

evaporator, and this in turn operates through the power element to increase the flow of liquid refrigerant. A flooded evaporator reduces the discharge superheat, and thus tends to reduce the flow of liquid refrigerant. Such an expansion valve is satisfactory for operation with fluctuating loads since this type of control tends to keep the evaporator filled with refrigerant at all times.

Low-Side Float Valves. A liquid refrigerant control of the low-side float valve type consists of a ball float located in either a receiver or the evaporator itself on the low-pressure side of the system. A needle valve, operated through a simple lever mechanism attached to the float, permits the passage of more or less refrigerant, as the level in the receiver or the evaporator fluctuates. Such a control must be used in conjunction with a flooded evaporator, and has been applied extensively to household refrigerators and, to some extent, in commercial and industrial installations.

High-Side Float Valves. A high-side float valve differs from a low-side float valve in that the float is located in a receiver or container on the high-pressure side of the system. Proper operation again depends upon metering of the refrigerant through a controlled opening, depending upon the level of the liquid refrigerant in the container. Such a control has the disadvantage that the evaporator must be placed directly adjacent to the float container, or some intermediate pressure device must be applied to prevent flashing of the refrigerant upon pressure drop.

Capillary Tubes. A capillary tube may be used as a liquid refrigerant expanding device. Such a device consists of an extremely small bore tube (in the order of 0.04 inch in diameter) of five to twenty feet in length. Although such a restricting device operates as a very simple means of expanding the liquid refrigerant, it has the disadvantage that no modifications are possible to adjust the rate of expansion under various operating conditions. The bore and length of the tube, as well as the proportions of the rest of the system, are critical. It is for these reasons that its application has been limited to factory-assembled domestic and commercial units.

Refrigerant Flow Control

The capacity of the refrigeration system must be controlled in accordance with the load imposed on the system. Except for those cases where the compressor is started and stopped by a thermostat responding to load conditions, some form of control of the refrigerant flow is usually required. Controlling the flow of refrigerant in accordance with the load may be accomplished in a number of ways, some of which are:

1. **Solenoid valves** are frequently used for control of gas or liquid flow. A solenoid valve is placed in the liquid line ahead of the expansion valve. It is closed whenever the compressor is not in operation and thus leakage into the evaporator is prevented. In most cases it is controlled by a thermostat responding to the load. Solenoid liquid valves are widely used for control of refrigerant flow to individual evaporators in a multiple evaporator system operated by one compressor. The compressor capacity is controlled separately by a controller responding to suction pressure. Solenoid liquid and suction valves are sometimes used to isolate an evaporator completely for defrosting purposes.

When solenoid liquid valves are used for controlling refrigerant flow, a pump-down cycle of control is often employed. The thermostat closes the solenoid liquid valve and the compressor then pumps down until stopped by the low-suction pressure switch. The compressor cannot restart until the thermostat again opens the solenoid liquid valve.

2. **Expansion Valve Modulation.** This form of flow control utilizes the throttling characteristics of the expansion valve to starve the coil under reduced-load conditions. This may be accomplished by a device attached to the expansion valve or a pilot control which varies the setting of the expansion valve by changing the pressure in the equalizer line. Either of these are actuated by a controller sensitive to load conditions and the compressor capacity control is from suction pressure. Since short-cycling of the compressor is apt to occur if the flow of refrigerant is reduced too much, some limit to prevent this is required.

3. **Evaporator Pressure Control** is accomplished by back-pressure valves. These are placed in the suction line between the evaporator and the compressor, and maintain the evaporator pressure constant by regulating the amount of vapor drawn from the evaporator. The compressor capacity is controlled from suction pressure at the compressor. The use of back pressure valves permits a lower pressure at the compressor than would be permissible at the evaporator, thus minimizing short-cycling. On a multiple-evaporator system served by one compressor, the evaporators may be maintained at different temperatures. Back pressure valves can be equipped with an air connection for automatically varying the evaporator pressure in accordance with the demands of a pneumatic controller. Means for mechanically readjusting the valves by use of a proportioning electric operator also are available.

