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Cascade refrigerating systems—state-of-the-art

The cascade system, employing two separate refrigerating cycles with different refrigerants, connected by a heat exchanger, has largely supplanted the compound system as a means of attaining temperatures in the range of −40 to −120 F or lower. The reasons are partly economic: The low-stage compressor and motor for a low-temperature refrigerant will be smaller and lower priced than the low stage for one of the standard refrigerants, which may have a specific volume of 20 ft^3/lb or higher at very low evaporator temperatures. Other reasons include better inherent stability, lower cost of operation, ease of maintenance and better service experience.

When compression ratios exceed 10 or 15, volumetric efficiencies get very low, and discharge temperatures go up. Most of our current reciprocating compressors are built basically for air-conditioning or commercial applications, where the design compression ratios are from 3:1 to 8:1. So, when the refrigerant evaporating temperatures get down below —30 or —40 F, the standard vapor compression cycle becomes impractical and we go to staging, preferably cascade.

The high stage is fairly standard. Depending on the level of the low stage evaporating temperature, the high stage evaporating temperature will be in the range of −30 to +20 F, and a standard commercial or air-conditioning compressor will operate satisfactorily.

High stage refrigerants may be Refrigerants 12 or 502. Refrigerant 22 is still in use, but Refrigerant 502 offers the advantage of lower discharge temperatures at the fairly high compression ratios sometimes necessary, and is recommended over Refrigerant 22 for this reason. The normal precautions of Refrigerant 502 practice should be followed.

The cascade condenser is the component which ties together the high stage and the low stage. It is a heat exchanger; on one side the high-stage refrigerant evaporates, on the other the low-stage refrigerant condenses. In other words, the high stage is the "evaporative condenser" for the low stage. While many methods of construction have been suggested and manufactured, the preferred design for small systems seems to be a vertical shell and coil, fabricated of all non-corrosive metals. The coil is the high-stage evaporator, the shell is the low-stage condenser-receiver. Adequate insulation must be provided for 0 F or lower.

The flow diagram, Fig. 1, shows the high stage on the right and the low stage on the left. Several low-stage components not found in single-stage systems are of interest:

1. The desuperheater, here shown as air cooled. The expected discharge temperature of the compressor is 150 to 200 F, and an air-cooled desuperheater can probably reduce this to 100 F, so here is 8 to 15 Btu/lb of refrigerant circulated that will go into the air instead of into the high stage. If the high stage is water cooled, a small water heat exchanger can be substituted for the air-cooled exchanger. If the high stage is air cooled, the desuperheater can be made a part of the condenser assembly, eliminating the need for an extra fan.

2. The discharge-suction heat exchanger. Here we're doing two things, heating up the suction vapor and further desuperheating the discharge vapor. The objective is to raise the suction temperature to somewhere between 20 and 60 F, and to reduce the discharge temperature

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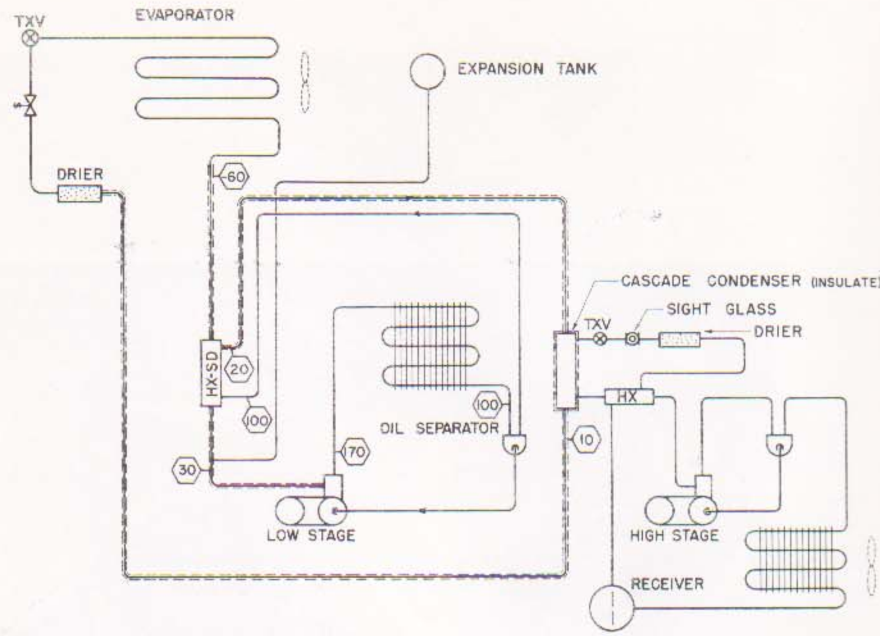

