77	1.94	3.48	2.44	4.32	3.03	5.30	3.73	6.47	4.55
2	40	5.35	3.75	6.72	4.72	8.33	5.86	10.30	7.20	12.50	8.78
2 1/2	40	8.53	5.99	10.70	7.53	13.30	9.35	16.30	11.50	19.90	14.00
3	40	15.10	10.60	18.90	13.30	23.50	16.50	28.90	20.30	35.20	24.80
4	40	30.80	21.70	38.70	27.20	48.00	33.80	58.80	41.50	71.60	50.50
5	40	55.60	39.20	69.80	49.10	86.50	60.93	106.00	74.95	129.00	91.00
6	40	89.90	63.40	113.00	79.60	140.00	98.50	172.00	121.00	209.00	148.00

Δp = pressure drop from line friction, psi per 100 ft equivalent line length

Δt = change in saturation temperature corresponding to pressure drop, °F per 100 ft

Table 12 Suction Line Capacities in Tons for Refrigerant 404a (Single- or High-Stage Applications)

Line Size		Saturated Suction Temperature, °F											
Type L Copper, OD		–60		–40		–20		0		20		40	
		Δt = 1°F Δp = 0.32	Δt = 0.5°F Δp = 0.16	Δt = 1°F Δp = 0.485	Δt = 0.5°F Δp = 0.243	Δt = 1°F Δp = 0.705	Δt = 0.5°F Δp = 0.353	Δt = 1°F Δp = 0.98	Δt = 0.5°F Δp = 0.49	Δt = 1°F Δp = 1.31	Δt = 0.5°F Δp = 0.655	Δt = 1°F Δp = 1.72	Δt = 0.5°F Δp = 0.86
1/2		0.03	0.02	0.06	0.04	0.10	0.07	0.16	0.11	0.25	0.17	0.37	0.25
5/8		0.06	0.04	0.11	0.08	0.19	0.13	0.30	0.21	0.46	0.32	0.69	0.47
3/4		0.10	0.07	0.19	0.13	0.32	0.22	0.52	0.35	0.79	0.54	1.17	0.80
7/8		0.16	0.11	0.29	0.20	0.50	0.34	0.80	0.55	1.22	0.84	1.81	1.24
1 1/8		0.33	0.23	0.60	0.41	1.02	0.70	1.63	1.12	2.48	1.70	3.66	2.52
1 3/8		0.59	0.40	1.05	0.72	1.78	1.22	2.84	1.95	4.33	2.98	6.38	4.40
1 5/8		0.93	0.63	1.67	1.14	2.82	1.93	4.50	3.09	6.84	4.71	10.08	6.95
2 1/8		1.94	1.33	3.48	2.38	5.86	4.02	9.33	6.42	14.19	9.78	20.86	14.42
2 5/8		3.45	2.36	6.17	4.23	10.38	7.13	16.50	11.37	25.04	17.30	36.79	25.48
3 1/8		5.53	3.78	9.87	6.78	16.57	11.40	26.36	18.17	39.90	27.63	58.65	40.65
3 5/8		8.24	5.64	14.70	10.09	24.66	16.98	39.19	27.05	59.27	41.08	86.99	60.38
4 1/8		11.66	7.99	20.74	14.27	34.82	24.00	55.29	38.19	83.67	57.95	122.65	85.08
5 1/8		20.91	14.35	37.20	25.58	62.32	43.03	98.68	68.35	149.15	103.62	218.80	151.93
6 1/8		33.70	23.17	59.82	41.25	100.16	69.25	158.78	109.86	239.61	166.38	350.99	244.04
8 1/8		70.16	48.33	124.35	85.75	207.70	143.94	329.02	228.24	496.00	344.71	725.34	504.94
Steel													
IPS	SCH												
3/8	80	0.03	0.02	0.05	0.03	0.08	0.06	0.13	0.09	0.19	0.13	0.27	0.19
1/2	80	0.05	0.04	0.09	0.07	0.16	0.11	0.25	0.17	0.37	0.26	0.54	0.38
3/4	80	0.12	0.08	0.21	0.15	0.36	0.25	0.56	0.39	0.83	0.59	1.21	0.85
1	80	0.24	0.17	0.42	0.29	0.70	0.49	1.09	0.77	1.63	1.15	2.37	1.67
1 1/4	40	0.52	0.36	0.91	0.63	1.50	1.05	2.34	1.65	3.50	2.46	5.07	3.57
1 1/2	40	0.80	0.55	1.39	0.97	2.29	1.60	3.57	2.51	5.33	3.76	7.74	5.45
2	40	1.86	1.30	3.24	2.26	5.32	3.74	8.30	5.85	12.40	8.73	17.96	12.66
2 1/2	40	2.96	2.07	5.16	3.61	8.48	5.96	13.23	9.32	19.71	13.92	28.57	20.17
3	40	5.25	3.68	9.13	6.41	15.01	10.54	23.37	16.47	34.83	24.59	50.48	35.63
4	40	10.75	7.53	18.64	13.06	30.59	21.53	47.64	33.61	71.01	50.12	102.93	72.64
5	40	19.42	13.61	33.64	23.67	55.22	38.85	86.00	60.66	128.09	90.47	185.40	130.81
6	40	31.37	22.07	54.45	38.36	89.29	62.97	139.08	98.09	207.08	146.31	299.84	211.53
8	40	64.28	45.29	111.50	78.62	182.58	128.75	284.48	200.61	423.62	299.27	613.41	433.35
10	40	116.63	82.09	201.92	142.37	330.75	233.20	514.60	363.34	766.32	541.35	1108.13	783.91
12	ID*	186.39	131.47	322.98	227.70	528.22	373.02	823.24	580.40	1224.19	866.05	1772.90	1252.32
14	30	241.28	170.14	418.14	294.77	683.87	482.92	1064.28	751.41	1585.02	1119.62	2295.51	1621.44
16	30	348.15	245.48	602.49	424.62	985.62	695.84	1533.35	1082.76	2284.15	1613.40	3302.98	2336.63

Notes:

1. Δt = change in saturation temperature corresponding to pressure drop, °F per 100 ft.
2. Tons based on standard refrigerant cycle of 105°F liquid and saturated evaporator outlet temperature. Liquid tons based on 20°F evaporator temperature.
3. Thermophysical properties and viscosity data based on calculations from NIST REFPROP program Version 6.01.
4. Values based on 105°F condensing temperature. Multiply table capacities by the following factors for other condensing temperatures.

*Pipe inside diameter is same as nominal pipe size.

Condensing Temperature, °F	Suction Line
80	1.246
90	1.150
100	1.051
110	0.948
120	0.840
130	0.723

Table 13 Suction Line Capacities in Tons for Refrigerant 507 (Single- or High-Stage Applications)

Line Size		Saturated Suction Temperature, °F											
Type L Copper, OD		−60		−40		−20		0		20		40	
		Δt = 1°F Δp = 0.335	Δt = 0.5°F Δp = 0.168	Δt = 1°F Δp = 0.505	Δt = 0.5°F Δp = 0.253	Δt = 1°F Δp =0.73	Δt = 0.5°F Δp = 0.365	Δt = 1°F Δp = 1.01	Δt = 0.5°F Δp = 0.505	Δt = 1°F Δp = 1.355	Δt = 0.5°F Δp = 0.678	Δt = 1°F Δp = 1.8	Δt = 0.5°F Δp = 0.9
1/2		0.03	0.02	0.06	0.04	0.10	0.07	0.16	0.11	0.25	0.17	0.37	0.26
5/8		0.06	0.04	0.11	0.08	0.19	0.13	0.31	0.21	0.47	0.32	0.70	0.48
3/4		0.11	0.07	0.19	0.13	0.33	0.22	0.52	0.36	0.80	0.55	1.20	0.82
7/8		0.17	0.11	0.30	0.20	0.51	0.35	0.81	0.56	1.24	0.85	1.85	1.27
1 1/8		0.34	0.23	0.61	0.42	1.03	0.71	1.65	1.13	2.52	1.73	3.74	2.57
1 3/8		0.60	0.41	1.07	0.73	1.81	1.24	2.88	1.97	4.39	3.02	6.51	4.49
1 5/8		0.95	0.65	1.70	1.16	2.87	1.96	4.56	3.13	6.94	4.78	10.28	7.09
2 1/8		1.99	1.36	3.55	2.43	5.95	4.09	9.44	6.51	14.37	9.92	21.28	14.70
2 5/8		3.53	2.42	6.29	4.31	10.54	7.24	16.70	11.51	25.40	17.54	37.53	25.99
3 1/8		5.66	3.88	10.06	6.90	16.82	11.57	26.69	18.40	40.48	27.98	59.74	41.41
3 5/8		8.43	5.78	14.98	10.29	25.04	17.24	39.62	27.38	60.13	41.61	88.62	61.51
4 1/8		11.92	8.18	21.16	14.53	35.37	24.37	55.91	38.62	84.73	58.69	124.94	86.66
5 1/8		21.40	14.71	37.90	26.11	63.19	43.71	99.99	69.10	151.06	104.94	222.92	154.78
6 1/8		34.51	23.73	60.98	42.03	101.79	70.33	160.57	111.29	242.71	168.52	357.63	248.63
8 1/8		71.74	49.40	126.80	87.41	210.91	145.98	332.73	230.83	502.46	349.13	739.16	514.43
Steel													
IPS	SCH												
3/8	80	0.03	0.02	0.05	0.03	0.08	0.06	0.13	0.09	0.19	0.13	0.28	0.19
1/2	80	0.06	0.04	0.10	0.07	0.16	0.11	0.25	0.17	0.37	0.26	0.55	0.38
3/4	80	0.13	0.09	0.22	0.15	0.36	0.25	0.56	0.39	0.84	0.59	1.23	0.87
1	80	0.25	0.17	0.43	0.30	0.71	0.50	1.10	0.77	1.65	1.16	2.41	1.70
1 1/4	40	0.53	0.37	0.93	0.65	1.52	1.07	2.37	1.66	3.54	2.49	5.16	3.63
1 1/2	40	0.81	0.57	1.41	0.99	2.32	1.63	3.61	2.54	5.39	3.80	7.87	5.54
2	40	1.90	1.33	3.29	2.31	5.39	3.79	8.38	5.90	12.53	8.83	18.26	12.86
2 1/2	40	3.03	2.12	5.25	3.68	8.58	6.04	13.35	9.40	19.93	14.07	29.05	20.51
3	40	5.37	3.76	9.29	6.52	15.19	10.69	23.58	16.64	35.22	24.85	51.33	36.23
4	40	10.95	7.69	18.93	13.32	30.96	21.79	48.07	33.92	71.78	50.66	104.65	73.86
5	40	19.77	13.90	34.20	23.98	55.89	39.37	86.80	61.22	129.59	91.45	188.50	133.18
6	40	32.06	22.52	55.36	38.95	90.37	63.73	140.36	98.99	209.38	147.89	304.85	215.38
8	40	65.72	46.21	113.19	79.83	185.07	130.51	287.10	202.42	428.18	302.50	623.68	440.60
10	40	119.01	83.76	205.02	144.56	334.75	236.03	519.34	366.70	774.58	547.19	1126.66	797.04
12	ID*	190.34	134.16	327.88	231.16	535.50	377.46	830.83	585.64	1237.39	875.38	1802.55	1273.31
14	30	246.21	173.66	424.56	299.30	693.31	488.76	1074.09	758.20	1602.11	1131.69	2333.91	1648.57
16	30	354.73	250.14	611.65	431.90	997.35	704.12	1550.19	1092.75	2308.78	1630.79	3358.23	2375.74

Notes:

1. Δt = change in saturation temperature corresponding to pressure drop, °F per 100 ft.
2. Tons based on standard refrigerant cycle of 105°F liquid and saturated evaporator outlet temperature. Liquid tons based on 20°F evaporator temperature.
3. Thermophysical properties and viscosity data based on calculations from NIST REFPROP program Version 6.01.
4. Values based on 105°F condensing temperature. Multiply table capacities by the following factors for other condensing temperatures.

*Pipe inside diameter is same as nominal pipe size.

Condensing Temperature, °F	Suction Line
80	1.267
90	1.163
100	1.055
110	0.944
120	0.826
130	0.701

Table 14 Suction Line Capacities in Tons for Refrigerant 410a (Single- or High-Stage Applications)

Line Size		Saturated Suction Temperature, °F											
Type L Copper, OD		–60		–40		–20		0		20		40	
		Δt = 1°F Δp = 0.42	Δt = 0.5°F Δp = 0.21	Δt = 1°F Δp = 0.635	Δt = 0.5°F Δp = 0.318	Δt = 1°F Δp = 0.925	Δt = 0.5°F Δp = 0.463	Δt = 1°F Δp = 1.285	Δt = 0.5°F Δp = 0.643	Δt = 1°F Δp = 1.73	Δt = 0.5°F Δp = 0.865	Δt = 1°F Δp = 2.25	Δt = 0.5°F Δp = 1.125
1/2		0.06	0.04	0.11	0.08	0.18	0.13	0.29	0.20	0.43	0.29	0.61	0.42
5/8		0.12	0.08	0.21	0.14	0.35	0.24	0.54	0.37	0.80	0.55	1.15	0.79
3/4		0.21	0.14	0.36	0.25	0.60	0.41	0.92	0.63	1.37	0.94	1.96	1.34
7/8		0.33	0.22	0.57	0.38	0.92	0.63	1.43	0.98	2.12	1.45	3.02	2.08
1 1/8		0.67	0.46	1.15	0.79	1.88	1.28	2.90	1.99	4.29	2.95	6.12	4.22
1 3/8		1.18	0.80	2.02	1.38	3.28	2.25	5.06	3.47	7.49	5.15	10.65	7.34
1 5/8		1.87	1.27	3.20	2.19	5.20	3.56	8.00	5.50	11.84	8.16	16.82	11.62
2 1/8		3.90	2.66	6.66	4.57	10.80	7.42	16.60	11.43	24.53	16.94	34.82	24.06
2 5/8		6.92	4.74	11.81	8.11	19.15	13.16	29.37	20.24	43.30	29.96	61.42	42.54
3 1/8		11.10	7.59	18.88	12.98	30.56	21.03	46.84	32.36	69.12	47.78	97.93	67.88
3 5/8		16.54	11.32	28.12	19.33	45.48	31.32	69.66	48.14	102.68	71.03	145.29	100.82
4 1/8		23.37	16.04	39.75	27.34	64.13	44.26	98.29	67.89	144.70	100.22	204.80	142.08
5 1/8		41.90	28.80	71.16	49.04	114.79	79.27	175.44	121.50	257.95	179.21	365.02	253.76
6 1/8		67.56	46.54	114.71	79.08	184.50	127.75	282.30	195.66	414.50	287.76	586.12	407.59
8 1/8		140.71	96.90	238.00	164.42	382.64	265.15	583.63	405.01	858.05	596.10	1208.61	843.44
Steel													
IPS	SCH												
3/8	80	0.05	0.04	0.09	0.06	0.15	0.10	0.22	0.16	0.32	0.23	0.45	0.32
1/2	80	0.11	0.07	0.18	0.13	0.29	0.20	0.44	0.31	0.64	0.45	0.89	0.63
3/4	80	0.25	0.17	0.41	0.29	0.65	0.46	0.99	0.69	1.44	1.01	2.01	1.42
1	80	0.48	0.34	0.81	0.57	1.28	0.90	1.94	1.36	2.81	1.98	3.94	2.78
1 1/4	40	1.04	0.73	1.74	1.22	2.76	1.94	4.16	2.92	6.04	4.25	8.45	5.96
1 1/2	40	1.60	1.11	2.66	1.86	4.21	2.96	6.35	4.46	9.20	6.48	12.90	9.09
2	40	3.73	2.60	6.19	4.34	9.79	6.88	14.72	10.38	21.40	15.08	29.94	21.09
2 1/2	40	5.94	4.16	9.85	6.93	15.59	10.98	23.46	16.53	34.03	24.02	47.62	33.61
3	40	10.52	7.37	17.43	12.25	27.60	19.43	41.47	29.26	60.13	42.44	84.14	59.39
4	40	21.48	15.08	35.60	25.06	56.24	39.58	84.52	59.63	122.57	86.51	171.56	121.08
5	40	38.84	27.30	64.25	45.21	101.52	71.51	152.52	107.63	221.30	156.17	309.01	218.33
6	40	62.85	44.23	104.14	73.26	164.15	115.77	246.64	174.04	357.45	252.55	499.76	353.09
8	40	128.81	90.62	212.93	150.18	336.18	236.70	504.51	355.89	731.21	516.58	1022.43	722.30
10	40	233.22	164.52	385.68	271.93	608.06	428.73	912.58	644.70	1322.74	934.44	1847.00	1306.62
12	ID*	372.99	263.04	616.79	434.92	972.73	685.64	1459.96	1029.64	2113.09	1494.90	2955.02	2087.38
14	30	483.55	340.47	798.65	563.02	1259.39	887.82	1887.38	1333.03	2735.91	1932.59	3826.11	2702.56
16	30	696.69	491.23	1150.59	812.45	1811.67	1279.02	2724.04	1921.21	3942.69	2784.92	5505.32	3894.62

Notes:

1. Δt = change in saturation temperature corresponding to pressure drop, °F per 100 ft.
2. Tons based on standard refrigerant cycle of 105°F liquid and saturated evaporator outlet temperature. Liquid tons based on 20°F evaporator temperature.
3. Thermophysical properties and viscosity data based on calculations from NIST REFPROP program Version 6.01.
4. Values based on 105°F condensing temperature. Multiply table capacities by the following factors for other condensing temperatures.

*Pipe inside diameter is same as nominal pipe size.

