

other part of the system. It may be drained into a run down receiver which is disconnected from the suction pressure when the level has increased to a preset point and connected to the discharge pressure which serves to push the liquid back to a point where liquid is wanted, such as the low temperature evaporator, the receiver or liquid line.

In some cases a heater is installed to evaporate the liquid. The heat might be provided by a coil, using the warm condenser liquid, if the quantity of liquid from the knock-out drum is not too great as compared to the quantity of warm liquid available.

If the refrigerant is boiled out of the liquid which has drained into the knock-out drum, there is a probability of a quantity of oil remaining which must be returned to the compressor, by injecting it into the suction line or, if relative pressures permit, directly to the crankcase. It is good practice to heat the oil or otherwise insure that it is above the temperature corresponding to the saturation pressure at the crankcase before it is returned. This heat can be, and often is, supplied by warm liquid by means of a heat exchanger. Sometimes heat is supplied from another source, such as electricity. This is not as efficient but its convenience may make it preferable.

Horizontal knock-out drums are also sometimes used. They may have baffles or mesh eliminators to remove the liquid from the gas flow.

PIPING

Pipe Sizing

Liquid lines are generally sized for limits of friction loss and possibly to avoid excessive velocities, particularly where quick shutoff valves such as solenoid valves are used. The discharge and suction gas lines are generally sized for reasonable friction losses in the case of horizontal or downward flow lines. Other treatment may be necessary where the lines result in an upward flow. This case is discussed in a later paragraph on arrangement and size of piping for oil return.

Pipe friction data are available for Refrigerants 12, 22 and 717 in Chapters 70 and 71 of this book, and in a number of other sources. Similar data for Refrigerants 50, 170, 290, 600, 601 or 1270 are available in petroleum industry handbooks. For Refrigerant 13 and some of the newer refrigerants, there is not as yet much prepared data available. Friction losses can be calculated from basic data such as the Moody curves when Reynolds number and relative roughness can be determined, but this takes more time than is desired in many cases.

It is often assumed that friction for Refrigerant 13 is reasonably close to that of the other halocarbon refrigerants under conditions where gases at similar specific volumes are compared. Results obtained from such an assumption will probably be useful but it should be always recognized that it is an approximation and consideration must be given to the effect of deviations of several percent from the true values.

Arrangement and Size of Piping for Oil Return

Oil return in the suction line of multi-stage systems encounters the same difficulties as in other systems as long as the oil remains reasonably fluid. For applications where the oil is at such a low temperature that its viscosity is high enough so that it will not run, it cannot be expected to return with the gas stream and must be separated, warmed up, and then returned to the compressor through separate means.

Oil which is low enough in viscosity so that it will flow will

Table 1 Sizing Data for Oil Return in Discharge or Suction Lines With Flow Vertically Upward^{a,b}

Saturation Temp, F	Line Size	
	2 in. and less	Above 2 in.
50 and above	0.25 psi/100 ft.	0.15 psi/100 ft.
0	0.35 psi/100 ft.	0.20 psi/100 ft.
-50	0.45 psi/100 ft.	0.25 psi/100 ft.

^a Friction values, at minimum operating flow rate, should not be less than these values.
^b From System Design and Installation Manual, Worthington Corporation, P. A. 37a-5.

encounter no problem in returning with the gas stream through horizontal lines or lines which are pitching downward in the direction of flow. Where the oil must flow up in an upwardly pitched or vertical line, it is necessary to maintain a certain minimum gas flow velocity which will exert sufficient friction drag on the oil at the pipe wall to cause it to move in the same direction. There are curves available for the minimum upward velocities for Refrigerants 12 and 22 (Chapter 70) but these are not generally available for other refrigerants.

To cause oil to flow upward along the walls of a pipe requires a certain minimum drag of the gas flow against the pipe. This is represented by some minimum friction gradient of the gas flow in the pipe line. Table 1 shows values for minimum friction gradients which have been found to be satisfactory in practice. Velocities which cause these friction gradients are very similar to those given in Chapter 70 on refrigerant piping systems for Refrigerants 12, 22 and 500.

To illustrate the use of Table 1, assume that the refrigerant flowing upward in a vertical line is at a pressure corresponding to zero F. If the line is likely to be larger than 2 in. diameter, oil can be expected to flow in the direction of gas flow if the line is sized to result in a friction of at least 0.20 psi per 100 ft.

CONTROL

Liquid Feed Control

In addition to the discussion in Chapter 36 of the 1961 GUIDE AND DATA BOOK the following comments apply, particularly to low temperature or low absolute pressure applications.

If liquid flow is controlled by means of low-side float valves, having their floats directly in the chamber where the level is being controlled, the low pressures and temperatures have no appreciable effect on their operation. External float chambers, however, must be thoroughly insulated to prevent heat influx which might cause boiling and an unstable level affecting the float response. Equalizing lines to external float chambers, particularly the upper line, must be generously sized so that liquid can reach the float chamber and the gas which results from any evaporation can return to the vessel with no appreciable pressure loss. (A loss of 0.1 psi corresponds to about a 2 in. column of Refrigerants 12, 22, or 13 and a 4 in. column of Refrigerant 717.)

Constant pressure expansion valves can be used if the system has only one evaporator, but it must be recognized that at low absolute pressures very small differences in pressure will have significant effects on the saturation temperature of the refrigerant.

