CHAPTER C9
REFRIGERATION SYSTEMS PIPING

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The broad term *refrigeration* refers to a general science concerned with the use of producing temperatures below normal for commercial or other useful purposes. Refrigeration piping is used in conjunction with refrigeration equipment. Refrigerants are fluids which absorb heat by evaporating at a lower temperature and pressure and transfer heat out when they condense at a higher pressure and temperature. The increase in pressure necessary to elevate the temperature level is produced by a compressor of the reciprocating, rotary, or centrifugal type. In the case of an absorption system, the transfer of heat, thereby boiling the more volatile refrigerant out of a solution.

Fluids which do not change state are sometimes used to transfer heat in an indirect system. Such fluids are called *secondary coolants*. To be classified as a secondary coolant, the fluid must be used for the transfer of heat without a change in its state. Brine, a solution of salt and water, is a secondary coolant.

Many fluids have been used as volatile refrigerants in the evaporation, compression, condensing, and expansion cycle. This chapter will deal with application and structural design of piping for the more commonly used volatile refrigerants such as ammonia and some of the halogenated hydrocarbons. It will also cover general methods for other refrigerants where specific tables are not presented. Since volatile refrigerants are used in the liquid, vapor, and mixture phases, each of these will be treated separately.

Many fluids have been used for brines. Originally, the term *brines* applied to salt solutions such as calcium chloride or sodium chloride. The use of such salt brines permitted the transfer of heat at lower temperature levels without introducing refrigerant of the volatile type into refrigerated spaces. These brines were commonly used for cold storage plants, ice plants, or commercial and process refrigeration.

Solutions of glycols are also used as secondary coolants. Ethylene glycol and propylene glycol are most commonly used for this purpose. Several other compounds or mixtures have been developed specifically for the purpose of heat-transfer media. These compounds are specifically designed to have high thermal capacities, low viscosities, and other desirable properties for high heat transfer and low pressure losses.

Two major codes relate to refrigeration piping. One of these is the American Standard Safety Code for Mechanical Refrigeration. This code is reviewed and revised periodically. The most recent edition at the time of this writing was issued in 1994. This code is sponsored by the American Society of Heating, Refrigerating, and Air Conditioning Engineers and has been adopted by many states and municipalities to be the existing law in these localities. This Code will be referred to frequently in this section and will be designated as ANSI/ASHRAE 15.[1]

Another important Code on piping is the American Standard B31 Code for Pressure Piping.[2] Section B31.5 covers refrigeration piping and relates to structural design rules, fabrication, construction, and testing. The code in this section will be referred to as the ASME B31.5 Code.

This section recognizes and uses the definitions included in Sec. 3 of ASHRAE 15. The definitions included in Sec. 500.2 of the ASME B31.5 Code are also used. In general, the definitions in these two Codes coincide with definitions in the ASME Boiler Code, Sec. VIII, usually called the Pressure Vessel Code.[3] In addition, the ASME B31.5 Code recognizes and refers to the basic definitions of the American Welding Society.

Other basic definitions accepted by the refrigeration and air-conditioning industry are given in ASHRAE *Terminology of Heating, Ventilation, Air-Conditioning and Refrigeration*, published by the American Society of Heating, Refrigerating, and Air Conditioning Engineers.[4]

REFRIGERATION CYCLES

Compression System

Figure C9.1*a* shows a typical single-stage refrigeration cycle plotted on a pressure-enthalpy chart. Figure C9.1*b* is a typical diagram of a single-stage compression system and shows a compressor, a condenser, an optional receiver, an expansion device, and an evaporator. The state points on the line diagram of Fig. C9.1*b* are numbered to correspond to the same points on the pressure-enthalpy chart of Fig. C9.1*a*.

In a typical system, P_1 represents the pressure in the evaporator corresponding to the temperature at which the refrigerant is evaporating. From point 1 to point 2, the refrigerant vapor is carried to the compressor in a suction line. The pressure-enthalpy chart indicates a small pressure drop in this line. From point 2 to point 3, the refrigerant vapor is compressed. The connection between points 3 and 4 represents the discharge, or hot-gas, line, and the pressure-enthalpy chart also indicates the pressure drop due to friction in this line. Desuperheating and condensation of the refrigerant at constant pressure in the condenser occur between points 4 and 5. The liquid line is represented by the section between point 5 and point 6. From point 6 to point 7, there is a representation of the pressure reduction or ''expansion'' through the expansion device. From point 5 to point 8, the refrigerant is a mixture of liquid and vapor because of the expansion at constant enthalpy. At point 8, the liquid and vapor mixture enters the evaporator; heat transferred from the evaporator results in evaporation of the liquid in the liquid-vapor mixture and, as indicated, a slight superheating of the resultant vapor. At point 1, the cycle is completed.

In this chapter, the line between points 1 and 2 will be referred to as the *suction line*. The pipe or tubing between points 3 and 4 will be designated as the *hot-gas*

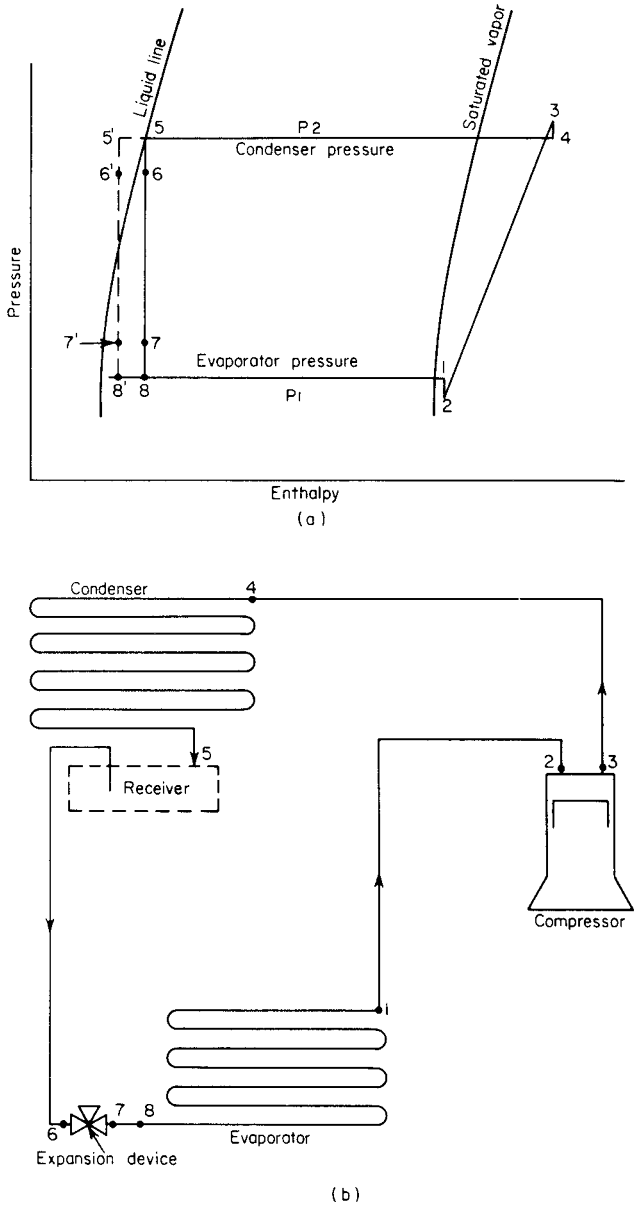


FIGURE C9.1 Compression refrigeration cycle. *(Courtesy Carrier Air Conditioning Co.)*

line. The piping between points 5 and 6 will be termed the *liquid line before expansion,* and that between points 7 and 8 will be called the *liquid line after expansion*.

Absorption System

Figure C9.2 shows a typical piping arrangement for an absorption-refrigeration cycle in which a lithium salt solution is used with water as the refrigerant. In such

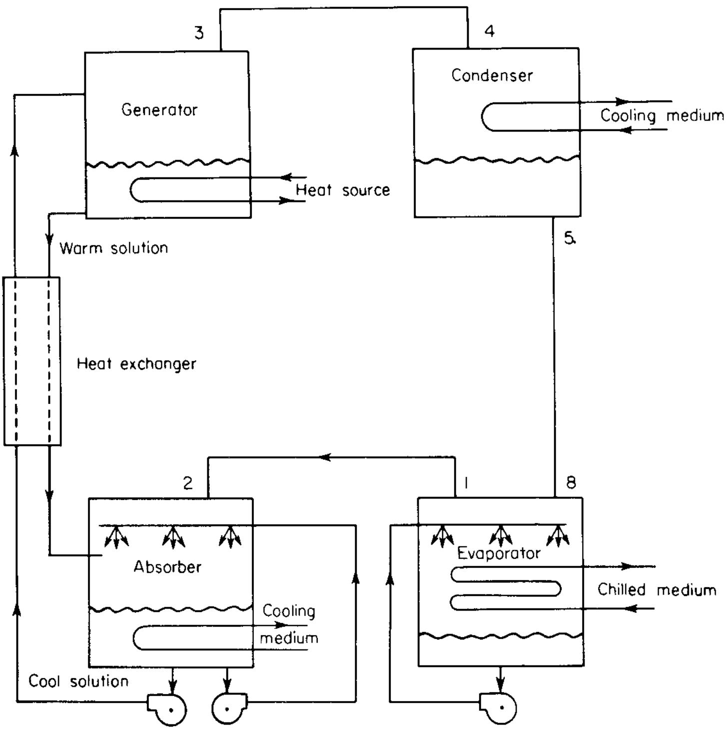


FIGURE C9.2 Absorption refrigeration cycle, lithium bromide type. *(Courtesy Carrier Air Conditioning Co.)*

a system, the absorber and generator serve the same purpose as the compressor in a compression system. The refrigerant is absorbed in solution at low pressure in the absorber and is pumped to the generator, where the refrigerant is boiled out of the solution at high pressure. The state points are numbered in Fig. C9.2 to correspond to the similar points of the compression system cycle in Fig. C9.1*a*.

For other types of absorption systems (such as ammonia and water), a rectifier or a fractionating-column type of purifier would be installed between the generator and condenser. Also, the evaporator might be remote or of some other type, such as a flooded cooller or a direct expansion type in which the refrigerant passes through coils.

In a consideration of the piping of an absorption system, the piping between points 3 and 4, 5 and 8, and 1 and 2 may be handled in a manner similar to those of a compression system when the different fluids and flow rates which might be encountered are taken into account. The solution lines between the absorber and generator and through the heat exchanger would be treated as brine lines in the case of lithium bromide or lithium chloride systems. For ammonia-water systems, properties of aqueous solutions of ammonia are available, and these lines would be desinged in the same manner as would be brine or water lines.

Present-day absorption-refrigeration systems are integrally designed as complete units, and in many instances the enitre assembly including the piping is made at the factory. For these reasons it is seldom necessary to consider piping for an absorption system as a separate design problem. However, the principles involved in this chapter could be applied to the corresponding sections of an absorption system.

Flow Rates

Refrigeration is usually measured in tons (kW) of refrigeration or in Btus per hour. The relationship between these is

$$1 \text{ ton of refrigeration} = 200 \text{ Btu/min or } 12{,}000 \text{ Btu/h} = 3.517 \text{ kW}$$

$$(1 \text{ kW} = 1 \text{ KJ/s} = 0.2843 \text{ Tons})$$

For selection of piping, it is necessary to relate refrigeration rate to flow rate of the refrigerant. Since the refrigerant changes state in the process, it is customary to calculate refrigerant flow in pounds per hour or per minute (kg/s). This weight flow rate for a constant rate of refrigeration will be constant throughout the system (for a single-stage system). The volume of refrigerant handled at the various points of the cycle can be determined if the density at these various points is known. Since the volume handled in a suction line is important not only for the piping selection but also for the compressor selection, charts are frequently made showing the cubic feet per minute per ton ($M^3/s \cdot kW$) of refrigerant gas or vapor at this point.

To calculate the flow rate of refrigerant for a given rate of refrigeration the following procedure is used:

$$\frac{\text{lb}}{(\text{min ton})} = \frac{200}{(h_g - h_f)} \tag{C9.1}$$

$$\left(\frac{\text{Kg}}{\text{s} \cdot \text{kW}} = \frac{\text{Kg}}{\text{KJ}} = \frac{1}{hg - hf}\right) \tag{C9.1M}$$

where h_g is the enthalpy of dry saturated vapor at the evaporator outlet pressure or temperature and h_f is the enthalpy of the liquid refrigerant at the expansion device inlet.

To calculate the flow rate it is necessary to have thermodynamic properties of the refrigerant available. After the mass-flow rate has been determined as shown in the equation, it is possible to determine the volume flowing at various points in the system.

If the specific volume of the refrigerant is known at the state point leaving the evaporator corresponding to point 1 in Fig. C9.1*a*, the volume at that point can be determined as follows:

$$\frac{\text{cfm}}{\text{ton}} = \frac{200\,V}{(h_g - h_f)} \tag{C9.2}$$

$$\left[\frac{\text{M}^3}{\text{s}\cdot\text{kW}} = \frac{V}{(h_g - h_f)}\right] \tag{C9.2M}$$

where V is the specific volume of the vapor at the evaporator suction temperature. For purposes of design, this value may be taken as the specific volume of dry, saturated vapor.

In the previous equations, h_g corresponds to the enthalpy of the refrigerant vapor at point 1 leaving the evaporator. In the flooded type of cooler, this state point is close to saturation, but for a direct-expansion coil-type evaporator, the vapor may be slightly superheated. For low amounts of superheat, it is permissible to ignore the superheat for purposes of selecting the piping, although this should not be done for compressor selection or for very accurate determination of the velocity.

The value h_f corresponds to the enthalpy of the liquid entering the expansion device. Some subcooling of liquid may exist at this point, in which case point 6′ would result in a more accurate figure. However, for selection of piping, it is customary to use the enthalpy of the saturated liquid at the condensing temperature (point 5) for determining the flow rate. Later considerations will show when the actual state of the liquid may have an effect on the liquid line size. A later example will illustrate methods of calculation for flow rate by weight and by volume in the suction line for a typical refrigerant.

Table C9.1 shows flow rates in pounds per minute per ton and cubic feet per minute per ton at the suction condition for three common refrigerants: ammonia, refrigerant R134a (tetrafluoroethane) and refrigerant 22 (monochlorodifluoromethane). This table is based on enthalpies read at saturated conditions of refrigerant vapor and liquid as previously mentioned.

The cubic feet per minute per ton (dm^3 per second per kW) in the discharge line cannot be calculated readily because the actual temperature at the end of the compression is a function of the compressor design and efficiency and these will vary among various manufacturers. The discharge cubic feet per minute per ton (dm^3 per second per kW) for single-stage systems can be approximated by the following formula:

$$\text{Discharge cfm/ton} = \text{suction cfm/ton} \times \frac{P_1}{P_2} \times 1.2 \tag{C9.3}$$

$$\left(\text{Discharge dm}^3/\text{s}\cdot\text{kW} = \text{suction dm}^3/\text{s}\cdot\text{kW} \times \frac{P_1}{P_2} \times 1.2\right) \tag{C9.3M}$$

where P_1 = absolute pressure at suction
P_2 = absolute pressure at discharge

This formula is not exact, but it will serve as an approximation when it is necessary to determine approximate velocities in discharge lines.

TABLE C9.1 Refrigerant Flow Rate

Refrigerant chemical formula (common name)		R-717 NH3 ammonia			R134a CH2FCF3 1,1,1,2-Tetrafluoroethane			R-22 CHCLF2 Chlordifluormethane		
Condensing temperature °C		30	35	40	30	35	40	30	35	40
°F		86	95	104	86	95	104	86	95	104
Evaporating temperature °C	°F	lbs/min per ton*								
0	32	0.410	0.420	0.429	2.89	3.02	3.17	2.70	2.81	2.92
−10	14	0.415	0.424	0.434	3.00	3.14	3.31	2.76	2.87	3.00
−20	−4	0.420	0.429	0.440	3.12	3.28	3.46	2.84	2.95	3.08
−30	−22	0.422	0.432	0.442	3.26	3.44	3.63	2.91	3.04	3.17
		Cfm/ton (suction line)†								
0	32	1.95	2.00	2.04	3.23	3.38	3.56	2.09	2.80	2.27
−10	14	2.85	2.92	2.99	4.87	5.11	5.38	2.97	3.09	3.22
−20	−4	4.31	4.40	4.51	7.52	7.91	8.34	4.32	4.50	4.69
−30	−22	6.68	6.84	7.00	12.04	12.69	13.43	6.50	6.77	7.08

* To convert to kg/s per kw, multiply by 2.1496×10^{-3}
† To convert to dm^3/s per kw, multiply by 0.1342

ALTERNATIVE REFRIGERANTS

Because of the depletion of stratospheric ozone, which is partly attributed to chlorine released to the atmosphere from chloro-fluro-carbon (CFC) chemicals, many alternative refrigerants made from fluorinated hydrocarbon gases are being evaluated within the refrigeration industry. The common CFC refrigerants R-12 and R-502 are no longer produced. The hydro-chloro-fluro-carbon (HCFC) refrigerants, such as R-22, are to be phased out of production in the United States by the year 2015, and earlier phaseouts are scheduled in Europe.

The natural refrigerants ammonia (R-717), propane (R-290), and carbon dioxide (R-744) are being promoted in new applications because when released they do not contribute to ozone depletion or to global warming. In addition to the customary use of ammonia in industrial food processing and storage, ammonia is now being widely used in district cooling plants for cooling of commercial buildings. Some food processing plants, now using R-12, R-22, or R-502, are converting or planning to eventually convert to ammonia. Some new processing plants are being designed to use R-22 currently and to convert to ammonia in the future. Propane is now being used in domestic refrigerators in Europe and Asia. Carbon dioxide is now being used as a low-stage refrigerant in supermarkets.

In the United States, the Environmental Protection Agency evaluates and regulates substitutes for the CFC ozone-depleting chemicals. The Significant New Alternatives Policy (SNAP) rule lists substitutes for R-11, R-12, and R-502 in both retrofit and new equipment.

- For new centrifugal chillers which previously used R-11, R-12, R-113, R-114, or R-500, the substitute refrigerants are R-123, R-124, R-22, R-134a, R-227ea, R-717, and absorption systems.
- For new reciprocating chillers, which previously used R-12, the substitutes are R-22, R-134a, and R-227ea.
- For direct air-conditioning applications, previously using R-12, the popular substitute is R-134a.
- For cold storage and retail food refrigeration formerly using R-12 and R-502, the substitutes are R-22, R-134a, R-227ea, R-402A, R-402B, R-404A, R-507, and R-717 vapor compression.

TABLE C9.2*a* Temperature-Pressure Chart for Constant Boiling Point Refrigerants

Temp°C	R-717 psia	R-134a psia	R-22 psia	R-507 psia	R-290 psia	R-744 psia	Temp°F
−90		0.222	0.696	1.049	0.933		−130
−80		0.535	1.501	2.197	1.887		−112
−70	1.5825	1.162	2.965	4.228	3.532		−94
−65	2.2608	1.738	4.200	5.704	4.856		−85
−60	3.1688	2.312	5.434	7.571	6.180		−76
−55	4.3636	3.294	7.394	9.899	8.200	80.47	−67
−50	5.9112	4.276	9.354	12.76	10.22	90.03	−58
−45	7.8870	5.852	12.02	16.24	13.16	120.8	−49
−40	10.376	7.429	15.26	20.42	16.10	145.8	−40
−35	13.473	9.832	19.16	25.41	19.90	174.5	−31
−30	17.281	12.26	23.76	31.27	24.32	207.1	−22
−25	21.915	15.45	29.23	38.16	29.50	244.1	−13
−20	27.498	19.25	35.58	46.11	35.45	285.7	−4
−15	34.163	23.77	42.99	55.30	42.29	332.3	5
−10	42.050	29.08	51.46	51.29	50.07	384.2	14
−5	51.311	35.30	61.21	77.75	58.90	441.8	23
0	62.102	42.45	72.24	91.23	68.79	505.5	32
5	74.591	50.72	84.77	106.4	79.94	575.7	41
10	88.950	60.12	98.80	123.4	92.32	653.0	50
15	105.36	70.84	115.9	142.5	106.1	737.7	59
20	124.01	82.90	132.0	163.5	121.3	803.8	68
25	145.09	96.52	151.6	186.8	138.1	933.2	77
30	168.80	111.7	172.9	212.5	156.5	1045.9	86
35	195.35	128.6	196.7	240.8	176.7		95
40	224.94	147.4	222.5	271.8	198.6		104
45	257.80	168.2	250.9	305.7	222.5		113
50	294.15	191.2	281.8	342.7	248.6		122
55	334.21	216.4	315.5	383.3	276.6		131
60	378.23	243.9	352.0	427.7	306.9		140
65	426.45	274.1	391.7	476.3	339.8		149
70	479.12	306.9	434.7		375.1		158
80	598.88	381.9	531.3		454.3		176
90	739.77	470.5	644.3		545.9		194
100	904.32	576.1					212
110	1095.5						230

Pressure is in psia; subtract 14.696 for gauge pressure (for kPa, multiply psia times 6.8947, subtract 101.33 for gauge pressure).