Condensing Temperature, °F	Suction Line
80	1.170
90	1.104
100	1.035
110	0.964
120	0.889
130	0.808

Table 15 Suction Line Capacities in Tons for Refrigerant 407c (Single- or High-Stage Applications)

Line Size		Saturated Suction Temperature, °F											
Type L Copper, OD		–60		–40		–20		0		20		40	
		Δt = 1°F Δp = 0.218	Δt = 0.5°F Δp = 0.109	Δt = 1°F Δp = 0.35	Δt = 0.5°F Δp = 0.175	Δt = 1°F Δp = 0.53	Δt = 0.5°F Δp = 0.265	Δt = 1°F Δp = 0.775	Δt = 0.5°F Δp = 0.388	Δt = 1°F Δp = 1.08	Δt = 0.5°F Δp = 0.54	Δt = 1°F Δp = 1.46	Δt = 0.5°F Δp = 0.73
1/2		0.03	0.02	0.05	0.04	0.09	0.06	0.16	0.11	0.25	0.17	0.37	0.25
5/8		0.05	0.04	0.10	0.07	0.18	0.12	0.30	0.20	0.46	0.32	0.70	0.48
3/4		0.09	0.06	0.17	0.12	0.31	0.21	0.51	0.34	0.79	0.54	1.19	0.81
7/8		0.15	0.10	0.27	0.18	0.48	0.32	0.78	0.53	1.23	0.84	1.84	1.26
1 1/8		0.30	0.20	0.56	0.38	0.97	0.66	1.60	1.09	2.49	1.71	3.74	2.56
1 3/8		0.52	0.36	0.98	0.66	1.70	1.16	2.79	1.91	4.35	2.98	6.52	4.48
1 5/8		0.83	0.57	1.56	1.06	2.69	1.84	4.43	3.03	6.89	4.72	10.30	7.09
2 1/8		1.75	1.19	3.24	2.22	5.61	3.84	9.20	6.32	14.31	9.84	21.36	14.73
2 5/8		3.11	2.12	5.77	3.94	9.94	6.82	16.30	11.20	25.30	17.42	37.75	26.08
3 1/8		5.00	3.41	9.24	6.32	15.91	10.92	26.05	17.90	40.34	27.82	60.23	41.58
3 5/8		7.45	5.09	13.77	9.44	23.72	16.29	38.75	26.70	59.97	41.40	89.47	61.81
4 1/8		10.57	7.22	19.47	13.36	33.52	23.03	54.73	37.71	84.60	58.46	126.06	87.32
5 1/8		19.00	13.00	35.00	24.01	60.09	41.35	97.90	67.62	151.22	104.74	225.14	156.10
6 1/8		30.67	21.04	56.41	38.77	96.82	66.66	157.64	108.96	243.24	168.35	361.69	251.08
8 1/8		63.98	43.94	117.40	80.79	201.22	138.52	326.82	226.06	503.94	349.69	748.45	520.10
Steel													
IPS	SCH												
3/8	80	0.02	0.02	0.05	0.03	0.08	0.05	0.13	0.09	0.19	0.13	0.28	0.20
1/2	80	0.05	0.03	0.09	0.06	0.15	0.11	0.25	0.17	0.38	0.27	0.56	0.39
3/4	80	0.11	0.08	0.21	0.14	0.35	0.24	0.56	0.39	0.86	0.60	1.26	0.88
1	80	0.22	0.15	0.40	0.28	0.68	0.48	1.10	0.77	1.68	1.18	2.47	1.73
1 1/4	40	0.48	0.33	0.87	0.61	1.48	1.03	2.36	1.66	3.60	2.53	5.28	3.72
1 1/2	40	0.74	0.51	1.34	0.93	2.25	1.57	3.61	2.53	5.49	3.86	8.06	5.67
2	40	1.73	1.20	3.12	2.18	5.25	3.68	8.39	5.90	12.77	8.98	18.71	13.18
2 1/2	40	2.77	1.92	4.98	3.49	8.36	5.85	13.39	9.41	20.35	14.30	29.82	21.00
3	40	4.92	3.42	8.82	6.16	14.81	10.39	23.66	16.66	35.96	25.31	52.68	37.18
4	40	10.07	7.04	18.05	12.62	30.24	21.26	48.33	34.01	73.25	51.63	107.39	75.79
5	40	18.24	12.74	32.63	22.88	54.64	38.40	87.11	61.38	132.31	93.20	193.65	136.64
6	40	29.56	20.67	52.87	37.10	88.32	62.14	141.05	99.22	213.85	150.90	313.17	221.26
8	40	60.72	42.53	108.35	76.09	181.10	127.34	288.49	203.43	437.43	308.63	640.64	452.60
10	40	110.03	77.14	196.18	137.91	327.97	230.56	522.51	368.41	791.25	558.98	1158.92	817.38
12	ID*	176.17	123.83	314.18	220.86	523.74	369.31	834.64	588.33	1265.85	892.90	1851.38	1307.58
14	30	228.34	160.22	406.04	286.29	678.86	477.98	1080.58	762.91	1636.44	1155.99	2397.05	1693.24
16	30	329.42	231.48	585.97	413.05	978.61	689.83	1557.07	1099.28	2358.16	1665.74	3454.36	2440.00

Notes:

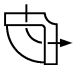
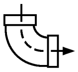
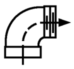
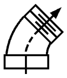
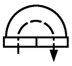
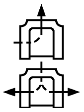
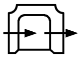
1. Δt = change in saturation temperature corresponding to pressure drop, °F per 100 ft.
2. Tons based on standard refrigerant cycle of 105°F liquid and saturated evaporator outlet temperature. Liquid tons based on 20°F evaporator temperature.
3. Thermophysical properties and viscosity data based on calculations from NIST REFPROP program Version 6.01.
4. Values based on 105°F condensing temperature. Multiply table capacities by the following factors for other condensing temperatures.

*Pipe inside diameter is same as nominal pipe size.

Condensing Temperature, °F	Suction Line
80	1.163
90	1.099
100	1.033
110	0.966
120	0.896
130	0.824

Table 16 Fitting Losses in Equivalent Feet of Pipe

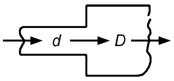
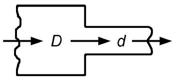
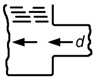
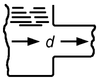
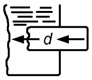
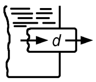
(Screwed, Welded, Flanged, Flared, and Brazed Connections)

	Smooth Bend Elbows						Smooth Bend Tees			
	90° Std[a]	90° Long-Radius[b]	90° Street[a]	45° Std[a]	45° Street[a]	180° Std[a]	Flow Through Branch	Straight-Through Flow		
								No Reduction	Reduced 1/4	Reduced 1/2
Nominal Pipe or Tube Size, in.										
3/8	1.4	0.9	2.3	0.7	1.1	2.3	2.7	0.9	1.2	1.4
1/2	1.6	1.0	2.5	0.8	1.3	2.5	3.0	1.0	1.4	1.6
3/4	2.0	1.4	3.2	0.9	1.6	3.2	4.0	1.4	1.9	2.0
1	2.6	1.7	4.1	1.3	2.1	4.1	5.0	1.7	2.2	2.6
1 1/4	3.3	2.3	5.6	1.7	3.0	5.6	7.0	2.3	3.1	3.3
1 1/2	4.0	2.6	6.3	2.1	3.4	6.3	8.0	2.6	3.7	4.0
2	5.0	3.3	8.2	2.6	4.5	8.2	10.0	3.3	4.7	5.0
2 1/2	6.0	4.1	10.0	3.2	5.2	10.0	12.0	4.1	5.6	6.0
3	7.5	5.0	12.0	4.0	6.4	12.0	15.0	5.0	7.0	7.5
3 1/2	9.0	5.9	15.0	4.7	7.3	15.0	18.0	5.9	8.0	9.0
4	10.0	6.7	17.0	5.2	8.5	17.0	21.0	6.7	9.0	10.0
5	13.0	8.2	21.0	6.5	11.0	21.0	25.0	8.2	12.0	13.0
6	16.0	10.0	25.0	7.9	13.0	25.0	30.0	10.0	14.0	16.0
8	20.0	13.0	—	10.0	—	33.0	40.0	13.0	18.0	20.0
10	25.0	16.0	—	13.0	—	42.0	50.0	16.0	23.0	25.0
12	30.0	19.0	—	16.0	—	50.0	60.0	19.0	26.0	30.0
14	34.0	23.0	—	18.0	—	55.0	68.0	23.0	30.0	34.0
16	38.0	26.0	—	20.0	—	62.0	78.0	26.0	35.0	38.0
18	42.0	29.0	—	23.0	—	70.0	85.0	29.0	40.0	42.0
20	50.0	33.0	—	26.0	—	81.0	100.0	33.0	44.0	50.0
24	60.0	40.0	—	30.0	—	94.0	115.0	40.0	50.0	60.0

[a] *R/D* approximately equal to 1.

[b] *R/D* approximately equal to 1.5.

Table 17 Special Fitting Losses in Equivalent Feet of Pipe

	Sudden Enlargement, *d/D*			Sudden Contraction, *d/D*			Sharp Edge		Pipe Projection	
	1/4	1/2	3/4	1/4	1/2	3/4	Entrance	Exit	Entrance	Exit
Nominal Pipe or Tube Size, in.										
3/8	1.4	0.8	0.3	0.7	0.5	0.3	1.5	0.8	1.5	1.1
1/2	1.8	1.1	0.4	0.9	0.7	0.4	1.8	1.0	1.8	1.5
3/4	2.5	1.5	0.5	1.2	1.0	0.5	2.8	1.4	2.8	2.2
1	3.2	2.0	0.7	1.6	1.2	0.7	3.7	1.8	3.7	2.7
1 1/4	4.7	3.0	1.0	2.3	1.8	1.0	5.3	2.6	5.3	4.2
1 1/2	5.8	3.6	1.2	2.9	2.2	1.2	6.6	3.3	6.6	5.0
2	8.0	4.8	1.6	4.0	3.0	1.6	9.0	4.4	9.0	6.8
2 1/2	10.0	6.1	2.0	5.0	3.8	2.0	12.0	5.6	12.0	8.7
3	13.0	8.0	2.6	6.5	4.9	2.6	14.0	7.2	14.0	11.0
3 1/2	15.0	9.2	3.0	7.7	6.0	3.0	17.0	8.5	17.0	13.0
4	17.0	11.0	3.8	9.0	6.8	3.8	20.0	10.0	20.0	16.0
5	24.0	15.0	5.0	12.0	9.0	5.0	27.0	14.0	27.0	20.0
6	29.0	22.0	6.0	15.0	11.0	6.0	33.0	19.0	33.0	25.0
8	—	25.0	8.5	—	15.0	8.5	47.0	24.0	47.0	35.0
10	—	32.0	11.0	—	20.0	11.0	60.0	29.0	60.0	46.0
12	—	41.0	13.0	—	25.0	13.0	73.0	37.0	73.0	57.0
14	—	—	16.0	—	—	16.0	86.0	45.0	86.0	66.0
16	—	—	18.0	—	—	18.0	96.0	50.0	96.0	77.0
18	—	—	20.0	—	—	20.0	115.0	58.0	115.0	90.0
20	—	—	—	—	—	—	142.0	70.0	142.0	108.0
24	—	—	—	—	—	—	163.0	83.0	163.0	130.0

Note: Enter table for losses at smallest diameter *d*.

compressor will operate. When suction or evaporator pressure regulators are used, suction risers must be sized for actual gas conditions in the riser.

For a single compressor with capacity control, the minimum capacity is the lowest capacity at which the unit can operate. For multiple compressors with capacity control, the minimum capacity is the lowest at which the last operating compressor can run.

Riser Sizing. The following example demonstrates the use of Table 19 in establishing maximum riser sizes for satisfactory oil transport down to minimum partial loading.

Example 3. Determine the maximum size suction riser that will transport oil at minimum loading, using R-22 with a 40 ton compressor with capacity in steps of 25, 50, 75, and 100%. Assume the minimum system loading is 10 tons at 40°F suction and 105°F condensing temperatures with 15°F superheat.

Solution: From Table 19, a 2 1/8 in. OD pipe at 40°F suction and 90°F liquid temperature has a minimum capacity of 7.5 tons. When corrected to 105°F liquid temperature using the chart at the bottom of Table 19, minimum capacity becomes 7.2 tons. Therefore, 2 1/8 in. OD pipe is suitable.

Based on Table 19, the next smaller line size should be used for marginal suction risers. When vertical riser sizes are reduced to provide satisfactory minimum gas velocities, pressure drop at full load increases considerably; horizontal lines should be sized to keep total pressure drop within practical limits. As long as horizontal lines are level or pitched in the direction of the compressor, oil can be transported with normal design velocities.

Because most compressors have multiple capacity-reduction features, gas velocities required to return oil up through vertical suction risers under all load conditions are difficult to maintain. When the suction riser is sized to allow oil return at the minimum operating capacity of the system, pressure drop in this portion of the line may be too great when operating at full load. If a correctly sized suction riser imposes too great a pressure drop at full load, a double suction riser should be used.

Table 18 Valve Losses in Equivalent Feet of Pipe

Nominal Pipe or Tube Size, in.	Globe[a]	60° Wye	45° Wye	Angle[a]	Gate[b]	Swing Check[c]	Lift Check
3/8	17	8	6	6	0.6	5	Globe and vertical lift same as globe valve[d]
1/2	18	9	7	7	0.7	6	
3/4	22	11	9	9	0.9	8	
1	29	15	12	12	1.0	10	
1 1/4	38	20	15	15	1.5	14	
1 1/2	43	24	18	18	1.8	16	
2	55	30	24	24	2.3	20	
2 1/2	69	35	29	29	2.8	25	
3	84	43	35	35	3.2	30	
3 1/2	100	50	41	41	4.0	35	
4	120	58	47	47	4.5	40	
5	140	71	58	58	6.0	50	
6	170	88	70	70	7.0	60	
8	220	115	85	85	9.0	80	Angle lift same as angle valve
10	280	145	105	105	12.0	100	
12	320	165	130	130	13.0	120	
14	360	185	155	155	15.0	135	
16	410	210	180	180	17.0	150	
18	460	240	200	200	19.0	165	
20	520	275	235	235	22.0	200	
24	610	320	265	265	25.0	240	

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

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

[b] Regular and short pattern plug cock valves, when fully open, have same loss as gate valve. For valve losses of short pattern plug cocks above 6 in., check with manufacturer.

[c] Losses also apply to inline, ball check valve.

[d] For *Y* pattern globe lift check valve with seat approximately equal to nominal pipe diameter, use values of 60° wye valve for loss.

Oil Return up Suction Risers: Multistage Systems. Oil movement in the suction lines of multistage systems requires the same design approach as that for single-stage systems. For oil to flow up along a pipe wall, a certain minimum drag of gas flow is required. Drag can be represented by the friction gradient. The following sizing data may be used for ensuring oil return up vertical suction lines for refrigerants other than those listed in Tables 19 and 20. The line size selected should provide a pressure drop equal to or greater than that shown in the chart.

Saturation Temperature, °F	Line Size	
	2 in. or less	Above 2 in.
0	0.35 psi/100 ft	0.20 psi/100 ft
–50	0.45 psi/100 ft	0.25 psi/100 ft

Double Suction Risers. Figure 3 shows two methods of double suction riser construction. Oil return in this arrangement is accomplished at minimum loads, but it does not cause excessive pressure drops at full load. Sizing and operation of a double suction riser are as follows:

1. Riser A is sized to return oil at minimum load possible.
2. Riser B is sized for satisfactory pressure drop through both risers at full load. The usual method is to size riser B so that the combined cross-sectional area of A and B is equal to or slightly greater than the cross-sectional area of a single pipe sized for acceptable pressure drop at full load without regard for oil return at minimum load. The combined cross-sectional area, however, should not be greater than the cross-sectional area of a single pipe that would return oil in an upflow riser under maximum load.
3. A trap is introduced between the two risers, as shown in both methods. During part-load operation, gas velocity is not sufficient to return oil through both risers, and the trap gradually fills up with oil until riser B is sealed off. The gas then travels up riser A only with enough velocity to carry oil along with it back into the horizontal suction main.

The oil holding capacity of the trap is limited to a minimum by close-coupling the fittings at the bottom of the risers. If this is not done, the trap can accumulate enough oil during part-load operation to lower the compressor crankcase oil level. Note in Figure 3 that riser lines A and B form an inverted loop and enter the horizontal suction line from the top. This prevents oil drainage into the risers, which may be idle during part-load operation. The same purpose can be served by running risers horizontally into the main, provided that the main is larger in diameter than either riser.

Often, double suction risers are essential on low-temperature systems that can tolerate very little pressure drop. Any system using

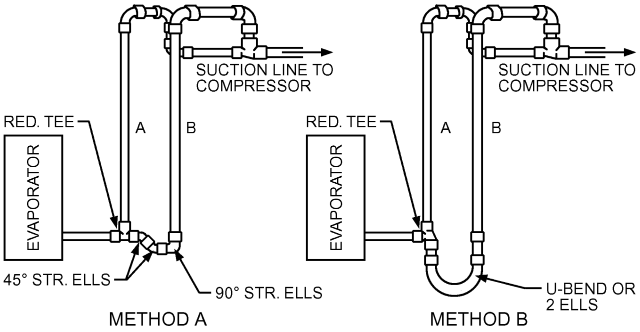


Fig. 3 Double-Suction Riser Construction

Table 19 Minimum Refrigeration Capacity in Tons for Oil Entrainment up Suction Risers (Type L Copper Tubing)

Refrigerant	Saturated Suction Temp., °F	Suction Gas Temp., °F	Pipe OD, in.											
			1/2	5/8	3/4	7/8	1 1/8	1 3/8	1 5/8	2 1/8	2 5/8	3 1/8	3 5/8	4 1/8
			Area, in^2											
			0.146	0.233	0.348	0.484	0.825	1.256	1.780	3.094	4.770	6.812	9.213	11.970
22	–40.0	–30.0	0.067	0.119	0.197	0.298	0.580	0.981	1.52	3.03	5.20	8.12	11.8	16.4
		–10.0	0.065	0.117	0.194	0.292	0.570	0.963	1.49	2.97	5.11	7.97	11.6	16.1
		10.0	0.066	0.118	0.195	0.295	0.575	0.972	1.50	3.00	5.15	8.04	11.7	16.3
	–20.0	–10.0	0.087	0.156	0.258	0.389	0.758	1.28	1.98	3.96	6.80	10.6	15.5	21.5
		10.0	0.085	0.153	0.253	0.362	0.744	1.26	1.95	3.88	6.67	10.4	15.2	21.1
		30.0	0.086	0.154	0.254	0.383	0.747	1.26	1.95	3.90	6.69	10.4	15.2	21.1
	0.0	10.0	0.111	0.199	0.328	0.496	0.986	1.63	2.53	5.04	8.66	13.5	19.7	27.4
		30.0	0.108	0.194	0.320	0.484	0.942	1.59	2.46	4.92	8.45	13.2	19.2	26.7
		50.0	0.109	0.195	0.322	0.486	0.946	1.60	2.47	4.94	8.48	13.2	19.3	26.8
	20.0	30.0	0.136	0.244	0.403	0.608	1.18	2.00	3.10	6.18	10.6	16.6	24.2	33.5
		50.0	0.135	0.242	0.399	0.603	1.17	1.99	3.07	6.13	10.5	16.4	24.0	33.3
		70.0	0.135	0.242	0.400	0.605	1.18	1.99	3.08	6.15	10.6	16.5	24.0	33.3
	40.0	50.0	0.167	0.300	0.495	0.748	1.46	2.46	3.81	7.60	13.1	20.4	29.7	41.3
		70.0	0.165	0.296	0.488	0.737	1.44	2.43	3.75	7.49	12.9	20.1	29.3	40.7
		90.0	0.165	0.296	0.488	0.738	1.44	2.43	3.76	7.50	12.9	20.1	29.3	40.7
134a	0.0	10.0	0.089	0.161	0.259	0.400	0.78	1.32	2.03	4.06	7.0	10.9	15.9	22.1
		30.0	0.075	0.135	0.218	0.336	0.66	1.11	1.71	3.42	5.9	9.2	13.4	18.5
		50.0	0.072	0.130	0.209	0.323	0.63	1.07	1.64	3.28	5.6	8.8	12.8	17.8
	10.0	20.0	0.101	0.182	0.294	0.453	0.88	1.49	2.31	4.61	7.9	12.4	18.0	25.0
		40.0	0.084	0.152	0.246	0.379	0.74	1.25	1.93	3.86	6.6	10.3	15.1	20.9
		60.0	0.081	0.147	0.237	0.366	0.71	1.21	1.87	3.73	6.4	10.0	14.6	20.2
	20.0	30.0	0.113	0.205	0.331	0.510	0.99	1.68	2.60	5.19	8.9	13.9	20.3	28.2
		50.0	0.095	0.172	0.277	0.427	0.83	1.41	2.17	4.34	7.5	11.6	17.0	23.6
		70.0	0.092	0.166	0.268	0.413	0.81	1.36	2.10	4.20	7.2	11.3	16.4	22.8
	30.0	40.0	0.115	0.207	0.335	0.517	1.01	1.70	2.63	5.25	9.0	14.1	20.5	28.5
		60.0	0.107	0.193	0.311	0.480	0.94	1.58	2.44	4.88	8.4	13.1	19.1	26.5
		80.0	0.103	0.187	0.301	0.465	0.91	1.53	2.37	4.72	8.1	12.7	18.5	25.6
	40.0	50.0	0.128	0.232	0.374	0.577	1.12	1.90	2.94	5.87	10.1	15.7	22.9	31.8
		70.0	0.117	0.212	0.342	0.528	1.03	1.74	2.69	5.37	9.2	14.4	21.0	29.1
		90.0	0.114	0.206	0.332	0.512	1.00	1.69	2.61	5.21	8.9	14.0	20.4	28.3

Notes:

1. Refrigeration capacity in tons is based on 90°F liquid temperature and superheat as indicated by listed temperature. For other liquid line temperatures, use correction factors in table at right.
2. Values computed using ISO 32 mineral oil for R-22. R-134a computed using ISO 32 ester-based oil.