Condenser Control

The majority of refrigeration systems, other than fractional horsepower, use water-cooled rather than air-cooled condensers. Condenser control is used for the purpose of conserving water or limiting the condenser pressure, or both. Solenoid water valves on small compressors, or pneumatic or electric valves on larger compressors, are usually controlled simultaneously with starting or stopping of the compressor. A pressure-operated valve will provide proportional control of water flow in response to condenser pressure. Similar water valves controlled thermostatically by the temperature of water discharged from the condenser are sometimes used. Evaporative condensers may be controlled by an automatic damper responding to a condenser-pressure controller. Cooling tower fans sometimes are started and stopped by a thermostat in the condensing water or a pressure controller in the condenser. Some form of freeze protection for evaporative condensers and cooling towers is necessary if they are required to operate in cold weather.

Refrigeration Control for Air Conditioning

When refrigerating equipment is used for space cooling, two major control problems exist: one is control of the temperature and the other, control of the humidity. In some applications the amount of latent heat to be removed is small compared with the sensible heat. In such cases, sufficient dehumidification will usually occur without any special provisions. In other cases, such as theaters, where the latent load is relatively high, the air must be cooled below its dew-point temperature, and sometimes rewarmed to return it to the comfort range. Refer to Chapter 43 for general information on the subject of controls and for applications relating to air-conditioning systems.

REFRIGERATION PIPING

The pressure drop which occurs during passage of the refrigerant through connecting piping is similar in effect to that which occurs through suction and discharge valves of the compressor. Thus, the effect of the pressure drop in the suction line between evaporator and compressor requires that a lower pressure be maintained inside the compressor during suction than is maintained in the evaporator. The pressure drop through the connecting piping between the compressor and condenser requires that a higher pressure be maintained inside the compressor during discharge than in the condenser. These losses result in a greater compression ratio, and therefore greater power requirements, as well as a lower volumetric efficiency and higher displacement requirements. Pressure losses in the liquid line between condenser or receiver and the expansion valve may

Table 5.... Dichlorodifluoromethane* (CCl₂F₂) Liquid Lines, Tons Capacity per 100 Ft Equivalent Length

Line Size, Inches	Pressure Drop per 100 Ft Equivalent Length, Psi			
	3	5	10	20
1/4 OD	0.88	1.14	1.80	2.58
1/4 OD	2.89	3.64	5.56	8.50
1/2 IPS	4.56	6.81	10.2	15.8
1/2 OD	4.56	6.81	10.2	15.8
3/4 IPS	9.73	12.6	18.5	27.0
3/4 OD	9.73	12.6	18.5	27.0
1 IPS	10.5	14.1	21.8	33.0
1 IPS	21.4	28.2	41.3	60.8
1 1/4 OD	21.4	28.2	41.3	60.8
1 1/4 IPS	36.9	48.1	70.5	101.
1 1/2 OD	36.9	48.1	70.5	101.
1 1/2 IPS	62.0	80.2	114.	160.
1 3/4 OD	62.0	80.2	114.	160.
2 IPS	124.	161.	231.	328.
2 1/4 IPS	230.	297.	426.	607.
3 IPS	364.	469.	676.	972.
3 1/4 IPS	539.	704.	1005.	1430.
4 IPS	753.	972.	1385.	1945.

Note: Tonnage values above those underlined give velocities of 300 fpm or less.
*Refrigerant 12.

result in some flashing of the liquid refrigerant, unless the liquid is subcooled. In all cases, friction losses should be kept to a minimum, and piping should be selected which will give the smallest loss consistent with overall economy in the system.

Refrigerant liquid lines from the receiver to the expansion valve should preferably be designed with a pressure drop of less than 5 psi, and with 10 psi as the maximum. A velocity of 100 to 250 fpm is recommended to prevent a pressure drop great enough to cause vaporization of the refrigerant ahead of the expansion valve. If the evaporator is to be located at a higher elevation than the condenser or receiver, account should be taken of the pressure drop for each foot of static liquid lift. Approximate values are 0.26 psi per foot for ammonia, 0.57 psi per foot for dichlorodifluoromethane, and 0.64 psi per foot for monochlorodifluoromethane. Where there is a possibility of vaporization of some of the liquid before reaching the expansion valves, means for subcooling should be provided.