TABLE C9.2*b* Temperature-Pressure Chart for Zerotropic Mixtures with Temperature Glide

Pressure psia	R404A Temp °F bubble	Dew	R407C Temp °F bubble	Dew	R410A Temp °F bubble	Dew	Pressure kPa
2.0	−113.74	−111.89	−108.95	−94.98	−119.71	−119.62	13.79
4.0	−94.84	−93.16	−89.91	−76.32	−101.65	−101.56	27.58
6.0	−82.60	−81.02	−77.61	−64.27	−90.00	−89.90	41.37
8.0	−73.32	−71.80	−68.30	−55.15	−81.18	−81.08	55.16
10	−65.74	−64.28	−47.71	−73.99	−73.89	−68.95	68.95
14	−53.64	−52.25	−48.60	−35.84	−62.53	−62.42	96.53
18	−44.02	−42.69	−38.99	−26.43	−53.45	−53.33	124.1
22	−35.95	−34.66	−30.94	−18.55	−45.84	−45.72	151.7
26	−28.95	−27.71	−23.97	−11.72	−39.26	−39.13	179.3
30	−22.75	−21.54	−17.80	−5.67	−33.43	−33.30	206.8
34	−17.15	−15.97	−12.23	−0.23	−28.17	−28.04	234.4
38	−12.04	−10.88	−7.15	4.78	−23.38	−23.25	262.0
42	−7.33	−6.19	−2.47	9.32	−18.97	−18.83	289.6
46	−2.94	−1.83	1.88	13.58	−14.87	−14.73	317.2
50	1.16	2.25	5.95	17.55	−11.04	−10.89	344.7
60	10.41	11.46	15.13	26.52	−2.40	−2.24	413.7
70	18.57	19.58	23.20	34.41	5.20	5.36	482.6
80	25.88	26.86	30.44	41.47	12.00	12.17	551.6
90	32.54	33.49	37.03	47.89	18.18	18.36	620.5
100	38.66	39.58	43.08	53.79	23.86	24.04	689.5
120	49.65	50.52	53.93	64.35	34.03	34.22	827.4
140	59.34	60.17	63.50	73.65	42.99	43.19	965.3
160	68.05	68.83	72.10	81.99	51.02	51.23	1103
180	75.97	76.73	79.92	89.57	58.33	58.44	1241
200	83.27	83.99	87.13	96.53	65.04	65.26	1379
220	90.04	90.72	93.81	102.98	71.27	71.49	1517
240	96.36	97.01	100.06	108.99	77.04	77.30	1655
260	102.29	102.92	105.93	114.64	82.53	82.75	1793
280	107.90	108.49	111.48	119.95	87.67	87.89	1931
320	118.24	118.78	121.74	129.75	97.19	97.39	2206
360	127.63	128.12	131.09	138.63	105.80	106.01	2482
400	136.24	136.67	139.70	146.75	113.71	113.91	2758
450	146.06	146.42	149.59	156.00	122.77	122.95	3103
500	154.99	155.25	158.69	164.41	131.05	131.22	3447
550			167.13	138.82	138.68	138.82	3792
600			175.04	179.00	145.72	145.83	4137
650			182.58	185.04	152.22	152.29	4481

For gauge pressure in psig, subtract 14.696: For gauge pressure in kPa, subtract 101.33. For temperature in °C = 5/9(°F-32)

The marketplace is determining the popular substitutes for the CFCs. The HCFCs such as R-22, R-402A, and R-402B will be eventually phased out. The present choices being made in the United States, as reported in the commercial news, are:

- R-134a and R-22 to replace R-12 in refrigeration systems
- R-134a, R-407C, and R-410A to replace R-22 in air-conditioning equipment
- R-404A, R-407A, R-507, and R-717 to replace R-22 and R-502 in refrigeration systems

Tables C9.2*a* and C9.2*b* show the temperature-pressure relation for the main candidates. The pressures shown may be expected in refrigerant lines when both liquid and vapor are present. Propane (R-290) is included; however flammable options are not used in equipment exposed to the public in the United States because of liability laws. Table C9.2*b* shows the pressure-temperature relation for blends of two or more components. These mixtures have temperature changes, called *temperature glides,* during vaporizing and condensing. Note that R-404A and R-410A have very little temperature glide, so that they may be treated similarly to azeotropes (mixtures with a constant boiling temperature, such as R-507). These and similar mixtures with very small glides are informally referred to as *near-aseotropic refrigerant mixtures* (NARMs). R-407A, R-407C, and many other blends have significant temperature glide. With careful design of evaporators and condensers, the glide can be used to improve efficiency. However, blends with significant temperature glide must be charged into the system in the liquid phase in order to maintain the percentage of the components. Also, blends with glide are not suitable for use in flooded evaporators.

PRESSURE DROP IN REFRIGERATION PIPING

There are two separate tasks when designing a refrigeration piping system. The first task is to select the size of a pipe for a given flow. The second task, determining pressure drop, is accomplished after the pipe size is selected. Both tasks use the same equation, which describes the pressure drop in fluid flow within piping.

However, the tasks must be addressed separately. Customary piping tables, such as C9.4 through C9.9, list capacities at specified temperature or pressure drops per unit length of piping. These tables are an aid to determining the pressure drop after the equivalent length is known. But the equivalent length depends upon the diameter selected. Thus it is clear that the listed capacities are only correct for the unit length basis used in developing the table and are not to be used as recommended capacities.

Darcy-Weisbach Equation

The pressure drop of most fluids in piping is described by the Darcy-Weisbach equation:

$$\Delta p = f\left(\frac{L}{D}\right)\left(\rho\frac{V^2}{2g}\right) \tag{C9.4}$$

$$\left(\Delta p = f\left(\frac{L}{D}\right)\left(\rho\frac{V^2}{2}\right)\right) \tag{C9.4M}$$

where Δp = pressure, psf (pa)
f = Moody friction factor, dimensionless
L = length of pipe, ft (m)
D = inside diameter of pipe, ft (m)
ρ = density, lb/ft^3 (kg/m^3)

V = velocity, ft/s (m/s)
g = gravity acceleration, 32.174 ft/s^2 (9.807 m/s^2)

Optimum Sizing

In order to select a pipe size, some previous experience or criteria is needed, other than the customary capacity tables. Figure C9.3 illustrates the economics of selecting

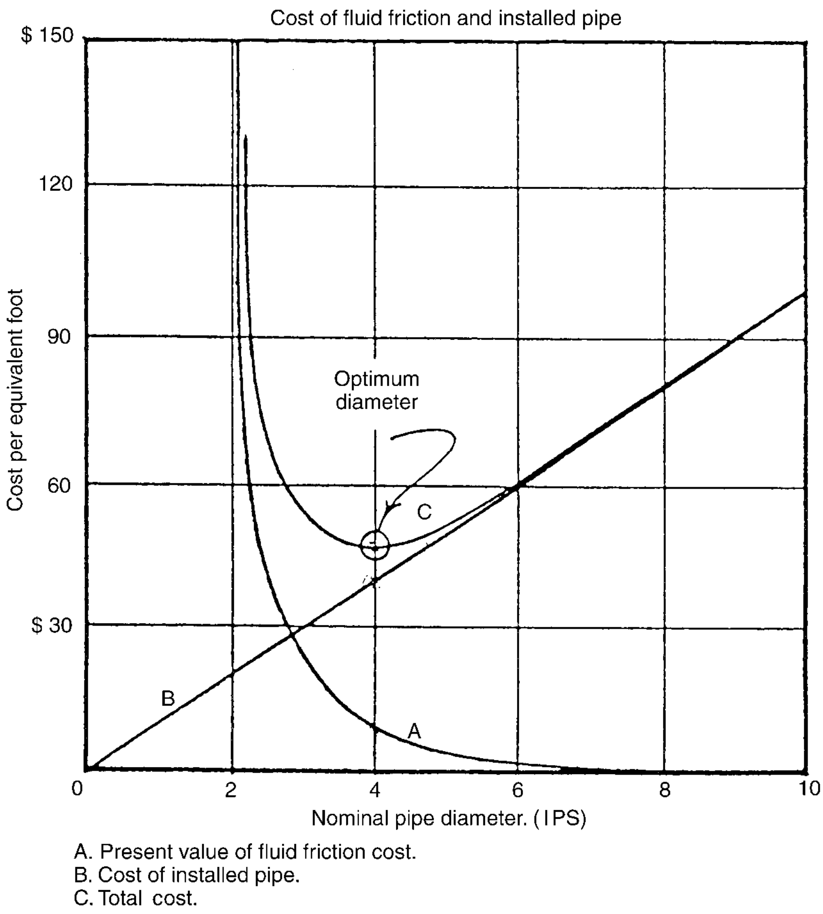


FIGURE C9.3 Cost of fluid friction and installed pipe.

a pipe size. Initially, it may seem that for a given flow and refrigerant condition, the optimum diameter of a long pipe will be greater than that of a short one. Richards[7] shows, however, that the optimum diameter is independent of length. The method sets the present installed cost of pipe minus the present worth of the lifetime pumping costs equal to zero, and then differentiates with respect to diameter to determine the optimum diameter. The length cancels in this analysis, which proves that the optimum diameter is independent of length. The effect of longer lengths is to increase the power cost for operation and may, for extreme lengths, require a slightly larger pump or compressor. The first cost of the next-larger pipe size will always exceed the savings represented by the present worth of future power costs. The total owning costs of a smaller pipe size will always be greater than for an optimum choice, because the present worth of the future cost of operation will always exceed the first cost savings.

Studies[8,9] recently made for R-717 and R-22 give some guidance for recommended piping capacities. Conservative conditions are assumed in ammonia industrial applications. Assume typical equipment efficiencies, running 40 percent of the time at full load for 15 years, with power cost of $0.05 per kW, the cost of money 5 percent higher than inflation, and with installed insulated pipe costs of $13.00 per in of dia per ft of length ($1.68 per mm of dia per m of length). The results of such studies can be expressed in recommended velocity pressures per velocity head, $\Delta p/k = \rho V^2/2g$ ($\Delta p/k = \rho\, V^2/2$). A more convenient form is to state the recommended equivalent temperature drop in a globe valve.

Recommended Temperature Drop Gradients for Vapor Flow in Industrial Applications

- For R-717 vapor, the result of such studies suggests that the temperature drop in a globe valve at low temperatures should be about 0.25°F (0.14°C), that is at −40°F (−40°C), and at high temperatures about 0.125°F (0.07°C), i.e. ≥ at 41°F (≥5°C).
- For R-22 vapor, the suggested temperature drop in a globe valve at low temperatures should be about 0.5°F (0.28°C), and at high temperatures about 0.25°F (0.14°C).

Practical temperature drop gradients can be much greater with higher installation and operating costs, with less running time, and with shorter capital recovery life. For each doubling of any combination of increased costs or reduced running time or life expentancy, the optimum capacity increases by 28 percent, the pressure gradient increases by 58 percent in copper tube and by 62 percent in steel pipe. The natural refrigerants R-290 and R-744 use similar pressure gradients as R-717. All halocarbon refrigerants use similar pressure gradients to R-22.

Comparing Pipe Flow Capacities

To compare pipe flow capacities we can rewrite the Darcy equation in terms of mass flow. Since mass flow equals area times velocity times density, we can insert the mass flow divided by area and density in place of velocity.

$$\Delta p = f\left(\frac{L}{D}\right)\left(\frac{w^2}{2 * A^2 * \rho * g}\right) \tag{C9.5}$$

$$\left(\Delta p = f\left(\frac{L}{D}\right)\left(\frac{w^2}{2 * A^2 * \rho}\right)\right) \tag{C9.5M}$$

where w = flow, lbm/s (kg/s)
A = inside flow area of pipe, ft^2 (m^2)

Rearranging the equation to show the mass flow capacity:

$$w = A\left(\frac{2 * g * \Delta p * \rho * D}{f * L}\right)^{0.5} \tag{C9.6}$$

$$\left(w = A\left(\frac{2 * \Delta p * \rho * D}{f * L}\right)^{0.5}\right) \tag{C9.6M}$$

Now, since it can be shown that all pipe sizes have similar L/D ratios and since the pressure drop and density are relatively constant for a given application, we can show that the mass flow is proportional to the area divided by the square root of the friction factor.

$$w \approx \frac{A}{\sqrt{f}} \tag{C9.7}$$

The relative values of friction factors for fully turbulent flow extend reasonably well in the area of the Moody chart between laminar and fully turbulent flow where most applications occur. Table C9.3 compares optimum pipe flow capacities for R-717 suction lines compared to capacities listed in Table C9.7. The friction factors used are those for fully turbulent flow.

The obvious point to be made in the previous comparison is that the listed capacities in Tables C9.4 through C9.9 are not recommended capacities. These tables merely provide a convenient starting point for adjusting the capacity and temperature drop to other lengths and to other capacities. After a pipe size is

TABLE C9.3 Ammonia Vapor Flow Methods Compared

IPS	1	1½	2	3	4	5	6	8	10	12
DN	25	40	50	80	100	125	150	200	250	300
d(inch)	*0.957*	*1.50*	*2.067*	*3.067*	*4.026*	*5.045*	*6.065*	*7.981*	*10.02*	*12.00*
d(mm)	*26.64*	*40.89*	*52.50*	*77.93*	*102.3*	*128.2*	*154.1*	*202.7*	*254.5*	*304.8*
f	*0.0225*	*0.0202*	*0.0190*	*0.0173*	*0.0163*	*0.0155*	*0.0149*	*0.0141*	*0.0134*	*0.0130*
multiplier[1]	*0.0481*	*0.1247*	*0.2441*	*0.5632*	*I.0000*	*1.6109*	*2.3725*	*4.2280*	*6.8205*	*9.9599*
Optimum[2] tons −30°F	3.3	8.6	17.1	**39.4**	69.9	113	166	302	477	696
Table C9.3 tons −30°F ½ psi/100 ft	1.9	7.3	14.1	**40.1**	83.5	150	244	500	900	1450
Optimum[2] tons + 40°F	9.0	23.3	46.0	**106.2**	189	304	447	815	1286	1878
Table C9.3 tons + 40°F ½ psi/100 ft	4.3	17.1	32.8	**92.5**	190	342	558	1135	2040	3325

Note 1: The multiplier is $A/f^{0.5}$, relative to IPS 4 (DN 100) at 1.0000.
Note 2: Optimum capacity is from Reference 9.

selected, the equivalent lengths including the valves and fittings are totaled from Tables C9.10*a*, *b*, and *c*. The pressure drop or temperature drop can be found by using the formulas in Notes 2 and 3 under Tables C9.7, C9.8, and C9.9. There are factors other than economics, that affect line sizes. These factors are discussed in the following paragraphs.

Suction Lines

In Fig. C9.1*a*, the evaporator pressure between points 8 and 1 is established by the heat load in the evaporator. The compressor pumps the refrigerant from point 2 to point 3. A pressure loss in the suction line between point 1 and 2 causes the compressor to operate at a lower suction pressure. Since the vapor expands at a lower pressure and since the compressor is essentially a fixed-volume device, a reduction in suction pressure causes a decrease in mass flow. The increased pressure differential due to pressure loss also results in more power per ton (per kW) of refrigeration.

Pressure drop and costs are not the only considerations in the selection of suction lines for refrigeration systems. The chlorinated hydrocarbons (CFCs and HCFCs) are miscible to some degree with mineral oil. The newer fluorinated hydrocarbons (HFCs) require synthetic lubricants in most cases to provide for mutual solubility that will keep the oil in circulation to enable the return of oil to the compressor. The sizing of oil return lines for vertical suction risers is discussed later in connection with Table C9.13.

It is customary to size suction lines so that the total loss in pressure doesn't exceed the equivalent of about 4°F (2.2°C) in saturated temperatures for halocarbon refrigerants and about 2°F (1.1°C) for ammonia. Greater equivalent pressure drops are tolerated in very long piping systems. Tables C9.2*a* and C9.2*b* reveal the relation between pressure change and temperature change for common refrigerants. Keep in mind that the system capacity loss, or the increased cost of refrigeration due to suction line pressure loss is about 2.2 percent per°F (4 percent per°C). Tables C9.4, C9.5, and C9.6 show suction line capacities in tons for two temperature loss rates per 100 feet (kW per K/m). The capacities are shown for single-stage systems with condensing temperatures at 86°F (30°C) for R-717 and 104°F (40°C) for R-134a and R-22.

When the liquid feed is precooled in economized or two-stage systems, the system capacities increase in proportion to the enthalpy increase between liquid and suction. Tables of the thermal properties of the refrigerant are required to assist in such an analysis. Table C9.7 for ammonia lists suction line capacities. Tables C9.8 for R-134a and C9.9 for R-22 list suction line capacities. The formulas in the table notes provide the means of adjusting the line capacity or temperature drop from the table values to actual values. The factors in Note 4 provide for the effect of different condensing temperatures.

The application of two-phase flow or the simultaneous liquid and vapor flow in return lines from evaporators to accumulators is not treated in this discussion. Suffice it to say that the pressure drop in horizontal two-phase flow increases the pressure drop approximately in proportion to the increase in the mass flow. This means that the velocity in the suction line for liquid overfeed applications should be reduced approximately by the reciprocal of the square root of the circulating number. The circulating number is the reciprocal of the refrigerant quality at the evaporator exit. For example, the velocity in return piping, for a circulating number of 4 (vapor quality at the exit of the evaporator is 0.25 or 3 parts liquid and 1 part

TABLE C9.4 (IP) Suction Line Capacities in Tons for Ammonia with Pressure Drops of 0.25 and 0.50°F per 100 ft Equivalent

Steel line size		Saturated suction temperature, °F					
		−60		−40		−20	
NPS	SCH	Δt = 0.025°F Δp = 0.046	Δt = 0.50°F Δp = 0.092	Δt = 0.25°F Δp = 0.077	Δt = 0.50°F Δp = 0.155	Δt = 0.25°F Δp = 0.123	Δt = 0.50°F Δp = 0.245
⅜	80	0.03	0.05	0.06	0.09	0.11	0.16
½	80	0.06	0.10	0.12	0.18	0.22	0.32
¾	80	0.15	0.22	0.28	0.42	0.50	0.73
1	80	0.30	0.45	0.57	0.84	0.99	1.44
1¼	40	0.82	1.21	1.53	2.24	2.65	3.84
1½	40	1.25	1.83	2.32	3.38	4.00	5.80
2	40	2.43	3.57	4.54	6.59	7.79	11.26
2½	40	3.94	5.78	7.23	10.56	12.50	18.03
3	40	7.10	10.30	13.00	18.81	22.23	32.09
4	40	14.77	21.21	26.81	38.62	45.66	65.81
5	40	26.66	38.65	48.68	70.07	82.70	119.60
6	40	43.48	62.83	79.18	114.26	134.37	193.44
8	40	90.07	129.79	163.48	235.38	277.80	397.55
10	40	164.26	236.39	297.51	427.71	504.98	721.08
12	ID	264.07	379.88	477.55	686.10	808.93	1157.59

Steel line size		Saturated suction temperature, °F					
		0		20		40	
NPS	SCH	Δt = 0.025°F Δp = 0.184	Δt = 0.50°F Δp = 0.368	Δt = 0.25°F Δp = 0.265	Δt = 0.50°F Δp = 0.530	Δt = 0.25°F Δp = 0.366	Δt = 0.50°F Δp = 0.582
⅜	80	0.18	0.26	0.28	0.40	0.41	0.53
½	80	0.36	0.52	0.55	0.80	0.82	1.05
¾	80	0.82	1.18	1.26	1.83	1.87	2.38
1	40	1.62	2.34	2.50	3.60	3.68	4.69
1¼	40	4.30	6.21	6.63	9.52	9.76	12.42
1½	40	6.49	9.34	9.98	14.34	14.68	18.64
2	40	12.57	18.12	19.35	27.74	28.45	36.08
2½	40	20.19	28.94	30.98	44.30	45.37	57.51
3	40	35.87	51.35	54.98	78.50	80.40	101.93
4	40	73.56	105.17	112.34	160.57	164.44	208.34
5	40	133.12	190.55	203.53	289.97	296.88	376.18
6	40	216.05	308.62	329.59	469.07	480.96	609.57
8	40	444.56	633.82	676.99	962.47	985.55	1250.34
10	40	806.47	1148.72	1226.96	1744.84	1786.55	2263.99
12	ID	1290.92	1839.28	1964.56	2790.37	2862.23	3613.23

Note: Capacities are in tons of refrigeration resulting in a line friction loss (Δp in psi per 100 ft equivalent pipe length), with corresponding change (Δt in °F per 100 ft) in saturation temperature.

vapor) should be reduced to 50 percent of the velocity allowed for the vapor flow alone. Vertical risers in two-phase flow require higher velocities or multiple risers to carry the liquid upward with reduced pressure losses. Pressure losses in two-phase flow are reduced when accumulators are located below and close to evaporators. When accumulators are located above evaporators, arrangements for separating and pumping the liquid return improve efficiency.

TABLE C9.4M (SI) Suction Line Capacities (kW) for Ammonia for Pressure Drops of 0.005 and 0.01 K/m Equivalent

Steel line size nominal DN	Saturated suction temperature, °C					
	−50		−40		−30	
	Δt = 0.005 K/m Δp = 12.1 Pa/m	Δt = 0.01 K/m Δp = 24.2 Pa/m	Δt = 0.005 K/m Δp = 19.2 Pa/m	Δt = 0.01 K/m Δp = 38.4 Pa/m	Δt = 0.005 K/m Δp = 29.1 Pa/m	Δt = 0.01 K/m Δp = 58.2 Pa/m
10	0.19	0.29	0.35	0.51	0.58	0.85
15	0.37	0.55	0.65	0.97	1.09	1.60
20	0.80	1.18	1.41	2.08	2.34	3.41
25	1.55	2.28	2.72	3.97	4.48	6.51
32	3.27	4.80	5.71	8.32	9.36	13.58
40	4.97	7.27	8.64	12.57	14.15	20.49
50	9.74	14.17	16.84	24.50	27.57	39.82
65	15.67	22.83	27.13	39.27	44.17	63.77
80	28.08	40.81	48.36	69.99	78.68	113.30
100	57.95	84.10	99.50	143.84	161.77	232.26
125	105.71	153.05	181.16	261.22	293.12	420.83
150	172.28	248.91	294.74	424.51	476.47	683.18
200	356.67	514.55	609.20	874.62	981.85	1402.03
250	649.99	937.58	1107.64	1589.51	1782.31	2545.46
300	1045.27	1504.96	1777.96	2550.49	2859.98	4081.54

TABLE C9.4M (SI) Suction Line Capacities (kW) for Ammonia for Pressure Drops of 0.005 and 0.01 K/m Equivalent *(Continued)*

Steel line size nominal DN	Saturated suction temperature, °C					
	−20		−5		+5	
	Δt = 0.005 K/m Δp = 42.2 Pa/m	Δt = 0.01 K/m Δp = 84.4 Pa/m	Δt = 0.005 K/m Δp = 69.2 Pa/m	Δt = 0.01 K/m Δp = 138.3 Pa/m	Δt = 0.005 K/m Δp = 92.6 Pa/m	Δt = 0.01 K/m Δp = 185.3 Pa/m
10	0.91	1.33	1.66	2.41	2.37	3.42
15	1.72	2.50	3.11	4.50	4.42	6.37
20	3.66	5.31	6.61	9.53	9.38	13.46
25	6.98	10.10	12.58	18.09	17.79	25.48
32	14.58	21.04	26.17	37.56	36.94	52.86
40	21.99	31.73	39.40	56.39	55.53	79.38
50	42.72	61.51	76.29	109.28	107.61	153.66
65	68.42	98.23	122.06	174.30	171.62	245.00
80	121.52	174.28	216.15	308.91	304.12	433.79
100	249.45	356.87	442.76	631.24	621.94	885.81
125	452.08	646.25	800.19	1139.74	1124.47	1598.31
150	733.59	1046.77	1296.07	1846.63	1819.59	2590.21
200	1506.11	2149.60	2662.02	3784.58	3735.65	5303.12
250	2731.90	3895.57	4818.22	6851.91	6759.98	9589.56
300	4378.87	6237.23	7714.93	10973.55	10810.65	15360.20

Note: Capacities are in kilowatts of refrigeration resulting in a line friction loss per unit equivalent pipe length (Δp in Pa/m), with corresponding change in saturation temperature per unit length (Δt in K/m).