Refrigerant	Liquid Temperature, °F								
	50	60	70	80	100	110	120	130	140
22	1.17	1.14	1.10	1.06	0.98	0.94	0.89	0.85	0.80
134a	1.26	1.20	1.13	1.07	0.94	0.87	0.80	0.74	0.67

these risers should include a suction trap (accumulator) and a means of returning oil gradually.

For systems operating at higher suction temperatures, such as for comfort air conditioning, single suction risers can be sized for oil return at minimum load. Where single compressors are used with capacity control, minimum capacity is usually 25 or 33% of maximum displacement. With this low ratio, pressure drop in single suction risers designed for oil return at minimum load is rarely serious at full load.

When multiple compressors are used, one or more may shut down while another continues to operate, and the maximum-to-minimum ratio becomes much larger. This may make a double suction riser necessary.

The remaining suction line portions are sized to allow a practical pressure drop between the evaporators and compressors because oil is carried along in horizontal lines at relatively low gas velocities. It is good practice to give some pitch to these lines toward the compressor. Traps should be avoided, but when that is impossible, the risers from them are treated the same as those leading from the evaporators.

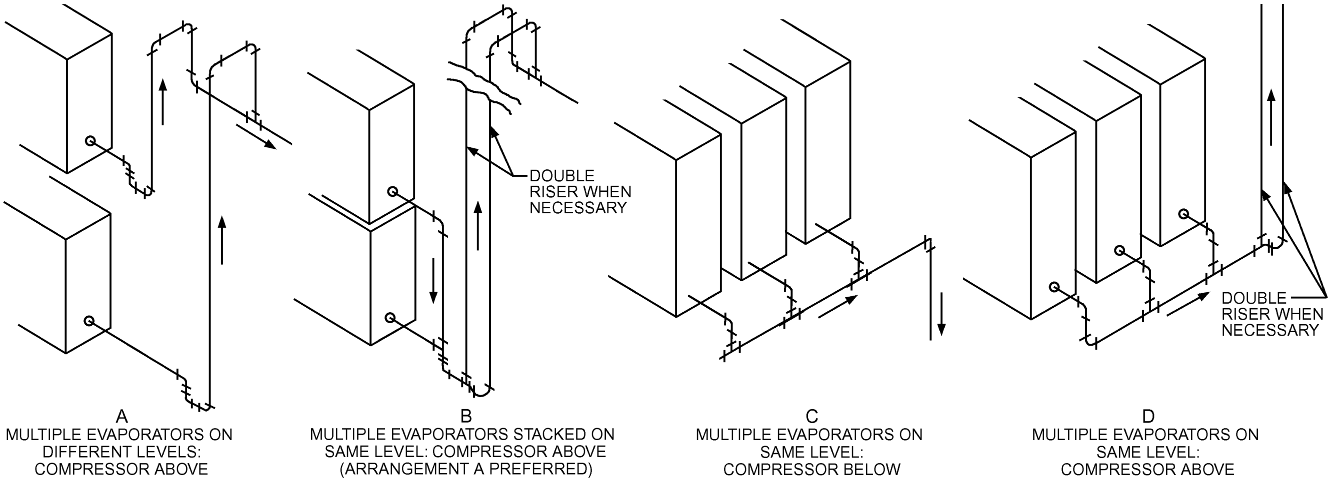
Preventing Oil Trapping in Idle Evaporators. Suction lines should be designed so that oil from an active evaporator does not drain into an idle one. Figure 4A shows multiple evaporators on different floor levels with the compressor above. Each suction line is brought upward and looped into the top of the common suction line to prevent oil from draining into inactive coils.

Figure 4B shows multiple evaporators stacked on the same level, with the compressor above. Oil cannot drain into the lowest evaporator because the common suction line drops below the outlet of the lowest evaporator before entering the suction riser.

Figure 4C shows multiple evaporators on the same level, with the compressor located below. The suction line from each evaporator drops down into the common suction line so that oil cannot drain into an idle evaporator. An alternative arrangement is shown in Figure 4D for cases where the compressor is above the evaporators.

Figure 5 illustrates typical piping for evaporators above and below a common suction line. All horizontal runs should be level or pitched toward the compressor to ensure oil return.

Traps shown in the suction lines after the evaporator suction outlet are recommended by thermal expansion valve manufacturers to prevent erratic operation of the thermal expansion valve. Expansion valve bulbs are located on the suction lines between the evaporator and these traps. The traps serve as drains and help prevent liquid from accumulating under the expansion valve bulbs during compressor off cycles. They are useful only where straight runs or risers are encountered in the suction line leaving the evaporator outlet.



A
MULTIPLE EVAPORATORS ON
DIFFERENT LEVELS:
COMPRESSOR ABOVE

B
MULTIPLE EVAPORATORS STACKED ON
SAME LEVEL: COMPRESSOR ABOVE
(ARRANGEMENT A PREFERRED)

C
MULTIPLE EVAPORATORS ON
SAME LEVEL:
COMPRESSOR BELOW

D
MULTIPLE EVAPORATORS ON
SAME LEVEL:
COMPRESSOR ABOVE

Note: All arrangements should include a pumpdown cycle.

Fig. 4 Suction Line Piping at Evaporator Coils

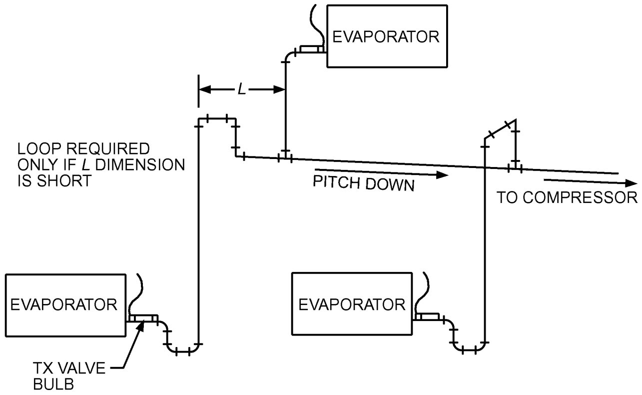


Fig. 5 Typical Piping from Evaporators Located above and below Common Suction Line

DISCHARGE (HOT-GAS) LINES

Hot-gas lines should be designed to

- Avoid trapping oil at part-load operation
- Prevent condensed refrigerant and oil in the line from draining back to the head of the compressor
- Have carefully selected connections from a common line to multiple compressors
- Avoid developing excessive noise or vibration from hot-gas pulsations, compressor vibration, or both

Oil Transport up Risers at Normal Loads. Although a low pressure drop is desired, oversized hot-gas lines can reduce gas velocities to a point where the refrigerant will not transport oil. Therefore, when using multiple compressors with capacity control, hot-gas risers must transport oil at all possible loadings.

Minimum Gas Velocities for Oil Transport in Risers. Minimum capacities for oil entrainment in hot-gas line risers are shown in Table 20. On multiple-compressor installations, the lowest possible system loading should be calculated and a riser size selected to give at least the minimum capacity indicated in the table for successful oil transport.

In some installations with multiple compressors and with capacity control, a vertical hot-gas line, sized to transport oil at minimum load, has excessive pressure drop at maximum load. When this problem exists, either a double riser or a single riser with an oil separator can be used.

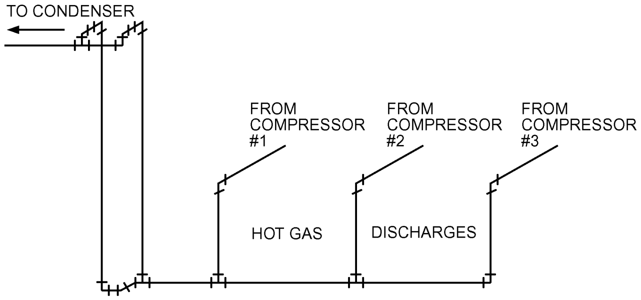


Fig. 6 Double Hot-Gas Riser

Double Hot-Gas Risers. A double hot-gas riser can be used the same way it is used in a suction line. Figure 6 shows the double riser principle applied to a hot-gas line. Its operating principle and sizing technique are described in the section on Double Suction Risers.

Single Riser and Oil Separator. As an alternative, an oil separator in the discharge line just before the riser allows sizing the riser for a low pressure drop. Any oil draining back down the riser accumulates in the oil separator. With large multiple compressors, separator capacity may dictate the use of individual units for each compressor located between the discharge line and the main discharge header. Horizontal lines should be level or pitched downward in the direction of gas flow to facilitate travel of oil through the system and back to the compressor.

Piping to Prevent Liquid and Oil from Draining to Compressor Head. Whenever the condenser is located above the compressor, the hot-gas line should be trapped near the compressor before rising to the condenser, especially if the hot-gas riser is long. This minimizes the possibility of refrigerant, condensed in the line during off cycles, draining back to the head of the compressor. Also, any oil traveling up the pipe wall will not drain back to the compressor head.

The loop in the hot-gas line (Figure 7) serves as a reservoir and traps liquid resulting from condensation in the line during shutdown, thus preventing gravity drainage of liquid and oil back to the compressor head. A small high-pressure float drainer should be installed at the bottom of the trap to drain any significant amount of refrigerant condensate to a low-side component such as a suction accumulator or low-pressure receiver. This float prevents excessive build-up of liquid in the trap and possible liquid hammer when the compressor is restarted.

Table 20 Minimum Refrigeration Capacity in Tons for Oil Entrainment up Hot-Gas Risers (Type L Copper Tubing)

Refrigerant	Saturated Temp., °F	Discharge Gas Temp., °F	Pipe OD, in.											
			1/2	5/8	3/4	7/8	1 1/8	1 3/8	1 5/8	2 1/8	2 5/8	3 1/8	3 5/8	4 1/8
			Area, in^2											
			0.146	0.233	0.348	0.484	0.825	1.256	1.780	3.094	4.770	6.812	9.213	11.970
22	80.0	110.0	0.235	0.421	0.695	1.05	2.03	3.46	5.35	10.7	18.3	28.6	41.8	57.9
		140.0	0.223	0.399	0.659	0.996	1.94	3.28	5.07	10.1	17.4	27.1	39.6	54.9
		170.0	0.215	0.385	0.635	0.960	1.87	3.16	4.89	9.76	16.8	26.2	38.2	52.9
	90.0	120.0	0.242	0.433	0.716	1.06	2.11	3.56	5.50	11.0	18.9	29.5	43.0	59.6
		150.0	0.226	0.406	0.671	1.01	1.97	3.34	5.16	10.3	17.7	27.6	40.3	55.9
		180.0	0.216	0.387	0.540	0.956	1.88	3.18	4.92	9.82	16.9	26.3	38.4	53.3
	100.0	130.0	0.247	0.442	0.730	1.10	2.15	3.83	5.62	11.2	19.3	30.1	43.9	60.8
		160.0	0.231	0.414	0.884	1.03	2.01	3.40	5.26	10.5	18.0	28.2	41.1	57.0
		190.0	0.220	0.394	0.650	0.982	1.91	3.24	3.00	9.96	17.2	26.8	39.1	54.2
	110.0	140.0	0.251	0.451	0.744	1.12	2.19	3.70	5.73	11.4	19.6	30.6	44.7	62.0
		170.0	0.235	0.421	0.693	1.05	2.05	3.46	3.35	10.7	18.3	28.6	41.8	57.9
		200.0	0.222	0.399	0.658	0.994	1.94	3.28	5.06	10.1	17.4	27.1	39.5	54.8
	120.0	150.0	0.257	0.460	0.760	1.15	2.24	3.78	5.85	11.7	20.0	31.3	45.7	63.3
		180.0	0.239	0.428	0.707	1.07	2.08	3.51	5.44	10.8	18.6	29.1	42.4	58.9
		210.0	0.225	0.404	0.666	1.01	1.96	3.31	5.12	10.2	17.6	27.4	40.0	55.5
134a	80.0	110.0	0.199	0.360	0.581	0.897	1.75	2.96	4.56	9.12	15.7	24.4	35.7	49.5
		140.0	0.183	0.331	0.535	0.825	1.61	2.72	4.20	8.39	14.4	22.5	32.8	45.6
		170.0	0.176	0.318	0.512	0.791	1.54	2.61	4.02	8.04	13.8	21.6	31.4	43.6
	90.0	120.0	0.201	0.364	0.587	0.906	1.76	2.99	4.61	9.21	15.8	24.7	36.0	50.0
		150.0	0.184	0.333	0.538	0.830	1.62	2.74	4.22	8.44	14.5	22.6	33.0	45.8
		180.0	0.177	0.320	0.516	0.796	1.55	2.62	4.05	8.09	13.9	21.7	31.6	43.9
	100.0	130.0	0.206	0.372	0.600	0.926	1.80	3.05	4.71	9.42	16.2	25.2	36.8	51.1
		160.0	0.188	0.340	0.549	0.848	1.65	2.79	4.31	8.62	14.8	23.1	33.7	46.8
		190.0	0.180	0.326	0.526	0.811	1.58	2.67	4.13	8.25	14.2	22.1	32.2	44.8
	110.0	140.0	0.209	0.378	0.610	0.942	1.83	3.10	4.79	9.57	16.5	25.7	37.4	52.0
		170.0	0.191	0.346	0.558	0.861	1.68	2.84	4.38	8.76	15.0	23.5	34.2	47.5
		200.0	0.183	0.331	0.534	0.824	1.61	2.72	4.19	8.38	14.4	22.5	32.8	45.5
	120.0	150.0	0.212	0.383	0.618	0.953	1.86	3.14	4.85	9.69	16.7	26.0	37.9	52.6
		180.0	0.194	0.351	0.566	0.873	1.70	2.88	4.44	8.88	15.3	23.8	34.7	48.2
		210.0	0.184	0.334	0.538	0.830	1.62	2.74	4.23	8.44	14.5	22.6	33.0	45.8

Notes:

1. Refrigeration capacity in tons based on saturated suction temperature of 20°F with 15°F superheat at indicated saturated condensing temperature with 15°F subcooling. For other saturated suction temperatures with 15°F superheat, use correction factors in the table at right.
2. Table computed using ISO 32 mineral oil for R-22, and ISO 32 ester-based oil for R-134a.

Refrigerant	Saturated Suction Temperature, °F			
	–40	–20	0	+40
22	0.92	0.95	0.97	1.02
134a	—	—	0.96	1.04

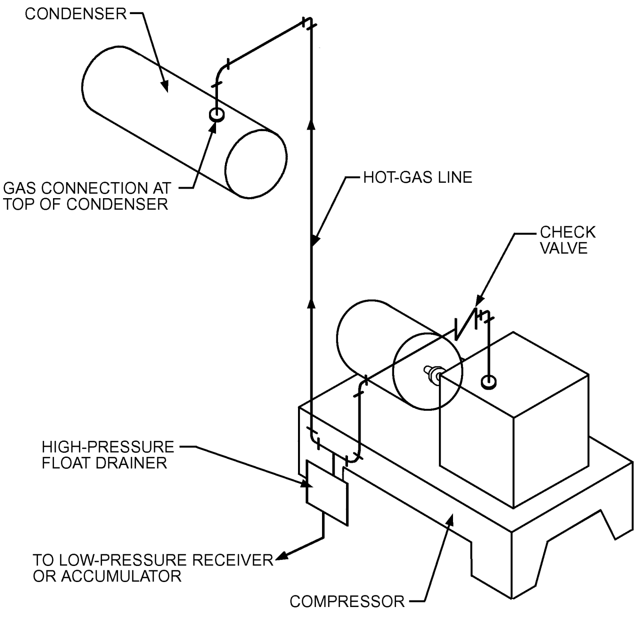


Fig. 7 Hot-Gas Loop

For multiple-compressor arrangements, each discharge line should have a check valve to prevent gas from active compressors from condensing on heads of idle compressors.

For single-compressor applications, a tightly closing check valve should be installed in the hot-gas line of the compressor whenever the condenser and the receiver ambient temperature are higher than that of the compressor. The check valve prevents refrigerant from boiling off in the condenser or receiver and condensing on the compressor heads during off cycles.

This check valve should be a piston type, which will close by gravity when the compressor stops running. The use of a spring-loaded check may incur chatter (vibration), particularly on slow-speed reciprocating compressors.

For compressors equipped with water-cooled oil coolers, a water solenoid and water-regulating valve should be installed in the water line so that the regulating valve maintains adequate cooling during operation, and the solenoid stops flow during the off cycle to prevent localized condensing of the refrigerant.

Hot-Gas (Discharge) Mufflers. Mufflers can be installed in hot-gas lines to dampen discharge gas pulsations, reducing vibration and noise. Mufflers should be installed in a horizontal or downflow portion of the hot-gas line immediately after it leaves the compressor.

