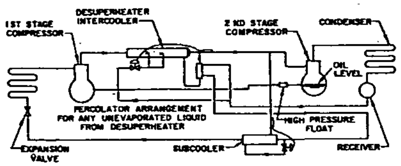

Fig. 2 .... Two-Stage Compound System with Subcooling and Intercooling

orating pressure at the refrigeration temperature requirements, it may be found that it has a pressure at the condensing temperature higher than that acceptable for the compressor or shells. When this occurs a cascade system rather than a compound system would be indicated.

Often there are additional refrigeration loads requiring evaporating temperatures corresponding to intermediate suction pressures which can be most efficiently handled by having some of the compression equipment (high stage) operating at or near this equivalent suction pressure. Then the upper stage or stages are used for the total refrigeration load and the low stage or stages as boosters for the low temperature load. See Fig. 4.

With compound units it is possible to subcool the liquid refrigerant to nearly the inter-stage temperature equivalent. (Low-stage compressor ratings are usually based on this being achieved by refrigeration effect from the high-stage compressor.) Also the discharge from the lower stage compressor can be sometimes desuperheated by available cooling water or other means besides the refrigerant. These features of the multi-stage system make it possible to get more refrigerating effect from a given power input. However, they add somewhat to the first cost since additional equipment is required, and proper economic study is indicated.

REFRIGERANTS

All known compression refrigerants can be, and probably have been used in multi-stage systems. Fig. 5 gives an approximate range of the temperature limits within which some of the more common refrigerants might be used.

If the refrigerant used is dangerously flammable (such as the hydrocarbons), it is recommended that their use be limited to pressures above atmospheric unless special precautions are taken to assure that no air enters the refrigerant circuit. If any air enters, it is likely to result in an explosion hazard. Some compressors are provided with special packing arrangements or double seals to permit the maintenance of a slight positive pressure next to the ambient even with subatmospheric suction pressure.

It must be pointed out that although it is common to think of very low temperatures with condensing in the normal range of around 100 F when discussing multi-stage systems, the present trend toward the use of heat pumps makes it not unusual to have to provide multi-stage systems for especially high condensing temperatures as well. The discussions in this chapter do not refer particularly to these high condensing temperature systems, although it should be realized that these conditions often make multi-stage application desirable.

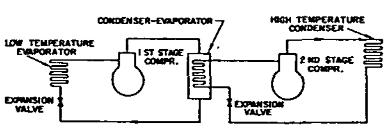

Fig. 3 .... Basic Elements of a Cascade System

EQUIPMENT CONSIDERATIONS

Compressors

The most common type compressor unit in multi-stage systems at the present time is the reciprocating compressor. If for some reason it is desirable to use a single refrigerant at

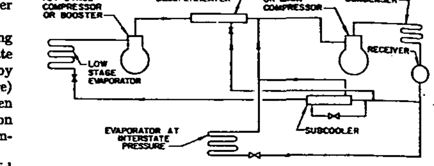

Fig. 4 .... Compound System with Evaporator at Interstage Pressure

a very low evaporating pressure, particularly below 2 psia, it is likely that a rotary or a centrifugal compressor will be necessary. Although the constant displacement type of compressor, which includes the rotary and reciprocating, will be the main one considered in this chapter, there are many practical and successful compound systems using centrifugal compressors. It is not uncommon to find applications of centrifugal compressors for the lower stages with reciprocating compressors on the upper stages on applications where Refrigerant 22, for instance, is evaporating as low as −130 F and condensing at about 105 F.

This chapter is not concerned with the details involved in the selection of a proper compressor size. It is quite important to realize, however, that at low absolute pressure levels a small increase in suction pressure will result in a rather large increase in the capacity of the compressor. The higher compressor capacities are accompanied by an increase in heat rejection which usually results in a significant increase in condensing temperature and a larger amount of power is required to drive the compressor. To avoid trouble, the motor must either be selected for considerably more power and the condenser for greater heat rejecting capacity than will be required from it at the normal operating condition, or means must be provided to reduce the capacity of the compressor during periods of pull down when the evaporating termperature will be higher than normal.

Condenser

Ordinarily the types of condensers used in multi-stage systems are the same as might be used for any single-stage system. However, since the heat rejection per ton of refrigeration varies over a wide range, no attempt should be made to select a condenser on the basis of tons of refrigeration but only on the basis of actual heat rejection. The cascade condenser, which operates between the two separate refrigerant circuits, is different in that to serve as a means for discharging the heat from the first circuit it must act as an evaporator of the second or high temperature circuit. The design of this piece of equipment must thus be such that it considers both condenser and evaporator design practice.