TABLE C9.5 (IP) Suction Line Capacities in Tons for Refrigeration 134a (Single- or High-Stage Applications)

Line size		Saturated suction temperature, °F									
		0		10		20		30		40	
Type L copper, OD		$\Delta t = 1$°F $\Delta p = 0.50$	$\Delta t = 0.5$°F $\Delta p = 0.25$	$\Delta t = 1$°F $\Delta p = 0.60$	$\Delta t = 0.5$°F $\Delta p = 0.30$	$\Delta t = 1$°F $\Delta p = 0.71$	$\Delta t = 0.5$°F $\Delta p = 0.35$	$\Delta t = 1$°F $\Delta p = 0.83$	$\Delta t = 0.5$°F $\Delta p = 0.42$	$\Delta t = 1$°F $\Delta p = 0.97$	$\Delta t = 0.5$°F $\Delta p = 0.48$
½		0.10	0.07	0.12	0.08	0.16	0.11	0.19	0.13	0.24	0.16
⅝		0.18	0.12	0.23	0.16	0.29	0.20	0.37	0.25	0.45	0.31
⅞		0.48	0.33	0.62	0.42	0.78	0.53	0.97	0.66	1.20	0.82
1⅛		0.99	0.67	1.26	0.86	1.59	1.08	1.97	1.35	2.43	1.66
1⅜		1.73	1.18	2.21	1.51	2.77	1.89	3.45	2.36	4.25	2.91
1⅝		2.75	1.88	3.50	2.40	4.40	3.01	5.46	3.75	6.72	4.61
2⅛		5.73	3.92	7.29	5.00	9.14	6.27	11.40	7.79	14.00	9.59
2⅝		10.20	6.97	12.90	8.87	16.20	11.10	20.00	13.80	24.70	17.00
3⅛		16.20	11.10	20.60	14.20	25.90	17.80	32.10	22.10	39.40	27.20
3⅝		24.20	16.60	30.80	21.20	38.50	26.50	47.70	32.90	58.70	40.40
4⅛		34.20	23.50	43.40	29.90	54.30	37.40	67.30	46.50	82.60	57.10
5⅛		61.30	42.20	77.70	53.60	97.20	67.10	121.00	83.20	148.00	102.00
6⅛		98.80	68.00	125.00	86.30	157.00	108.00	194.00	134.00	237.00	165.00
Steel											
NPS	SCH										
½	80	0.16	0.11	0.20	0.14	0.25	0.17	0.30	0.21	0.37	0.26
¾	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
1¼	40	1.84	1.29	2.31	1.62	2.87	2.02	3.54	2.48	4.31	3.03
1½	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
2½	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 due to line friction, psi per 100 ft equivalent line length
Δt = change in saturation temperature corresponding to pressure drop, °F per 100 ft

TABLE C9.5M (SI) Suction Line Capacities in Kilowatts for Refrigeration 134a (Single- or High-Stage Applications) for Pressure Drops of 0.02 and 0.01 K/m Equivalent

Nominal line size,	Saturated suction temperature, °C									
	−10		−5		0		5		10	
	Δt = 0.02 K	Δt = 0.01 K	Δt = 0.02 K	Δt = 0.01 K	Δt = 0.02 K	Δt = 0.01 K	Δt = 0.02 K	Δt = 0.01 K	Δt = 0.02 K	Δt = 0.01 K
	Δp = 159	Δp = 79.3	Δp = 185	Δp = 92.4	Δp = 212	Δp = 106	Δp = 243	Δp = 121	Δp = 278	Δp = 139
Copper O.D. mm										
12	0.42	0.28	0.52	0.35	0.63	0.43	0.76	0.51	0.91	0.62
15	0.81	0.55	0.99	0.67	1.20	0.82	1.45	0.99	1.74	1.19
18	1.40	0.96	1.73	1.18	2.09	1.43	2.53	1.72	3.03	2.07
22	2.48	1.69	3.05	2.08	3.69	2.52	4.46	3.04	5.34	3.66
28	4.91	3.36	6.03	4.13	7.31	5.01	8.81	6.02	10.60	7.24
35	9.05	6.18	11.10	7.60	13.40	9.21	16.20	11.10	19.40	13.30
42	15.00	10.30	18.40	12.60	22.30	15.30	26.90	18.40	32.10	22.10
54	30.00	20.50	36.70	25.20	44.40	30.50	53.40	36.70	63.80	44.00
67	53.40	36.70	65.40	44.90	79.00	54.40	95.00	65.40	113.00	78.30
79	82.80	56.90	101.00	69.70	122.00	84.30	147.00	101.00	176.00	122.00
105	178.00	122.00	217.00	149.00	262.00	181.00	315.00	217.00	375.00	260.00
Steel DN										
10	0.61	0.42	0.74	0.52	0.89	0.62	1.06	0.74	1.27	0.89
15	1.13	0.79	1.38	0.96	1.65	1.16	1.97	1.38	2.35	1.65
20	2.39	1.67	2.91	2.03	3.49	2.44	4.17	2.92	4.94	3.47
25	4.53	3.17	5.49	3.85	6.59	4.62	7.86	5.52	9.33	6.56
32	9.37	6.57	11.40	7.97	13.60	9.57	16.30	11.40	19.30	13.60
40	14.10	9.86	17.10	12.00	20.50	14.40	24.40	17.10	28.90	20.40
50	27.20	19.10	32.90	23.10	39.50	27.70	47.00	33.10	55.80	39.40
65	43.30	30.40	52.50	36.90	62.90	44.30	75.00	52.70	88.80	62.70
80	76.60	53.80	92.80	65.30	111.00	78.30	133.00	93.10	157.00	111.00
100	156.00	110.00	189.00	133.00	227.00	160.00	270.00	190.00	320.00	226.00

Δp = pressure drop per unit equivalent length of line, Pa/m
Δt = corresponding change in saturation temperature, K/m

TABLE C9.6 (IP) 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	
		$\Delta t = 1$°F $\Delta p = 0.393$	$\Delta t = 0.5$°F $\Delta p = 0.197$	$\Delta t = 1$°F $\Delta p = 0.577$	$\Delta t = 0.5$°F $\Delta p = 0.289$	$\Delta t = 1$°F $\Delta p = 0.813$	$\Delta t = 0.5$°F $\Delta p = 0.406$	$\Delta t = 1$°F $\Delta p = 1.104$	$\Delta t = 0.5$°F $\Delta p = 0.552$	$\Delta t = 1$°F $\Delta p = 1.455$	$\Delta t = 0.5$°F $\Delta p = 0.727$
½		0.07	0.05	0.12	0.08	0.18	0.12	0.27	0.19	0.40	0.27
⅝		0.13	0.09	0.22	0.15	0.34	0.23	0.52	0.35	0.75	0.51
¾		0.22	0.15	0.37	0.25	0.58	0.39	0.86	0.59	1.24	0.85
⅞		0.35	0.24	0.58	0.40	0.91	0.62	1.37	0.93	1.97	1.35
1⅛		0.72	0.49	1.19	0.81	1.86	1.27	2.77	1.90	3.99	2.74
1⅜		1.27	0.86	2.09	1.42	3.25	2.22	4.84	3.32	6.96	4.78
1⅝		2.02	1.38	3.31	2.26	5.16	3.53	7.67	5.26	11.00	7.57
2⅛		4.21	2.88	6.90	4.73	10.71	7.35	15.92	10.96	22.81	15.73
2⅝		7.48	5.13	12.23	8.39	18.97	13.04	28.19	19.40	40.38	27.84
3⅛		11.99	8.22	19.55	13.43	30.31	20.85	44.93	31.00	64.30	44.44
3⅝		17.89	12.26	29.13	20.00	45.09	31.03	66.81	46.11	95.68	66.09
4⅛		25.29	17.36	41.17	28.26	63.71	43.85	94.25	65.12	134.81	93.22
Steel											
NPS	SCH										
⅜	80	0.06	0.04	0.10	0.07	0.15	0.10	0.21	0.15	0.30	0.21
½	80	0.12	0.08	0.19	0.13	0.29	0.20	0.42	0.30	0.60	0.42
¾	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
1¼	40	1.38	0.96	2.21	1.55	3.37	2.36	4.91	3.45	6.93	4.88
1½	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
2½	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 due to line friction, psi per 100 ft equivalent line length
Δt = change in saturation temperature corresponding to pressure drop, °F per 100 ft

TABLE C9.6M (SI) Suction Line Capacities in Kilowatts for Refrigerant 22 (Single- or High-Stage Applications) for Pressure Drops of 0.02 and 0.01 K/m Equivalent

Nominal line size,	Saturated suction temperature, °C									
	−40		−30		−20		−5		+5	
	$\Delta t = 0.02$ $\Delta p = 97.9$	$\Delta t = 0.01$ $\Delta p = 49.0$	$\Delta t = 0.02$ $\Delta p = 138$	$\Delta t = 0.01$ $\Delta p = 69.2$	$\Delta t = 0.02$ $\Delta p = 189$	$\Delta t = 0.01$ $\Delta p = 94.6$	$\Delta t = 0.02$ $\Delta p = 286$	$\Delta t = 0.01$ $\Delta p = 143$	$\Delta t = 0.02$ $\Delta p = 366$	$\Delta t = 0.01$ $\Delta p = 183$
O.D. mm	Copper line									
12	0.21	0.14	0.34	0.23	0.51	0.34	0.87	0.59	1.20	0.82
15	0.41	0.28	0.65	0.44	0.97	0.66	1.67	1.14	2.30	1.56
18	0.72	0.49	1.13	0.76	1.70	1.15	2.91	1.98	4.00	2.73
22	1.28	0.86	2.00	1.36	3.00	2.04	5.14	3.50	7.07	4.82
28	2.54	1.72	3.97	2.70	5.95	4.06	10.16	6.95	13.98	9.56
35	4.69	3.19	7.32	4.99	10.96	7.48	18.69	12.80	25.66	17.59
42	7.82	5.32	12.19	8.32	18.20	12.46	31.03	21.27	42.59	29.21
54	15.63	10.66	24.34	16.65	36.26	24.88	61.79	42.43	84.60	58.23
67	27.94	19.11	43.48	29.76	64.79	44.48	110.05	75.68	150.80	103.80
79	43.43	29.74	67.47	46.26	100.51	69.04	170.64	117.39	233.56	161.10
105	93.43	63.99	144.76	99.47	215.39	148.34	365.08	251.92	499.16	344.89
DN	Steel line									
10	0.33	0.23	0.50	0.35	0.74	0.52	1.25	0.87	1.69	1.18
15	0.61	0.42	0.94	0.65	1.38	0.96	2.31	1.62	3.15	2.20
20	1.30	0.90	1.98	1.38	2.92	2.04	4.87	3.42	6.63	4.65
25	2.46	1.71	3.76	2.62	5.52	3.86	9.22	6.47	12.52	8.79
32	5.11	3.56	7.79	5.45	11.42	8.01	19.06	13.38	25.88	18.20
40	7.68	5.36	11.70	8.19	17.16	12.02	28.60	20.10	38.89	27.35
50	14.85	10.39	22.65	14.86	33.17	23.27	55.18	38.83	74.92	52.77
65	23.74	16.58	36.15	25.30	52.84	37.13	87.91	61.89	119.37	84.05
80	42.02	29.43	63.95	44.84	93.51	65.68	155.62	109.54	211.33	148.77
100	85.84	60.16	130.57	91.69	190.95	134.08	317.17	223.47	430.77	303.17
125	155.21	108.97	235.58	165.78	344.66	242.47	572.50	403.23	776.67	547.16
150	251.47	176.49	381.78	268.72	557.25	391.95	925.72	652.73	1255.93	885.79
200	515.37	362.01	781.63	550.49	1141.07	803.41	1895.86	1336.79	2572.39	1813.97
250	933.07	656.12	1413.53	996.65	2063.66	1454.75	3429.24	2417.91	4646.48	3280.83
300	1494.35	1050.57	2264.54	1593.85	3305.39	2330.50	5477.74	3867.63	7433.20	5248.20

TABLE C9.7 (IP) Suction, Discharge, and Liquid Line Capacities in Tons for Ammonia (Single- or High-Stage Applications)

Steel line size		Suction lines (Δt = 1°F/100 ft) Saturated suction temperature, °F					Discharge lines	Steel line size		Liquid lines	
NPS	SCH	−40 Δp = 0.31	−20 Δp = 0.49	0 Δp = 0.73	20 Δp = 1.06	40 Δp = 1.46	Δt = 1°F Δp = 2.95	IPS	SCH	Velocity = 100 fpm	Δp = 2.0 psi Δt = 0.7°F
⅜	80	—	—	—	—	—	—	⅜	80	8.6	12.1
½	80	—	—	—	—	—	3.1	½	80	14.2	24.0
¾	80	—	—	—	2.6	3.8	7.1	¾	80	26.3	54.2
1	80	—	2.1	3.4	5.2	7.6	13.9	1	80	43.8	106.4
1¼	40	3.2	5.6	8.9	13.6	19.9	36.5	1¼	80	78.1	228.6
1½	40	4.9	8.4	13.4	20.5	29.9	54.8	1½	80	107.5	349.2
2	40	9.5	16.2	26.0	39.6	57.8	105.7	2	40	204.2	811.4
2½	40	15.3	25.9	41.5	63.2	92.1	168.5	2½	40	291.1	1292.6
3	40	27.1	46.1	73.5	111.9	163.0	297.6	3	40	449.6	2287.8
4	40	55.7	94.2	150.1	228.7	333.0	606.2	4	40	774.7	4662.1
5	40	101.1	170.4	271.1	412.4	600.9	1095.2	5	40	—	—
6	40	164.0	276.4	439.2	667.5	971.6	1771.2	6	40	—	—
8	40	337.2	566.8	901.1	1366.6	1989.4	3623.0	8	40	—	—
10	40	611.6	1027.2	1634.3	2474.5	3598.0	—	10	40	—	—
12	ID	981.6	1644.5	2612.4	3963.5	5764.6	—	12	ID	—	—

Notes:

1. Table capacities are in tons of refrigeration.
   Δp = pressure drop due to 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

$$\text{Line capacity} = \text{Table capacity}\left(\frac{\text{Table } L_e}{\text{Actual } L_e} \times \frac{\text{Actual } \Delta t}{\text{Table } \Delta t}\right)^{0.51}$$

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

$$\Delta t = \text{Table } \Delta t \left(\frac{\text{Actual } L_e}{\text{Table } L_e}\right)\left(\frac{\text{Actual capacity}}{\text{Table capacity}}\right)^{1.96}$$

4. Values in the table are based on 90°F condensing temperature. Multiply table capacities by the following factors for other condensing temperatures.

Condensing temperature, °F	Suction lines	Discharge lines
70	1.05	0.78
80	1.02	0.89
90	1.00	1.00
100	0.98	1.11

5. Discharge and liquid line capacities are based on 20°F suction. Evaporator temperature is 0°F. The capacity is affected less than 3% when applied from −40 to +40°F extremes.

TABLE C9.7M (SI) Suction, Discharge, and Liquid Line Capacities for Ammonia (Single- or High-Stage Applications)

Steel line size	Suction lines $\Delta t = 0.02$ K/m					Discharge lines $\Delta t = 0.02$ K/m, $\Delta p = 684.0$ Pa/m			Steel line size		
	Saturated suction temperature, °C										
Nominal DN	−40 $\Delta p = 76.9$	−30 $\Delta p = 116.3$	−20 $\Delta p = 168.8$	−5 $\Delta p = 276.6$	5 $\Delta p = 370.5$	−40	−20	+5	Nominal mm	velocity = 0.5 m/s	$\Delta t = 0.013$ K/100 m $\Delta p = 450$ Pa/m
10	0.8	1.2	1.9	3.5	4.9	8.0	8.3	8.5	10	3.9	63.8
15	1.4	2.3	3.6	6.5	9.1	14.9	15.3	15.7	15	63.2	118.4
20	3.0	4.9	7.7	13.7	19.3	31.4	32.3	33.2	20	110.9	250.2
25	5.8	9.4	14.6	25.9	36.4	59.4	61.0	62.6	25	179.4	473.4
32	12.1	19.6	30.2	53.7	75.4	122.7	126.0	129.4	32	311.0	978.0
40	18.2	29.5	45.5	80.6	113.3	184.4	189.4	194.5	40	423.4	1469.4
50	35.4	57.2	88.1	155.7	218.6	355.2	364.9	374.7	50	697.8	2840.5
65	56.7	91.6	140.6	248.6	348.9	565.9	581.4	597.0	65	994.8	4524.8
80	101.0	162.4	249.0	439.8	616.9	1001.9	1029.3	1056.9	80	1536.3	8008.8
100	206.9	332.6	509.2	897.8	1258.6	2042.2	2098.2	2154.3	—	—	—
125	375.2	601.8	902.6	1622.0	2271.4	3682.1	3783.0	3884.2	—	—	—
150	608.7	975.6	1491.4	2625.4	3672.5	5954.2	6117.4	6281.0	—	—	—
200	1252.3	2003.3	3056.0	5382.5	7530.4	12195.3	12529.7	12864.8	—	—	—
250	2271.0	3625.9	5539.9	9733.7	13619.6	22028.2	22632.2	23237.5	—	—	—
300	3640.5	5813.5	8873.4	15568.9	21787.1	35239.7	36206.0	37174.3	—	—	—

Notes:

1. Table capacities are in kilowatts of refrigeration resulting in a line friction loss per unit equivalent pipe length, with corresponding change in saturation temperature, where

$$\Delta p = \text{pressure drop due to line friction, Pa/m}$$

$$\Delta t = \text{change in saturation temperature, K/m}$$

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

$$\text{Line capacity} = \text{Table capacity}\left(\frac{\text{Table } L_e}{\text{Actual } L_e} \times \frac{\text{Actual } \Delta t}{\text{Table } \Delta t}\right)^{0.51}$$

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

$$\Delta t = \text{Table } \Delta t \,\frac{\text{Actual } L_e}{\text{Table } L_e}\left(\frac{\text{Actual capacity}}{\text{Table capacity}}\right)^{1.96}$$

4. Values in the table are based on 30°C condensing temperature. Multiply table capacities by the following factors for other condensing temperatures:

Condensing temperature, °C	Suction lines	Discharge lines
20	1.04	0.86
30	1.00	1.00
40	0.96	1.24
50	0.91	1.43

5. Discharge and liquid line capacities are based on −5°C suction.

TABLE C9.8 (IP) Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 134a (Single- or High-Stage Applications)

Line size	Suction lines (Δt = 2°F/100 ft)					Discharge lines (Δt = 1°F, Δp = 2.2 psi/100 ft)			Line size	Liquid lines	
	Saturated suction temperature, °F					Saturated suction temperature, °F				See notes a and b	
	0	10	20	30	40						
Type L copper OD	Corresponding Δp, psi/100 ft								Type L Copper, OD		
	1.00	1.19	1.41	1.66	1.93	0	20	40		Vel. = 100 fpm	Δt = 1°F Δp = 2.2
½	0.14	0.18	0.23	0.29	0.35	0.54	0.57	0.59	½	2.13	2.79
⅝	0.27	0.34	0.43	0.54	0.66	1.01	1.07	1.12	⅝	3.42	5.27
⅞	0.71	0.91	1.14	1.42	1.75	2.67	2.81	2.94	⅞	7.09	14.00
1⅛	1.45	1.84	2.32	2.88	3.54	5.40	5.68	5.95	1⅛	12.10	28.40
1⅜	2.53	3.22	4.04	5.02	6.17	9.42	9.91	10.40	1⅜	18.40	50.00
1⅝	4.02	5.10	6.39	7.94	9.77	14.90	15.70	16.40	1⅝	26.10	78.60
2⅛	8.34	10.60	13.30	16.50	20.20	30.80	32.40	34.00	2⅛	45.30	163.00
2⅝	14.80	18.80	23.50	29.10	35.80	54.40	57.20	59.90	2⅝	69.90	290.00
3⅛	23.70	30.00	37.50	46.40	57.10	86.70	91.20	95.50	3⅛	100.00	462.00
3⅝	35.10	44.60	55.80	69.10	84.80	129.00	135.00	142.00	3⅝	135.00	688.00
4⅛	49.60	62.90	78.70	97.40	119.43	181.00	191.00	200.00	4⅛	175.00	971.00
5⅛	88.90	113.00	141.00	174.00	213.00	323.00	340.00	356.00	—	—	—
6⅛	143.00	181.00	226.00	280.00	342.00	518.00	545.00	571.00	—	—	—

TABLE C9.8 (IP) Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 134a (Single- or High-Stage Applications) *(Continued)*

Line size		Suction lines (Δt = 2°F/100 ft)					Discharge lines (Δt = 1°F, Δp = 2.2 psi/100 ft)			Line size		Liquid lines	
		Saturated suction temperature, °F					Saturated suction temperature, °F					See notes a and b	
		0	10	20	30	40							
		Corresponding Δp, psi/100 ft											
OD		1.00	1.19	1.41	1.66	1.93	0	20	40	OD		Vel. = 100 fpm	Δt = 1°F Δp = 2.2
Steel										Steel			
NPS	SCH									NPS	SCH		
½	80	0.22	0.28	0.35	0.43	0.53	0.79	0.84	0.88	½	80	3.43	4.38
¾	80	0.51	0.64	0.79	0.98	1.19	1.79	1.88	1.97	¾	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
1¼	40	2.62	3.30	4.09	5.03	6.12	9.20	9.68	10.10	1¼	80	18.80	41.80
1½	40	3.94	4.95	6.14	7.54	9.18	13.80	14.50	15.20	1½	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
2½	40	12.10	15.20	18.90	23.10	28.20	42.40	44.60	46.70	2½	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 due to 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

$$\text{Line capacity} = \text{Table capacity}\left(\frac{\text{Table } L_e}{\text{Actual } L_e} \times \frac{\text{Actual } \Delta t}{\text{Table } \Delta t}\right)^{0.541} \text{(0.51 for steel)}$$

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

$$\Delta t = \text{Table } \Delta t\left(\frac{\text{Actual } L_e}{\text{Table } L_e}\right)\left(\frac{\text{Actual capacity}}{\text{Table capacity}}\right)^{1.85} \text{(1.96 for steel)}$$

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

[b] The line pressure drop Δp is conservative; if subcooling is substantial or the 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 C9.8M (SI) Suction, Discharge, and Liquid Line Capacities in Kilowatts for Refrigerant 134a (Single- or High-Stage Applications)

Nominal line size, Type L copper O.D. mm	Suction lines (Δt = 0.04 K/m)					Discharge lines (Δt = 0.02 K/m, Δp = 538 Pa/m)			Liquid lines	
	Saturated suction temperature at Corresponding Δp, Pa/m					Saturated suction temperature			See notes a and b	
	−10°C Δp = 318	−5°C Δp = 368	0°C Δp = 425	5°C Δp = 487	10°C Δp = 555	−10°C	0°C	10°C	Vel. = 0.5 m/s	Δt = 0.02 K/m Δp = 538 Pa/m
12	0.62	0.76	0.92	1.11	1.33	1.69	1.77	1.84	6.51	8.50
15	1.18	1.45	1.76	2.12	2.54	3.23	3.37	3.51	10.60	16.30
18	2.06	2.52	3.60	3.69	4.42	5.61	5.85	6.09	16.00	28.40
22	3.64	4.45	5.40	6.50	7.77	9.87	10.30	10.70	24.50	50.10
28	7.19	8.80	10.70	12.80	15.30	19.50	20.30	21.10	41.00	99.50
35	13.20	16.10	19.50	23.50	28.10	35.60	37.20	38.70	64.90	183.00
42	21.90	26.80	32.40	39.00	46.50	59.00	61.60	64.10	95.20	304.00
54	43.60	53.20	64.40	77.30	92.20	117.00	122.00	127.00	160.00	605.00
67	77.70	94.60	115.00	138.00	164.00	208.00	217.00	226.00	248.00	1080.00
79	120.00	147.00	177.00	213.00	253.00	321.00	335.00	349.00	346.00	1670.00
105	257.00	313.00	379.00	454.00	541.00	686.00	715.00	744.00	618.00	3580.00

TABLE C9.8M (SI) Suction, Discharge, and Liquid Line Capacities in Kilowatts for Refrigerant 134a (Single- or High-Stage Applications) *(Continued)*

Nominal line size,	Suction lines (Δt = 0.04 K/m)					Discharge lines (Δt = 0.02 K/m, Δp = 538 Pa/m)			Liquid lines	
	Saturated suction temperature at Corresponding Δp, Pa/m					Saturated suction temperature			See notes a and b	
O.D. mm	−10°C Δp = 318	−5°C Δp = 368	0°C Δp = 425	5°C Δp = 487	10°C Δp = 555	−10°C	0°C	10°C	Vel. = 0.5 m/s	Δt = 0.02 K/m Δp = 538 Pa/m
Steel NPS										
10	0.87	1.06	1.27	1.52	1.80	2.28	2.38	2.47	9.81	12.30
15	1.62	1.96	2.36	2.81	3.34	4.22	4.40	4.58	15.60	22.80
20	3.41	4.13	4.97	5.93	7.02	8.88	9.26	9.64	27.40	48.20
25	6.45	7.81	9.37	11.20	13.30	16.70	17.50	18.20	44.40	91.00
32	13.30	16.10	19.40	23.10	27.40	34.60	36.10	37.50	76.90	188.00
40	20.00	24.20	29.10	34.60	41.00	51.90	54.10	56.30	105.00	283.00
50	38.60	46.70	56.00	66.80	79.10	100.00	104.00	108.00	173.00	546.00
65	61.50	74.30	89.30	106.00	126.00	159.00	166.00	173.00	246.00	871.00
80	109.00	131.00	158.00	288.00	223.00	281.00	294.00	306.00	380.00	1540.00
100	222.00	268.00	322.00	383.00	454.00	573.00	598.00	622.00	655.00	3140.00

Notes:

1. Table capacities are in kilowatts of refrigeration.

Δp = pressure drop per equivalent length of line, Pa/m
Δt = corresponding change in saturation temperature, K/m

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

$$\text{Line capacity} = \text{Table capacity}\left(\frac{\text{Table } L_e}{\text{Actual } L_e} \times \frac{\text{Actual } \Delta t}{\text{Table } \Delta t}\right)^{0.541} \text{(0.51 for steel)}$$

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

$$\Delta t = \text{Table } \Delta t\left(\frac{\text{Actual } L_e}{\text{Table } L_e}\right)\left(\frac{\text{Actual capacity}}{\text{Table capacity}}\right)^{1.85} \text{(1.96 for steel)}$$

4. Values in the table are based on 40°C condensing temperature. Multiply table capacities by the following factors for other condensing temperatures.