Because gas velocity through the muffler is substantially lower than that through the hot-gas line, the muffler may form an oil trap.

The muffler should be installed to allow oil to flow through it and not be trapped.

DEFROST GAS SUPPLY LINES

Sizing refrigeration lines to supply defrost gas to one or more evaporators is not an exact science. The parameters associated with sizing the defrost gas line are related to allowable pressure drop and refrigerant flow rate during defrost.

Engineers use an estimated two times the evaporator load for effective refrigerant flow rate to determine line sizing requirements. Pressure drop is not as critical during the defrost cycle, and many engineers use velocity as the criterion for determining line size. The effective condensing temperature and average temperature of the gas must be determined. The velocity determined at saturated conditions gives a conservative line size.

Some controlled testing (Stoecker 1984) has shown that, in small coils with R-22, the defrost flow rate tends to be higher as the condensing temperature is increased. The flow rate is on the order of two to three times the normal evaporator flow rate, which supports the estimated two times used by practicing engineers.

Table 21 provides guidance on selecting defrost gas supply lines based on velocity at a saturated condensing temperature of 70°F. It is recommended that initial sizing be based on twice the evaporator flow rate and that velocities from 1000 to 2000 fpm be used for determining the defrost gas supply line size.

Gas defrost lines must be designed to continuously drain any condensed liquid.

RECEIVERS

Refrigerant receivers are vessels used to store excess refrigerant circulated throughout the system. Their purpose is to

- Provide pumpdown storage capacity when another part of the system must be serviced or the system must be shut down for an extended time. In some water-cooled condenser systems, the condenser also serves as a receiver if the total refrigerant charge does not exceed its storage capacity.
- Handle the excess refrigerant charge that occurs with air-cooled condensers using flooding condensing pressure control (see the section on Head Pressure Control for Refrigerant Condensers).
- Accommodate a fluctuating charge in the low side and drain the condenser of liquid to maintain an adequate effective condensing surface on systems where the operating charge in the evaporator and/or condenser varies for different loading conditions. When an evaporator is fed with a thermal expansion valve, hand expansion valve, or low-pressure float, the operating charge in the evaporator varies considerably depending on the loading. During low load, the evaporator requires a larger charge because boiling is not as intense. When load increases, the operating charge in the evaporator decreases, and the receiver must store excess refrigerant.
- Hold the full charge of the idle circuit on systems with multicircuit evaporators that shut off liquid supply to one or more circuits during reduced load and pump out the idle circuit.

Connections for Through-Type Receiver. When a through-type receiver is used, liquid must always flow from condenser to receiver. Pressure in the receiver must be lower than that in the condenser outlet. The receiver and its associated piping provide free flow of liquid from the condenser to the receiver by equalizing pressures between the two so that the receiver cannot build up a higher pressure than the condenser.

If a vent is not used, piping between condenser and receiver (condensate line) is sized so that liquid flows in one direction and gas flows in the opposite direction. Sizing the condensate line for 100 fpm liquid velocity is usually adequate to attain this flow. Piping should slope at least 0.25 in/ft and eliminate any natural liquid traps. Figure 8 illustrates this configuration.

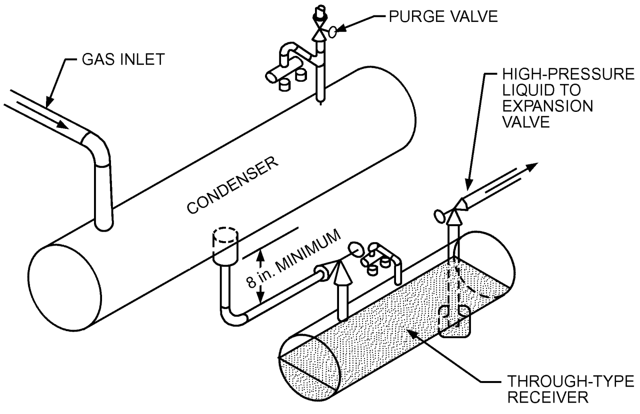


Fig. 8 Shell-and-Tube Condenser to Receiver Piping (Through-Type Receiver)

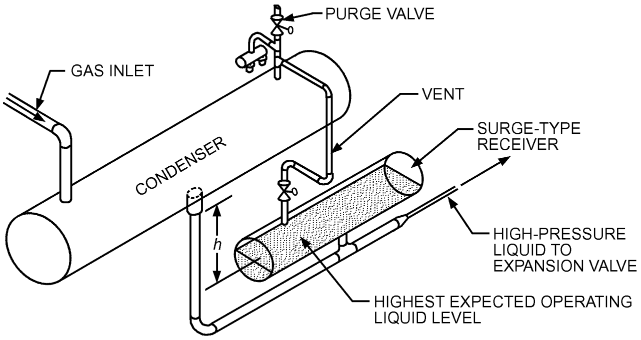


Fig. 9 Shell-and-Tube Condenser to Receiver Piping (Surge-Type Receiver)

Piping between the condenser and the receiver can be equipped with a separate vent (equalizer) line to allow receiver and condenser pressures to equalize. This external vent line can be piped either with or without a check valve in the vent line (see Figures 10 and 11). If there is no check valve, prevent discharge gas from discharging directly into the vent line; this should prevent a gas velocity pressure component from being introduced on top of the liquid in the receiver. When the piping configuration is unknown, install a check valve in the vent with flow in the direction of the condenser. The check valve should be selected for minimum opening pressure (i.e., approximately 0.5 psi). When determining condensate drop leg height, allowance must be made to overcome both the pressure drop across this check valve and the refrigerant pressure drop through the condenser. This ensures that there will be no liquid backup into an operating condenser on a multiple-condenser application when one or more of the condensers is idle. The condensate line should be sized so that velocity does not exceed 150 fpm.

The vent line flow is from receiver to condenser when receiver temperature is higher than condensing temperature. Flow is from condenser to receiver when air temperature around the receiver is below condensing temperature. Flow rate depends on this temperature difference as well as on the receiver surface area. Vent size can be calculated from this flow rate.

Connections for Surge-Type Receiver. The purpose of a surge-type receiver is to allow liquid to flow to the expansion valve without exposure to refrigerant in the receiver, so that it can remain subcooled. The receiver volume is available for liquid that is to be removed from the system. Figure 9 shows an example of connections for a surge-type receiver. Height *h* must be adequate for a liquid pressure at least as large as the pressure loss through the condenser, liquid line, and vent line at the maximum temperature difference between

Table 21 Refrigerant Flow Capacity Data For Defrost Lines

Pipe Size Copper[a]		R-22 Mass Flow Data, lb/h			R-134a Mass Flow Data, lb/h			R-404a Mass Flow Data, lb/h			R-507 Mass Flow Data, lb/h			R-410a Mass Flow Data, lb/h			R-407c Mass Flow Data, lb/h		
		Velocity, fpm			Velocity, fpm			Velocity, fpm			Velocity, fpm			Velocity, fpm			Velocity, fpm		
		1000	2000	3000	1000	2000	3000	1000	2000	3000	1000	2000	3000	1000	2000	3000	1000	2000	3000
1/2		150	300	450	110	220	330	220	440	660	233	465	698	221	442	662	147	294	441
5/8		240	480	720	170	350	520	354	707	1061	374	747	1121	355	709	1064	236	472	708
3/4		350	710	1060	260	510	770	528	1056	1584	558	1116	1674	530	1059	1589	352	705	1057
7/8		500	1000	1500	360	720	1090	734	1467	2201	775	1550	2325	736	1471	2207	490	979	1469
1 1/8		850	1700	2550	620	1230	1850	1251	2502	3752	1321	2643	3964	1254	2509	3763	835	1670	2504
1 3/8		1300	2590	3890	940	1880	2820	1905	3810	5715	2013	4025	6037	1910	3821	5731	1272	2543	3814
1 5/8		1840	3670	5510	1330	2660	3990	2697	5393	8090	2849	5697	8546	2704	5408	8112	1800	3599	5399
2 1/8		3190	6390	9580	2310	4630	6940	4691	9382	14,073	4955	9911	14,866	4704	9408	14,112	3131	6262	9392
2 5/8		4930	9850	14,800	3570	7140	10,700	7234	14,468	21,702	7642	15,283	22,925	7254	14,508	21,762	4828	9656	14,484
3 1/8		7030	14,100	21,100	5100	10,200	15,300	10,326	20,651	30,977	10,907	21,815	32,722	10,354	20,708	31,062	6891	13,783	20,674
3 5/8		9510	19,000	28,500	6900	13,800	20,700	13,966	27,932	41,897	14,753	29,505	44,258	14,004	28,008	42,012	9321	18,641	27,962
4 1/8		12,400	24,700	37,100	9000	17,900	26,900	18,155	36,309	54,464	19,178	38,355	57,533	18,204	36,409	54,613	12,116	24,233	36,349
5 1/8		19,300	38,500	57,800	14,000	27,900	41,900	28,294	56,588	84,882	29,888	59,776	89,665	28,372	56,743	85,115	18,883	37,767	56,650
6 1/8		27,700	55,400	83,100	20,100	40,100	60,200	40,674	81,347	122,021	42,965	85,931	128,896	40,785	81,571	122,356	27,146	54,291	81,436
8 1/8		48,400	96,700	145,100	35,100	70,100	105,200	71,046	142,092	213,138	75,049	150,099	225,148	71,241	142,483	213,724	47,416	94,832	142,248
Steel																			
IPS	**SCH**																		
3/8	80	150	290	440	110	210	320	213	426	639	225	450	675	214	427	641	142	284	427
1/2	80	240	480	720	180	350	530	355	710	1,065	375	750	1125	356	712	1,068	237	474	711
3/4	80	450	890	1340	320	650	970	656	1,311	1,966	692	1385	2077	657	1,315	1,972	438	875	1312
1	80	740	1480	2230	540	1080	1610	1090	2181	3271	1152	2304	3455	1093	2187	3280	728	1455	2183
1 1/4	80	1540	3090	4630	1120	2240	3360	1945	3889	5833	2054	4108	6162	1950	3900	5850	1298	2596	3893
1 1/2	80	2100	4200	6300	1520	3050	4570	2679	5357	8036	2830	5659	8489	2686	5372	8058	1788	3576	5363
2	40	3460	6930	10,400	2510	5020	7530	5087	10,173	15,260	5373	10,746	16,120	5101	10,201	15,302	3395	6790	10,184
2 1/2	40	4940	9870	14,800	3580	7160	10,700	7252	14,503	21,755	7660	15,320	22,981	7272	14,543	21,815	4840	9679	14,519
3	40	7620	15,200	22,900	5530	11,100	16,600	11,199	22,398	33,596	11,830	23,660	35,489	11,230	22,459	33,689	7474	14,948	22,422
4	40	13,100	26,300	39,400	9520	19,000	28,600	19,297	38,594	57,891	20,384	40,769	61,153	19,350	38,700	58,050	12,879	25,758	38,636
5	40	20,600	41,300	61,900	15,000	29,900	44,900	30,302	60,603	90,905	32,009	64,018	96,027	30,385	60,770	91,155	20,223	40,447	60,670
6	40	29,800	59,600	89,400	21,600	43,200	64,800	43,793	87,586	131,379	46,261	92,521	138,782	43,913	87,827	131,740	29,227	58,455	87,682
8	40	51,600	103,300	154,900	37,400	74,800	112,300	75,833	151,666	227,498	80,106	160,212	240,318	76,041	152,083	228,124	50,611	101,222	151,832
10	40	81,400	162,800	244,100	59,000	118,00	176,900	119,530	239,061	358,591	126,266	252,532	378,797	119,859	239,718	359,576	79,775	159,549	239,323
12	ID[b]	116,700	233,400	350,200	84,600	169,200	253,800	171,437	342,874	514,311	181,098	362,195	543,293	171,908	343,817	515,725	114,417	228,834	343,251
14	30	—	—	—	—	—	—	209,013	418,027	627,040	220,791	441,582	662,374	209,588	419,176	628,764	139,496	278,991	418,486
16	30	—	—	—	—	—	—	276,874	553,748	830,622	292,476	584,951	877,427	277,635	555,270	832,905	184,786	369,571	554,357

Note: Refrigerant flow data based on saturated condensing temperature of 70°F.

[a] For brazed Type L copper tubing for defrost service, see Safety Requirements section.

[b] Pipe inside diameter is same as nominal pipe size.

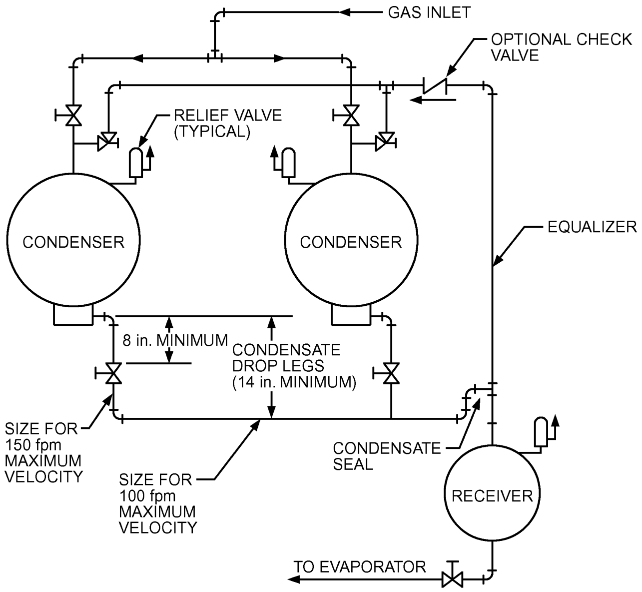


Fig. 10 Parallel Condensers with Through-Type Receiver

the receiver ambient and the condensing temperature. Condenser pressure drop at the greatest expected heat rejection should be obtained from the manufacturer. The minimum value of *h* can then be calculated and a decision made as to whether the available height will permit the surge-type receiver.

Multiple Condensers. Two or more condensers connected in series or in parallel can be used in a single refrigeration system. If connected in series, the pressure losses through each condenser must be added. Condensers are more often arranged in parallel. Pressure loss through any one of the parallel circuits is always equal to that through any of the others, even if it results in filling much of one circuit with liquid while gas passes through another.

Figure 10 shows a basic arrangement for parallel condensers with a through-type receiver. Condensate drop legs must be long enough to allow liquid levels in them to adjust to equalize pressure losses between condensers at all operating conditions. Drop legs should be 6 to 12 in. higher than calculated to ensure that liquid outlets remain free-draining. This height provides a liquid pressure to offset the largest condenser pressure loss. The liquid seal prevents gas blow-by between condensers.

Large single condensers with multiple coil circuits should be piped as though the independent circuits were parallel condensers. For example, if the left condenser in Figure 10 has 2 psi more pressure drop than the right condenser, the liquid level on the left is about 4 ft higher than that on the right. If the condensate lines do not have enough vertical height for this level difference, liquid will back up into the condenser until pressure drop is the same through both circuits. Enough surface may be covered to reduce condenser capacity significantly.

Condensate drop legs should be sized based on 150 fpm velocity. The main condensate lines should be based on 100 fpm. Depending on prevailing local and/or national safety codes, a relief device may have to be installed in the discharge piping.

Figure 11 shows a piping arrangement for parallel condensers with a surge-type receiver. When the system is operating at reduced load, flow paths through the circuits may not be symmetrical. Small pressure differences are not unusual; therefore, the liquid line junction should be about 2 or 3 ft below the bottom of the condensers. The exact amount can be calculated from pressure loss through each path at all possible operating conditions.

When condensers are water-cooled, a single automatic water valve for the condensers in one refrigeration system should be used. Individual valves for each condenser in a single system cannot maintain the same pressure and corresponding pressure drops.

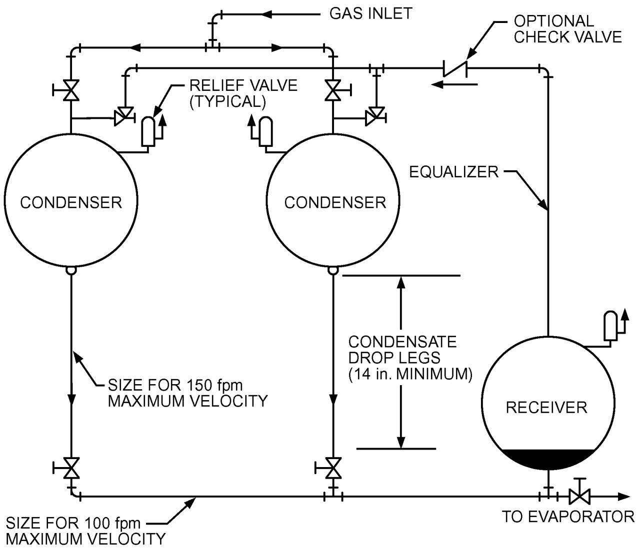


Fig. 11 Parallel Condensers with Surge-Type Receiver

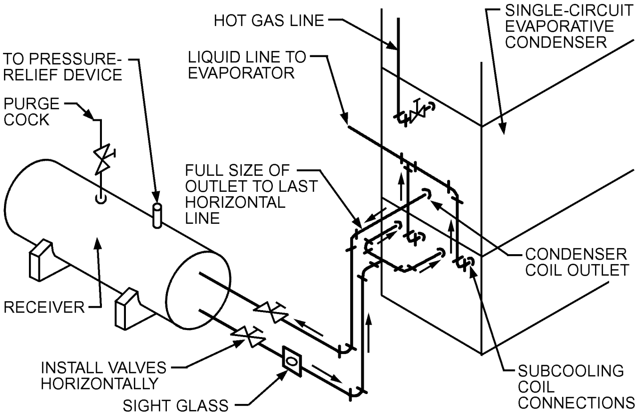


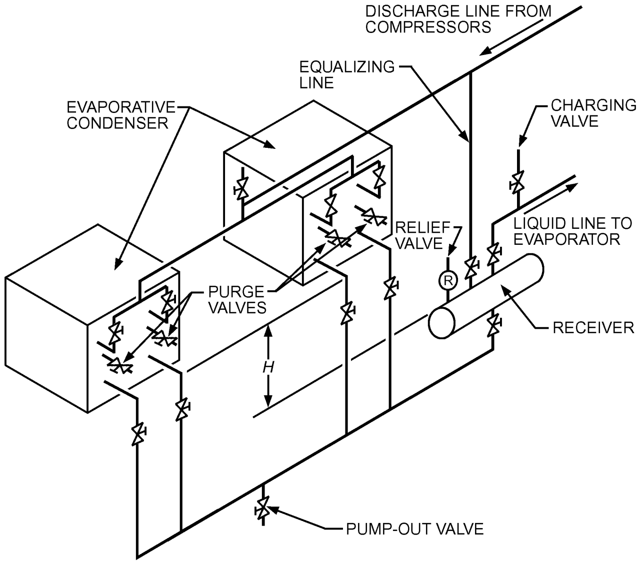
Fig. 12 Single-Circuit Evaporative Condenser with Receiver and Liquid Subcooling Coil

With evaporative condensers (Figure 12), pressure loss may be high. If parallel condensers are alike and all are operated, the differences may be small, and condenser outlets need not be more than 2 or 3 ft above the liquid line junction. If fans on one condenser are not operated while the fans on another condenser are, then the liquid level in the one condenser must be high enough to compensate for the pressure drop through the operating condenser.