For the direct-expansion type of evaporator the superheat controlled expansion valve (thermal or thermostatic) is most

likely to be used. It is important to realize that this type of valve actually operates on a pressure difference between that of the bulb which is responsive to the suction temperature and that below the diaphragm which is actually the suction pressure. Thus a Refrigerant 12 valve which operates with 10 F deg superheat at suction pressures corresponding to 40 F would require about 16 F deg superheat at an evaporating pressure corresponding to 0 F. Since it is often the case that the fluid being cooled is less than 16 F deg warmer than the evaporating temperature, there would be no possibility of providing 16 F deg of superheat to make the valve operate.

There are two means of avoiding this problem. One is to use certain cross charges (charged with a fluid other than the system refrigerant) in the power elements for the superheat control valve, for which one is referred to the manufacturer's data.

Another solution is to obtain the necessary superheat by providing a liquid to suction heat exchanger at the outlet of the evaporator. The expansion valve bulb is then placed on the compressor side of the heat exchanger so that the warm liquid being supplied to the expansion valve will give up some of its heat to the suction gas and thus provide superheat for the valve. Within the limits of satisfactory oil return as previously mentioned, this liquid-suction heat exchanger should be pitched slightly upward in the direction of flow. The gas flow cross section in this heat exchanger should be approximately the same as that of the main suction line. Double-pipe heat exchangers and some other types satisfying these requirements have proven satisfactory for this service. Shell-and-coil or shell-and-tube heat exchangers with liquid in the coil or tubes are usually unsatisfactory.

With a properly sized and designed liquid-suction heat exchanger it is practicable to use a liquid-charged type of superheat controlled valve to rather low temperatures. This is desirable in that the same valve is satisfactory over a considerable temperature range.

There is sometimes excessive hunting with this control, particularly if the exchanger is not of the best type. A part capacity bypass around the expansion valve or a constant source of heat (such as electric heat) to the suction line preceding the bulb, reduces the hunting tendency. Sizing and selection is critical and adjustment is usually required.

Although the thermal expansion or superheat controlled valve is designed to maintain a preset superheat in the suction gas, it must be kept in mind that the pressure sensing part of the system responds almost immediately to any change in conditions whereas the temperature sensing bulb must overcome a definite amount of thermal inertia before its effect is felt on the power element of the valve. For this reason when sudden compressor capacity increases occur, it is possible for the evaporator to boil over and the expansion valve to continue overfeeding for a few seconds before the bulb has sensed the presence of liquid in the suction line and cut down on the feed. It is for this particular reason that it is necessary to use knock-out drums, which have previously been discussed, on direct-expansion systems, especially low temperature systems.

Controlling Load During Pull Down

Since a compressor can handle a greatly increased mass flow of refrigerant at the higher absolute pressures corresponding to the starting conditions than it can handle at the low absolute pressures in the final operating conditions, some degree of capacity control must be included if very large motors and very large condensers are not desired. If the

motors and condensers are sized for the maximum rate of flow that can be expected during all stages of the pulldown periods, they will be considerably oversized for the final low temperature operating condition. In addition, if the compressor is allowed to operate at its full capacity during the pull down, the resultant rapid removal of the refrigerant gas from the evaporator will result in seriously increased probabilities of suction line stop over.

One method of avoiding these problems is to use compressor capacity control or compressor cylinder unloading. Properly controlled, this will reduce the capacity of the compressor to stay within the capacities of the motor and the condenser.

Another method is the use of a hold-back valve installed in the suction line which will throttle the flow as required to limit the suction pressure at the compressor to a predetermined maximum pressure. This would cause the compressor to operate at a controlled maximum suction pressure throughout the pull down period.

Use of the hold-back valve in multi-stage systems to prevent overloading during the pull down period has one distinct disadvantage. Such a valve will impose some friction resistance in the suction line even when the valve is fully opened. Auxiliary controls may be used so that the pressure drop will be only that required to pass through the fully open valve and not that required to hold the valve in the open position. Even this amount of pressure drop can be significant in reducing a compressor capacity on very low absolute pressure systems.

If pull down is required only occasionally the cost of the additional condenser and driving equipment can hardly be justified for the amount of time which must be invested for the pull down. In that case a hold-back valve which does not impose too much pressure drop in the suction line or unloading of the compressor becomes the proper answer.

With proper pilot connections to a large size hold-back valve, which is simply a pilot controlled piston valve, it is possible to use this same valve as a back pressure regulator which controls the evaporating pressure after the system has pulled down to the operating level. It is also possible by putting a solenoid valve in the pilot line to use the hold-back valve as a suction line stop valve.

Operation at Varying Loads and Temperatures

The means of controlling compressor capacity and evaporator capacity is about the same with multi-stage systems as with single-stage systems. This may be by compressor unloaders, hot gas bypasses, back pressure regulators, etc. One effect of low pressure on control systems is that the more significant increase in specific volumes of the refrigerant gas with respect to pressure drop results in a much greater capacity reduction when a decrease in load is accompanied by a decrease in evaporator pressure. Therefore a small pressure reduction may cause a very large percentage capacity reduction. These controls may be accompanied by hot gas bypasses into the evaporator or by the use of balanced loaders.

It is important to realize that when the capacity on one part of a cascade system is reduced by any means the load on the upper cascade will be also reduced and means must be made available to take care of the system control problems which this might create.

MOISTURE IN THE REFRIGERANT CIRCUIT

The subject of moisture in the refrigerant circuit and descriptions of the testing and drying procedures for refrigerant systems are discussed in Chapters 59 and 61 of the 1961