Condensing temperature, °C	Suction line	Discharge line
20	1.239	0.682
30	1.120	0.856
40	1.0	1.0
50	0.888	1.110

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

[b] The line pressure drop Δp is conservative; if subcooling is substantial or the 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 C9.9 (IP) Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 22 (Single- or High-Stage Applications)

Line size	Suction lines (Δt = 2°F/100 ft)					Discharge lines (Δt = 1°F, Δp = 3.05 psi)		Line size	Liquid lines	
	Saturated suction temperature, °F					Saturated suction temperature, °F			See notes a and b	
	−40	−20	0	20	40					
Type L copper OD	Corresponding Δp, psi/100 ft							Type L Copper, OD		
	0.79	1.15	1.6	2.22	2.91	−40	40		Vel. = 100 fpm	Δt = 1°F Δp = 3.05
½	—	—	—	0.40	0.6	0.75	0.85	½	2.3	3.6
⅝	—	0.32	0.51	0.76	1.1	1.4	1.6	⅝	3.7	6.7
⅞	0.52	0.86	1.3	2.0	2.9	3.7	4.2	⅞	7.8	18.2
1⅛	1.1	1.7	2.7	4.0	5.8	7.5	8.5	1⅛	13.2	37.0
1⅜	1.9	3.1	4.7	7.0	10.1	13.1	14.8	1⅜	20.2	64.7
1⅝	3.0	4.8	7.5	11.1	16.0	20.7	23.4	1⅝	28.5	102.5
2⅛	6.2	10.0	15.6	23.1	33.1	42.8	48.5	2⅛	49.6	213.0
2⅝	10.9	17.8	27.5	40.8	58.3	75.4	85.4	2⅝	76.5	376.9
3⅛	17.5	28.4	44.0	65.0	92.9	120.2	136.2	3⅛	109.2	601.5
3⅝	26.0	42.3	65.4	96.6	137.8	178.4	202.1	3⅝	147.8	895.7
4⅛	36.8	59.6	92.2	136.3	194.3	251.1	284.4	4⅛	192.1	1263.2

TABLE C9.9 (IP) Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 22 (Single- or High-Stage Applications) *(Continued)*

Line size		Suction lines (Δt = 2°F/100 ft)					Discharge lines (Δt = 1°F, Δp = 3.05 psi)		Line size		Liquid lines	
		Saturated suction temperature, °F					Saturated suction temperature, °F				See notes a and b	
		−40	−20	0	20	40						
		Corresponding Δp, psi/100 ft										
OD		0.79	1.15	1.6	2.22	2.91	−40	40	OD		Vel. = 100 fpm	Δt = 1°F Δp = 3.05
Steel									Steel			
NPS	SCH								IPS	SCH		
½	40	—	0.38	0.58	0.85	1.2	1.5	1.7	½	80	3.8	5.7
¾	40	0.50	0.8	1.2	1.8	2.5	3.3	3.7	¾	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
1¼	40	2.0	3.2	4.8	7.0	9.9	12.6	14.3	1¼	80	20.6	54.1
1½	40	3.0	4.7	7.2	10.5	14.8	19.0	21.5	1½	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
2½	40	9.2	14.6	22.1	32.2	45.4	58.1	65.9	2½	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 due to 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

$$\text{Line capacity} = \text{Table capacity} \left(\frac{\text{Table } L_e}{\text{Actual } L_e} \times \frac{\text{Actual } \Delta t}{\text{Table } \Delta t} \right)^{0.541} \quad (0.51 \text{ for steel})$$

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

$$\Delta t = \text{Table } \Delta t \left(\frac{\text{Actual } L_e}{\text{Table } L_e} \right) \left(\frac{\text{Actual capacity}}{\text{Table capacity}} \right)^{1.85} \quad (1.96 \text{ for steel})$$

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

[b] The line pressure drop Δp is conservative; if subcooling is substantial or the 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 C9.9M (SI) Suction, Discharge, and Liquid Line Capacities in Kilowatts for Refrigerant 22 (Single- or High-Stage Applications)

Nominal line size, mm	Suction lines (Δt = 0.04 K/m)					Discharge lines (Δt = 0.02 K/m, Δp = 74.90)			Liquid lines	
	Saturated suction temperature, °C					Saturated suction temp., °C				
	−40	−30	−20	−5	+5	−40	−20	+5	Velocity = 0.5 m/s[a]	Δt = 0.02 K/m Δp = 74.9 Pa/m
	Corresponding Δp, Pa/m									
	196	277	378	572	731					
O.D.	Copper line									
12	0.32	0.50	0.75	1.28	1.76	2.30	2.44	2.60	7.08	11.24
15	0.61	0.95	1.43	2.45	3.37	4.37	4.65	4.95	11.49	21.54
18	1.06	1.66	2.49	4.26	5.85	7.59	8.06	8.59	17.41	37.49
22	1.88	2.93	4.39	7.51	10.31	13.32	14.15	15.07	26.66	66.18
28	3.73	5.82	8.71	14.83	20.34	26.24	27.89	29.70	44.57	131.0
35	6.87	10.70	15.99	27.22	37.31	48.03	51.05	54.37	70.52	240.07
42	11.44	17.80	26.56	45.17	61.84	79.50	84.52	90.00	103.4	399.3
54	22.81	35.49	52.81	89.69	122.7	157.3	167.2	178.1	174.1	794.2
67	40.81	63.34	94.08	159.5	218.3	279.4	297.0	316.3	269.9	1415.0
79	63.34	98.13	145.9	247.2	337.9	431.3	458.5	488.2	376.5	2190.9
105	136.0	210.3	312.2	527.8	721.9	919.7	977.6	1041.0	672.0	4697.0

TABLE C9.9M (SI) Suction, Discharge, and Liquid Line Capacities in Kilowatts for Refrigerant 22 (Single- or High-Stage Applications) *(Continued)*

Nominal line size, mm	Suction lines (Δt = 0.04 K/m)					Discharge lines (Δt = 0.02 K/m, Δp = 74.90)			Liquid lines	
	Saturated suction temperature, °C					Saturated suction temp., °C			Velocity = 0.5 m/s[a]	Δt = 0.02 K/m Δp = 74.9 Pa/m
	−40	−30	−20	−5	+5	−40	−20	+5		
	Corresponding Δp, Pa/m									
	196	277	378	572	731					
NPS	Steel line									
10	0.47	0.72	1.06	1.78	2.42	3.04	3.23	3.44	10.66	15.96
15	0.88	1.35	1.98	3.30	4.48	5.62	5.97	6.36	16.98	29.62
20	1.86	2.84	4.17	6.95	9.44	11.80	12.55	13.36	29.79	62.55
25	3.52	5.37	7.87	13.11	17.82	22.29	23.70	25.24	48.19	118.2
32	7.31	11.12	16.27	27.11	36.79	46.04	48.94	52.11	83.56	244.4
40	10.98	16.71	24.45	40.67	55.21	68.96	73.31	78.07	113.7	366.6
50	21.21	32.23	47.19	78.51	106.4	132.9	141.3	150.5	187.5	707.5
65	33.84	51.44	75.19	124.8	169.5	211.4	224.7	239.3	267.3	1127.3
80	59.88	90.95	132.8	220.8	299.5	373.6	397.1	422.9	412.7	1991.3
100	122.3	185.6	270.7	450.1	610.6	761.7	809.7	862.2	711.2	4063.2

Notes:

1. Table capacities are in kilowatts of refrigeration.

Δp = pressure drop per unit equivalent length of line, Pa/m
Δt = corresponding change in saturation temperature, K/m

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

$$\text{Line capacity} = \text{Table capacity}\left(\frac{\text{Table } L_e}{\text{Actual } L_e} \times \frac{\text{Actual } \Delta t}{\text{Table } \Delta t}\right)^{0.541} \quad \text{(0.51 for steel)}$$

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

$$\Delta t = \text{Table } \Delta t \left(\frac{\text{Actual } L_e}{\text{Table } L_e}\right)\left(\frac{\text{Actual capacity}}{\text{Table capacity}}\right)^{1.85} \quad \text{(1.96 for steel)}$$

4. Values in the table are based on 40°C condensing temperature. Multiply table capacities by the following factors for other condensing temperatures.

Condensing temp., °C	Suction line	Discharge line
20	1.18	0.80
30	1.10	0.88
40	1.00	1.00
50	0.91	1.11

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

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

Discharge Lines

In Figure C9.1*a*, the condenser pressure between points 4 and 5 is established by the temperature of the cooling medium. A pressure loss in the discharge line, shown between points 3 and 4, increases the pressure ratio in the compressor, shown between points 2 and 3. With higher discharge pressures, the compressor must pump against a higher pressure differential. This increases the power required, reduces the volumetric efficiency, and reduces the capacity of the system. Similarly as with suction lines, the sizing of oil return lines for vertical risers with halocarbon refrigerants is discussed later in connection with Table C9.14 for discharge (hot gas) risers.

It is desirable to not exceed 2.0°F (1.1°C) equivalent loss in the discharge line. Greater equivalent pressure drops are tolerated in large systems. The discharge lines are listed for capacities at 1°F per 100 feet (0.02 K/m) in Tables C9.7, C9.8, and C9.9*a*. The information in the notes is useful in adjusting the line capacities and pressure drops to other conditions.

Liquid Lines

When a system is equipped with a refrigerant receiver, the liquid line entering the receiver is usually sized generously to assure free flow from the condenser. An allowable velocity of 100 ft/min (0.5 m/s) is typical. However, Table C9.15 provides a rational sizing of condenser drain lines when the line also provides equalization between the receiver and condenser. Velocities of 150 percent of Table C9.15 are used when a separate equalizing line, from the top of the receiver to the condenser inlet, allows the condenser drain line to run full of liquid.

Piping for condensers is a complex subject, requiring consideration of pressure drop, trapping of individual circuits, subcooling of liquid, provision for purging of air, and accommodation of temperature differences between the receiver and the condensers, among other issues. A single condenser coil can be connected to the receiver without an equalizer, and with no liquid head *H*, provided the drain line is sized in accord with Table C9.15 capacities so as to allow vapor flow from the receiver back to the condenser.

Multiple condensers can be connected in series, but are usually connected as shown in Fig. C9.4. This parallel arrangement requires that the condenser coils be of equal size, that the units operate together, and that the receiver be located in an ambient temperature that is equal to or lower than the inlet air temperature to the condenser. Some liquid subcooling occurs, as the liquid tends to hang up in the lower tubes. The amount of liquid head *H* under these restraints is only that which will induce flow. If the coil sizes are unequal or do not always operate together, or if the receiver is not located such that it is below the ambient temperature of the inlet air to the condenser, additional height *H* is needed to make up for differences in coil pressure drops at full load, or to overcome the pressure equivalent of the temperature difference between the receiver and the condenser.

Two methods of piping parallel evaporative condensers are illustrated in Figs. C9.5 and C9.6. Both methods require equalizer lines from the receiver to the condenser inlet piping. When adequate equalizer lines are used, the drain lines can be sized for 150 percent of the flow suggested in Table C9.15. One advantage of the surge-type hookup in Fig. C9.5 is that subcooling of liquid accomplished in the condenser is delivered directly to the liquid line feeding the evaporators. This can be a measurable increase in efficiency, since the heating of liquid in a warm receiver

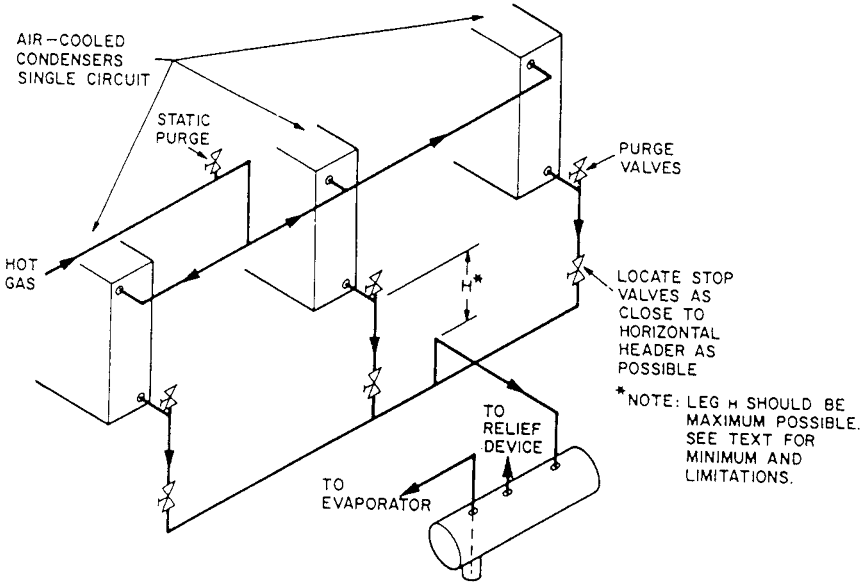


FIGURE C9.4 Multiple air-cooled condensers.

is bypassed. Another feature of the surge-type arrangement is that the individual drop legs are isolated from each other by the liquid trapped at the liquid level in the receiver. Also, as the liquid rises in the drain leg to accomodate increased load and reduced pressure at the coil outlet, the receiver can usually provide plenty of liquid to fill the individual drop legs. When coils are not of equal size or when some condensers may be shut off, the static pressure of the liquid height in the drains of the working coils equalizes the pressure drop across all of the coils and provides individual pumping force to drain each condenser coil. With through-type receivers, shown in Fig. C9.6, the horizontal portion of the drain line should be limited to the flow in Table C9.15, as the liquid volume in the horizontal line should provide for filling the individual drop legs of the working coils. The liquid head *h* in either case needs to be equal to the differences in pressure drops in the parallel coils under all conditions of load. Typical height *h* with ammonia is 4 to 6 ft (1200 to 1800 mm). Typical height *h* with halocarbons is 8 ft (2400 mm).

The design constraint on liquid line sizes between points 5 and 6 in Fig. C9.1*a*, when the liquid is near to saturation or at the bubble point, is that too much pressure drop will result in liquid flashing to vapor. The object is to deliver liquid to the expansion valve. Excessive pressure drop in high-pressure liquid lines causes formation of vapor bubbles or *flash gas*. This flash gas can severely erode the valve seat or needle in an expansion valve. Also, since vapor occupies a much greater volume than liquid, the remaining liquid will be forced along the pipe at a much higher velocity. The increased velocity causes a greater pressure drop, which also restricts the mass flow. A rise in elevation of the line, or the pressure drop due to pipe friction, may lower the pressure below the *bubble point*. In some cases, a means of cooling the liquid may be necessary to avoid forming flash gas. The liquid cooling can be obtained in an extra circuit in the condenser, or may be provided with

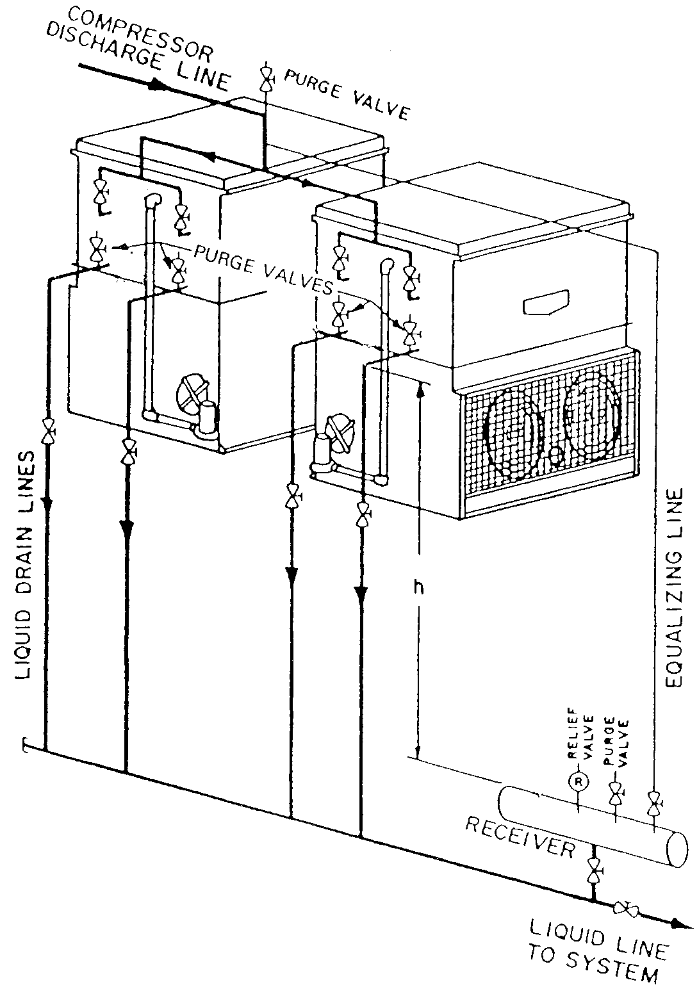


FIGURE C9.5 Piping for parallel condensers with surge-type receiver.

suction/liquid heat exchangers or separately refrigerated liquid subcooling heat exchangers. In condenser arrangements similar to Fig. C9.4, there is some liquid subcooling accomplished in the bottom of the condensers. However, in equalized condenser arrangements as shown in Fig. C9.5, there is little or no liquid subcooling. For receivers located above evaporators, there is little concern about liquid flashing because the liquid gains in static pressure as it flows downward, which provides effective subcooling. For evaporators located above and remote from liquid receivers, the pressure loss in the liquid line must be kept to a very low amount. Alternatively, the liquid may be delivered with pumps as is common in many large piping systems. In Tables C9.7, C9.8, and C9.9, the listed capacities of large liquid lines suffers some overstatement compared to recommended capacities, while the listed

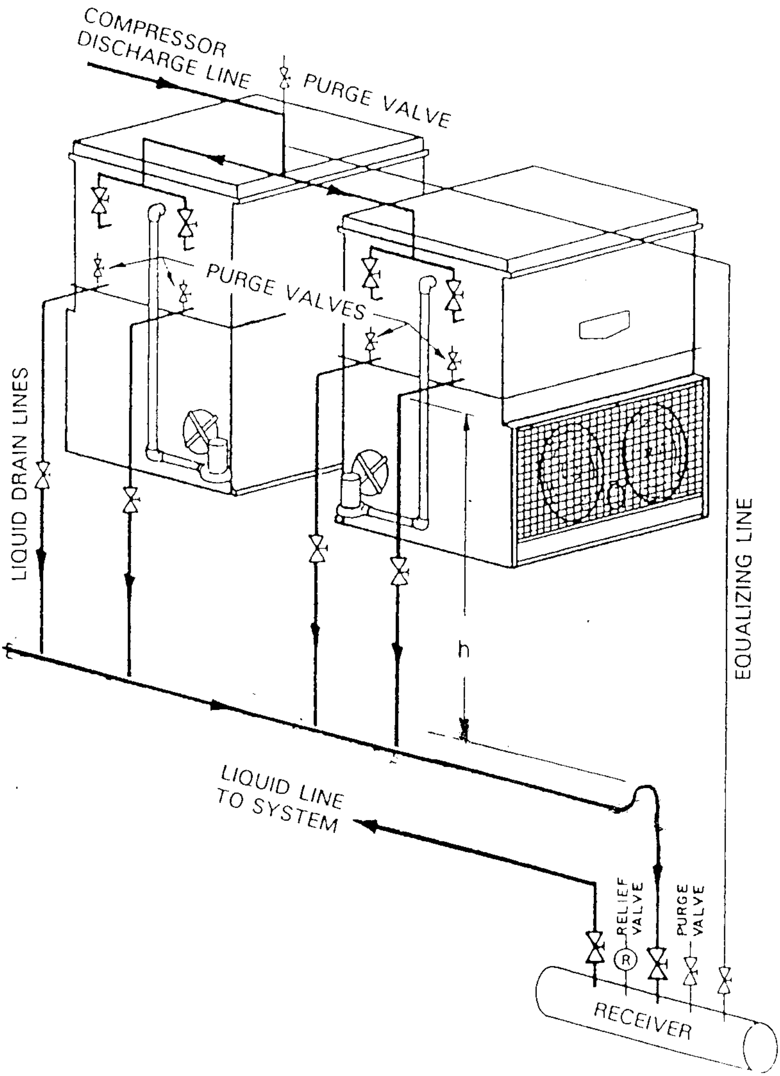


FIGURE C9.6 Piping for parallel condensers with top inlet receiver.

capacities of smaller lines are understated. As a guide for selecting the size of high-pressure liquid lines, the velocity pressure in one velocity head for the natural refrigerants, R-717, R-290, and R-744 should be about 0.055 psi (375 pa). The velocity pressure in one velocity head for high-pressure liquid halocarbon refrigerants should be about 0.11 psi (750 pa). See the definition of velocity pressure per velocity head in the second paragraph under ''Optimum Sizing.''