When the available level difference between condenser outlets and the liquid-line junction is sufficient, the receiver may be vented to the condenser inlets (Figure 13). In this case, the surge-type receiver can be used. The level difference must then be at least equal to the greatest loss through any condenser circuit plus the greatest vent line loss when the receiver ambient is greater than the condensing temperature.

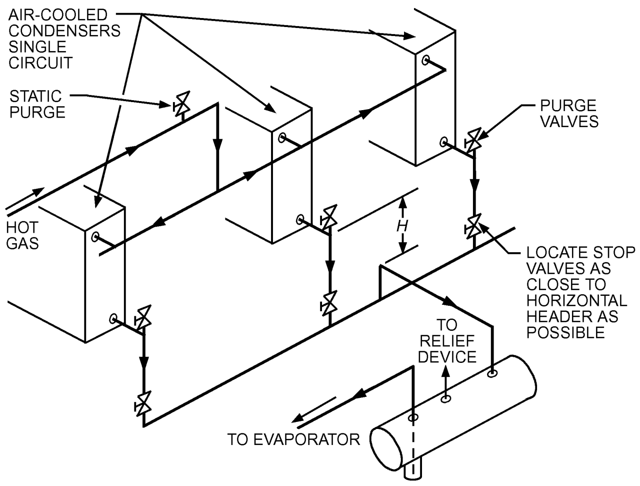
AIR-COOLED CONDENSERS

Refrigerant pressure drop through air-cooled condensers must be obtained from the supplier for the particular unit at the specified load. If refrigerant pressure drop is low enough and the arrangement is practical, parallel condensers can be connected to allow for



All outlets must be trapped. Trap leg height must be such that when one or more units are idle (fan or pump stopped), liquid may rise in operating unit leg so that static head equals pressure drop in operating unit at all conditions. Trap height should be 6 to 12 in. greater than calculated height to ensure that liquid outlets remain free-draining.

Fig. 13 Multiple Evaporative Condensers with Equalization to Condenser Inlets



Note: Leg *H* should be maximum possible. See text for minimum and limitations.

Fig. 14 Multiple Air-Cooled Condensers

capacity reduction to zero on one condenser without causing liquid backup in active condensers (Figure 14). Multiple condensers with high pressure drops can be connected as shown in Figure 14, provided that (1) the ambient at the receiver is equal to or lower than the inlet air temperature to the condenser; (2) capacity control affects all units equally; (3) all units operate when one operates, unless valved off at both inlet and outlet; and (4) all units are of equal size.

A single condenser with any pressure drop can be connected to a receiver without an equalizer and without trapping height if the condenser outlet and the line from it to the receiver can be sized for sewer flow without a trap or restriction, using a maximum velocity of 100 fpm. A single condenser can also be connected with an equalizer line to the hot-gas inlet if the vertical drop leg is sufficient to balance refrigerant pressure drop through the condenser and liquid line to the receiver.

If unit sizes are unequal, additional liquid height *H*, equivalent to the difference in full-load pressure drop, is required. Usually, condensers of equal size are used in parallel applications.

If the receiver cannot be located in an ambient temperature below the inlet air temperature for all operating conditions, sufficient extra height of drop leg *H* is required to overcome the equivalent differences in saturation pressure of the receiver and the condenser. Subcooling by the liquid leg tends to condense vapor in the receiver to reach a balance between rate of condensation, at an intermediate saturation pressure, and heat gain from ambient to the receiver. A relatively large liquid leg is required to balance a small temperature difference; therefore, this method is probably limited to marginal cases. Liquid leaving the receiver is nonetheless saturated, and any subcooling to prevent flashing in the liquid line must be obtained downstream of the receiver. If the temperature of the receiver ambient is above the condensing pressure only at part-load conditions, it may be acceptable to back liquid into the condensing surface, sacrificing the operating economy of lower part-load head pressure for a lower liquid leg requirement. The receiver must be adequately sized to contain a minimum of the backed-up liquid so that the condenser can be fully drained when full load is required. If a low-ambient control system of backing liquid into the condenser is used, consult the system supplier for proper piping.

PIPING AT MULTIPLE COMPRESSORS

Multiple compressors operating in parallel must be carefully piped to ensure proper operation.

Suction Piping

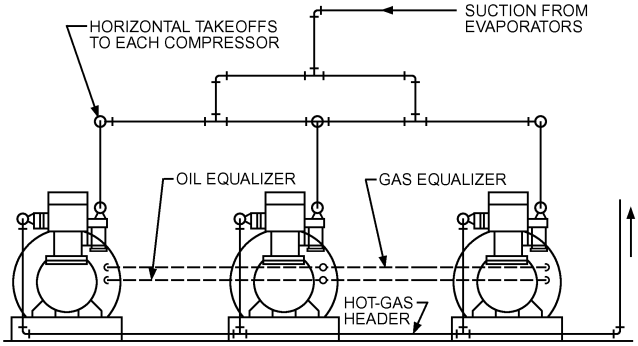
Suction piping should be designed so that all compressors run at the same suction pressure and so that oil is returned in equal proportions. All suction lines should be brought into a common suction header to return oil to each crankcase as uniformly as possible. Depending on the type and size of compressors, oil may be returned by designing the piping in one or more of the following schemes:

- Oil returned with the suction gas to each compressor
- Oil contained with a suction trap (accumulator) and returned to the compressors through a controlled means
- Oil trapped in a discharge line separator and returned to the compressors through a controlled means (see the section on Discharge Piping)

The suction header is a means of distributing suction gas equally to each compressor. Header design can be to freely pass the suction gas and oil mixture or to provide a suction trap for the oil. The header should be run above the level of the compressor suction inlets so oil can drain into the compressors by gravity.

Figure 15 shows a pyramidal or yoke-type suction header to maximize pressure and flow equalization at each of three compressor suction inlets piped in parallel. This type of construction is recommended for applications of three or more compressors in parallel. For two compressors in parallel, a single feed between the two compressor takeoffs is acceptable. Although not as good for equalizing flow and pressure drops to all compressors, one alternative is to have the suction line from evaporators enter at one end of the header instead of using the yoke arrangement. Then the suction header may have to be enlarged to minimize pressure drop and flow turbulence.

Suction headers designed to freely pass the gas/oil mixture should have branch suction lines to compressors connected to the side of the header. Return mains from the evaporators should not be connected into the suction header to form crosses with the branch suction lines to the compressors. The header should be full size based on the largest mass flow of the suction line returning to the compressors. The takeoffs to the compressors should either be the



Note: Gas equalizer must be large enough to approximate identical crankcase pressure in all compressors with any combination of idle and operating compressors (any pressure difference is reflected by difference in oil level).

Fig. 15 Suction and Hot-Gas Headers for Multiple Compressors

same size as the suction header or be constructed so that the oil will not trap within the suction header. The branch suction lines to the compressors should not be reduced until the vertical drop is reached.

Suction traps are recommended wherever (1) parallel compressors, (2) flooded evaporators, (3) double suction risers, (4) long suction lines, (5) multiple expansion valves, (6) hot-gas defrost, (7) reverse-cycle operation, or (8) suction-pressure regulators are used.

Depending on system size, the suction header may be designed to function as a suction trap. The suction header should be large enough to provide a low-velocity region in the header to allow suction gas and oil to separate. See the section on Low-Pressure Receiver Sizing in Chapter 1 to arrive at recommended velocities for separation. Suction gas flow for individual compressors should be taken off the top of the suction header. Oil can be returned to the compressor directly or through a vessel equipped with a heater to boil off refrigerant and then allow oil to drain to the compressors or other devices used to feed oil to the compressors.

The suction trap must be sized for effective gas and liquid separation. Adequate liquid volume and a means of disposing of it must be provided. A liquid transfer pump or heater may be used. Chapter 1 has further information on separation and liquid transfer pumps.

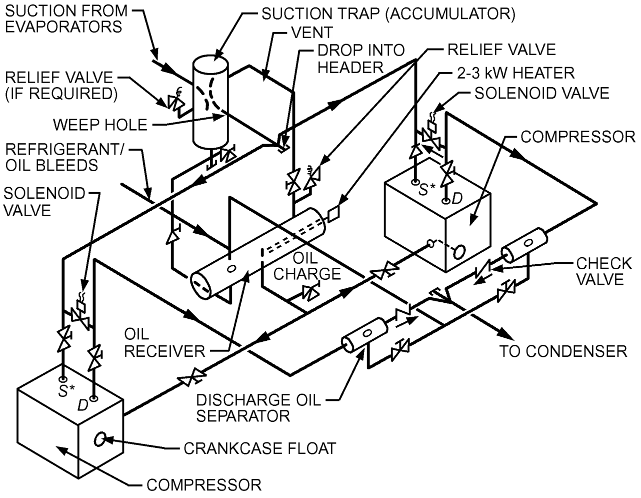
An oil receiver equipped with a heater effectively evaporates liquid refrigerant accumulated in the suction trap. It also assumes that each compressor receives its share of oil. Either crankcase float valves or external float switches and solenoid valves can be used to control the oil flow to each compressor.

A gravity-feed oil receiver should be elevated to overcome the pressure drop between it and the crankcase. The oil receiver should be sized so that a malfunction of the oil control mechanism cannot overfill an idle compressor.

Figure 16 shows a recommended hookup of multiple compressors, suction trap (accumulator), oil receiver, and discharge line oil separators. The oil receiver also provides a reserve supply of oil for compressors where oil in the system outside the compressor varies with system loading. The heater mechanism should always be submerged.

Discharge Piping

The piping arrangement in Figure 15 is suggested for discharge piping. The piping must be arranged to prevent refrigerant liquid and oil from draining back into the heads of idle compressors. A check valve in the discharge line may be necessary to prevent refrigerant and oil from entering the compressor heads by migration. It is recommended that, after leaving the compressor head, the piping be routed to a lower elevation so that a trap is formed to allow for drainback of refrigerant and oil from the discharge line when flow rates are reduced or the compressors are off. If an oil separator is used in the discharge line, it may suffice as the trap for drainback for the discharge line.



**Note*: Solenoid valve open when compressor is idle to avoid check valve leakage from condensing on heads. Not needed if oil separator has bleeder to outlet.

Fig. 16 Parallel Compressors with Gravity Oil Flow

A bullheaded tee at the junction of two compressor branches and the main discharge header should be avoided because it causes increased turbulence, increased pressure drop, and possible hammering in the line.

When an oil separator is used on multiple-compressor arrangements, oil must be piped to return to the compressors. This can be done in various ways, depending on the oil management system design. Oil may be returned to an oil receiver that is the supply for control devices feeding oil back to the compressors.

Interconnection of Crankcases

When two or more compressors are to be interconnected, a method must be provided to equalize the crankcases. Some compressor designs do not operate correctly with simple equalization of the crankcases. For these systems, it may be necessary to design a positive oil float control system for each compressor crankcase. A typical system allows oil to collect in a receiver that, in turn, supplies oil to a device that meters it back into the compressor crankcase to maintain a proper oil level (Figure 16).

Compressor systems that can be equalized should be placed on foundations so that all oil equalizer tapping locations are exactly level. If crankcase floats (as in Figure 16) are not used, an oil equalization line should connect all crankcases to maintain uniform oil levels. The oil equalizer may be run level with the tapping, or, for convenient access to compressors, it may be run at the floor (Figure 17). It should never be run at a level higher than that of the tapping.

For the oil equalizer line to work properly, equalize the crankcase pressures by installing a gas equalizer line above the oil level. This line may be run to provide head room (Figure 17) or run level with tapping on the compressors. It should be piped so that oil or liquid refrigerant will not be trapped.

Both lines should be the same size as the tapping on the largest compressor and should be valved so that any one machine can be taken out for repair. The piping should be arranged to absorb vibration.

PIPING AT VARIOUS SYSTEM COMPONENTS

Flooded Fluid Coolers

For a description of flooded fluid coolers, see Chapter 37 of the 2004 *ASHRAE Handbook—HVAC Systems and Equipment*.

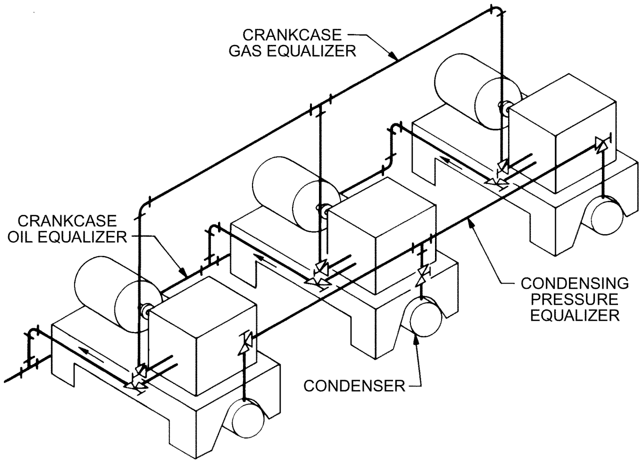
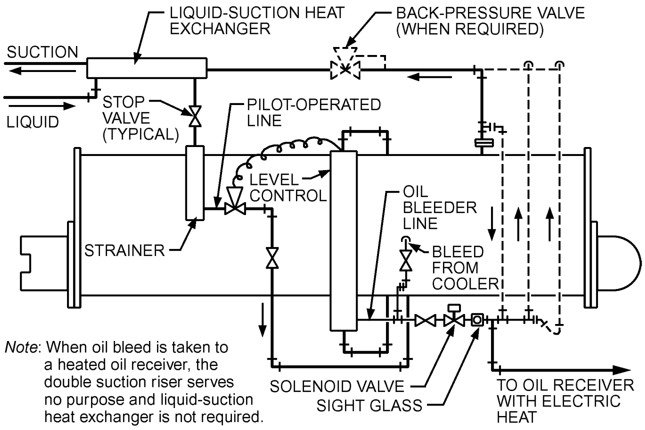


Fig. 17 Interconnecting Piping for Multiple Condensing Units



Note: When oil bleed is taken to a heated oil receiver, the double suction riser serves no purpose and liquid-suction heat exchanger is not required.

Fig. 18 Typical Piping at Flooded Fluid Cooler

Shell-and-tube flooded coolers designed to minimize liquid entrainment in the suction gas require a continuous liquid bleed line (Figure 18) installed at some point in the cooler shell below the liquid level to remove trapped oil. This continuous bleed of refrigerant liquid and oil prevents the oil concentration in the cooler from getting too high. The location of the liquid bleed connection on the shell depends on the refrigerant and oil used. For refrigerants that are highly miscible with the oil, the connection can be anywhere below the liquid level.

Refrigerant 22 can have a separate oil-rich phase floating on a refrigerant-rich layer. This becomes more pronounced as evaporating temperature drops. When R-22 is used with mineral oil, the bleed line is usually taken off the shell just slightly below the liquid level, or there may be more than one valved bleed connection at slightly different levels so that the optimum point can be selected during operation. With alkyl benzene lubricants, oil/refrigerant miscibility may be high enough that the oil bleed connection can be anywhere below the liquid level. The solubility charts in Chapter 7 give specific information.

Where the flooded cooler design requires an external surge drum to separate liquid carryover from suction gas off the tube bundle, the richest oil concentration may or may not be in the cooler. In some cases, the surge drum has the highest concentration of oil. Here, the refrigerant and oil bleed connection is taken from the surge drum. The refrigerant and oil bleed from the cooler by gravity. The bleed sometimes drains into the suction line so oil can be returned to the compressor with the suction gas after the accompanying liquid refrigerant is vaporized in a liquid-suction heat interchanger. A better method is to drain the refrigerant/oil bleed into a heated receiver that boils refrigerant off to the suction line and drains oil back to the compressor.

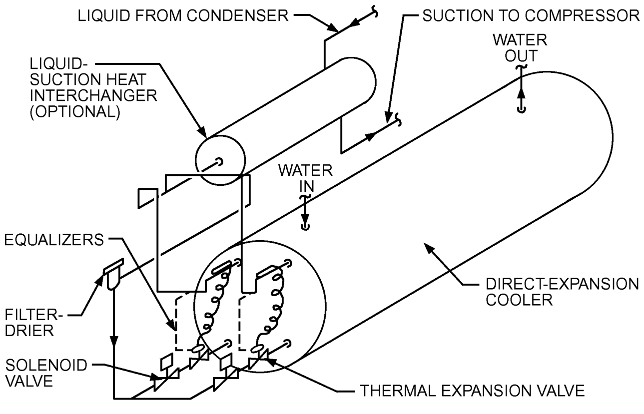


Fig. 19 Two-Circuit Direct-Expansion Cooler Connections (for Single-Compressor System)

Refrigerant Feed Devices

For further information on refrigerant feed devices, see Chapter 44. The pilot-operated low-side float control (Figure 18) is sometimes selected for flooded systems using halocarbon refrigerants. Except for small capacities, direct-acting low-side float valves are impractical for these refrigerants. The displacer float controlling a pneumatic valve works well for low-side liquid level control; it allows the cooler level to be adjusted within the instrument without disturbing the piping.

High-side float valves are practical only in single-evaporator systems, because distribution problems result when multiple evaporators are used.

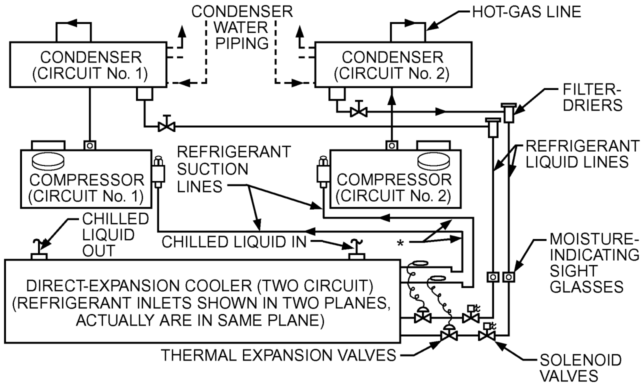
Float chambers should be located as near the liquid connection on the cooler as possible because a long length of liquid line, even if insulated, can pick up room heat and give an artificial liquid level in the float chamber. Equalizer lines to the float chamber must be amply sized to minimize the effect of heat transmission. The float chamber and its equalizing lines must be insulated.

Each flooded cooler system must have a way of keeping oil concentration in the evaporator low, both to minimize the bleedoff needed to keep oil concentration in the cooler low and to reduce system losses from large stills. A highly efficient discharge gas/oil separator can be used for this purpose.

