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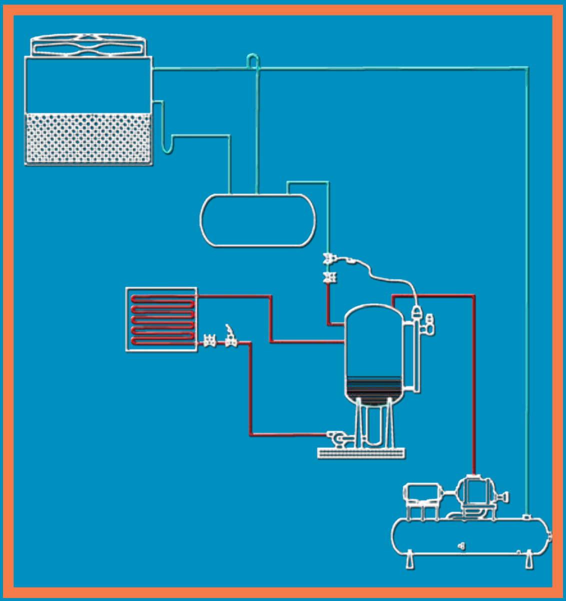

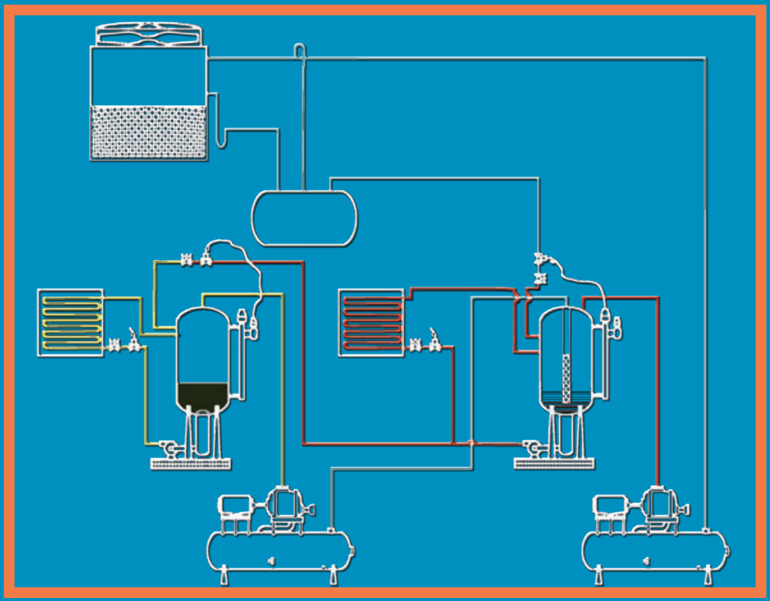

Single- or Two-Stage Compression

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Historically, the move toward multistage compression systems in industrial refrigeration applications was rooted in demand for lower operating temperatures. First generation compression technologies consisted of reciprocating compressors, but later rotary vane machines also were used. These two machines faced some key constraints in their operation at low temperatures. Reciprocating and rotary vane compressors have physical compression ratio (ratio of absolute discharge to absolute suction pressure) limits that are quickly approached as operating temperatures decrease. Additionally, compressors and the oil used in the compressors have limits on discharge gas temperature.

Most reciprocating compressors are limited to compression ratios on the order of 8:1. A common design saturated condensing temperature for an ammonia system is 95°F (35°C), which corresponds to a saturation pressure of 196 psia (1351 kPa). This fixes the high-pressure side of the compressor. The design low-side pressure of the compressor would be established to avoid exceeding the compression ratio limit of 8:1. With a discharge

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pressure of 196 psia (1351 kPa) and a compression ratio limit of 8:1, the resulting lower limit on suction pressure would be 24.5 psia (169 kPa), which corresponds to a saturated suction temperature of approximately –8.5°F (–22.5°C). With many food processing facilities operating low-temperature freezing systems below –40°F (–40°C), it is clear that reciprocating compressors only could be used in a two-stage or compound arrangement to overcome the compression ratio limit. Although modern day screw compressors are capable of serving a –40°F (–40°C) load in a single stage of compression, the overall system operating efficiency may be significantly lower than a two-stage arrangement.

The second factor limiting single-stage compression systems is the behavior exhibited by industrial refrigerants such as ammonia during the compression process. As ammonia is compressed, its pressure increases and because of its low heat capacity, it experiences a dramatic increase in temperature. With reciprocating compression technologies, high compression ratio operation requires an external source of cooling for the compressor (water or refrigerant-cooled heads). A high discharge temperature for a reciprocating compressor would be 250°F (121°C). High discharge temperatures tend to increase the rate of compressor lubricating oil breakdown as well as increasing the likelihood of compressor material fatigue.

The combination of compression ratio limits and refrigerant discharge superheat conspire to limit our ability to provide useful refrigeration in a single-stage compression arrangement with reciprocating and rotary vane compressors. As a result, early refrigeration pioneers overcame these constraints by conceiving, implementing, and refining multistage (compound) compression systems.

In this article, we explore single versus two-stage compression arrangements from an efficiency perspective. Operating efficiency changes with varying suction and discharge pressures are determined.

System Configurations

Before proceeding with an evaluation of the energy-efficiency characteristics of two-stage compression systems, we need to establish some terminology that will be used during our discussion. Multiple *stages of compression* are often combined with *multiple stages of liquid expansion* and *intercooling*. *Stages of compression* represent the number of compression steps required to raise the refrigerant pressure from suction to condensing. The term *liquid expansion* used here refers to the number of times liquid refrigerant expanded (reduced in pressure) from the condensing pressure until it reaches the lowest pressure level in the system. At each stage of liquid expansion (or throttling), the resulting flash gas is recompressed to a higher pressure level within the system. Two stages of liquid expansion can be implemented on two-stage compression systems and single-stage compression systems configured with two or more suction pressure levels. The term *intercooling* only applies to compound systems and represents the process of desuperheating the discharge gas from the low stage (or booster) compressors by direct-contact with liquid refrigerant maintained at the intercooling (high-stage suction) pressure.