Pressure Loss in Valves and Fittings

Two methods are used for determining the pressure losses in valves and fittings. The most convenient one to use is the equivalent-length method, which expresses the flow resistance of valves and fittings in terms of length of the same size of pipe. Tables C9.10*a*, *b*, and *c* show equivalent lengths for valves and fittings. The total of the actual run of pipe plus the equivalent lengths for the valves and fittings gives the length used in the calculation of pressure loss.

The flow resistance of pipe and fittings can also be expressed in velocity heads k, where one velocity head is $k = f(L/D)$. This method is more complex, but it may be more accurate. In order to use this method, the length of pipe that has a resistance of one velocity head is determined so that the actual run of pipe can be expressed as a number of velocity heads (k). In order to accurately calculate the number of velocity heads in a length of pipe, the friction factor must either be calculated or determined from a Moody chart. To accurately estimate the number

TABLE C9.10*a* (IP) 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
⅜	17	8	6	6	0.6	5	Globe and vertical lift same as globe valve[d]
½	18	9	7	7	0.7	6	
¾	22	11	9	9	0.9	8	
1	29	15	12	12	1.0	10	
1¼	28	20	15	15	1.5	14	
1½	43	24	18	18	1.8	16	
2	55	30	24	24	2.3	20	
2½	69	35	29	29	2.8	25	
3	84	43	35	35	3.2	30	
3½	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 the in-line, ball-type check valve.

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

TABLE C9.10*aM* (SI) Valve Losses in Equivalent Meters of Pipe

Nominal pipe or tube size, DN	Globe[a]	60° − Y	45° − Y	Angle[a]	Gate[b]	Swing check[c]	Lift check
10	5.2	2.4	1.8	1.8	0.2	1.5	Globe and vertical lift same as globe valve[d]
15	5.5	2.7	2.1	2.1	0.2	1.8	
20	6.7	3.4	2.1	2.1	0.3	2.2	
25	8.8	4.6	3.7	3.7	0.3	3.0	
32	12	6.1	4.6	4.6	0.5	4.3	
40	13	7.3	5.5	5.5	0.5	4.9	
50	17	9.1	7.3	7.3	0.73	6.1	
65	21	11	8.8	8.8	0.9	7.6	
80	26	13	11	11	1.0	9.1	
90	30	15	13	13	1.2	10	
100	37	18	14	14	1.4	12	
125	43	22	18	18	1.8	15	
150	52	27	21	21	2.1	18	Angle lift same as angle valve
200	62	35	26	26	2.7	24	
250	85	44	32	32	3.7	30	
300	98	50	40	40	4.0	37	
350	110	56	47	47	4.6	41	
400	125	64	55	55	5.2	46	
450	140	73	61	61	5.8	50	
500	160	84	72	72	6.7	61	
600	186	98	81	81	7.6	73	

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 150 mm, check with manufacturer.

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

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

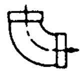
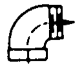
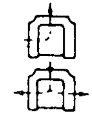
of velocity heads in a length of piping, the Reynolds number and the pipe or tube roughness must be determined. Tables C9.11*a* and *b* show representative velocity heads or *k* factors for valves and fittings.

It should be noted that sizes of copper tubes are referred to by nominal inside diameter in Tables C9.10 and C9.11. In all other capacity tables, the nominal outside diameter is used. This practice is probably carried over from the industry usage of referring to ASTM-280 or AC&R tubing by nominal outside diameters, and referring to other copper tubing used in plumbing by nominal inside diameters.

Example C9.1

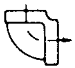
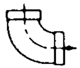
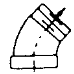
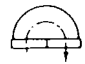
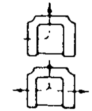
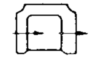
Example of Pipe-Sizing Process. The following example details methods for determining flow rates, volume of flow, line sizing, and pressure-drop for R-22.

TABLE C9.10*b* (IP) Fitting Losses in Equivalent Feet of Pipe (Screwed, Welded, Flanged, Flared, and Brazed Connections)

	Smooth bend elbows						Smooth bend tees			
							Flow-through branch	Straight-through flow		
Nominal pipe or tube size, in	90° Std[a] 	90° Long-radius[b] 	90° Street[a] 	45° Std[a] 	45° Street[a] 	180° Std[a] 		No reduction 	Reduced ¼ 	Reduced ½ 
⅜	1.4	0.9	2.3	0.7	1.1	2.3	2.7	0.9	1.2	1.4
½	1.6	1.0	2.5	0.8	1.3	2.5	3.0	1.0	1.4	1.6
¾	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
1¼	3.3	2.3	5.6	1.7	3.0	5.6	7.0	2.3	3.1	3.3
½	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
2½	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
3½	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

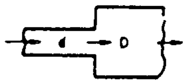
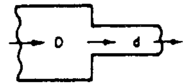
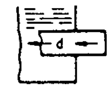
[a] R/D approximately equal to 1.
[b] R/D approximately equal to 1.5.

TABLE C9.10*bM* (SI) Fitting Losses in Equivalent Meters of Pipe (Screwed, Welded, Flanged, Flared, and Brazed Connections)

	Smooth bend elbows						Smooth bend tees			
							Flow-through branch	Straight-through flow		
Nominal pipe or tube size, DN	90° Std[a] 	90° Long-rad.[b] 	90° Street[a] 	45° Std[a] 	45° Street[a] 	180° Std[a] 		No reduction 	Reduced ¼ 	Reduced ½ 
10	0.4	0.3	0.7	0.2	0.3	0.7	0.8	0.3	0.4	0.4
15	0.5	0.3	0.8	0.2	0.4	0.8	0.9	0.3	0.4	0.5
20	0.6	0.4	1.0	0.3	0.5	1.0	1.2	0.4	0.6	0.6
25	0.8	0.5	1.2	0.4	0.6	1.2	1.5	0.5	0.7	0.8
32	1.0	0.7	1.7	0.5	0.9	1.7	2.1	0.7	0.9	1.0
40	1.2	0.8	1.9	0.6	1.0	1.9	2.4	0.8	1.1	1.2
50	1.5	1.0	2.5	0.8	1.4	2.5	3.0	1.0	1.4	1.5
65	1.8	1.2	3.0	1.0	1.6	3.0	3.7	1.2	1.7	1.8
80	2.3	1.5	3.7	1.2	2.0	3.7	4.6	1.5	2.1	2.3
90	2.7	1.8	4.6	1.4	2.2	4.6	5.5	1.8	2.4	2.7
100	3.0	2.0	5.2	1.6	2.6	5.2	6.4	2.0	2.7	3.0
125	4.0	2.5	6.4	2.0	3.4	6.4	7.6	2.5	3.7	4.0
150	4.9	3.0	7.6	2.4	4.0	7.6	9	3.0	4.3	4.9
200	6.1	4.0	—	3.0	—	10	12	4.0	5.5	6.1
250	7.6	4.9	—	4.0	—	13	15	4.9	7.0	7.6
300	9.1	5.8	—	4.9	—	15	18	5.8	7.9	9.1
350	10	7.0	—	5.5	—	17	21	7.0	9.1	10
400	12	7.9	—	6.1	—	19	24	7.9	11	12
450	13	8.8	—	7.0	—	21	26	8.8	12	13
500	15	10	—	7.9	—	25	30	10	13	15
600	18	12	—	9.1	—	29	35	12	15	18

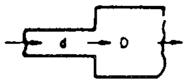
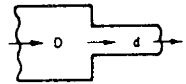
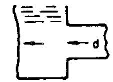
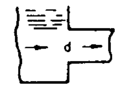
[a] *R*/*D* approximately equal to 1.
[b] *R*/*D* approximately equal to 1.5.

TABLE C9.10c (IP) Special Fitting Losses in Equivalent Feet of Pipe

Nominal pipe or tube size, in	Sudden enlargement, d/D			Sudden contraction, d/D			Sharp edge		Pipe projection	
	¼	½	¾	¼	½	¾	Entrance	Exit	Entrance	Exit
										
⅜	1.4	0.8	0.3	0.7	0.5	0.3	1.5	0.8	1.5	1.1
½	1.8	1.1	0.4	0.9	0.7	0.4	1.8	1.0	1.8	1.5
¾	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
1¼	4.7	3.0	1.0	2.3	1.8	1.0	5.3	2.6	5.3	4.2
1½	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
2½	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
3-½	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*.

TABLE C9.10*cM* (SI) Special Fitting Losses in Equivalent Meters of Pipe

Nominal pipe or tube size, DN	Sudden enlargement, d/D			Sudden contraction, d/D			Sharp edge		Pipe projection	
	¼	½	¾	¼	½	¾	Entrance	Exit	Entrance	Exit
										
10	0.4	0.2	0.1	0.2	0.2	0.1	0.5	0.2	0.5	0.3
15	0.5	0.3	0.1	0.3	0.3	0.1	0.5	0.3	0.5	0.5
20	0.8	0.5	0.2	0.4	0.3	0.2	0.9	0.4	0.9	0.7
25	1.0	0.6	0.2	0.5	0.4	0.2	1.1	0.5	1.1	0.8
32	1.4	0.9	0.3	0.7	0.5	0.3	1.6	0.8	1.6	1.3
40	1.8	1.1	0.4	0.9	0.7	0.4	2.0	1.0	2.0	1.5
50	2.4	1.5	0.5	1.2	0.9	0.5	2.7	1.3	2.7	2.1
65	3.0	1.9	0.6	1.5	1.2	0.6	3.7	1.7	3.7	2.7
80	4.0	2.4	0.8	2.0	1.5	0.8	4.3	2.2	4.3	3.8
90	4.6	2.8	0.9	2.3	1.8	0.9	5.2	2.6	5.2	4.0
100	5.2	3.4	1.2	2.7	2.1	1.2	6.1	3.0	6.1	4.9
125	7.3	4.6	1.5	3.7	2.7	1.5	8.2	4.3	8.2	6.1
150	8.8	6.7	1.8	4.6	3.4	1.8	10	5.8	10	7.6
200	—	7.6	2.6	—	4.6	2.6	14	7.3	14	10
250	—	9.8	3.4	—	6.1	3.4	18	8.8	18	14
300	—	12.4	4.0	—	7.6	4.0	22	11	22	17
350	—	—	4.9	—	—	4.9	26	14	26	20
400	—	—	5.5	—	—	5.5	29	15	29	23
450	—	—	6.1	—	—	6.1	35	18	35	27
500	—	—	—	—	—	—	43	21	43	33
600	—	—	—	—	—	—	50	25	50	40

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

TABLE C9.11*a* (IP) *K* Factors—Screwed Pipe Fittings

Nominal pipe size (NPS)	90° ell reg.	90° ell long	45° ell	Return bend	Tee-line	Tee-branch	Globe valve	Gate valve	Angle valve	Swing check valve	Bell mouth inlet	Square inlet	Projected inlet
⅜	2.5	—	0.38	2.5	0.90	2.7	20	0.40	—	8.0	0.05	0.5	1.0
½	2.1	—	0.37	2.1	0.90	2.4	14	0.33	—	5.5	0.05	0.5	1.0
¾	1.7	0.92	0.35	1.7	0.90	2.1	10	0.28	6.1	3.7	0.05	0.5	1.0
1	1.5	0.78	0.34	1.5	0.90	1.8	9	0.24	4.6	3.0	0.05	0.5	1.0
1¼	1.3	0.65	0.33	1.3	0.90	1.7	8.5	0.22	3.6	2.7	0.05	0.5	1.0
1½	1.2	0.54	0.32	1.2	0.90	1.6	8	0.19	2.9	2.5	0.05	0.5	1.0
2	1.0	0.42	0.31	1.0	0.90	1.4	7	0.17	2.1	2.3	0.05	0.5	1.0
2½	0.85	0.35	0.30	1.85	0.90	1.3	6.5	0.16	1.6	2.2	0.05	0.5	1.0
3	0.80	0.31	0.29	1.80	0.90	1.2	6	0.14	1.3	2.1	0.05	0.5	1.0
4	0.70	0.24	0.28	0.70	0.90	1.1	5.7	0.12	1.0	2.0	0.05	0.5	1.0

Source: *Engineering Data Book* (HI 1979).

TABLE C9.11*b* *K* Factors—Flanged Welded Pipe Fittings

Nominal pipe size (NPS)	90° Ell reg.	90° Ell long	45° Ell long	Return bend reg.	Return bend long	Tee-line	Tee-branch	Glove valve	Gate valve	Angle valve	Swing check valve
1	0.43	0.41	0.22	0.43	0.43	0.26	1.0	13	—	4.8	2.0
1¼	0.41	0.37	0.22	0.41	0.38	0.25	0.95	12	—	3.7	2.0
1½	0.40	0.35	0.21	0.40	0.35	0.23	0.90	10	—	3.0	2.0
2	0.38	0.30	0.20	0.38	0.30	0.20	0.84	9	0.34	2.5	2.0
2½	0.35	0.28	0.19	0.35	0.27	0.18	0.79	8	0.27	2.3	2.0
3	0.34	0.25	0.18	0.34	0.25	0.17	0.76	7	0.22	2.2	2.0
4	0.31	0.22	0.18	0.31	0.22	0.15	0.70	6.5	0.16	2.1	2.0
6	0.29	0.18	0.17	0.29	0.18	0.12	0.62	6	0.10	2.1	2.0
8	0.27	0.16	0.17	0.27	0.15	0.10	0.58	5.7	0.08	2.1	2.0
10	0.25	0.14	0.16	0.25	0.14	0.09	0.53	5.7	0.06	2.1	2.0
12	0.24	0.13	0.16	0.24	0.13	0.08	0.50	5.7	0.05	2.1	2.0

Source: *Engineering Data Book* (HI 1979).

TABLE C9.11*a*(M) (SI) *K* Factors—Screwed Pipe Fittings

Nominal pipe size, DN	90° Ell reg.	90° Ell long	45° Ell	Return bend	Tee-line	Tee-branch	Globe valve	Gate valve	Angle valve	Swing check valve	Bell mouth inlet	Square inlet	Projected inlet
10	2.5	—	0.38	2.5	0.90	2.7	20	0.40	—	8.0	0.05	0.5	1.0
15	2.1	—	0.37	2.1	0.90	2.4	14	0.33	—	5.5	0.05	0.5	1.0
20	1.7	0.92	0.35	1.7	0.90	2.1	10	0.28	6.1	3.7	0.05	0.5	1.0
25	1.5	0.78	0.34	1.5	0.90	1.8	9	0.24	4.6	3.0	0.05	0.5	1.0
32	1.3	0.65	0.33	1.3	0.90	1.7	8.5	0.22	3.6	2.7	0.05	0.5	1.0
40	1.2	0.54	0.32	1.2	0.90	1.6	8	0.19	2.9	2.5	0.05	0.5	1.0
50	1.0	0.42	0.31	1.0	0.90	1.4	7	0.17	2.1	2.3	0.05	0.5	1.0
65	0.85	0.35	0.30	0.85	0.90	1.3	6.5	0.16	1.6	2.2	0.05	0.5	1.0
80	0.80	0.31	0.29	0.80	0.90	1.2	6	0.14	1.3	2.1	0.05	0.5	1.0
100	0.70	0.24	0.28	0.70	0.90	1.1	5.7	0.12	1.0	2.0	0.05	0.5	1.0

Source: *Engineering Data Book* (HI 1979).

TABLE C9.11*b*(M) (SI) *K* Factors—Flanged Welded Pipe Fittings

Nominal pipe size, DN	90° Ell reg.	90° Ell long	45° Ell long	Return bend reg.	Return bend long	Tee-line	Tee-branch	Globe valve	Gate valve	Angle valve	Swing check valve
25	0.43	0.41	0.22	0.43	0.43	0.26	1.0	13	—	4.8	2.0
32	0.41	0.37	0.22	0.41	0.38	0.25	0.95	12	—	3.7	2.0
40	0.40	0.35	0.21	0.40	0.35	0.23	0.90	10	—	3.0	2.0
50	0.38	0.30	0.20	0.38	0.30	0.20	0.84	9	0.34	2.5	2.0
65	0.35	0.28	0.19	0.35	0.27	0.18	0.79	8	0.27	2.3	2.0
80	0.34	0.25	0.18	0.34	0.25	0.17	0.76	7	0.22	2.2	2.0
100	0.31	0.22	0.18	0.31	0.22	0.15	0.70	6.5	0.16	2.1	2.0
150	0.29	0.18	0.17	0.29	0.18	0.12	0.62	6	0.10	2.1	2.0
200	0.27	0.16	0.17	0.27	0.15	0.10	0.58	5.7	0.08	2.1	2.0
250	0.25	0.14	0.16	0.25	0.14	0.09	0.53	5.7	0.06	2.1	2.0
300	0.24	0.13	0.16	0.24	0.13	0.08	0.50	5.7	0.05	2.1	2.0

Source: *Engineering Data Book* (HI 1979).

Given:

A refrigeration system uses refrigerant R-22 with a capacity of 100 tons (352 kW) at an evaporating temperature of 23°F (−5°C) and a water-cooled condensing temperature of 104° (40°C). The suction line is 82 ft (25 m) long, with two angle valves, one swing check valve, and three elbows. The discharge line is 98 ft (30 m) long, with one globe valve, one lift check valve, and three elbows. The liquid line is 180 ft (55 m) long, with two angle valves and three elbows.

From Table C9.1, the refrigerant flow rate is 2.96 lb/min per ton (6.36×10^{-3} kg/s per kW)

Flow for 100 tons $100 \times 2.96 = 296$ lbm/min ($352 \times 6.36 \times 10^{-3} = 2.24$ kg/s)

Suction specific volume at 23°F (−5°C) from table of thermophysical properties: $V_g = 0.8856$ lbs/ft^3 (14.19 kg/m^3).

Actual suction volume: $296/0.8856 = 334$ cfm ($2.24/14.19 = 0.158$ m^3/s)

Approximate discharge volume from Eq. C9.3:

Absolute pressure of R-22 @ +104°F from Table C9.2*a* (IP) = 222.5 psia

Absolute pressure of R-22 @ 23°F from Table C9.2a (IP) = 61.21 psia

Approximate discharge volume = suction volume $\times P_1/P_2 \times 1.2$

$262 \times (61.21/222.5) \times 1.2 = 86.5$ cfm

(Absolute pressure of R-22 @ +40°C from Table C9.2*a* SI = 1534 kPa)

(Absolute pressure of R-22 @ −5°C from Table C9.2*a* SI = 422 kPa)

(Approximate discharge volume m^3/kW = suction volume $\times P_1/P_2 \times 1.2$)

($0.124 \times 422/1545 \times 1.2 = 0.0406$ m^3/s)

Liquid density at 104°F (+40°C) from table of thermophysical properties is: 70.44 lb/ft^3 (1.128 kg/m^3)

Actual liquid flow volume = mass flow/liquid density

$296/70.44 = 4.2$ cfm ($2.24/1{,}128 = 1.98$ dm^3/s)

Suction Line. For the suction line, try the nominal NPS 4 steel pipe: (See Tables C9.10a and b; note that the IP and SI valve and fitting losses are not exactly equivalent because of rounding.)

Length of run	=	82 ft	(25 m)
Two angle valves	=	94 ft	(28 m)
One swing check	=	40 ft	(12 m)
Three elbows	=	20.1 ft	(6 m)
Total equivalent length	=	236.1 ft	(71 m)

Suction line temperature and pressure drop using Note 2 under Table C9.9

Interpolate tons capacity at + 23°F . . . . . . ~ 123 tons (−5°C at 433 kW)

$$\Delta t = 2 \times 236.1/100 \times (100/123)^{1.96}$$

$$= 3.15°\text{F} \qquad (1.74°C)$$

Interpolate pressure drop at + 23°F . . . . . . ~ 2.32 psi (16 kPa)

$$\Delta p = 2.32 \times 236.1/100 \times (100/123)$$

$$= 4.33 \text{ psi} \qquad (30.68 \text{ kPa})$$

The NPS 4 steel pipe is satisfactory for the pressure drop. The rated pressures from Table C9.20 for ERW pipe is $732 \times 0.85 = 622$ psig (4277 kPa), while the suggested test pressure for R22 in Table C9.19 for the low side is 182 psig (1255 kPa). Either seamless or ERW NPS 4 steel pipe is satisfactory for the required capacity and pressure.

Discharge Line. For the discharge line, try the 3 IPS steel pipe. (See Tables C9.10*a* and *b*, note that the IP and SI equivalent lengths are not exactly equal because of rounding.)

Length of line	= 98 ft	(30 m)
One globe valve	= 69 ft	(21 m)
One lift check	= 69 ft	(21 m)
Three long radius elbows	= 15 ft	(4.6 m)
Total equivalent length	= 251 ft	(76.6 m)

Using Table C9.9: The capacity of a NPS 3 pipe is 116.4 tons @ 1°F temperature drop.

$$\Delta t = 1.0(251/100)(100/116.4)^{1.96}$$

$$= 1.86°\text{F} \qquad (1.03°\text{C})$$

$$\Delta p = 3.05 \times (244.1/100)(100/116.4)^{1.96}$$

$$= 5.6 \text{ psig} \qquad (39.2 \text{ kPa})$$

The 3 IPS steel pipe is satisfactory for pressure drop. For strength, Table C9.20 shows the pressure rating as 664 psig (4564 kPa) for ERW schedule 40 pipe. Table C9.19 shows a suggested minimum field test pressure of 235 psig, (1628 kPa) while Table C9.18 shows the ANSI minimum design pressure as 208 psig (1433 kPa). Actual U.S. industry practice is to use 250 psig (1724 kPa) design pressure for water-cooled R-22 systems, and 300 psig (2068 kPa) or higher for air-cooled R-22 systems.

Liquid Lines. For the condenser drain, use 100 fpm (0.5 m/s) for sizing, when vapor from the receiver may flow in opposition with the free-draining liquid. Table C9.9 indicates the NPS 3 (DN 80) has a capacity of 118.5 tons at 100 fpm (412.7 kPa @ 0.5 m/s). When an equalizer line is used to vent the receiver to the top of the condenser, a velocity of 150 fpm (0.75 m/s) may be used for full liquid flow in the drain. A NPS 2½ (DN 65) has a capacity of 115 tons at 150 fpm (401 kPa at 0.75 m/s).

For the liquid line to the evaporator, try a NPS 2 steel pipe. From Tables C9.10*a* IP and C9.10*b* IP:

Line length	= 180 ft	(55 m)
Two angle valves	= 48 ft	(14.6 m)
Three elbows	= 9.9 ft	(3.0 m)
Total equivalent length	= 237.9 ft	(72.6 m)

From the receiver to the evaporator, check the equivalent temperature drop in a NPS 2 steel liquid line. Table C9.9 indicates a 1°F temperature drop per 100 feet of NPS 2 line (0.02°K/m in DN50.)