Fig. 1 Typical cascade system

to about 20 F above the condensing temperature. About 15 Btu/lb is removed from the low-stage discharge, so the heat remaining is not greatly in excess of the latent heat of condensation. This heat exchanger should not condense refrigerant. If the exchanger is pitched in the direction of suction flow, any liquid condensed will be trapped, since it will not flow uphill.

At discharge, the enthalpy of the vapor is about 75 Btu/lb, and the enthalpy of saturated liquid is about 10 Btu/lb. By use of the desuperheater and the discharge suction exchanger, we have reduced the enthalpy differential from 65 to 40 Btu/lb. In other words, we have reduced the high-stage load (and size) by nearly 40%. This is a worthwhile saving, applying to all of the high-stage components, including the cascade condenser.

The effect on the suction vapor is thermodynamically undesirable, but physically beneficial. The increase of specific volume amounts to 20 or 30%, while the temperature increases from 10 or 20 F above evaporating temperature to, say, 40 F. With 100 or more F of superheat, the vapor is certain to be dry. Furthermore, it will enter the compressor from an almost unfrosted suction line, not far from the temperature for which the compressor was designed. The advantages are greater than one might think: there is no need to run a compressor with a frosted head, something that always causes a service man to recoil in horror and phone for a replacement compressor. If we did not superheat the suction vapor in this way, it would be superheated anyway, from heat gain from the suction line and compressor cylinders.

3. An oil separator is essential.

4. There is no sight glass, since one cannot be read after operating temperatures are reached, and unsafe refrigerant charges are possible if one clears a bubbling, low-stage liquid sight glass.

5. If this is a small system, with a compressor of 10 hp or less, built-in modulation will not normally be available. Cycling the compressor for temperature control is undesirable, and in all probability the low stage will have been sized for fast pull-down rather than for a constant load at low temperature, so some form of capacity control will be needed. Just allowing the compressor to pump down is not satisfactory; on suction-cooled hermetics we need to keep vapor flowing through the motor. A good solution is to have two solenoid valves, one on the main liquid line, the other on a bypass line feeding a thermal expansion valve connected to the suction line upstream of the heat exchanger. The bulb of the expansion valve is attached to the suction line downstream of the heat exchanger. Interestingly, a Refrigerant 114 expansion valve works very satisfactorily in this service. The two solenoid valves work alternately; the bypass is open when the main liquid feed to the evaporator is closed.

6. An expansion tank is connected to the suction line, in order to create additional volume for the low-stage refrigerant to expand into during shutdown periods. The sizing can be determined from the pressure desired. Suppose we want to limit the maximum pressure to 150 psig at an ambient temperature of 100 F, at which the specific volume is .30 ft^3/lb. If the refrigerant charge is 6 lb, the system requires 1.8 ft^3 to contain the refrigerant vapor. If the system without the expansion tank doesn't have this volume, an expansion tank must be added to increase the volume to this value.

It will be noted that there is no liquid-suction exchanger on the low stage. In a system requiring frequent pull-down, such an exchanger will have such hot suction vapor at the beginning of operation that nearly all the liquid will flash in the exchanger, and a low rate of pull-down will result.

The conventional low-stage refrigerant is Refrigerant 13, on which information has been available for nearly 15 years. It is $CClF_3$, it boils at −114.6 F, and its critical temperature is 83.9 F, at which the critical pressure is 561 psia. Recently an azeotrope, Refrigerant 23/13, has been introduced. It will be called Refrigerant 503, it boils at −128 F, and its specific volume is somewhat less than that of Refrigerant 13. Its use will result in an increase in low-stage capacity, particularly at very low evaporating temperatures.