At low temperatures, periodic warm-up of the evaporator allows recovery of oil accumulation in the chiller. If continuous operation is required, dual chillers may be needed to deoil an oil-laden evaporator, or an oil-free compressor may be used.

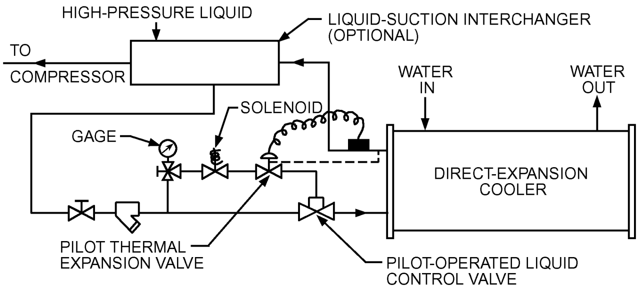
Direct-Expansion Fluid Chillers

For further information on these chillers, see Chapter 38 in the 2004 *ASHRAE Handbook—HVAC Systems and Equipment*. Figure 19 shows typical piping connections for a multicircuit direct-expansion chiller. Each circuit contains its own thermostatic expansion and solenoid valves. One solenoid valve can be wired to close at reduced system capacity. The thermostatic expansion valve bulbs should be located between the cooler and the liquid-suction interchanger, if used. Locating the bulb downstream from the interchanger can cause excessive cycling of the thermostatic expansion valve because the flow of high-pressure liquid through the interchanger ceases when the thermostatic expansion valve closes; consequently, no heat is available from the high-pressure liquid, and the



**Note*: Two liquid-suction heat exchangers can be used, one in each line.

Fig. 20 Typical Refrigerant Piping in Liquid Chilling Package with Two Completely Separate Circuits



Note: Solenoid valve to be located in pilot line to close main liquid control valve on system shutdown.

Fig. 21 Direct-Expansion Cooler with Pilot-Operated Control Valve

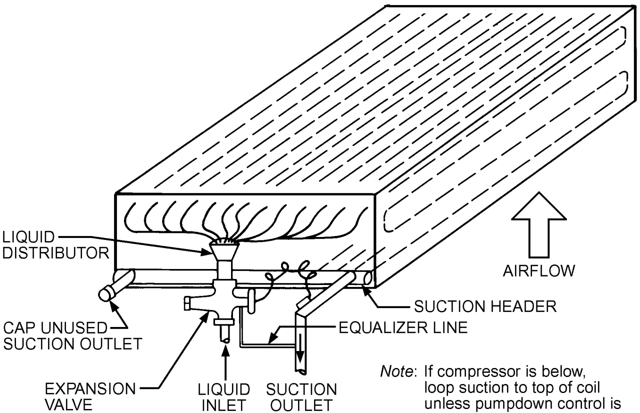
cooler must starve itself to obtain the superheat necessary to open the valve. When the valve does open, excessive superheat causes it to overfeed until the bulb senses liquid downstream from the interchanger. Therefore, the remote bulb should be positioned between the cooler and the interchanger.

Figure 20 shows a typical piping arrangement that has been successful in packaged water chillers having direct-expansion coolers. With this arrangement, automatic recycling pumpdown is needed on the lag compressor to prevent leakage through compressor valves, allowing migration to the cold evaporator circuit. It also prevents liquid from slugging the compressor at start-up.

On larger systems, the limited size of thermostatic expansion valves may require use of a pilot-operated liquid valve controlled by a small thermostatic expansion valve (Figure 21). The small thermostatic expansion valve pilots the main liquid control valve. The equalizing connection and bulb of the pilot thermostatic expansion valve should be treated as a direct-acting thermal expansion valve. A small solenoid valve in the pilot line shuts off the high side from the low during shutdown. However, the main liquid valve does not open and close instantaneously.

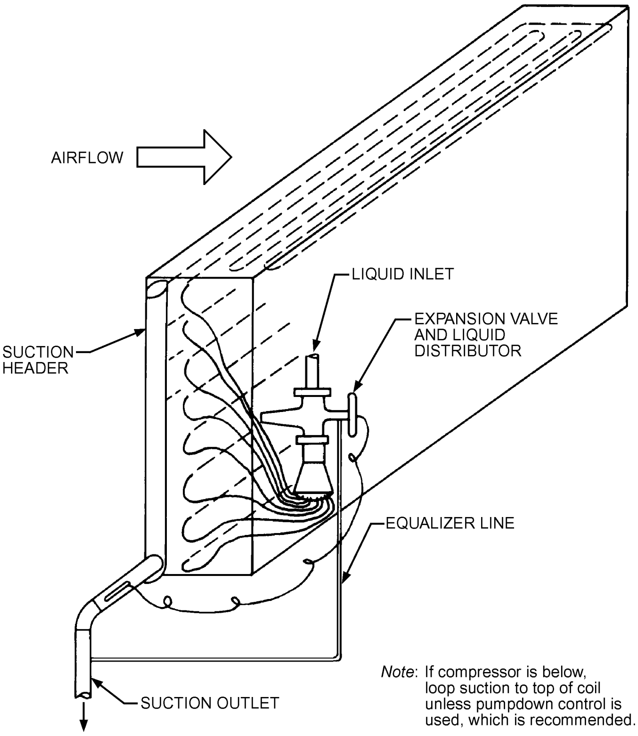
Direct-Expansion Air Coils

For further information on these coils, see Chapter 21 of the 2004 *ASHRAE Handbook—HVAC Systems and Equipment*. The most common ways of arranging direct-expansion coils are shown in Figures 22 and 23. The method shown in Figure 23 provides the superheat needed to operate the thermostatic expansion valve and is effective for heat transfer because leaving air contacts the coldest evaporator surface. This arrangement is advantageous on low-temperature applications, where the coil pressure drop represents an appreciable change in evaporating temperature.



Note: If compressor is below, loop suction to top of coil unless pumpdown control is used, which is recommended.

Fig. 22 Direct-Expansion Evaporator (Top-Feed, Free-Draining)



Note: If compressor is below, loop suction to top of coil unless pumpdown control is used, which is recommended.

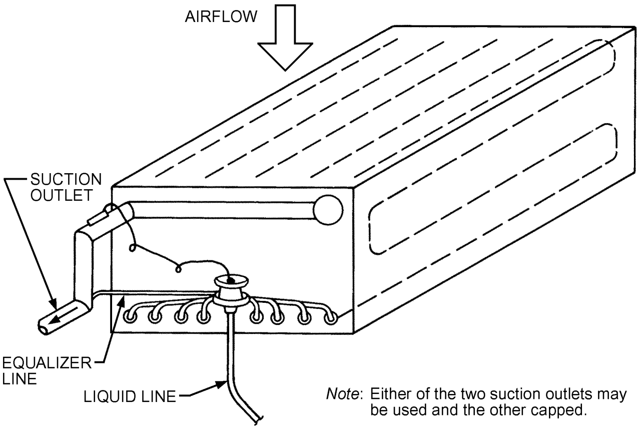
Fig. 23 Direct-Expansion Evaporator (Horizontal Airflow)

Direct-expansion air coils can be located in any position as long as proper refrigerant distribution and continuous oil removal facilities are provided.

Figure 22 shows top-feed, free-draining piping with a vertical up-airflow coil. In Figure 23, which illustrates a horizontal-airflow coil, suction is taken off the bottom header connection, providing free oil draining. Many coils are supplied with connections at each end of the suction header so that a free-draining connection can be used regardless of which side of the coil is up; the other end is then capped.

In Figure 24, a refrigerant upfeed coil is used with a vertical downflow air arrangement. Here, the coil design must provide sufficient gas velocity to entrain oil at lowest loadings and to carry it into the suction line.

Pumpdown compressor control is desirable on all systems using downfeed or upfeed evaporators, to protect the compressor against



Note: Either of the two suction outlets may be used and the other capped.

Fig. 24 Direct-Expansion Evaporator (Bottom-Feed)

a liquid slugback in cases where liquid can accumulate in the suction header and/or the coil on system off cycles. Pumpdown compressor control is described in the section on Keeping Liquid from Crankcase During Off Cycles.

Thermostatic expansion valve operation and application are described in Chapter 44. Thermostatic expansion valves should be sized carefully to avoid undersizing at full load and oversizing at partial load. The refrigerant pressure drops through the system (distributor, coil, condenser, and refrigerant lines, including liquid lifts) must be properly evaluated to determine the correct pressure drop available across the valve on which to base the selection. Variations in condensing pressure greatly affect the pressure available across the valve, and hence its capacity.

Oversized thermostatic expansion valves result in cycling that alternates flooding and starving the coil. This occurs because the valve attempts to throttle at a capacity below its capability, which causes periodic flooding of the liquid back to the compressor and wide temperature variations in the air leaving the coil. Reduced compressor capacity further aggravates this problem. Systems having multiple coils can use solenoid valves located in the liquid line feeding each evaporator or group of evaporators to close them off individually as compressor capacity is reduced.

For information on defrosting, see Chapter 42.

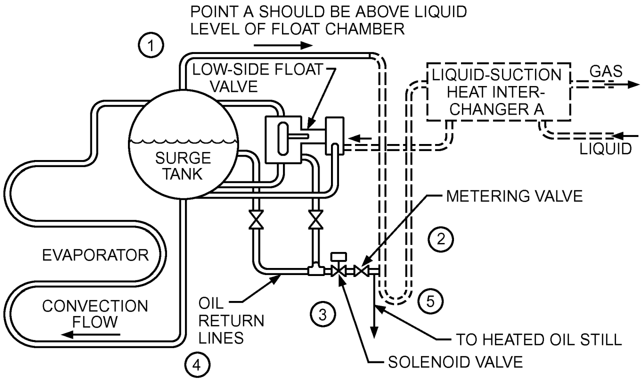
Flooded Evaporators

Flooded evaporators may be desirable when a small temperature differential is required between the refrigerant and the medium being cooled. A small temperature differential is advantageous in low-temperature applications.

In a flooded evaporator, the coil is kept full of refrigerant when cooling is required. The refrigerant level is generally controlled through a high- or low-side float control. Figure 25 represents a typical arrangement showing a low-side float control, oil return line, and heat interchanger.

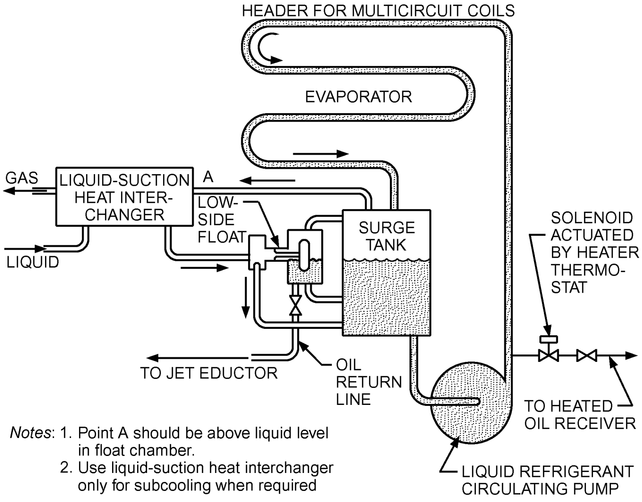
Circulation of refrigerant through the evaporator depends on gravity and a thermosiphon effect. A mixture of liquid refrigerant and vapor returns to the surge tank, and the vapor flows into the suction line. A baffle installed in the surge tank helps prevent foam and liquid from entering the suction line. A liquid refrigerant circulating pump (Figure 26) provides a more positive way of obtaining a high circulation rate.

Taking the suction line off the top of the surge tank causes difficulties if no special provisions are made for oil return. For this reason, the oil return lines in Figure 25 should be installed. These lines are connected near the bottom of the float chamber and also just below the liquid level in the surge tank (where an oil-rich liquid refrigerant exists). They extend to a lower point on the suction line to allow gravity flow. Included in this oil return line is (1) a solenoid valve that is open only while the compressor is running and (2) a metering valve that is adjusted to allow a constant but small-volume return to the suction line. A liquid-line sight glass may be installed downstream from the metering valve to serve as a convenient check on liquid being returned.



Notes: 1. Looped suction line is not needed when oil bleed goes to still. Heat exchanger is optional.
2. Trap allows gravity return of oil-rich liquid refrigerant to suction line ahead of interchanger.
3. Solenoid valve normally closed except when compressor is operating and heater is off.
4. Do not use pilot-operated solenoid valve.
5. To heated oil still is a better arrangement.

Fig. 25 Flooded Evaporator (Gravity Circulation)



Notes: 1. Point A should be above liquid level in float chamber.
2. Use liquid-suction heat interchanger only for subcooling when required

Fig. 26 Flooded Evaporator (Forced Circulation)

Oil can be returned satisfactorily by taking a bleed of refrigerant and oil from the pump discharge (Figure 26) and feeding it to the heated oil receiver. If a low-side float is used, a jet ejector can be used to remove oil from the quiescent float chamber.

REFRIGERATION ACCESSORIES

Liquid-Suction Heat Exchangers

Generally, liquid-suction heat exchangers subcool liquid refrigerant and superheat suction gas. They are used for one or more of the following functions:

- *Increasing efficiency of the refrigeration cycle.* Efficiency of the thermodynamic cycle of certain halocarbon refrigerants can be increased when the suction gas is superheated by removing heat from the liquid. This increased efficiency must be evaluated against the effect of pressure drop through the suction side of the exchanger, which forces the compressor to operate at a lower suction pressure. Liquid-suction heat exchangers are most beneficial at low suction temperatures. The increase in cycle efficiency for systems operating in the air-conditioning range (down to about 30°F evaporating temperature) usually does not justify their use. The heat exchanger can be located wherever convenient.

- *Subcooling liquid refrigerant to prevent flash gas at the expansion valve.* The heat exchanger should be located near the condenser or receiver to achieve subcooling before pressure drop occurs.
- *Evaporating small amounts of expected liquid refrigerant returning from evaporators in certain applications.* Many heat pumps incorporating reversals of the refrigerant cycle include a suction-line accumulator and liquid-suction heat exchanger arrangement to trap liquid floodbacks and vaporize them slowly between cycle reversals.

If an evaporator design makes a deliberate slight overfeed of refrigerant necessary, either to improve evaporator performance or to return oil out of the evaporator, a liquid-suction heat exchanger is needed to evaporate the refrigerant.

A flooded water cooler usually incorporates an oil-rich liquid bleed from the shell into the suction line for returning oil. The liquid-suction heat exchanger boils liquid refrigerant out of the mixture in the suction line. Exchangers used for this purpose should be placed in a horizontal run near the evaporator. Several types of liquid-suction heat exchangers are used.

Liquid and Suction Line Soldered Together. The simplest form of heat exchanger is obtained by strapping or soldering the suction and liquid lines together to obtain counterflow and then insulating the lines as a unit. To maximize capacity, the liquid line should always be on the bottom of the suction line, because liquid in a suction line runs along the bottom (Figure 27). This arrangement is limited by the amount of suction line available.

Shell-and-Coil or Shell-and-Tube Heat Exchangers (Figure 28)**.** These units are usually installed so that the suction outlet drains the shell. When the units are used to evaporate liquid refrigerant returning in the suction line, the free-draining arrangement is not recommended. Liquid refrigerant can run along the bottom of the heat exchanger shell, having little contact with the warm liquid coil, and drain into the compressor. By installing the heat exchanger at a slight angle to the horizontal (Figure 29) with gas entering at the bottom and leaving at the top, any liquid returning in the line is trapped in the shell and held in contact with the warm liquid coil, where most of it is vaporized. An oil return line, with a metering valve and solenoid valve (open only when the compressor is running), is required to return oil that collects in the trapped shell.

Concentric Tube-in-Tube Heat Exchangers. The tube-in-tube heat exchanger is not as efficient as the shell-and-finned-coil type. It is, however, quite suitable for cleaning up small amounts of excessive liquid refrigerant returning in the suction line. Figure 30 shows typical construction with available pipe and fittings.

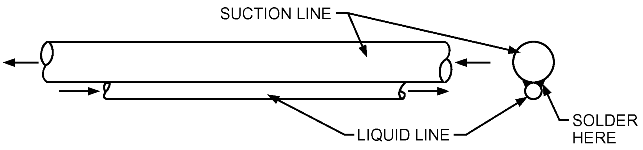


Fig. 27 Soldered Tube Heat Exchanger

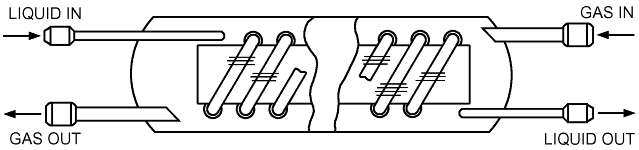


Fig. 28 Shell-and-Finned-Coil Heat Exchanger

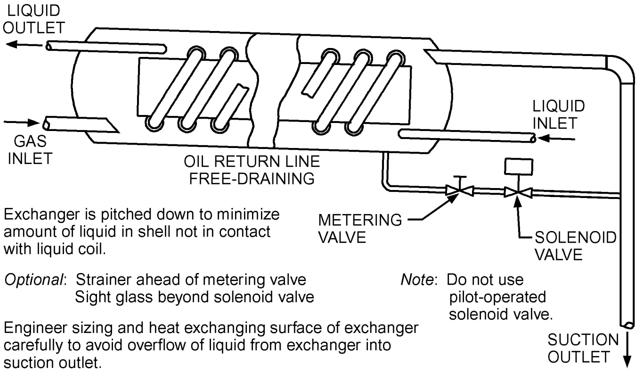
Plate Heat Exchangers. Plate heat exchangers provide high-efficiency heat transfer. They are very compact, have low pressure drop, and are lightweight devices. They are good for use as liquid subcoolers.

For air-conditioning applications, heat exchangers are recommended for liquid subcooling or for clearing up excess liquid in the suction line. For refrigeration applications, heat exchangers are recommended to increase cycle efficiency, as well as for liquid subcooling and removing small amounts of excess liquid in the suction line. Excessive superheating of the suction gas should be avoided.

Two-Stage Subcoolers

To take full advantage of the two-stage system, the refrigerant liquid should be cooled to near the interstage temperature to reduce the amount of flash gas handled by the low-stage compressor. The net result is a reduction in total system power requirements. The amount of gain from cooling to near interstage conditions varies among refrigerants.

Figure 31 illustrates an open or flash-type cooler. This is the simplest and least costly type, which has the advantage of cooling liquid to the saturation temperature of the interstage pressure. One disadvantage is that the pressure of cooled liquid is reduced to interstage pressure, leaving less pressure available for liquid transport. Although the liquid temperature is reduced, the pressure drops correspondingly, and the expansion device controlling flow to the cooler must be large enough to pass all the liquid refrigerant flow. Failure of this valve could allow a large flow of liquid to the upper-stage compressor suction, which could seriously damage the compressor.