Temperature and pressure drop in NPS 2 liquid line for 100 tons (352 kPa in DN50):

$$\Delta t = 1.0 \times 237.9/100 \times (100/192)^{1.96}$$
$$= 0.662°\text{F} \qquad (0.368°\text{C})$$
$$\Delta p = 3.05 \times 237.9/100 \times (100/192)^{1.96}$$
$$= 2.02 \text{ psi} \qquad (13.93 \text{ kPa})$$

For the illustrated calculations, the selected line sizes are as follows:

Suction line	NPS 4	(DN 100)
Discharge line	NPS 3	(DN 80)
Liquid line to receiver	NPS 3	(DN 80)
Liquid line to expansion valve	NPS 2	(DN 50)

The compressor selection must be adjusted for the suction and discharge line losses, so the compressor should be selected for the following gauge pressures:

Suction pressure	$= 46.54 - 4.45$	$= 42.09$ psig	(290 kPa)
Discharge pressure	$= 222.56 + 5.68$	$= 228.2$ psig	(1575 kPa)

Secondary Coolants

Piping for secondary coolant service may be designed by the methods used for water.

Consideration must be given to an economical choice of pressure loss and pump power. In evaluation of pressure drop, the density and viscosity have a direct effect. An additional consideration is the compatibility of materials with the fluid.

In the selection of the concentration of water-based secondary coolants, it is desirable to select a mixture which has a freezing point at least 18 to 20°F (10 to 11°C) below the operating temperature expected in the system. Use of lower freezing point mixtures provides a larger margin to prevent freezing. However, this is costly because of reduced heat transfer and increased pumping costs. The methods in Chap. B8 may be applied for sizing of secondary coolant piping. For four common water-based secondary coolants, Table C9.12 in IP shows the specific gravity, freezing point, and viscosity at various concentrations. Table C9.12 in SI shows percent by mass, density, specific heat, and viscosity at a temperature 18°F (10°C) above the freezing point. This information enables determination of the friction factor and the subsequent pressure loss.

Oil Return

The CFC and HCFC and hydrocarbon refrigerants are miscible with mineral oil. These refrigerants carry some oil in the liquid phase without separation, whereas the newer HFC refrigerants require synthetic oil to provide mutual solubility. Some practitioners suggest the use of small additions of R-290 or other hydrocarbons to improve the oil circulation when using conventional mineral oils with HFC

TABLE C9.12 (IP) Secondary Coolant Properties

Concentration %	Calcium chloride Specific gravity*	Calcium chloride Freezing point, F	Calcium chloride Viscosity†	Sodium chloride Specific gravity*	Sodium chloride Freezing point, F	Sodium chloride Viscosity†	Ethylene glycol Specific gravity*	Ethylene glycol Freezing point, F	Ethylene glycol Viscosity†	Propylene glycol Specific gravity*	Propylene glycol Freezing point, F	Propylene glycol Viscosity†
0	1.00	32°	1.27	1.00	32°	1.27	1.00	32°	1.27	1.00	32°	1.27
5	1.044	27.7	1.45	1.035	27.0	1.45	1.005	30	1.50	1.004	29.5	1.60
10	1.087	22.3	1.80	1.072	20.4	1.75	1.012	25.2	1.80	1.009	26.0	2.0
15	1.133	13.5	2.5	1.111	12.0	2.3	1.019	21	2.03	1.014	24.5	2.6
20	1.182	−0.4	4.1	1.15	1.8	3.4	1.026	15.8	2.9	1.018	20.5	3.7
23.3‡	.....	.....	.....	1.175	−6.0	4.7	....	....	....	....	....	...
25	1.233	−21.0	9.7	1.191	16.0	3.0	1.033	10.5	3.7	1.022	15.6	5.2
29.87‡	1.290	−67.0	38.0	.....	....	....	....	....	....	....	....	...
30	1.295	−50.8	25.7	.....	....	....	1.039	3.2	5.0	1.027	10.0	7.7
35	.....	.....	.....	.....	....	....	1.046	−1.0	6.3	1.031	3.5	12.6
40	.....	.....	.....	.....	....	....	1.053	−12.0	11.0	1.035	−4.0	24.0
45	.....	.....	.....	.....	....	....	1.059	−17.0	15.0	1.038	−15.0	45.0
50	.....	.....	.....	.....	....	....	1.066	−33.0	29.0	1.041	−24.0	85.0
55	.....	.....	.....	.....	....	....	1.072	−43.0	55.0	....	....	...

* Specific gravities compared with water at 60°F.
† Viscosities in centipoises at 20°F above freezing point.
‡ Eutectic points.

TABLE C9.12*a* (SI) Secondary Coolant Properties of Calcium Chloride—Water

Freezing point °C	Percent by mass %	Density* kg/m³	Specific heat* J/kg·K	Dyn. viscosity* 10^{-3} Ns/m²
−5	9.0	1078	3640	1.86
−10	14.0	1127	3340	2.47
−15	18.0	1167	3132	3.44
−20	21.0	1200	2990	4.50
−25	24.0	1234	2868	6.28
−30	26.0	1259	2780	8.30
−35	28.0	1282	2712	12.2
−40	29.5	2660	2660	17.4
−45	31.0	1312	2612	28.0

TABLE C9.12*b* (SI) Secondary Coolant Properties of Sodium Chloride—Water

Freezing point °C	Percent mass %	Density* kg/m³	Specific heat* J/kg·K	Dyn. viscosity* 10^{-3} Ns/m²
−5	7.9	1060	3782	1.70
−10	14.1	1109	3545	2.25
−15	18.8	1149	3405	3.10
−20	22.6	1183	3310	4.20

* Properties at 10°C above the freezing point.

refrigerants. Some oil is carried with the discharge gas from most conventional vapor compressors. In halocarbon direct-expansion systems without oil separators, the oil will pass through the system, returning with the suction vapor. In fact, mineral oil in small amounts improves vaporizing heat transfer by reducing the surface tension of the boiling liquid. Oil in greater quantities drastically reduces the vaporizing heat transfer in an evaporator. Reducing the proportion of oil circulating in the system is accomplished by use of synthetic oil, by lower temperatures in the discharge gas to reduce the amount of vaporized oil, and by the use of high-efficiency separating elements.

Oil discharged from the compressor is carried through the discharge line to the condenser and through the liquid line to the receiver. It then proceeds through the liquid line from the receiver to the evaporator, where it will collect in flooded evaporators as it does not vaporize. Various methods are used to bleed a mixture of oil and liquid refrigerant from flooded evaporators into the suction line. Heat exchange between the return suction line and the warm liquid feed line is a common method used to ‘‘dry’’ the returning refrigerant. The cooling of the liquid feed by the evaporation of the returning refrigerant conserves the refrigeration effect. Other methods of oil return involve heating the return mixture in a separate line through heat exchange with the discharge gas. Still other successful methods involve automated refrigerant gas pumping devices such as ejectors or electrically heated oil stills that are intermittantly pressurized or are equipped with pumps to transfer distilled oil back into the suction of the compressor.

After oil is introduced into the suction line by a direct expansion system or by one of the just-noted methods from flooded evaporators, the gas velocity in vertical return risers must be maintained so that the oil will be carried up the riser. The necessary velocities expressed in tons (kW) capacity have been experimentally

TABLE C9.12*c* (SI) Secondary Coolant Properties of Ethylene Glycol—Water

Freezing point °C	Percent mass %	Density* kg/m^3	Specific heat* J/kg·K	Dyn. viscosity* 10^{-3} Ns/m^2
−5	14.0	1020	4005	2.285
−10	23.5	1035	3820	3.525
−15	30.5	1047	3650	5.58
−20	36.0	1058	3440	7.90
−25	41.0	1068	3340	12.4
−30	45.5	1077	3205	18.0
−35	49.5	1084	3082	30.4
−40	53.0	1090	2950	48.0
−45	56.0	1095	2840	120.0

TABLE C9.12*d* Secondary Coolant Properties of Propylene Glycol—Water

Freezing point °C	Percent mass %	Density* kg/m^3	Specific heat* J/kg·K	Dyn. viscosity* 10^{-3} Ns/m^2
−5	15	1014	4082	2.90
−10	25	1025	3975	5.45
−15	33	1036	3858	11.5
−20	39	1045	3738	21.0
−25	44	1051	3625	44.8
−30	48	1058	3530	76.0
−35	51	1063	3440	168.0
−40	54	1067	3365	290.0
−45	57	1070	3288	700.0

* Properties at 10°C above freezing point.

determined. Table C9.13 lists the minimum capacities in copper tubes that will carry oil up suction risers. Table C9.14 lists the minimum capacities in copper tubes that will carry oil up discharge or hot gas risers. Multipliers shown at the bottom of Table C9.14 can be used for sizing oil return risers at various temperatures.

When a system operates over a range of capacities, the riser sized for the full load condition may not be functional for oil return at part loads. Is such cases, a double riser will increase the range of loads that will provide oil return. See Fig. C9.7 for details of a double riser. Lines A and B are sized so the two together will handle the full load with proper oil return in accordance with Tables C9.13 or C9.14, or with equivalent velocities of other refrigerants. Line A should be sized for the minimum capacity.

Since mineral oil has a very limited solubility with liquid ammonia, and is not mutually soluble with vaporized ammonia, it does not travel back to the compressor in dry vapor suction lines. Control of oil in ammonia plants is usually accomplished with efficient oil separators, and manually operated oil drain pots, stills, and pump-out systems. Some degree of automation is successfully used to return oil from evaporators to ammonia accumulators, such as is used in hot-gas defrosting methods. However, the separation of oil in the low side means that the principal method of oil control in ammonia systems is provided by oil stills or oil pots at the accumulators in flooded or circulated system evaporators. Additional automation of oil return is

TABLE C9.13 (IP) Minimum Refrigeration Capacity in Tons for Oil Entrainment up Suction Risers (Type L Copper Tubing)

Refrig-erant	Satu-rated suction temp., °F	Suction gas temp., °F	Pipe OD, in											
			½	⅝	¾	⅞	1-⅛	1-⅜	1-⅝	2-⅛	2-⅝	3-⅛	3-⅝	4-⅛
			Area, in²											
			0.146	0.233	0.348	0.484	0.825	1.256	1.780	3.094	4.770	6.812	9.213	11.970
		−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
22	−40.0	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
		−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
	−20.0	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
		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
	0.0	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
		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
	20.0	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
		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
	40.0	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
		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
134a	0.0	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
		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
	10.0	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

TABLE C9.13 (IP) Minimum Refrigeration Capacity in Tons for Oil Entrainment up Suction Risers (Type L Copper Tubing) *(Continued)*

Refrigerant	Saturated suction temp., °F	Suction gas temp., °F	Pipe OD, in: ½	⅝	¾	⅞	1-⅛	1-⅜	1-⅝	2-⅛	2-⅝	3-⅛	3-⅝	4-⅛
		Area, in²	0.146	0.233	0.348	0.484	0.825	1.256	1.780	3.094	4.770	6.812	9.213	11.970
134a	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
502	−40.0	−30.0	0.051	0.092	0.152	0.230	0.447	0.756	1.17	2.33	4.01	6.26	9.13	12.7
		−10.0	0.053	0.095	0.157	0.237	0.461	0.779	1.21	2.41	4.13	6.45	9.41	13.1
		10.0	0.055	0.098	0.163	0.246	0.476	0.809	1.25	2.50	4.29	6.39	9.76	13.5
	−20.0	−10.0	0.068	0.122	0.201	0.303	0.591	0.999	1.54	3.08	5.30	8.27	12.1	16.7
		10.0	0.070	0.125	0.207	0.312	0.608	1.03	1.59	3.17	5.45	8.51	12.4	17.2
		30.0	0.072	0.129	0.213	0.322	0.627	1.06	1.64	3.27	5.62	8.78	12.8	17.8
	0.0	10.0	0.087	0.157	0.259	0.391	0.761	1.29	1.99	3.97	6.82	10.6	15.5	21.5
		30.0	0.089	0.160	0.264	0.399	0.777	1.31	2.03	4.05	6.96	10.9	15.9	22.0
		50.0	0.092	0.165	0.273	0.412	0.802	1.36	2.10	4.19	7.19	11.2	16.4	22.7
	20.0	30.0	0.110	0.197	0.325	0.491	0.957	1.62	2.50	4.99	8.58	13.4	19.5	27.1
		50.0	0.112	0.201	0.331	0.501	0.975	1.65	2.55	5.09	8.74	13.6	19.9	27.6
		70.0	0.115	0.207	0.342	0.516	1.01	1.70	2.63	5.25	9.02	14.1	20.5	28.5
	40.0	50.0	0.136	0.243	0.401	0.606	1.18	2.00	3.09	6.16	10.6	16.5	24.1	33.4
		70.0	0.138	0.247	0.408	0.616	1.20	2.03	3.14	6.28	10.8	16.8	24.5	34.0
		90.0	0.142	0.254	0.420	0.634	1.23	2.09	3.23	6.44	11.1	17.3	25.2	35.0

Notes:

1. Refrigeration capacity in tons is based on 90°F liquid temperature and superheat as indicated by the listed temperature. For other liquid line temperatures, use correction factors in the table to the right.
2. This table has been computed using an ISO 32 mineral oil for R-22 and R-502. R-134a has been computed using an 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
502	1.24	1.18	1.12	1.06	0.94	0.87	0.81	0.74	0.67

TABLE C9.13 (SI) Minimum Refrigeration Capacity in Kilowatts for Oil Entrainment Up Suction Risers (Copper Tubing, ASTM B88M, Type B, Metric Size)

Refrig-erant	Saturated temp., °C	Suction gas temp., °C	Tubing diameter, nominal OD, mm											
			12	15	18	22	28	35	42	54	67	79	105	130
		−5.0	0.274	0.502	0.844	1.437	2.732	4.848	7.826	15.006	25.957	39.340	81.164	140.509
		5.0	0.245	0.450	0.756	1.287	2.447	4.342	7.010	13.440	23.248	35.235	72.695	125.847
134a	−10.0	15.0	0.238	0.436	0.732	1.247	2.370	4.206	6.790	13.019	22.519	34.129	70.414	121.898
		0.0	0.296	0.543	0.913	1.555	2.956	5.244	8.467	16.234	28.081	42.559	87.806	152.006
		10.0	0.273	0.500	0.840	1.431	2.720	4.827	7.792	14.941	25.843	39.168	80.809	139.894
134a	−5.0	20.0	0.264	0.484	0.813	1.386	2.634	4.674	7.546	14.468	25.026	37.929	78.254	135.471
		10.0	0.357	0.655	1.100	1.874	3.562	6.321	10.204	19.565	33.843	51.292	105.823	183.197
		20.0	0.335	0.615	1.033	1.761	3.347	5.938	9.586	18.380	31.792	48.184	99.412	172.098
134a	5.0	30.0	0.317	0.582	0.978	1.667	3.168	5.621	9.075	17.401	30.099	45.617	94.115	162.929
		15.0	0.393	0.721	1.211	2.063	3.921	6.957	11.232	21.535	37.250	56.456	116.479	201.643
		25.0	0.370	0.679	1.141	1.944	3.695	6.555	10.583	20.291	35.098	53.195	109.749	189.993
134a	10.0	35.0	0.358	0.657	1.104	1.881	3.576	6.345	10.243	19.640	33.971	51.486	106.224	183.891
		−35	0.182	0.334	0.561	0.956	1.817	3.223	5.203	9.977	14.258	26.155	53.963	93.419
		−25	0.173	0.317	0.532	0.907	1.723	3.057	4.936	9.464	16.371	24.811	51.189	88.617
22	−40	−15	0.168	0.307	0.516	0.880	1.672	2.967	4.791	9.185	15.888	24.080	49.681	86.006
		−15	0.287	0.527	0.885	1.508	2.867	5.087	8.213	15.748	27.239	41.283	85.173	147.449
		−5	0.273	0.501	0.841	1.433	2.724	4.834	7.804	14.963	25.882	39.226	80.929	140.102
22	−20	5	0.264	0.485	0.815	1.388	2.638	4.680	7.555	14.487	25.058	37.977	78.353	135.642
		0	0.389	0.713	1.198	2.041	3.879	6.883	11.112	21.306	36.854	55.856	115.240	199.499
		10	0.369	0.676	1.136	1.935	3.678	6.526	10.535	20.200	34.940	52.954	109.254	189.136
22	−5	20	0.354	0.650	1.092	1.861	3.537	6.275	10.131	19.425	33.600	50.924	105.065	181.884
		10	0.470	0.862	1.449	2.468	4.692	8.325	13.441	25.771	44.577	67.560	139.387	241.302
		20	0.440	0.807	1.356	2.311	4.393	7.794	12.582	24.126	41.731	63.246	130.488	225.896
22	5	30	0.422	0.774	1.301	2.217	4.213	7.476	12.069	23.141	40.027	60.665	125.161	216.675

TABLE C9.13 (SI) Minimum Refrigeration Capacity in Kilowatts for Oil Entrainment Up Suction Risers (Copper Tubing, ASTM B88M, Type B, Metric Size) *(Continued)*

Refrig-erant	Saturated temp., °C	Suction gas temp., °C	Tubing diameter, nominal OD, mm											
			12	15	18	22	28	35	42	54	67	79	105	130
502	−40	−35	0.129	0.236	0.397	0.676	1.284	2.279	3.679	7.054	12.201	18.492	38.152	66.048
		−25	0.125	0.229	0.385	0.657	1.248	2.215	3.575	6.855	11.858	17.972	37.079	64.190
		−15	0.121	0.223	0.374	0.638	1.212	2.151	3.472	6.658	11.516	17.453	36.009	62.337
502	−20	−15	0.210	0.385	0.647	1.102	2.096	3.718	6.003	11.510	19.909	30.173	62.253	107.769
		−5	0.204	0.374	0.628	1.070	2.033	3.607	5.823	11.166	19.314	29.272	60.392	104.549
		5	0.198	0.363	0.611	1.041	1.978	3.510	5.666	10.865	18.793	28.482	58.763	101.728
502	5	0	0.288	0.528	0.887	1.510	2.871	5.094	8.224	15.770	27.277	41.341	84.292	147.655
		10	0.279	0.511	0.859	1.464	2.783	4.937	7.970	15.282	26.434	40.063	82.656	143.091
		20	0.271	0.496	0.834	1.421	2.701	4.793	7.737	14.835	25.661	38.891	80.239	138.907
502	5	10	0.347	0.637	1.071	1.824	3.467	6.151	9.931	19.041	32.936	49.917	102.986	178.286
		20	0.336	0.617	1.036	1.765	3.356	5.954	9.613	18.431	31.881	48.318	99.688	172.577
		30	0.326	0.598	1.005	1.713	3.256	5.777	9.326	17.882	30.932	46.880	96.721	167.439

Notes:

1. Refrigeration capacity in kilowatts is based on saturated evaporator as shown in table and condensing temperature of 40°C. For other liquid line temperatures, use correction factors in the table below.
2. The tables have been computed using an ISO 32 mineral oil for R-22 and R-502. R-134a has been computed using an ISO 32 ester-based oil.

Refrigerant	Liquid temperature, °C		
	20	30	50
134a	1.20	1.10	0.89
22	1.17	1.08	0.91
502	1.26	1.12	0.86

TABLE C9.14 (IP) Minimum Refrigeration Capacity in Tons for Oil Entrainment up Hot-Gas Risers (Type L Copper Tubing)

Refrig-erant	Saturated temp., °F	Discharge gas temp., °F	Pipe OD, in											
			½	⅝	¾	⅞	1-⅛	1-⅜	1-⅝	2-⅛	2-⅝	3-⅛	3-⅝	4-⅛
			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
		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
22	80.0	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
		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
	90.0	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
		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
	100.0	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
		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
	110.0	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
		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
	120.0	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
		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
134a	80.0	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
		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
	90.0	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

TABLE C9.14 (IP) Minimum Refrigeration Capacity in Tons for Oil Entrainment up Hot-Gas Risers (Type L Copper Tubing) *(Continued)*

Refrigerant	Saturated temp., °F	Discharge gas temp., °F	Pipe OD, in											
			½	⅝	¾	⅞	1-⅛	1-⅜	1-⅝	2-⅛	2-⅝	3-⅛	3-⅝	4-⅛
			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
134a	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
502	80.0	110.0	0.192	0.344	0.567	0.857	1.67	2.82	4.36	8.71	15.0	23.4	34.1	47.3
		140.0	0.180	0.323	0.534	0.806	1.57	2.66	4.11	8.20	14.1	22.0	32.1	44.5
		170.0	0.173	0.310	0.512	0.773	1.50	2.54	3.94	7.85	13.5	21.1	30.7	42.8
	90.0	120.0	0.194	0.348	0.574	0.867	1.69	2.85	4.41	8.81	15.1	23.6	34.5	47.8
		150.0	0.182	0.326	0.538	0.813	1.58	2.68	4.14	8.26	14.2	22.2	32.3	44.8
		180.0	0.169	0.303	0.501	0.756	1.47	2.49	3.85	7.69	13.2	20.6	30.1	41.7
	100.0	130.0	0.194	0.348	0.575	0.869	1.69	2.86	4.42	8.83	15.2	23.7	34.5	47.9
		160.0	0.182	0.326	0.539	0.813	1.58	2.68	4.14	8.27	14.2	22.2	32.3	44.9
		190.0	0.170	0.304	0.503	0.739	1.48	2.50	3.87	7.71	13.3	20.7	30.2	41.9
	110.0	140.0	0.170	0.305	0.504	0.761	1.48	2.51	3.87	7.73	13.3	20.7	30.2	42.0
		170.0	0.162	0.291	0.481	0.726	1.41	2.39	3.70	7.38	12.7	19.8	28.9	40.1
		200.0	0.152	0.273	0.450	0.680	1.33	2.24	3.46	6.92	11.9	18.5	27.0	37.5
	120.0	150.0	0.170	0.305	0.503	0.760	1.48	2.50	3.87	7.73	13.3	20.7	30.2	41.9
		180.0	0.153	0.275	0.453	0.683	1.33	2.26	3.49	6.96	12.0	18.7	27.2	37.8
		210.0	0.149	0.267	0.440	0.665	1.30	2.19	3.39	6.76	11.6	18.1	26.4	36.7

Notes:

1. Refrigeration capacity in tons is based on a saturated suction temperature of 20°F with 15°F superheat at the indicated saturated condensing temperature with 15°F subcooling. For other saturated suction temperatures with 15°F superheat, use the correction factors in the table to the right.
2. This table has been computed using an ISO 32 mineral oil for R-22 and R-502. R-134a has been computed using an ISO 32 ester-based oil.