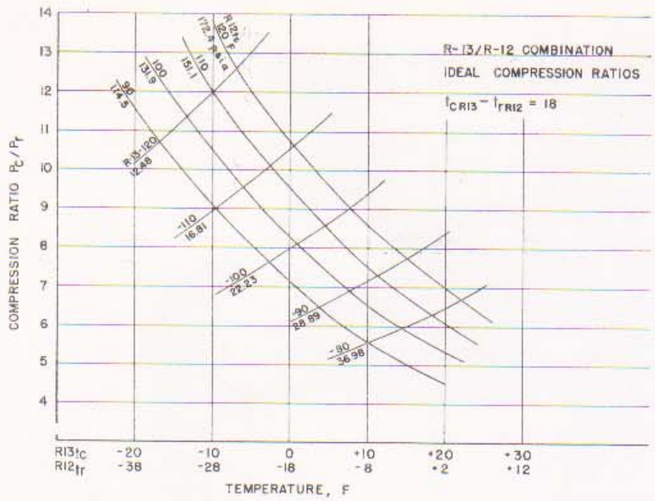

Fig. 2 Ideal compression ratios

The flow diagram shows expected temperatures at a number of points for a particular application with a chamber temperature of −65 F, a refrigerant temperature of −81 F. Note that the discharge line after desuperheating, the liquid line, and heat exchanger must all be insulated.

What probable compression ratios are likely to occur? Fig. 2 indicates the expected range for a Refrigerant 13/Refrigerant 12 combination, at low-stage refrigerant temperatures from −120 to −80 F, and high-stage condensing temperatures from 90 to 120 F. The range is from less than 6:1 to nearly 13:1. The curves are drawn on the assumption that the ideal occurs when the compression ratios of the low stage and high stage are equal.

Using ideal compression ratios, it may be of interest to see the effect of temperature on compressor displacement and horsepower. Table I shows the low-stage condensing temperature, compression ratio, refrigerating effect, specific volume of the suction vapor at 40 F, a probable volumetric efficiency, required compressor displacement and horsepower, all for a 6000 Btuh load. Refrigerant temperatures and horsepowers for the high-stage are also listed. As a rule of thumb, each reduction of 10 F in refrigerant evaporating temperature in this range adds 40% to the displacement and 50% to the horsepower.

It might be noted that field experience on 3500 rpm (2-pole) compressors at medium to high compression ratios has not been satisfactory; the volumetric efficiency falls off rapidly with increase of compression ratio.

Fig. 3 shows a graphical method of analyzing a particular cascade system. The system displayed has two 5-hp compressors, each with a displacement of 16.0 cfm. The high stage is air cooled, and it will be assumed that the condensing temperature is 100 F. The right hand curve is the capacity of such a condensing unit. The parallel diagonal lines represent evaporator capacity; a technique that most refrigerating engineers have used for determining the balance point of a particular condensing unit and evaporator for many years. The slope of the line, ordinate over abscissa, is the measure of evaporator capacity; the upper end is the temperature of evaporation, and the lower end is the temperature of the material being cooled. Since the evaporator in this case is the cascade condenser, the lower end of a line is the temperature of condensation of the low stage. Experience tells us that the condensing temperatures will range from −20 to +30 F. We should not operate much above +30 F, for at this point the condensing pressure of Refrigerant 13 is 278 psia. (The pressure can be controlled by a limit pressurestat in the low-stage suction which closes the main liquid solenoid valve on pre sure increase. Alternately, a high pressure cutout can be used to close the liquid solenoid valve on an increase in discharge pressure; this is less responsive.)

So we draw the parallel evaporator lines to the lower ends to intersect the abscissa at temperature decades from −20 to +30 F. The slope is determined so the temperature difference between condensation and evaporation near the midpoint is about 18 F—an experience number that gives reasonable size to the cascade condenser. The intersections of these lines with the capacity curve give the condensing capacity available at various condensing temperatures.