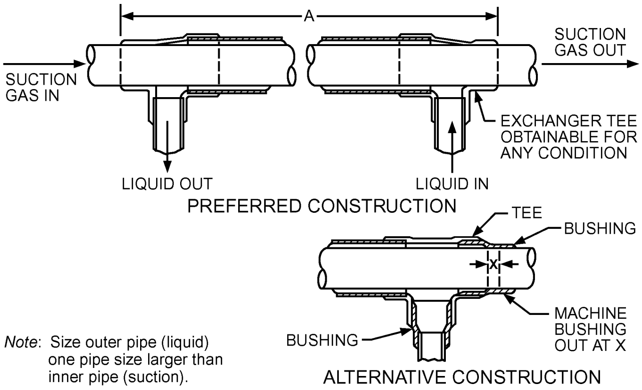
Exchanger is pitched down to minimize amount of liquid in shell not in contact with liquid coil.

Optional: Strainer ahead of metering valve
Sight glass beyond solenoid valve

Note: Do not use pilot-operated solenoid valve.

Engineer sizing and heat exchanging surface of exchanger carefully to avoid overflow of liquid from exchanger into suction outlet.

Fig. 29 Shell-and-Finned-Coil Exchanger Installed to Prevent Liquid Floodback



Note: Size outer pipe (liquid) one pipe size larger than inner pipe (suction).

Fig. 30 Tube-in-Tube Heat Exchanger

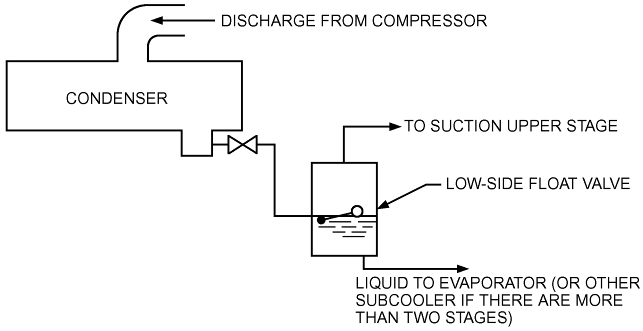


Fig. 31 Flash-Type Cooler

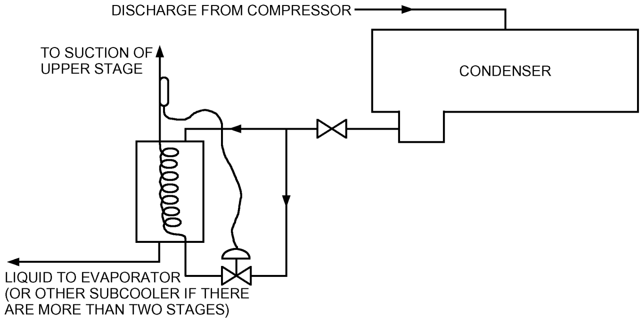


Fig. 32 Closed-Type Subcooler

Liquid from a flash cooler is saturated, and liquid from a cascade condenser usually has little subcooling. In both cases, the liquid temperature is usually lower than the temperature of the surroundings. Thus, it is important to avoid heat input and pressure losses that would cause flash gas to form in the liquid line to the expansion device or to recirculating pumps. Cold liquid lines should be insulated, because expansion devices are usually designed to feed liquid, not vapor.

Figure 32 shows the closed or heat exchanger type of subcooler. It should have sufficient heat transfer surface to transfer heat from the liquid to the evaporating refrigerant with a small final temperature difference. Pressure drop should be small, so that full pressure is available for feeding liquid to the expansion device at the low-temperature evaporator. The subcooler liquid control valve should be sized to supply only the quantity of refrigerant required for the subcooling. This prevents a tremendous quantity of liquid from flowing to the upper-stage suction in the event of a valve failure.

Discharge Line Oil Separators

Oil is always in circulation in systems using halocarbon refrigerants. Refrigerant piping is designed to ensure that this oil passes through the entire system and returns to the compressor as fast as it leaves. Although well-designed piping systems can handle the oil in most cases, a discharge-line oil separator can have certain advantages in some applications (see Chapter 44), such as

- In systems where it is impossible to prevent substantial absorption of refrigerant in the crankcase oil during shutdown periods. When the compressor starts up with a violent foaming action, oil is thrown out at an accelerated rate, and the separator immediately returns a large portion of this oil to the crankcase. Normally, the system should be designed with pumpdown control or crankcase heaters to minimize liquid absorption in the crankcase.
- In systems using flooded evaporators, where refrigerant bleedoff is necessary to remove oil from the evaporator. Oil separators reduce the amount of bleedoff from the flooded cooler needed for operation.
- In direct-expansion systems using coils or tube bundles that require bottom feed for good liquid distribution and where refrigerant carryover from the top of the evaporator is essential for proper oil removal.
- In low-temperature systems, where it is advantageous to have as little oil as possible going through the low side.
- In screw-type compressor systems, where an oil separator is necessary for proper operation. The oil separator is usually supplied with the compressor unit assembly directly from the compressor manufacturer.
- In multiple compressors operating in parallel. The oil separator can be an integral part of the total system oil management system.

In applying oil separators in refrigeration systems, the following potential hazards must be considered:

- Oil separators are not 100% efficient, and they do not eliminate the need to design the complete system for oil return to the compressor.
- Oil separators tend to condense out liquid refrigerant during compressor off cycles and on compressor start-up. This is true if the condenser is in a warm location, such as on a roof. During the off cycle, the oil separator cools down and acts as a condenser for refrigerant that evaporates in warmer parts of the system. A cool oil separator may condense discharge gas and, on compressor start-up, automatically drain it into the compressor crankcase. To minimize this possibility, the drain connection from the oil separator can be connected into the suction line. This line should be equipped with a shutoff valve, a fine filter, hand throttling and solenoid valves, and a sight glass. The throttling valve should be adjusted so that flow through this line is only a little greater than would normally be expected to return oil through the suction line.
- The float valve is a mechanical device that may stick open or closed. If it sticks open, hot gas will be continuously bypassed to the compressor crankcase. If the valve sticks closed, no oil is returned to the compressor. To minimize this problem, the separator can be supplied without an internal float valve. A separate external float trap can then be located in the oil drain line from the separator preceded by a filter. Shutoff valves should isolate the filter and trap. The filter and traps are also easy to service without stopping the system.

The discharge line pipe size into and out of the oil separator should be the full size determined for the discharge line. For separators that have internal oil float mechanisms, allow enough room to remove the oil float assembly for servicing.

Depending on system design, the oil return line from the separator may feed to one of the following locations:

- Directly to the compressor crankcase
- Directly into the suction line ahead of the compressor
- Into an oil reservoir or device used to collect oil, used for a specifically designed oil management system

When a solenoid valve is used in the oil return line, the valve should be wired so that it is open when the compressor is running. To minimize entrance of condensed refrigerant from the low side, a thermostat may be installed and wired to control the solenoid in the oil return line from the separator. The thermostat sensing element should be located on the oil separator shell below the oil level and set high enough so that the solenoid valve will not open until the separator temperature is higher than the condensing temperature. A superheat-controlled expansion valve can perform the same function. If a discharge line check valve is used, it should be downstream of the oil separator.

Surge Drums or Accumulators

A surge drum is required on the suction side of almost all flooded evaporators to prevent liquid slopover to the compressor. Exceptions include shell-and-tube coolers and similar shell-type evaporators, which provide ample surge space above the liquid level or contain eliminators to separate gas and liquid. A horizontal surge drum is sometimes used where headroom is limited.

The drum can be designed with baffles or eliminators to separate liquid from the suction gas. More often, sufficient separation space is allowed above the liquid level for this purpose. Usually, the design is vertical, with a separation height above the liquid level of 24 to 30 in. and with the shell diameter sized to keep suction gas velocity low enough to allow liquid droplets to separate. Because these vessels are also oil traps, it is necessary to provide oil bleed.

Although separators may be fabricated with length-to-diameter (*L*/*D*) ratios of 1/1 up to 10/1, the lowest-cost separators are usually for *L*/*D* ratios between 3/1 and 5/1.

Compressor Floodback Protection

Certain systems periodically flood the compressor with excessive amounts of liquid refrigerant. When periodic floodback through the suction line cannot be controlled, the compressor must be protected against it.

The most satisfactory method appears to be a trap arrangement that catches liquid floodback and (1) meters it slowly into the suction line, where the floodback is cleared up with a liquid-suction heat interchanger; (2) evaporates the liquid 100% in the trap itself by using a liquid coil or electric heater, and then automatically returns oil to the suction line; or (3) returns it to the receiver or to one of the evaporators. Figure 29 illustrates an arrangement that handles moderate liquid floodback, disposing of liquid by a combination of boiling off in the exchanger and limited bleedoff into the suction line. This device, however, does not have sufficient trapping volume for most heat pump applications or hot-gas defrost systems using reversal of the refrigerant cycle.

For heavier floodback, a larger volume is required in the trap. The arrangement shown in Figure 33 has been applied successfully in reverse-cycle heat pump applications using halocarbon refrigerants. It consists of a suction-line accumulator with enough volume to hold the maximum expected floodback and a large enough diameter to separate liquid from suction gas. Trapped liquid is slowly bled off through a properly sized and controlled drain line into the suction line, where it is boiled off in a liquid-suction heat exchanger between cycle reversals.

With the alternative arrangement shown, the liquid/oil mixture is heated to evaporate the refrigerant, and the remaining oil is drained into the crankcase or suction line.

Refrigerant Driers and Moisture Indicators

The effect of moisture in refrigeration systems is discussed in Chapters 5 and 6. Using a permanent refrigerant drier is recommended on all systems and with all refrigerants. It is especially important on low-temperature systems to prevent ice from forming at expansion devices. A **full-flow drier** is always recommended in hermetic compressor systems to keep the system dry and prevent decomposition products from getting into the evaporator in the event of a motor burnout.

Replaceable-element filter-driers are preferred for large systems because the drying element can be replaced without breaking any refrigerant connections. The drier is usually located in the liquid line near the liquid receiver. It may be mounted horizontally or vertically with the flange at the bottom, but it should never be mounted vertically with the flange on top because any loose material would then fall into the line when the drying element was removed.

A three-valve bypass is usually used, as shown in Figure 34, to provide a way to isolate the drier for servicing. The refrigerant charging connection should be located between the receiver outlet valve and liquid-line drier so that all refrigerant added to the system passes through the drier.

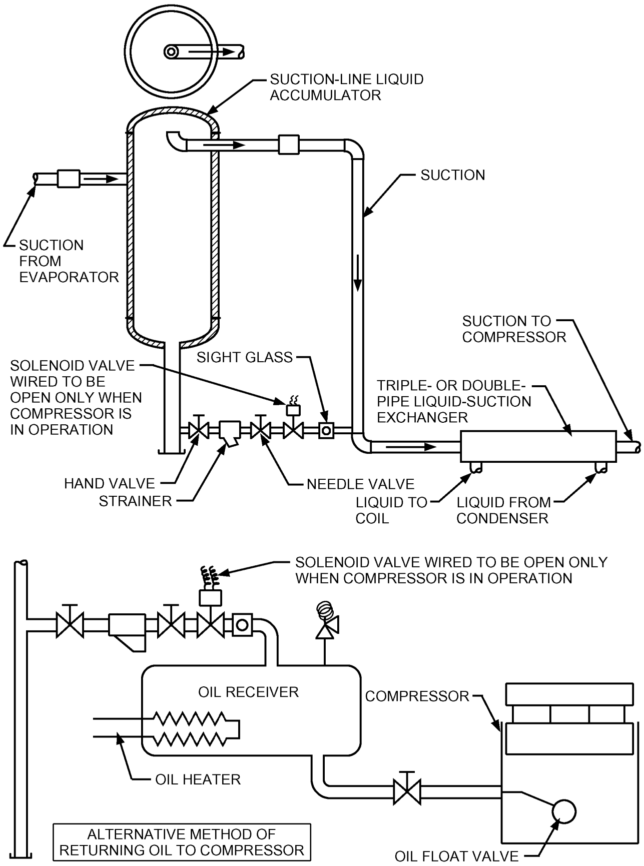
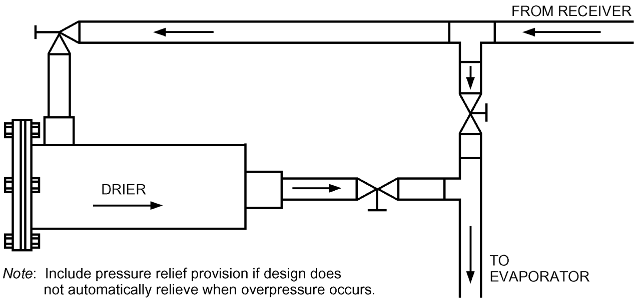


Fig. 33 Compressor Floodback Protection Using Accumulator with Controlled Bleed



Note: Include pressure relief provision if design does not automatically relieve when overpressure occurs.

Fig. 34 Drier with Piping Connections

Reliable moisture indicators can be installed in refrigerant liquid lines to provide a positive indication of when the drier cartridge should be replaced.

Strainers

Strainers should be used in both liquid and suction lines to protect automatic valves and the compressor from foreign material, such as pipe welding scale, rust, and metal chips. The strainer should be mounted in a horizontal line, oriented so that the screen can be replaced without loose particles falling into the system.

A liquid-line strainer should be installed before each automatic valve to prevent particles from lodging on the valve seats. Where multiple expansion valves with internal strainers are used at one location, a single main liquid-line strainer will protect all of these. The liquid-line strainer can be located anywhere in the line between the condenser (or receiver) and the automatic valves, preferably near the valves for maximum protection. Strainers should trap the particle size that could affect valve operation. With pilot-operated valves, a very fine strainer should be installed in the pilot line ahead of the valve.

Filter-driers dry the refrigerant and filter out particles far smaller than those trapped by mesh strainers. No other strainer is needed in the liquid line if a good filter-drier is used.

Refrigeration compressors are usually equipped with a built-in suction strainer, which is adequate for the usual system with copper piping. The suction line should be piped at the compressor so that the built-in strainer is accessible for servicing.

Both liquid- and suction-line strainers should be adequately sized to ensure sufficient foreign material storage capacity without excessive pressure drop. In steel piping systems, an external suction-line strainer is recommended in addition to the compressor strainer.

Liquid Indicators

Every refrigeration system should have a way to check for sufficient refrigerant charge. Common devices used are liquid-line sight glass, mechanical or electronic indicators, and an external gage glass with equalizing connections and shutoff valves. A properly installed sight glass shows bubbling when the charge is insufficient.

Liquid indicators should be located in the liquid line as close as possible to the receiver outlet, or to the condenser outlet if no receiver is used (Figure 35). The sight glass is best installed in a vertical section of line, far enough downstream from any valve that the resulting disturbance does not appear in the glass. If the sight glass is installed too far away from the receiver, the line pressure drop may be sufficient to cause flashing and bubbles in the glass, even if the charge is sufficient for a liquid seal at the receiver outlet.

When sight glasses are installed near the evaporator, often no amount of system overcharging will give a solid liquid condition at the sight glass because of pressure drop in the liquid line or lift. Subcooling is required here. An additional sight glass near the evaporator may be needed to check the refrigerant condition at that point.

Sight glasses should be installed full size in the main liquid line. In very large liquid lines, this may not be possible; the glass can then be installed in a bypass or saddle mount that is arranged so that any gas in the liquid line will tend to move to it. A sight glass with double ports (for back lighting) and seal caps, which provide added protection against leakage, is preferred. Moisture-liquid indicators large enough to be installed directly in the liquid line serve the dual purpose of liquid-line sight glass and moisture indicator.

Oil Receivers

Oil receivers serve as reservoirs for replenishing crankcase oil pumped by the compressors and provide the means to remove refrigerant dissolved in the oil. They are selected for systems having any of the following components:

- Flooded or semiflooded evaporators with large refrigerant charges
- Two or more compressors operated in parallel
- Long suction and discharge lines
- Double suction line risers

A typical hookup is shown in Figure 33. Outlets are arranged to prevent oil from draining below the heater level to avoid heater burnout and to prevent scale and dirt from being returned to the compressor.

Purge Units

Noncondensable gas separation using a purge unit is useful on most large refrigeration systems where suction pressure may fall below atmospheric pressure (see Figure 11 of Chapter 3).

HEAD PRESSURE CONTROL FOR REFRIGERANT CONDENSERS

For more information on head pressure control, see Chapter 35 of the 2004 *ASHRAE Handbook—HVAC Systems and Equipment*.

Water-Cooled Condensers

With water-cooled condensers, head pressure controls are used both to maintain condensing pressure and to conserve water. On cooling tower applications, they are used only where it is necessary to maintain condensing temperatures.

Condenser-Water-Regulating Valves

The shutoff pressure of the valve must be set slightly higher than the saturation pressure of the refrigerant at the highest ambient temperature expected when the system is not in operation. This ensures that the valve will not pass water during off cycles. These valves are usually sized to pass the design quantity of water at about a 25 to 30 psi difference between design condensing pressure and valve shutoff pressure. Chapter 44 has further information.

Water Bypass

In cooling tower applications, a simple bypass with a manual or automatic valve responsive to head pressure change can also be used to maintain condensing pressure. Figure 36 shows an automatic three-way valve arrangement. The valve divides water flow between the condenser and the bypass line to maintain the desired condensing pressure. This maintains a balanced flow of water on the tower and pump.

Evaporative Condensers

Among the methods used for condensing pressure control with evaporative condensers are (1) cycling the spray pump motor; (2) cycling both fan and spray pump motors; (3) throttling the spray water; (4) bypassing air around duct and dampers; (5) throttling air

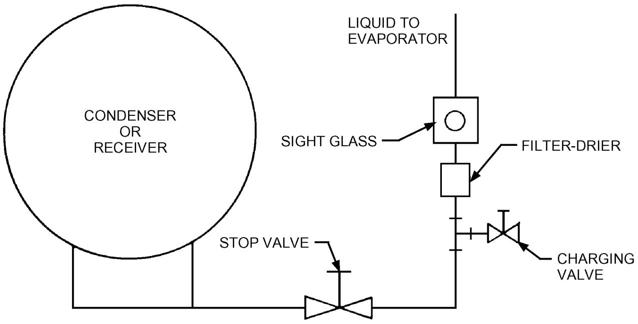
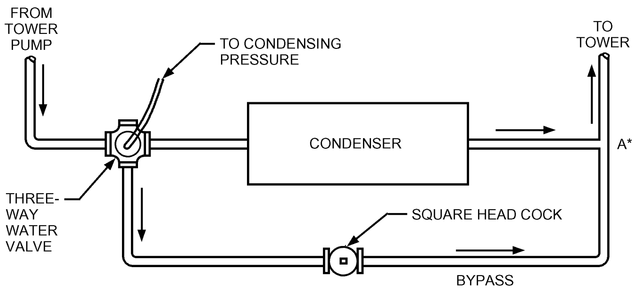


Fig. 35 Sight Glass and Charging Valve Locations



*A is alternative location for three-way valve, depending on valve design. Check manufacturer's recommendations.