Refrigerant	Saturated suction temperature, °F			
	−40	−20	0	+40
22	0.92	0.95	0.97	1.02
134a	—	—	0.96	1.04
502	0.85	0.91	0.95	1.04

TABLE C9.14 (SI) Minimum Refrigeration Capacity in Kilowatts for Oil Entrainment up Hot-Gas Risers (Copper Tubing, ASTM B88M, Type B, Metric Size)

Refrigerant	Saturated discharge temp., °C	Discharge gas temp., °C	Tubing diameter, nominal OD, mm											
			12	15	18	22	28	35	42	54	67	79	105	130
		60.0	0.469	0.860	1.445	2.462	4.681	8.305	13.408	25.709	44.469	67.396	139.050	240.718
		70.0	0.441	0.808	1.358	2.314	4.399	7.805	12.600	24.159	41.788	63.334	130.668	226.207
134a	20.0	80.0	0.431	0.790	1.327	2.261	4.298	7.626	12.311	23.605	40.830	61.881	127.671	221.020
		70.0	0.493	0.904	1.519	2.587	4.918	8.726	14.087	27.011	46.722	70.812	145.096	252.916
		80.0	0.463	0.849	1.426	2.430	4.260	8.196	13.232	25.371	43.885	66.512	137.225	237.560
134a	30.0	90.0	0.452	0.829	1.393	2.374	4.513	8.007	12.926	24.785	42.870	64.974	134.052	232.066
		80.0	0.507	0.930	1.563	2.662	5.061	8.979	14.496	27.794	48.075	72.863	150.328	260.242
		90.0	0.477	0.874	1.469	2.502	4.756	8.439	13.624	26.122	45.184	68.480	141.285	244.588
134a	40.0	100.0	0.465	0.852	1.432	2.439	4.637	8.227	13.281	25.466	44.048	66.759	137.735	238.443
		90.0	0.510	0.936	1.573	2.679	5.093	9.037	14.589	27.973	48.385	73.332	151.296	261.918
		100.0	0.479	0.878	1.476	2.514	4.779	8.480	13.690	26.248	45.402	68.811	141.969	245.772
134a	50.0	110.0	0.467	0.857	1.441	2.454	4.665	8.278	13.364	25.624	44.322	67.173	138.590	239.921
		60	0.563	0.032	0.735	2.956	5.619	9.969	16.094	30.859	43.377	80.897	116.904	288.938
		70	0.549	1.006	1.691	2.881	5.477	9.717	15.687	30.078	52.027	48.851	162.682	281.630
22	20	80	0.535	0.982	1.650	2.811	5.343	9.480	15.305	29.346	50.761	76.933	158.726	173.780
		70	0.596	1.092	1.836	3.127	5.945	10.547	17.028	32.649	56.474	85.591	176.588	305.702
		80	0.579	1.062	1.785	3.040	5.779	10.254	16.554	31.740	54.901	83.208	171.671	297.190
22	30	90	0.565	0.035	1.740	2.964	5.635	9.998	16.140	30.948	53.531	81.131	167.386	289.773
		80	0.618	1.132	1.903	3.242	6.163	10.934	17.653	33.847	58.546	88.732	183.069	316.922
		90	0.601	1.103	1.853	3.157	6.001	10.647	17.189	32.959	47.009	86.403	178.263	308.603
22	40	100	0.584	1.071	1.800	3.067	5.830	10.343	16.698	32.018	55.382	83.936	173.173	299.791
		90	0.630	1.156	1.943	3.310	6.291	11.162	18.020	34.552	59.766	90.580	186.882	323.523
		100	0.611	1.121	1.884	3.209	6.100	10.823	17.473	33.503	57.951	87.831	181.209	313.702
22	50	110	0.595	1.092	1.834	3.125	5.941	10.540	17.016	32.627	56.435	85.532	176.467	305.493

TABLE C9.14 (SI) Minimum Refrigeration Capacity in Kilowatts for Oil Entrainment up Hot-Gas Risers (Copper Tubing, ASTM B88M, Type B, Metric Size) *(Continued)*

Refrigerant	Saturated discharge temp., °C	Discharge gas temp., °C	Tubing diameter, nominal OD, mm											
			12	15	18	22	28	35	42	54	67	79	105	130
502	20	60	0.453	0.831	1.397	2.380	4.524	8.027	12.959	24.848	42.980	65.141	134.396	232.661
		70	0.440	0.807	1.357	2.311	4.393	7.795	12.585	24.130	41.737	63.257	130.509	225.933
		80	0.429	0.788	1.324	2.255	4.286	7.605	12.278	23.542	40.720	61.715	127.329	220.427
502	30	70	0.459	0.841	1.414	2.409	4.580	8.125	13.118	25.152	43.506	65.937	136.038	235.504
		80	0.446	0.818	1.375	2.343	4.454	7.902	12.757	24.461	42.311	54.126	132.302	229.036
		90	0.435	0.798	1.341	2.285	4.343	7.706	12.441	23.854	41.260	62.534	129.017	233.350
502	40	80	0.451	0.827	1.389	2.367	4.499	7.983	12.888	24.711	42.743	64.780	133.652	231.374
		90	0.439	0.804	1.352	2.303	4.378	7.767	12.540	24.044	41.589	63.031	130.044	225.127
		100	0.427	0.783	1.316	2.241	4.260	7.559	12.203	23.398	40.472	61.340	126.554	219.085
502	50	90	0.432	0.791	1.330	2.266	4.307	7.641	12.336	23.652	40.912	62.006	127.927	221.463
		100	0.418	0.767	1.289	2.196	4.174	7.406	11.956	22.925	39.654	60.100	123.996	214.657
		110	0.406	0.745	1.253	2.134	2.056	7.197	11.619	22.279	38.536	58.404	120.498	208.602

Notes:

1. Refrigeration capacities in kilowatts are based on saturated evaporator at −5°C, and condensing temperature as shown in table. For other liquid line temperatures, use correction factors in the table below.

Refrigerant	Saturated suction temperature, °C				
	−50	−40	−30	−20	+5
22	0.87	0.90	0.93	0.96	1.02
502	0.77	0.83	0.88	0.93	1.04
	0		5		10
134a	1.02		1.04		1.06

2. These tables have been computed using an ISO 32 mineral oil for R-22 and R-502. R-134a has been computed using an ISO 32 ester-based oil.

Sizing Data for Oil Return in Discharge or Suction Lines with Flow Vertically Upward

Saturation temperature, °C	Line size, DN 50 or less	Line size, above DN 50
−18	80 Pa/m	45 Pa/m
−46	100 Pa/m	57 Pa/m

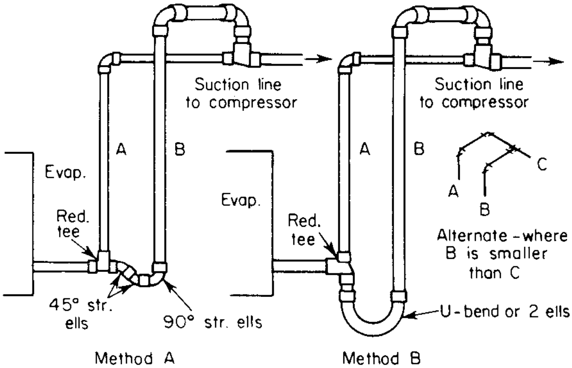


FIGURE C9.7 Double suction riser construction. *(Courtesy Carrier Air Conditioning Co.)*

practiced by equipping accumulators with oil stills and pumps that operate on demand from oil level controls to transfer oil to individual compressor packages.

Suction Line Design for Oil Return. Suction lines should be designed so that oil from an active evaporator does not drain into an idle one. Various multiple coil arrangements are shown in Figs. C9.8*a, b, c,* and *d*.

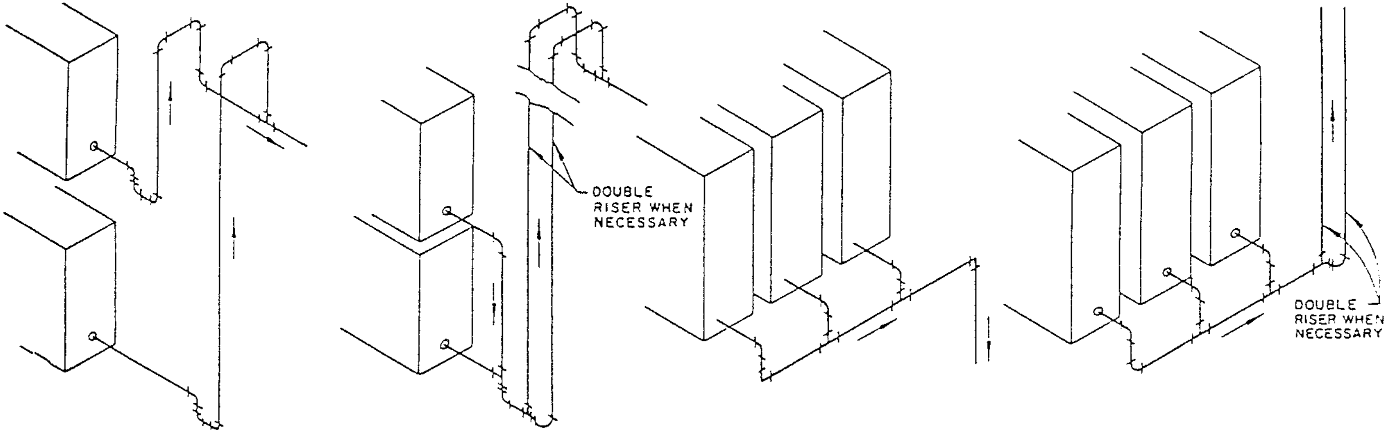
Figure C9.8*a* shows multiple evaporators on different floor levels, with the suction line above. Each suction line is brought upward and looped into the top of the common suction line to prevent oil from draining into inactive coils. Figure C9.8*b* shows multiple stacked coils on the same level, with the suction line above, using an alternative double riser method.

Figure C9.8*c* shows multiple coils on the same level, with the suction line below. Figure C9.8*d* shows multiple coils on the same level, with the suction line above.

In all cases that use thermostatic expansion valves, the control bulbs are located upstream of the liquid traps or the elbows which turn down, as illustrated in Fig. C9.9. This prevents erratic operation that may be caused by liquid holdup during compressor-off cycles.

Condenser Drains. In the section on selection of line sizes, liquid lines were sized on the basis of about 100 fpm (0.5 m/s). When liquid lines run vertically downward from the bottom of a horizontal shell-and-tube or shell-and-coil condenser to a receiver and where full equalization of the condenser and receiver is required through the condensate drain line, it may be necessary to maintain velocities less than 100 fpm (0.5 m/s). Table C9.15 shows the maximum velocity based on full pipe area which should be allowed in vertical outlets from condensers to assure full equalization and to prevent gas binding. This table is equally applicable to pipe or copper tubing, and is applicable to any refrigerant compatible with the materials, since it is based on the hydraulic effect of liquid flowing into vertical outlets. If the velocities exceed those shown in this table, separate vapor-equalizing lines should be used.

Marine Condensers. Refrigerant condensers for use on land usually have a single liquid outlet. Condensers intended for use on shipboard are usually mounted fore



A
MULTIPLE EVAPORATORS
ON DIFFERENT LEVELS–
COMPRESSOR ABOVE

B
ALTERNATE FOR 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

[a]A pumpdown cycle is recommended with all arrangements shown.

FIGURE C9.8 Arrangements of suction line loops.

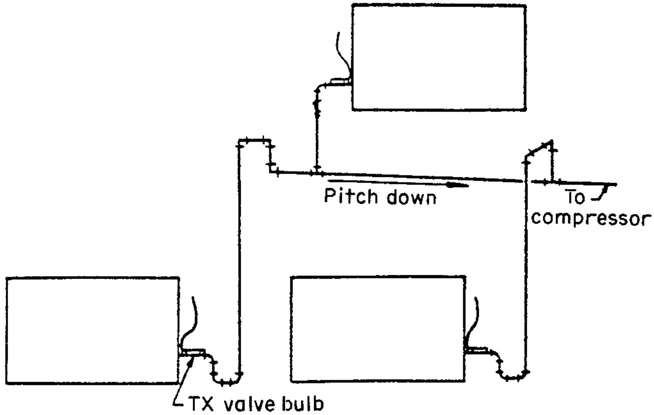


FIGURE C9.9 Typical piping from evaporators located above and below common suction line.

and aft to take advantage of the lower pitch angle. Outlets are provided at each end of the condenser. These are piped into a common header having a single outlet in the center. Each condenser outlet should be capable of handling the entire full capacity of the condenser.

Insulation. Since refrigerating systems are designed basically to produce temperatures below normal, many of the pipelines in a refrigeration system will be at temperatures below the dew point of the surrounding air. In addition to a heat gain from the surroundings, condensation, commonly called *sweating,* will form on

TABLE C9.15 Maximum Velocity in Condenser Vertical Drains to Assure Equalization

Inside diameter			
in	mm	Max. velocity, fpm	(m/s)
1	25	45.5	0.231
2	51	63.8	0.324
3	76	77.8	0.395
4	102	89.1	0.453
5	127	99.7	0.507
6	152	109.3	0.555
8	203	125.4	0.637
10	254	140.5	0.714
12	305	153.8	0.781

Table shows maximum velocity permitted in round vertical drains, based on full pipe area, to assure full equalization with receiver.

Based on the formula: Velocity (fpm) = $44.4\sqrt{d}$ where d is in inches ($0.04475\sqrt{d}$ where d is in mm).

these pipes. The condensation may be objectionable and even harmful. It is customary to insulate all refrigerant lines where there is a possibility of condensation on the refrigerant line.

The amount and type of insulation depend upon the operating-temperature level. Chapter B7 covers insulation in detail with recommendations for economical thicknesses, types, and application.

An important consideration in the application of any insulation to refrigerant piping is to assure that there is a vapor seal on the outside of the insulation. If the insulation type is such that moisture can enter it, the natural difference in vapor pressure between the surrounding atmosphere and the surface of the pipe will result in moisture migration into the insulation and eventually to the surface of the pipe. Vapor seals at all joints and on the outside of the insulation are essential to assure efficient performance and the avoidance of future trouble.

It is also necessary to avoid thermal bridges between the cold piping and the outside ambient air or atmosphere. Pipe hangers for supports should not contact the cold piping and should be arranged so that the supports bear against saddles of adequate area which are outside the insulation. Any metal in contact with the cold pipe wall, because of the higher conductivity of the metal hanger or rod, will cause condensation on the hangers.

MECHANICAL PIPING DESIGN

Code Requirements

The Safety Code for Mechanical Refrigeration, ANSI/ASHRAE 15[1], was written to assure the safe design, construction, installation, operation, and inspection of every refrigerating system using a volatile refrigerant. ANSI/ASHRAE 15 Standard covers refrigerant piping, valves, fittings, and related parts.

The ASME Code for Pressure Piping, Section B31.5,[2] is a standard covering the minimum requirements for materials, design, fabrication, assembly, examination, inspection, and testing of refrigeration piping, including limitations.

The ANSI/ASHRAE 15 and ASME B31.5 Codes recognize that refrigeration equipment of the self-contained or unit type which has been designed in accordance with good practice and which has been submitted to an approved, nationally recognized testing laboratory which provides uniform testing and examination procedures and which has a follow-up inspection service of current production of such units will be construed as meeting the requirements of either of these codes.

The ASME B31.5 Code also excludes water piping from consideration but does include secondary coolant piping.

Materials. Since volatile refrigerants have different chemical compositions, some materials are incompatible with certain refrigerants. Table C9.16 shows that several common materials are not compatible with certain of the refrigerants; it also indicates certain other limitations from a code standpoint. It will be noted that cast-iron pipe is not permitted by the Pressure Piping Code for any volatile refrigerant; case-iron valves and fittings of approved types are permitted.

Copper and brass are not compatible with ammonia. Aluminum, zinc, and magnesium are not suitable for use with methyl chloride. Zinc and magnesium are not suitable for use with any of the halogenated hydrocarbon refrigerants or ammonia.

In addition to the limitations and qualifications shown in Table C9.16, each of

TABLE C9.16 Material Compatibility

Material	Refrigerant number 22	134a	290	All 400's	All 500's	717	744
Carbon steel	S*	S	S	S	S	S	S
Wrought iron	S	S	S	S	S	S	S
Cast-iron pipe	NP*	NP	NP	NP	NP	NP	NP
Copper or brass	S	S	S	S	S	NS	S
Aluminum	Q*	Q	Q	Q	Q	Q	NS
Zinc	NS*	NS	NS	NS	NS	NS	NS
Magnesium	NS	NS	NS	NS	NS	NS	NS
ASHRAE 15-94 group	A1	A1	A1	A1	A1	B2	A1

* NP, not permitted by ASME B31.5 Code; NS, not satisfactory; Q, qualified—moist refrigerant may corrode—consult supplier; S, satisfactory.

the eight listed refrigerants is classified according to the Underwriters' Laboratories classification and according to a group classification system covered in the ANSI/ASHRAE 15 Safety Code.

The ANSI/ASHRAE 15 classification of Groups A1 and B1 include refrigerants which are not considered flammable toxic or in the ordinary sense (remember that many refrigerants can smother if present in heavy concentrations).

Groups A2 and B2 in the ASHRAE classification cover flammable or toxic refrigerants, and GROUPS A3 and B2 cover highly flammable or explosive refrigerants.

Limitations. Table C9.17 shows the limitations of various types of materials and group classifications with respect to the service with refrigerants. It will be noted that steel pipe must be Sch 40 or heavier for use with volatile refrigerants, with certain limitations with respect to size.

ASME B31.5 no longer permits the use of butt-welded carbon steel with refrigerants. However, it does permit the use of listed electric-resistance welded pipe and tube.

Cast-iron pipe is not allowed for any volatile, flammable, or toxic refrigerant but may be used for water or nonvolatile brines. Cast-iron is not allowed for temperatures below −150°F (−65°C).

Copper or brass tubing may be used with any refrigerant with which it is compatible and in any size or pressure when selected by the design rules. If copper tubing is erected on the premises, it must be Type *K*, *L*, or ACR.

The ANSI/ASHRAE 15 Safety Code has certain requirements for institutional, public assembly, residential, and commercial occupancies. These rules prohibit the carrying of refrigerant piping through floors except that it may be carried from the basement to the first floor or from the top floor to a machinery penthouse to the roof.

Refrigerant piping may be connected to a condenser on the roof if it is carried through an approved rigid and tight, continuous, fire-resistant pipe duct or shaft having no openings on intermediate floors, or it may be carried on the outer wall of the building provided it is not located in an air shaft, closed court, or any other similar opening enclosed within the outer walls of the building.

For Group A1 refrigerants, the refrigerant piping may be carried through floors intermediate between the first floor and the top floor provided it is enclosed in an

TABLE C9.17 Piping Limitations

Type or material	ASHRAE 15 group number	Line size NPS	Service	Limitation
Carbon steel or wrought iron	A2, B2, A3, B3	1½ or smaller	Refrigerant liquid	Sch 80 or heavier
	A1, B1	6 or smaller	Refrigerant liquid	Sch 40 or heavier
	A2, B2, A3, B3	2 through 6	Refrigerant liquid	Sch 40 or heavier
	All groups	6 or smaller	Refrigerant vapor	Sch 40 or heavier
Butt-welded carbon steel or wrought iron	Any	Any	Refrigerant liquid	Not permitted by code
Cast-iron pipe	Any	Any	Refrigerant	Not permitted for refrigerants
Cast-iron pipe	—	—	Brine	Permitted above −150°F (−65°C)
Cast-iron pipe	—	—	Any	Not permitted below −150°F (−65°C)
Copper or brass	Any (except ammonia)	Any	Any (except ammonia)	Type *K* or *L* if erected on premises; soft annealed may not exceed 1⅜ in OD (DN 32)
			Ammonia	Not compatible
Aluminum, zinc, magnesium	—	—	Methyl chloride	Not compatible
Magnesium	—	—	Halogenated hydrocarbons	Not compatible
Threaded joints	A3, B3	1 and smaller	Any	Seal welded or braze
		See limitation	Group A3 or B3 fluids	Not allowed over 1 in (DN 25)
			Salt brines	Not allowed over 6 in (DN 150)
			Any	Not lighter than Sch 40 up through 6 in (DN 150); not lighter than Sch 30 on 8, 10, or 12 in (DN 200, 250, 300)

approved, rigid and tight, continuous fire-resisting pipe or shaft where it passes through intermediate spaces not served by the system. The piping of direct systems need not be enclosed where it passes through space served by that system. The pipe duct or shaft must be vented to the outside or to a space served by the system.

The ANSI/ASHRAE 15 Code further requires rigid or flexible metal enclosures for soft, annealed copper tubing used for refrigerant piping which contains other than Group A1 refrigerants.

Schedule 40 wrought-steel or wrought-iron pipe may be used for design working pressure not exceeding 300 psig (2100 KPa) provided electric-resistance-welded, or seamless pipe is used.

For limitations on copper tubing, refer to ANSI/ASHRAE 15.

Joints on copper tubing containing Groups A2, A3, B1, B2 and B3 refrigerants as classified by the ANSI/ASHRAE 15 Code must be brazed. Solder joints are prohibited in such systems.

A brazed joint is obtained by the joining of metal parts with alloys which melt at temperatures higher than 800°F (425°C) but less than the melting temperatures of the joined parts.

A soldered joint is obtained by the joining of metal parts with metallic mixtures or alloys which melt at temperatures below 800 (below 425 and above 200°C) and above 400°F.

Systems containing 100 lb (45 kg) of refrigerant with positive-displacement compressors should have stop valves on each inlet of each liquid receiver and on each branch liquid and suction line except on receivers which are in a condensing unit or which are an integral part of the condenser.

Refrigerant piping crossing an open space which affords a passage way in any building must be not less than 7½ ft (2.3 m) above the floor unless against a ceiling of such space.

Free passageway must not be obstructed by refrigerant piping. Refrigerant piping must not be placed in any elevator, dumb-waiter, or other shaft containing a moving object or any shaft which has openings to living quarters or to main exit hallways. Refrigerant piping is not to be placed in public hallways, lobbies, or stairways except that such refrigerant piping may pass along a public hallway if there are no joints in the section in the public hallway and provided that nonferrous tubing of 1⅛ in OD and smaller is contained in a rigid metal pipe.

Limitations on Threaded Joints. Threaded joints should be seal welded or brazed for refrigerants. Threaded joints larger than NPS 1 (DN 25) in size should not be used for Group A3 and B3 fluids and must be no larger than NPS 6 (DN 150) in size for salt brine. Threaded joints must not be used on lighter than Sch 40 pipe up through NPS 6 (DN 150) in diameter or on lighter than Sch 30 pipe for NPS 8-, 10-, or 12 pipe (DN 200, 250, 300).