It is now necessary to calculate the capacity of the low stage at each decade of condensing temperature from

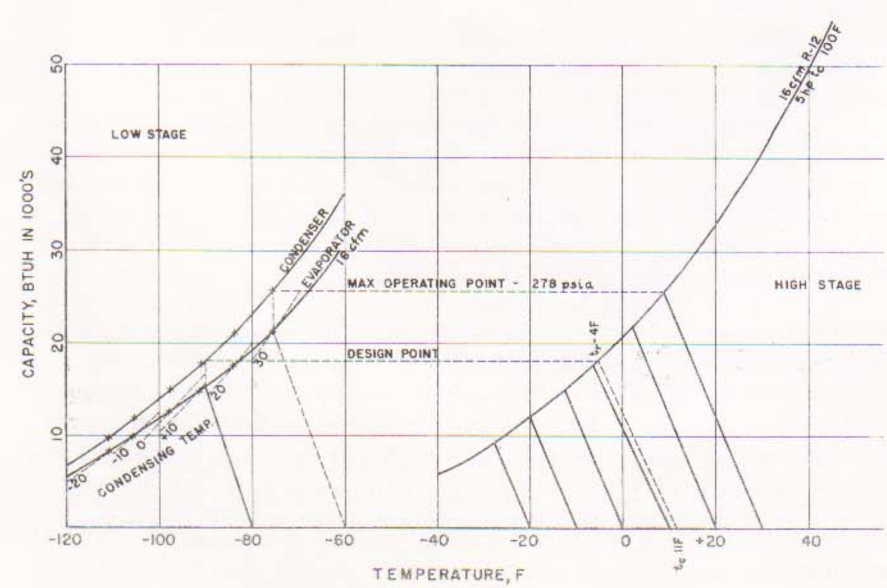

Fig. 3 Cascade system analysis

Table I. 6000 Btuh Load at Various Temperatures, Refrigerant 13/ Refrigerant 12 System

Low stage								High stage t_c 100 F, 7200 Btuh	
t_r	t_c	Compres ratio	Refrig effect Btu/lb	V_s at 40 F	η_v	Gross displacement	hp	t_r	hp
−120 F	−17 F	11.3	44.1	4.2	.44	21.6 cfm	7½	−31 F	7½
−110	− 6	9.5	41.9	3.0	.50	14.3	5	−24	4 or 5
−100	+ 1	8.1	41.2	2.3	.55	10.1	3	−17	3
− 90	8	6.9	40.4	1.8	.60	7.4	2	−10	2
− 80	14	6.0	39.7	1.4	.64	5.5	1½	− 4	1½

(Drop of 10 F adds 40% to displacement and 50% to hp)

−20 to +30 F, and to draw a series of parallel curves. The capacity is calculated by the relation

$$Q\ (\text{Btuh}) = \frac{q \times 60 \times \Delta h \times \eta_v}{V_s}$$

q is the displacement, cfm
Δh is the refrigerating effect, Btu/lb
η_v is the volumetric efficiency, decimal
V_s is the specific volume of the suction vapor at 40 F, assumed to be the temperature entering the compressor, ft³/lb

The volumetric efficiency varies in some inverse way with compression ratio, and can sometimes be obtained from the compressor manufacturer. If not available, it can be calculated from the displacement of the compressor and its capacities for a standard refrigerant, at known operating conditions.

After the parallel capacity curves are drawn, the final curve is determined as follows:

Draw a horizontal line from the high stage curve to a point 20% above the capacity curve for a particular condensing temperature. Mark the point and the point on the curve below it. Because of the large amount of desuperheating, the condensing effect (heat rejection) is assumed to be only 20% more than the refrigerating effect.

Other points are determined in the same way. We then have a series of points for the condensing capacity curve, and a series for the evaporating capacity. The points are joined into smooth curves. This process is necessary because the low-stage does not have a constant capacity heat sump, as is approximately the case in a single-stage system with a cooling tower or evaporative condenser.

With this diagram, one can determine the evaporating and condensing temperatures of the low stage over the entire range of air temperatures, and the evaporating temperature of the high stage. Because of the large number of variables, the original process is fairly laborious, but if a diagram is once drawn, the effects of changing the size of either compressor, or of changing the refrigerant, can be readily analyzed on the same diagram.

The cascade system is economical, stable and quite satisfactory, and these are good reasons for its increasing popularity.