Fig. 36 Head Pressure Control for Condensers Used with Cooling Towers (Water Bypass Modulation)

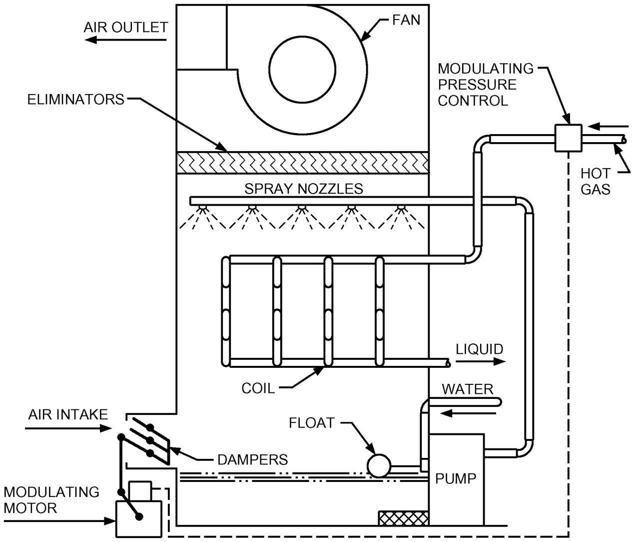


Fig. 37 Head Pressure Control for Evaporative Condenser (Air Intake Modulation)

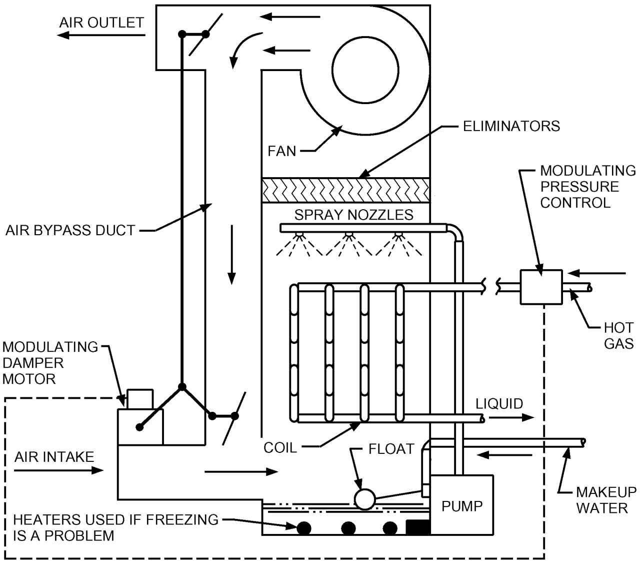


Fig. 38 Head Pressure Control for Evaporative Condenser (Air Bypass Modulation)

via dampers, on either inlet or discharge; and (6) combinations of these methods. For further information, see Chapter 35 of the 2004 *ASHRAE Handbook—HVAC Systems and Equipment.*

In water pump cycling, a pressure control at the gas inlet starts and stops the pump in response to head pressure changes. The pump sprays water over the condenser coils. As head pressure drops, the pump stops and the unit becomes an air-cooled condenser.

Constant pressure is difficult to maintain with coils of prime surface tubing because as soon as the pump stops, the pressure goes up and the pump starts again. This occurs because these coils have insufficient capacity when operating as an air-cooled condenser. The problem is not as acute with extended-surface coils. Short-cycling results in excessive deposits of mineral and scale on the tubes, decreasing the life of the water pump.

One method of controlling head pressure is using cycle fans and pumps. This minimizes water-side scaling. In colder climates, an indoor water sump with a remote spray pump(s) is required. The fan cycling sequence is as follows:

Upon dropping head pressure

- Stop fans.
- If pressure continues to fall, stop pumps.

Upon rising head pressure

- Start fans.
- If pressure continues to rise, start pumps.

Damper control (Figure 37) may be incorporated in systems requiring more constant head pressures (e.g., some systems using thermostatic expansion valves). One drawback of dampers is formation of ice on dampers and linkages.

Figure 38 incorporates an air bypass arrangement for controlling head pressure. A modulating motor, acting in response to a modulating pressure control, positions dampers so that the mixture of recirculated and cold inlet air maintains the desired pressure. In extremely cold weather, most of the air is recirculated.

Air-Cooled Condensers

Methods for condensing pressure control with air-cooled condensers include (1) cycling fan motor, (2) air throttling or bypassing, (3) coil flooding, and (4) fan motor speed control. The first two methods are described in the section on Evaporative Condensers.

The third method holds condensing pressure up by backing liquid refrigerant up in the coil to cut down on effective condensing surface. When head pressure drops below the setting of the modulating control valve, it opens, allowing discharge gas to enter the liquid drain line. This restricts liquid refrigerant drainage and causes the condenser to flood enough to maintain the condenser and receiver pressure at the control valve setting. A pressure difference must be available across the valve to open it. Although the condenser would impose sufficient pressure drop at full load, pressure drop may practically disappear at partial loading. Therefore, a positive restriction must be placed parallel with the condenser and the control valve. Systems using this type of control require extra refrigerant charge.

In multiple-fan air-cooled condensers, it is common to cycle fans off down to one fan and then to apply air throttling to that section or modulate the fan motor speed. Consult the manufacturer before using this method, because not all condensers are properly circuited for it.

Using ambient temperature change (rather than condensing pressure) to modulate air-cooled condenser capacity prevents rapid cycling of condenser capacity. A disadvantage of this method is that the condensing pressure is not closely controlled.

KEEPING LIQUID FROM CRANKCASE DURING OFF CYCLES

Control of reciprocating compressors should prevent excessive accumulation of liquid refrigerant in the crankcase during off cycles. Any one of the following control methods accomplishes this.

Automatic Pumpdown Control (Direct-Expansion Air-Cooling Systems)

The most effective way to keep liquid out of the crankcase during system shutdown is to operate the compressor on automatic pumpdown control. The recommended arrangement involves the following devices and provisions:

- A liquid-line solenoid valve in the main liquid line or in the branch to each evaporator.
- Compressor operation through a low-pressure cutout providing for pumpdown whenever this device closes, regardless of whether the balance of the system is operating.

- Electrical interlock of the liquid solenoid valve with the evaporator fan, so refrigerant flow stops when the fan is out of operation.
- Electrical interlock of refrigerant solenoid valve with safety devices (e.g., high-pressure cutout, oil safety switch, and motor overloads), so that the refrigerant solenoid valve closes when the compressor stops.
- Low-pressure control settings such that the cut-in point corresponds to a saturated refrigerant temperature lower than any expected compressor ambient air temperature. If the cut-in setting is any higher, liquid refrigerant can accumulate and condense in the crankcase at a pressure corresponding to the ambient temperature. Then, the crankcase pressure would not rise high enough to reach the cut-in point, and effective automatic pumpdown would not be obtained.

Crankcase Oil Heater (Direct-Expansion Systems)

A crankcase oil heater with or without single (nonrecycling) pumpout at the end of each operating cycle does not keep liquid refrigerant out of the crankcase as effectively as automatic pumpdown control, but many compressors equalize too quickly after stopping automatic pumpdown control. Crankcase oil heaters maintain the crankcase oil at a temperature higher than that of other parts of the system, minimizing the absorption of the refrigerant by the oil.

Operation with the single pumpout arrangement is as follows. Whenever the temperature control device opens the circuit, or the manual control switch is opened for shutdown purposes, the crankcase heater is energized, and the compressor keeps running until it cuts off on the low-pressure switch. Because the crankcase heater remains energized during the complete off cycle, it is important that a continuous live circuit be available to the heater during the off time. The compressor cannot start again until the temperature control device or manual control switch closes, regardless of the position of the low-pressure switch.

This control method requires

- A liquid-line solenoid valve in the main liquid line or in the branch to each evaporator
- Use of a relay or the maintained contact of the compressor motor auxiliary switch to obtain a single pumpout operation before stopping the compressor
- A relay or auxiliary starter contact to energize the crankcase heater during the compressor off cycle and deenergize it during the compressor on cycle
- Electrical interlock of the refrigerant solenoid valve with the evaporator fan, so that refrigerant flow is stopped when the fan is out of operation
- Electrical interlock of refrigerant solenoid valve with safety devices (e.g., high-pressure cutout, oil safety switch, and motor overloads), so that the refrigerant flow valve closes when the compressor stops

Control for Direct-Expansion Water Chillers

Automatic pumpdown control is undesirable for direct-expansion water chillers because freezing is possible if excessive cycling occurs. A crankcase heater is the best solution, with a solenoid valve in the liquid line that closes when the compressor stops.

Effect of Short Operating Cycle

With reciprocating compressors, oil leaves the crankcase at an accelerated rate immediately after starting. Therefore, each start should be followed by a long enough operating period to allow the oil level to recover. Controllers used for compressors should not produce short-cycling of the compressor. Refer to the compressor manufacturer’s literature for guidelines on maximum or minimum cycles for a specified period.

HOT-GAS BYPASS ARRANGEMENTS

Most large reciprocating compressors are equipped with unloaders that allow the compressor to start with most of its cylinders unloaded. However, it may be necessary to further unload the compressor to (1) reduce starting torque requirements so that the compressor can be started both with low-starting-torque prime movers and on low-current taps of reduced voltage starters and (2) allow capacity control down to 0% load conditions without stopping the compressor.

Full (100%) Unloading for Starting

Starting the compressor without load can be done with a manual or automatic valve in a bypass line between the hot-gas and suction lines at the compressor.

To prevent overheating, this valve is open only during the starting period and closed after the compressor is up to full speed and full voltage is applied to the motor terminals.

In the control sequence, the unloading bypass valve is energized on demand of the control calling for compressor operation, equalizing pressures across the compressor. After an adequate delay, a timing relay closes a pair of normally open contacts to start the compressor. After a further time delay, a pair of normally closed timing relay contacts opens, deenergizing the bypass valve.

Full (100%) Unloading for Capacity Control

Where full unloading is required for capacity control, hot-gas bypass arrangements can be used in ways that will not overheat the compressor. In using these arrangements, hot gas should not be bypassed until after the last unloading step.

Hot-gas bypass should (1) give acceptable regulation throughout the range of loads, (2) not cause excessive superheating of the suction gas, (3) not cause any refrigerant overfeed to the compressor, and (4) maintain an oil return to the compressor.

Hot-gas bypass for capacity control is an artificial loading device that maintains a minimum evaporating pressure during continuous compressor operation, regardless of evaporator load. This is usually done by an automatic or manual pressure-reducing valve that establishes a constant pressure on the downstream side.

Four common methods of using hot-gas bypass are shown in Figure 39. Figure 39A illustrates the simplest type; it will dangerously overheat the compressor if used for protracted periods of time. Figure 39B shows the use of hot-gas bypass to the exit of the evaporator. The expansion valve bulb should be placed at least 5 ft downstream from the bypass point of entrance, and preferably further, to ensure good mixing.

In Figure 39D, the hot-gas bypass enters after the evaporator thermostatic expansion valve bulb. Another thermostatic expansion valve supplies liquid directly to the bypass line for desuperheating. It is always important to install the hot-gas bypass far enough back in the system to maintain sufficient gas velocities in suction risers and other components to ensure oil return at any evaporator loading.

Figure 39C shows the most satisfactory hot-gas bypass arrangement. Here, the bypass is connected into the low side between the expansion valve and entrance to the evaporator. If a distributor is used, gas enters between the expansion valve and distributor. Refrigerant distributors are commercially available with side inlet connections that can be used for hot-gas bypass duty to a certain extent. Pressure drop through the distributor tubes must be evaluated to determine how much gas can be bypassed. This arrangement provides good oil return.

Solenoid valves should be placed before the constant-pressure bypass valve and before the thermal expansion valve used for liquid injection desuperheating, so that these devices cannot function until they are required.

Control valves for hot gas should be close to the main discharge line because the line preceding the valve usually fills with liquid when closed.

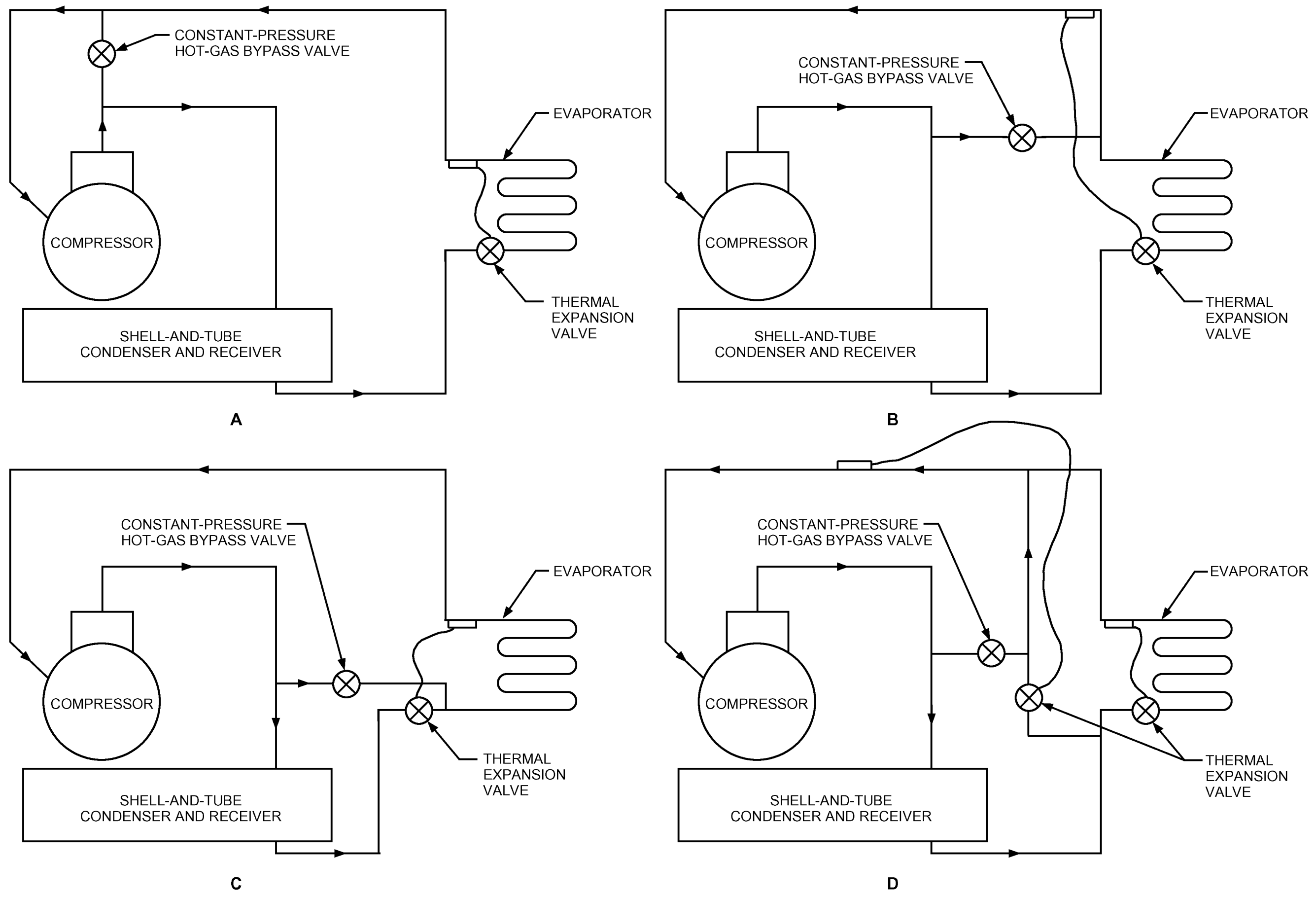


Fig. 39 Hot-Gas Bypass Arrangements

The hot-gas bypass line should be sized so that its pressure loss is only a small percentage of the pressure drop across the valve. Usually, it is the same size as the valve connections. When sizing the valve, consult a control valve manufacturer to determine the minimum compressor capacity that must be offset, refrigerant used, condensing pressure, and suction pressure.

When unloading (Figure 39C), head pressure control requirements increase considerably because the only heat delivered to the condenser is that caused by the motor power delivered to the compressor. Discharge pressure should be kept high enough that the hot-gas bypass valve can deliver gas at the required rate. The condenser head pressure control must be capable of meeting this condition.

Safety Requirements

ASHRAE *Standard* 15 and ASME *Standard* B31.5 should be used as guides for safe practice because they are the basis of most municipal and state codes. However, some ordinances require heavier piping and other features. The designer should know the specific requirements of the installation site. Only A106 Grade A or B or A53 Grade A or B should be considered for steel refrigerant piping.

The designer should know that the rated internal working pressure for Type L copper tubing decreases with (1) increasing metal operating temperature, (2) increasing tubing size (OD), and (3) increasing temperature of joining method. Hot methods used to join drawn pipe (e.g., brazing or welding) produce joints as strong as surrounding pipe, but reduce the strength of the heated pipe material to that of annealed material. Particular attention should be paid when specifying use of copper in conjunction with newer, high-pressure refrigerants (e.g., R-404a, R-507, R-410a, R-407c) because some of these refrigerants can achieve operating pressures as high as 500 psia and operating temperatures as high as 300°F at a typical saturated condensing condition of 130°F.

REFERENCES

Alofs, D.J., M.M. Hasan, and H.J. Sauer, Jr. 1990. Influence of oil on pressure drop in refrigerant compressor suction lines. *ASHRAE Transactions* 96:1.

ASHRAE. 2004. Safety standard for refrigeration systems. ANSI/ASHRAE *Standard* 15-2004.

ASME. 2001. Refrigeration piping and heat transfer components. ANSI/ASME *Standard* B31.5-2001. American Society of Mechanical Engineers, New York.

Atwood, T. 1990. Pipe sizing and pressure drop calculations for HFC-134a. *ASHRAE Journal* 32(4):62-66.

Colebrook, D.F. 1938, 1939. Turbulent flow in pipes. *Journal of the Institute of Engineers* 11.

Cooper, W.D. 1971. Influence of oil-refrigerant relationships on oil return. *ASHRAE Symposium Bulletin* PH71(2):6-10.

Jacobs, M.L., F.C. Scheideman, F.C. Kazem, and N.A. Macken. 1976. Oil transport by refrigerant vapor. *ASHRAE Transactions* 81(2):318-329.

Keating, E.L. and R.A. Matula. 1969. Correlation and prediction of viscosity and thermal conductivity of vapor refrigerants. *ASHRAE Transactions* 75(1).

Stoecker, W.F. 1984. Selecting the size of pipes carrying hot gas to defrosted evaporators. *International Journal of Refrigeration* 7(4):225-228.

Timm, M.L. 1991. An improved method for calculating refrigerant line pressure drops. *ASHRAE Transactions* 97(1):194-203.

Wile, D.D. 1977. *Refrigerant line sizing*. ASHRAE.

Related Commercial Resources