Design Pressures

Minimum design pressures for refrigerants are defined in the ANSI/ASHRAE-15 Safety Code as not less than pressure arising under maximum operating, standby, or shipping conditions, but not less than the saturated gauge pressure corresponding to certain specified temperatures. See Table C9.18 for minimum design gauge pressures of 9 refrigerants. Design pressures in other codes, as typified by the Safety Requirements in ISO-5149[6], specify higher temperatures than are used in the ANSI standards. The notes under the table explain the temperature basis for the minimum

TABLE C9.18 Minimum Design Gauge Pressures, psig[3]

				Low side			High side				
							Watercooled		Aircooled		
Refrigerant	Group ASHRAE	Name	Formula	80°F ANSI[1]	32°C ISO[2]	43°C ISO[3]	104°F ANSI[1]	43°C ISO[2]	122°F ANSI[1]	55°C ISO[2]	63°C ISO[3]
R-22	A1	Chlorodifluoromethane	CHCLF2	144	168	225	208	225	267	294	361
R-134a	A1	Tetrafluoroethane	CH_2FCF_3	87	104	145	133	145	177	202	247
R-290	A3	Propane	C_3H_8	129	150	198	184	198	234	262	312
R-404A	A1/A1	R-125/143a/134a	(44/52/4)	176	219	271	252	271	321	360	432
R-407A	A1/A1	R-32/125/134a	(20/40/40)	167	208	258	239	258	307	345	412
R-407C	A1/A1	R-32/125/134a	(23/25/52)	167	208	258	239	258	307	345	412
R-410A	A1/A1	R-32/125	(50/50)	236	274	363	337	363	431	485	580
R-507	A1	R-125/143a	(50/50)	180	209	292	257	292	328	369	440
R-717	B2	Ammonia	NH_3	138	165	230	210	230	280	320	392

Notes:

1. ANSI/ASHRAE-15 defines ''Design Pressures'' as not less than pressure arising under maximum operating, standby, or shipping conditions, but not less than the saturated gauge pressure corresponding to the following temperatures: For low sides, not less than 80°F (26.7°C); For water-cooled high sides, not less than 30°F (16.7°C) higher than the summer 1 percent wet-bulb, or 15°F (8.3°C) higher than the highest design leaving condensing water temperature for which the equipment is designed, or 104°F (40°C), whichever is greatest. For air cooled high sides, 30°F (16.7°C) higher than the highest summer 1 percent design dry-bulb for the location but not lower than 122°F (50°C).
2. ISO-5149 defines the pressures for systems only protected by pressure limiting devices, as the pressure of the refrigerant having at least the following temperatures: For ambient temperatures up to 32°C (89.6°F): 32°C (89.6°F) low side, 43°C (109.4°F) water-cooled high side, 55°C (131°F) air cooled high side.
3. ISO-5149 defines the pressures for ambient temperatures up to 43°C (109.4°F): 43°C (109.4°F) low side, 43°C (109.4°F) water cooled high side, 63°C (145.4°F) air cooled high side.
4. The gauge pressures shown in psig are minimums. To convert to kPa gauge, multiply by 6.89474.

TABLE C9.19 Suggested Minimum Field Test Pressure for Refrigerant Piping, psig

Refrigerant	Group	Name	Formula	Low side	High side water-cooled	High side air-cooled
R-22	A1	Chlorodifluoromethane	CHCLF2	185	250	325
R-134a	A1	Tetrafluoroethane	CH_2FCF_3	115	160	225
R-290	A3	Propane	C_3H_8	165	220	290
R-404A	A1/A1	R-125/143a/134a	(44/52/4)	240	300	400
R-407A	A1/A1	R-32/125/134a	(20/40/40)	230	285	380
R-407C	A1/A1	R-32/125/134a	(23/25/52)	230	285	380
R-410A	A1/A1	R-32/125	(50/50)	300	400	535
R-507A	A1	R-125/143a	(50/50)	230	325	405
R-717	B2	Ammonia	NH_3	200	275	385

Notes:
1. The suggested field test pressures are approximately 110 percent of the ISO minimum design pressures in Table C9.18 for all refrigerants, except approximately 120 percent of the ISO minimum design pressure for ammonia.
2. Most safety and jurisdictional mechanical codes no longer speficy field test pressures. The field test pressure should follow the manufacturer or designer's specification, or be guided by performance codes such as IIAR-2 for ammonia systems, or be adjusted to not more than 130 percent of the nameplate pressure rating of any components in the parts of the system being tested.
3. When not otherwise specified, the minimum suggested field test pressure is 110 percent of the design or maximum working pressure, so as to test the system at the rated operating pressure of the safety relief valve.
4. The gauge pressures are shown in psig. To convert to kPa gauge, multiply by 6.89474.

low-side and high-side pressures. The refrigeration industry in the United States usually designs to higher pressures than specified in ANSI codes. The principal reason for this practice is to provide safety relief valve set points at design pressures that are 25 percent higher than the maximum expected pressure. This practice enhances adequate closing pressure in conventional spring-loaded safety relief valves. System internal pressures within 90 percent of the set point can cause weeping or leaking of refrigerants from conventional relief valves.

The new International Mechanical Code which is replacing juridictional codes, references ANSI/ASHRAE-15 for safety requirements and references ANSI/IIAR-2 for equipment and design of ammonia refrigerating systems. There are no specified field test pressures in the International Mechanical Code. Table C9.19 lists suggested minimum field test pressure for refrigerating piping.

REQUIRED THICKNESS OF PIPE OR TUBE

The *required thickness* of pipe or tubing is determined from the following equations and nomenclature:

$$t_m = t + c \tag{C9.8}$$

$$t = \frac{PD_o}{2(S + Py)} = \frac{Pd}{2(S + Py - P)} \tag{C9.9}$$

$$P = \frac{2St}{(D_o - 2yt)} \tag{C9.10}$$

where t_m = minimum required thickness (in) (mm)
c = allowance for grooves, threads, tolerances, corrosion, erosion (in) (mm)
P = internal design pressure (psig)(kPa gauge)
D_o = outside diameter of pipe (in) (mm)
d = inside diameter of pipe (in) (mm)
S = allowable stress (psi) (kPa)
t = calculated thickness in (mm)

y is a material coefficient for which

Ductile nonferrous materials = 0.4

Ferritic steels = 0.4

Austenitic steels = 0.4

The design of piping for external pressure involves the use of charts to determine factors which are used to calculate the thickness or allowable working pressure. The method and charts referred to in paragraphs UG 28 and UG 31 of Section VIII of the ASME Boiler and Pressure Vessel Code[3] are acceptable for design of pipes and tubes subject to external pressure.

The Pressure Piping Code recognizes and permits the use of the design rules of the ASME Unfired Pressure Vessel Code for closures, flanges, and blind flanges. For blanks, the following equation should be used:

$$t = d_g\sqrt{3P/16S}$$

where t = required thickness (in) (mm)
d_g = inside diameter of gasket for raised or flat-face flanges or the pitch diameter of retained gasket flanges (in) (mm)
P = internal or external design pressure (psig) (kPa)
S = allowable stress (psi) (kPa)

Since the Pressure Piping Code[2] and the Safety Code for Mechanical Refrigeration both limit the minimum thickness of steel pipe and the minimum thickness for copper tubing for erection on the premises, it is possible to calculate the maximum working pressure for these commonly used weights of pipe or tubing.

Table C9.20 shows the maximum allowable internal working pressure for seamless steel pipe in the permitted schedule numbers. The maximum allowable external pressure has also been calculated for a length-over-diameter ratio in excess of 15 which will be common in most piping systems. It will be evident that the allowable working pressure for the permitted thicknesses of pipe is usually far in excess of that required by the design working pressure requirements for the various refrigerants. In most cases except in unusual circumstances, in cases where shock may be anticipated, or in the larger sizes, no further checking of allowable working pressure will be necessary.

Table C9.21 shows rated internal working pressures for type ACR copper tubing normally used for air conditioning and refrigeration service in the field. Ratings are given for annealed tubing.

The ASME B31.5 Pressure Piping Code establishes certain allowable working stresses for many grades of pipe and tubing in various materials. These allowable stresses are to be used in conjunction with the design equations listed above for special calculations.

TABLE C9.20 Allowable Working Pressures for Carbon Steel Refrigerant Piping

Nominal pipe size, NPS	(DN)	Schedule no.	Allowable internal working pressure, psig	Allowable external working pressure, psig
⅛	(3)	40	1,890	2,070
		80	3,510	2,860
¼	(6)	40	1,490	2,000
		80	2,880	2,700
⅜	(10)	40	1,300	1,660
		80	2,500	2,280
½	(15)	40	1,126	1,580
		80	2,210	2,140
¾	(20)	40	994	1,320
		80	1,890	1,800
1	(25)	40	866	1,210
		80	1,680	1,670
1¼	(32)	40	773	980
		80	1,470	1,410
1½	(40)	40	740	890
		80	1,390	1,270
2	(50)	40	670	750
2½	(65)	40	665	810
3	(80)	40	624	700
3½	(90)	40	600	640
4	(100)	40	580	580
6	(150)	40	534	450
8	(200)	40	515	390
10	(250)	40	496	340
12	(300)	STD.	436	260

For internal pressure:
- Based on minimum wall thickness; no corrosion or erosion allowance: thread allowance factor from ANSI/ASME B.1.201.
- Allowable stress = 12,000 psi.
- y (material coefficient) = 0.4.
- $P = 2St/(D_0 - 2yt)$.

For external pressure:
- Based on minimum wall thickness: no corrosion, erosion, threading, or grooving allowance.
- Yield: 24,000 psi to 30,000 psi.
- $L/D_o = 15$ or greater where D_o = outside pipe diameter in inches; L = maximum straight length of run between flanges, elbows, caps or stiffening rings (in).
- To convert psig to kPa gauge, multiply by 6.89474.

Low-Temperature Design Criteria. It is recognized that certain materials tend to become brittle at low temperatures and may be subject to failure which would not occur normally at usual temperatures or at elevated temperatures. The transition temperature at which certain materials become brittle is not well defined. Some ferrous materials may pass through the transition range at normal temperatures, while others may not become brittle until quite low temperatures are attained. The ASME Unfired Pressure Vessel Code[3] arbitrarily establishes a temperature of −20°F as a point below which all vessels constructed of carbon or low-alloy steels should be impact tested, with certain exemptions.

Refrigeration piping is frequently subject to temperatures below normal atmospheric temperatures to the degree that embrittlement may occur, and the ASME

TABLE C9.21 Rated Internal Working Pressures (psig) for Copper Tube Type ACR*

Size and wall thickness (in)	Rated internal working pressure (psig)						
	100°F (38°C)		200°F (93°C)		300°F (149°C)		400°F (204°C)
	Annealed	Drawn	Annealed	Drawn	Annealed	Drawn	Annealed or drawn
⅛ (.030)	3130	—	3090	—	2620	—	1310
3⁄16 (.030)	1990	—	1950	—	1650	—	820
¼ (.030)	1450	—	1420	—	1200	—	600
15⁄16 (.032)	1230	—	1200	—	1020	—	510
⅜ (.030)	900	1350	880	1300	740	1180	370
⅜ (.032)	1010	—	990	—	840	—	420
½ (.032)	740	—	730	—	610	—	300
½ (.035)	800	1200	780	1150	660	1060	330
⅝ (.035)	640	—	630	—	530	—	260
⅝ (.040)	740	1110	720	1060	610	980	300
¾ (.042)	650	980	630	930	530	850	260
⅞ (.045)	590	890	570	840	480	770	240
1⅛ (.050)	510	770	490	720	420	670	210
1⅜ (.055)	460	690	440	650	370	590	180
1⅝ (.060)	430	650	410	600	350	560	170
2⅛ (.070)	370	560	360	530	300	480	150
2⅝ (.080)	350	530	340	500	280	450	140
3⅛ (.090)	330	500	320	470	270	430	130
3⅝ (.100)	320	480	300	440	260	420	130
4⅛ (.110)	300	450	290	430	240	380	120

Based on *S* values as follows: 100°F—6,000 psi, annealed; 9,000 psi, drawn; 200°F—5,900 psi; annealed, 8,700 psi drawn; 300°F—5,000 psi, annealed, 8,000 psi, drawn; 400°F—2,500 psi, annealed or drawn, according to American National Standard Code for Pressure Piping, Refrigeration Piping, ANSI B31.5.

Source: Copper Development Association, Inc.

To convert inches to mm, multiply by 25.4.

To convert psig to kPa, multiply by 6.89474.

B31.5 Piping Code also requires impact tests on certain materias subject to temperature below −20°F. There are certain materials and certain conditions under which impact tests are not required. The exemptions are as follows:[2]

1. No impact tests are required for aluminum, austenitic stainless steel in grades 304, CF8, 304L, CF3, 316, CF8M, or 321, or copper, red brass, copper-nickel alloys, or nickel-copper alloys.
2. No impact tests are required for bolting material conforming with A193, Grade B7 for use at temperatures above −50°F.
3. No impact tests are required for bolting materials conforming with A320, Grades L7, L10, and L43 at temperatures above −150°F or above −225°F for A320, Grade L9.
4. No impact test is required for material used in fabricating a piping system for metal temperatures between −20 and −150°F when the most severe condition of pressure (internal if above atmospheric and external if below atmospheric) does not produce a stress exceeding 40% of the allowable material stress.

For low-temperature application, the use of nonferrous materials or the stainless steel mentioned will normally be satisfactory. The use of nickel-steel pipe in conjunction with the use of nickel-steel pressure vessels has long been an acceptable material for low-temperature when these materials are subjected to and pass the impact-testing requirements.

Impact tests, when conducted, shall follow the requirements of ASME B31.5, Paragraph 523.2.2.

The standard 10- by 10-mm specimen is used if the thickness of the material being tested is 7⁄16 in or greater. For material that is not of sufficient thickness to permit preparation of full-size specimens, tests may be made on the largest possible of the subsize specimens listed in the following table.

The impact properties for each size specimen are as follows:

Size of specimen (mm)	Minimum impact value required (ft · lb)	(J)
10 × 10	15	20.3
10 × 7.5	12.5	16.9
10 × 5	10	13.6
10 × 2.5	5	6.8

In welded fabrication, the weld also is required to meet the impact-test requirement.

Expansion and Contraction. Since refrigeration piping systems are subject to changes in temperature, some precautions must be taken to assure that these changes in temperature during operation or during shutdown are considered in the design of the piping and in the design of supports and flexibility.

Piping systems must be designed to have sufficient flexibility to prevent thermal expansion from causing

1. Failure of piping or anchors from overstress or overstrain
2. Leakage at joints
3. Detrimental distortion of connected equipment resulting from excessive thrusts or moments

Expansion strains are usually taken up by bending or torsion or by compression and tension. The concentration of stresses will be different in each case.

Bending or torsional flexibility may be provided by the use of bends, loops, or offsets. While swivel joints, ball joints, and corrugated expansion joints are recognized by the Pressure Piping Code, some of these are not considered desirable for volatile refrigerant piping. Bends, loops, and offsets are generally used to provide flexibility. Loops and cold springing also may be used in the design of piping. Chapter B4 covers in detail the general design considerations involved in expansion and flexibility of piping. These same principles must be applied to refrigeration piping, and the maximum temperature cycle involved in the installation should be taken into account in determining the nature and direction of the stresses which may be caused by temperature effects. As previously mentioned under ''Insulation,'' pipe hangers or supports for low-temperature piping normally will not be in direct contact with the metal portion of the piping. Consideration must be given to the insulation in types of support which are peculiar to refrigeration piping when the problems of expansion and flexibility are considered.

Miter Joints. The ASME B31.5 Pressure Piping Code gives details for design of branch connections where the angle between the axes of the branch and of the run is between 45 and 90°. Branch connections less than 45° impose special design and fabrication problems. The Code permits connections to be made by the use of tees and welding outlet fittings such as cast or forged nozzles or couplings, or by attaching a branch pipe directly to the run pipe by welding.

Normally the use of standard forged fittings of the butt-welding or socket types will provide sufficient strength in the case of a branch connection to permit application of these fittings without additional reinforcement. However, when a branch connection is welded into a hole cut into the main-run pipe, it is recognized that certain reinforcement may be required. The analysis of the extent of reinforcement, if required, is similar to that used on unfired pressure vessels for nozzle connections. The complete detailed analysis for such determinations and the means of determining the amount of reinforcement required are shown in the ASME B31.5 Pressure Piping Code or in the ASME Unfired Pressure Vessel Code. The general method is to calculate the amount of metal cut out of the pipe and to calculate the amount of metal which is added by extra thickness of the pipe wall over that required for strength, the extra metal in the nozzle or branch connection, and the extra metal which would be added within certain limiting geometric zones by welding. If the added metal provides the equivalent of the amount of metal which had been cut out, no additional reinforcement is necessary. Reinforcement metal usually is provided in the form of a ring or a saddle which is welded to the run pipe. This reinforcement material must be added within certain limiting dimensions as defined by the code. The use of ribs, gussets, or clamps is permissible to stiffen the branch connection, but their areas cannot be counted as contributing to the reinforcement area.

Welding, Brazing, and Soldering. Joints in piping which is to convey volatile refrigerants are usually made by welding, brazing, or soldering. This does not exclude the use of flanged connections, which are commonly used to connect valves or control devices in refrigeration piping. Flanged connections are commonly used to connect to pressure vessels and to compressors. Couplings of the friction type with seal rings may be used for refrigerants when the materials are compatible with the refrigerant and when the pressures permit. Use of such fittings is normally confined to low-pressure refrigerants. Limitations on threaded connections have already been listed, and frequently threaded connections, where used and where permissible, will be seal welded.

Welded joints may be used in any materials for which it is possible to qualify the welding procedures, the welders, and the welding operators.

Butt welds are permitted. Usually, backing rings are used in butt-welded joints, but where it is necessary to have a smooth interior surface or where the backing ring may result in severe corrosion or erosion, the joint may be welded without backing rings provided the piping is suitably cleaned. Socket welds are permitted under the Pressure Piping Code. The Pressure Piping Code defines in detail the required weld sizes and joint arrangements which are recommended for use in welded-part construction.

The ASME Boiler and Pressure Vessel Code, Section IX, defines in detail the qualification of welding procedures and welders' performance requirements for unfired pressure vessel construction. These rules have been adopted by the Pressure

Piping Code ASME B31.5 and form part of the requirements of that Code. The welders and the procedures should be qualified to assure that their quality is in conformance with these codes.

The ANSI/ASHRAE 15 Safety Code for Mechanical Refrigeration requires brazing of certain joints. Also certain joints in restricted areas may require high-melting-point filler material. The ASME B31.5 Pressure Piping Code defines the filler metal used in brazing to be nonferrous metal or alloy having a melting point above 800°F (425°C) but below that of the metal being joined. Good practice in cleanliness of joints and the use of proper fluxes is required for brazed joints.

Brazing procedures and operators, except for socket-type capillary joints, should be qualified in accordance with the requirements of Section IX of the ASME Boiler and Pressure Vessel Code.

For soldered joints, the ASME B31.5 Code defines the solder metal to be a nonferrous metal or alloy having a melting point below 800°F (425°C) and below that of the metal being joined. Good soldering technique requires proper cleanliness and preparation of the joints, proper joint clearances, and proper heating. Procedures to be used on soldering or brazing socket type joints are outlined in ASTM B828-92.[5]

Miscellaneous Considerations

Corrosion. The ASME B31.5 Pressure Piping Code recognizes that corrosion or erosion may be factors to be considered in piping design. When corrosion or erosion is expected, an increase in wall thickness of the components above that dictated by other design requirements is to be provided consistent with the expected life of the particular piping involved. In the basic equation for calculating the pressure ratings of pipe or in determining the required wall thickness of pipe, the Pressure Piping Code requires the addition of a factor to the calculated wall thickness to result in the actual thickness required. The factor includes allowance for threading, groove depth, and manufacturers' minus tolerance plus corrosion and erosion allowances.

Corrosion allowance on the inside of piping for volatile refrigerants is not mandatory. The refrigerant is recirculating and is usually charged into the system in a commercially pure state after thorough cleaning and evacuation of the entire system. When installed in accordance with good practice, a leaktight refrigerating system will not tend to corrode and it is not customary to add corrosion allowances. It is possible, with certain of the halogenated hydrocarbon refrigerants when contaminated with noncondensible gases or with water which may leak into a system under vacuum, to have corrosive products form. On some occasions, in hermetically sealed refrigeration systems, compressor motor burnouts have resulted in formation of contaminants which also may be damaging to the inside of the system. However, these considerations are not properly part of the piping design and are usually the result of carelessness or misapplication. It is ordinarily not necessary to add corrosion allowance to volatile-refrigerant piping.

For secondary coolant piping, especially with salt brines, the consideration of possible corrosion should be kept in mind in the design of the piping system. Ordinarily standard-weight pipe for either volatile-refrigerant use or for brine piping inherently has sufficient strength so that the normal wall thickness of pipes which are used are much heavier than are required for the actual pressure service, and therefore it may not be necessary to add additional allowances for corrosion.

Fittings. The Pressure Piping Code permits the use of standard fittings, provided they are compatible with the refrigerant or fluid. The standard ratings of forged steel flanges, fittings, and similar parts may be used for refrigerant service

Bell-and-spigot fittings may be used only for water and drainage service.

Couplings made of cast, malleable, or wrought iron may not be used on pipe containing flammable or toxic fluids. Wrought-iron couplings are subject to the same limitations in temperature, stress, and service which apply to cast-iron screwed fittings.

Valves. Cast-iron gate valves and plug cocks must not be used in liquid-refrigerant lines unless consideration is given to the expansion of liquid trapped in a space when the valve is closed.

Several manufacturers make standard lines of refrigeration fittings which do not fall into the classification of ANSI Standards for forged-steel valves. These valves and fittings over long years of usage have gained acceptability and are widely used and acceptable for refrigeration service to the degree recommended by the manufacturer.

Other Factors. The Pressure Piping Code lists the following dynamic effects which should be taken into account in the design of refrigerant piping.

1. Impact forces (including hydraulic shock) caused by either external or internal conditions.

2. The effect of wind loading on exposed piping.

3. Piping systems located in regions where earthquakes are a factor are to be designated for a horizontal force in conformity with the good engineering practice using governmental data as a guide in determining the earthquake force. However, this force is not to be considered as acting concurrently with lateral wind force.

4. Piping shall be arranged and supported with consideration for vibration.

The Pressure Piping Code also calls attention to the following weight effects which should be taken into account in the design of piping:

1. Live loads such as the weight of the fluid transported and snow and ice loads if the latter will be encountered. If low-temperature piping is not insulated, there can be a buildup of ice on the pipe even in high ambient temperatures.

2. Dead loads, consisting of the weight of the piping components and insulation and other superimposed loads.

3. Test loads which consist of the weight of the test fluid in the pipe.

REFERENCES

1. ASHRAE Safety Code for Mechanical Refrigeration, ANSI/ASHRAE-15, American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., 1791 Tullie Circle NE, Atlanta, GA 30329.
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3. ASME Boiler and Pressure Vessel Code, Section VIII, Unfired Pressure Vessels, ASME International, Three Park Avenue, New York, NY 10016-5990.
4. ASHRAE Handbook of Fundamentals, American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., 1791 Tullie Circle NE, Atlanta, GA 30329.
5. ASTM Standard B828–92, Standard Practice for Making Capillary Joints by Soldering of Copper and Copper Alloy Tube and Fittings, American Society for Testing Materials, 100 Barr Harbor Drive, West Conshohocken, PA 19428.
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7. W.V. Richards, Refrigerant Vapor Line Sizing Not Dependent on Length, 1983 World Congress of Refrigeration, International Institute of Refrigeration, 177, Boulevard Malesherbes, F-75017 Paris, France.
8. W.V. Richards, Practical Pipe Sizing for Refrigerant Vapor Lines, 1984 Annual Proceedings, International Institute of Ammonia Refrigeration, 1200 19th St. NW, Suite 300, Washington, DC 20036–2412.
9. *IIAR Piping Practices Handbook*, Chap. 1, ''Pipe Sizing Procedures,'' International Institute of Ammonia Refrigeration, 1200 19th St. NW, Suite 300, Washington, DC 20036–2412.