Since a reduction of suction pressure at the compressor results in an appreciable reduction in capacity and more power input per ton of refrigeration, great care should be given to the proper sizing of suction lines between the evaporator and the compressor. Although comparatively high velocities, 500 to 5000 fpm, may be used, the optimum value will depend upon the refrigerant and the operating-pressure range. Since return of the oil to the compressor must be considered in the case of the fluorinated hydrocarbons and methyl chloride, for these refrigerants the minimum velocity should be 500 fpm for horizontal runs and 1000 fpm for vertical runs. For the former, the usual design velocities range between 1000 and 2000 fpm. Too

Table 6.... Maximum Tons of Compressor Capacity for CCl₂F₂ Lines*

Line Size, Inches	Suction Lines Based on 105 F Condensing Temperature						Discharge Lines	
	Psi Pressure Drop per 100 Ft Equivalent Length at 40 F Saturation						Condensing Temperature	
	1	2	3	4	5	115 F	90 F	
1/4 OD	0.14	0.20	0.28	0.35	0.41	0.45		
1/4 IPS	0.17	0.24	0.34	0.42	0.49	0.54		
1/2 OD	0.25	0.35	0.51	0.62	0.73	0.81	1.43	1.15
1/2 IPS	0.35	0.45	0.65	0.79	0.93	1.03	1.87	1.50
3/4 OD	0.65	0.76	1.10	1.34	1.58	1.75	2.97	2.38
3/4 IPS	0.68	0.94	1.35	1.65	1.92	2.12	3.26	2.62
1 OD	1.28	1.80	2.57	3.17	3.76	4.15	5.05	4.05
1 IPS	1.43	2.01	2.89	3.54	4.17	4.60	5.29	4.25
1 1/4 OD	2.21	3.12	4.45	5.50	6.38	7.05	7.72	6.19
1 1/4 IPS	2.70	3.82	5.37	6.72	7.68	8.48	9.16	7.35
1 1/2 OD	3.40	4.78	6.79	8.42	9.77	10.8	10.92	8.75
1 1/2 IPS	4.05	5.75	8.10	10.12	11.6	12.8	12.5	10.0
2 OD	6.12	8.60	12.1	15.1	17.4	19.2	19.2	15.3
2 IPS	7.66	10.9	15.3	19.2	22.2	24.5	20.6	16.5
2 1/4 OD	12.0	17.1	24.0	30.1	34.6	38.2	32.2	25.9
2 1/4 IPS	12.0	17.1	24.0	30.1	34.6	38.2	32.2	25.9
3 OD	19.1	27.2	38.2	47.8	55.0	60.7	51.5	39.8
3 IPS	20.9	29.4	42.3	51.8	60.0	66.2	54.5	43.8
3 1/2 OD	27.8	39.7	55.7	69.8	80.3	88.7	72.0	57.6
3 1/2 IPS	30.2	43.2	61.0	76.1	87.0	96.0	78.8	63.3
4 OD	38.6	55.2	78.0	97.3	111	123	95.8	77.1
4 IPS	40.7	58.6	83.0	103	118	130	101.6	81.6
5 IPS	71.3	100	141	176	203	224	171.5	137.8
6 IPS	126	183	257	322	366	403	266	214
8 IPS	211	297	422	523	602	664	461	370
10 IPS	352	503	712	887	1024	1130	725	582
12 IPS	550	780	1106	1373	1582	1748	1041	836

*Refrigerant 12.

high velocities create noise problems and excessive pressure drops. The total pressure drop in the suction line should be between one and two psi, if the velocity can be kept within the specified limits.

Compressor discharge or hot gas lines may be designed with velocities from 1000 to 5000 fpm, except for dense gases such as carbon dioxide, where noise considerations will reduce the upper limit. A pressure drop of 2 to 4 psi is recommended for the discharge lines. Extensive tables are available in the literature for the determination of pressure drops through refrigerant lines with various refrigerants. The capacities listed in Tables 5, 6, and 7 are published in *ARI Equipment Standards 520* (1946), of the Air-Conditioning and Refrigeration Institute, and are used by permission. Table 5 shows the tonnage capacity normally allowed for CCl₂F₂ liquid lines per foot equivalent length of pipe, and Table 6, the maximum tonnage for suction