Since it is not at all unusual to find the heat rejection, at the start of pulldown, several times that of the normal low temperature operating condition, some thought must be given to sizing the condenser for the maximum amount of heat rejection that might be expected of it. Some means such as compressor unloading may be used to limit the maximum amount of rejection during pulldown. If the condenser is not sized for full load pulldown conditions and there is no means for limiting capacity of the compressor for this period, the result will be an extreme increase in condensing temperature through an undersized condenser and therefore, an undue increase in compressor discharge pressure which will probably result in shutdown of the system. Thus it would be possible to design a system which would work very satisfactorily and even economically at the design operating low temperature condition, but which, without external aid, could never reach that condition from normal ambient conditions.

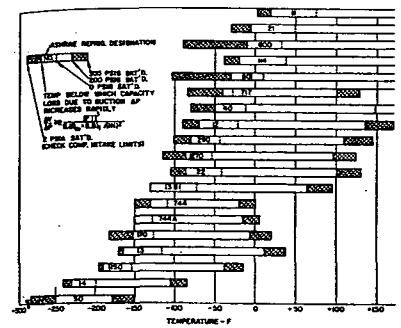

Fig. 5 .... Equivalent Temperature Limitations for Various Refrigerants in Reciprocating Compressors

Evaporator

There are several basic types of evaporators which may be used in multi-stage systems. Some particular problems become more acute in low temperature multi-stage systems than in the average higher temperature units. One is the ability to return oil from the evaporator to the compressor. Another is the increase in effect of the depth of the pool of liquid refrigerant in the evaporator or the pressure drop through the evaporator. Also the flow of evaporator gas (cfm) per ton increases with decreasing temperature.

Mainly because of the improved facility of oil return as well as smaller charge of refrigerant required, a direct-expansion type of evaporator is the most common. It is almost impossible, however, to design direct-expansion evaporators for some services and so a flooded type may appear more practical in these cases. The flooded evaporator should never be used without a full appreciation of the greater difficulties which will be experienced to effect oil return.

The effect of pressure losses in the evaporator is more acute in the low temperature systems because of the rather large change of both saturation temperature and specific volume with respect to change in pressure at these conditions. (Some refrigerants are more sensitive to these changes than others.) The depth of the pool of boiling refrigerant in a flooded evaporator results in a liquid head or static pressure which is exerted at some of the heat transfer surface. Therefore, the evaporating temperature at this surface is higher than that corresponding to the pressure in the suction line which is not under the effect of this static head. Direct-expansion evaporators, which generally do not have static liquid head to consider, do have pressure drops of the gas from the inlet to the outlet of the evaporator which results in much the same conditions. Therefore, some special consideration must be given to any evaporator which is to be used for low temperature service.

The liquid depth penalty of the flooded evaporator can be practically eliminated if the pool of liquid is below the heat transfer surface and a refrigerant pump is used to spray the liquid over the surface.

Another type of low temperature evaporator is the flash cooler, in which liquid refrigerant is itself cooled by the boiling off of some vapor. The liquid refrigerant may then be used as a brine. It is pumped from the flash cooler at the low temperature and absorbs heat during its passage through a secondary cooler or coil where heat is transferred to it from the product or material being cooled. The liquid refrigerant temperature is thus raised, but since its pressure is maintained sufficiently high by the circulation pump, it does not evaporate until it returns to the flash cooler. The significant difference between the flash cooling system and the other evaporators is that heat is transferred to the refrigerant by convection rather than by the evaporation of the refrigerant.

ACCESSORY EQUIPMENT

Subcoolers

In order to take full cycle advantage of the two-stage system, it is necessary to cool the refrigerant liquid to a temperature equivalent to the interstage pressure or nearly to that temperature. This significantly reduces the amount of flash gas which must be handled by the first stage compressor. The gas evolved from the subcooler is carried only by the higher stage or stages of compression. The net result is a reduction of total power requirements and of heat rejection to the second stage.

Fig. 6 illustrates a subcooler of the open or flash type. This is the simplest and least costly type of subcooler. It has the advantage of cooling the liquid to the temperature which is

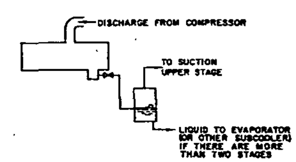

Fig. 6 .... Flash-Type Subcooler