We consider three system configurations in our efficiency comparison. The simplest configuration is a single-stage compression system with no loads present at the intermediate pressure. The next level of complexity is a single-stage compression system with the presence of loads at the intermediate pressure. The presence of intermediate loads allows evaluation of the efficiency benefits associated with two-stages of liquid expansion. And finally, a full-blown two-stage compression system with two-stages of liquid expansion and intercooling is considered. The three systems are shown in *Figure 1*. Another option that is not evaluated in this article is economized single-stage compression system arrangements. In some applications and operating situations, single-stage compression systems equipped with economized screw compressors can achieve efficiencies approaching two-stage compression arrangements.

In the present analysis, we focus on screw compressors equipped with external oil cooling because they are clearly the most commonly specified technology for industrial refrigeration applications today. All of the compressor performance information included in the analysis is based on manufacturers' selection programs across the ranges of operating pressures considered. The performance of the compressor is optimized (i.e., variable volume ratio or a properly chosen fixed volume ratio machine) for the given application conditions. In each case, the compressor's performance is based on the entire package, which includes pressure losses associated with suction and discharge trim. The refrigerant state entering the compressor is assumed to be saturated vapor and the liquid state leaving the evaporative condensers saturated.

Efficiency Comparison

The operating efficiency expressed as hp/ton for each of the three system arrangements shown in *Figure 1* will be necessarily affected by a number of factors. Some of these can be considered constraints such as the temperature requirements of the refrigeration loads at low suction and intermediate suction (if present). Other variables are uncontrolled such as the ambient conditions which will influence condensing pressure. We will investigate the comparison of single- versus two-stage compression arrangements over a range of suction pressures, intermediate pressures, and condensing pressures to accommodate this variability.

As a Function of Suction Pressure

The most obvious factor that affects the efficiency is the low-temperature suction pressure. *Figure 2* shows the effect of suction pressure on the efficiencies of the three systems for a fixed saturated intermediate pressure (SIP) of 30 psig (16.5°F saturated) (2 bar or –8.6°C saturated) and a fixed saturated condensing temperature (SCT) of 85°F (150 psig) (29°C or 10.3 bar). The condensing condition is chosen to approximate a yearly average condensing pressure for a refrigeration system and the 30 psig (2 bar) intermediate pressure as reflective of a typical high-stage suction pressure setpoint. The figure shows that as the compression ratio increases (suction pressure decreasing), the advantages of two-stages of compression and liquid expansion increases.

Two key points are identified in *Figure 2*. The point designated by ① indicates the point (–18.5°F [–25°C]) where the efficiency of the two-stage system (*Figure 1c*) is 10% better (lower hp/ton) than a single-stage system (*Figure 1a*) while ② indicates the point (–32°F [–35.6°C]) where the two-stage system is 10% better than the single-stage system with two-stages of liquid expansion (*Figure 1b*). It is also noteworthy that the efficiency advantage of the two-stage compression increases as the suction temperature decreases. At a saturated suction temperature of –12°F (–24°C) (8 psig [0.6 bar]) the two-stage and single-stage system with two-stages of liquid expansion have equal operating efficiency.

The advantage of a two-stage compression system is improved system efficiency, especially for lower temperature process requirements. The efficiency advantage does come with a price: more compressors. *Figure 3* shows the total compressor size (i.e., volume flow rate, cfm) per ton of low temperature refrigeration load. Across the range of saturated suction temperatures (–45°F to –5°F [–43°C to –21°C]), the two-stage system has an incremental compressor size of 1.5 cfm/ton to 2 cfm/ton (0.20 L/[s·kW] to (0.27 L/[s·kW]) compared to a single-stage system and 2.5 cfm/ton (0.34 L/[s·kW]) compared to a single-stage system with two-stages of liquid expansion. More compressors mean more maintenance, more machinery room floor space, etc. The two-stage system will result in lower compression ratio operation on each individual compressor which reduces wear and tear compared to a single stage system. These trade-offs are relatively complex and plant-dependent; however, they will affect the total life-cycle cost of the system.

As a Function of Intermediate Pressure

In the context of two-stage compression systems, the concept of an "optimum" interstage pressure arises. An optimum interstage pressure is one that will minimize the total power consumption for the system. A frequently applied approximation for this optimum intermediate pressure is given by:[1]

$$P_{opt,int} = \sqrt{P_{suction,sat} \cdot P_{discharge,sat}} \qquad (1)$$

where $P_{opt,int}$ represents an estimate of the optimum intermediate pressure, $P_{suction,sat}$ is the absolute saturation pressure corresponding to the suction conditions, and $P_{discharge,sat}$ is the absolute saturation pressure corresponding to the discharge or condensing conditions. This relationship equally divides the compression ratio between the low-stage and high-stage of a two-stage compression system. The actual optimum will depend on the operating efficiencies of individual compressors attached to each suction pressure level. In addition, there are other factors that may restrict the ability to vary the intermediate pressure such as temperature requirements for loads served by the intermediate pressure level of the system. Free from any constraints on intermediate pressure, *Figures 4* and *5* show the effect of intermediate pressure on the efficiencies of the three systems for a fixed saturated suction temperature of –40°F (–40°C) and –25°F

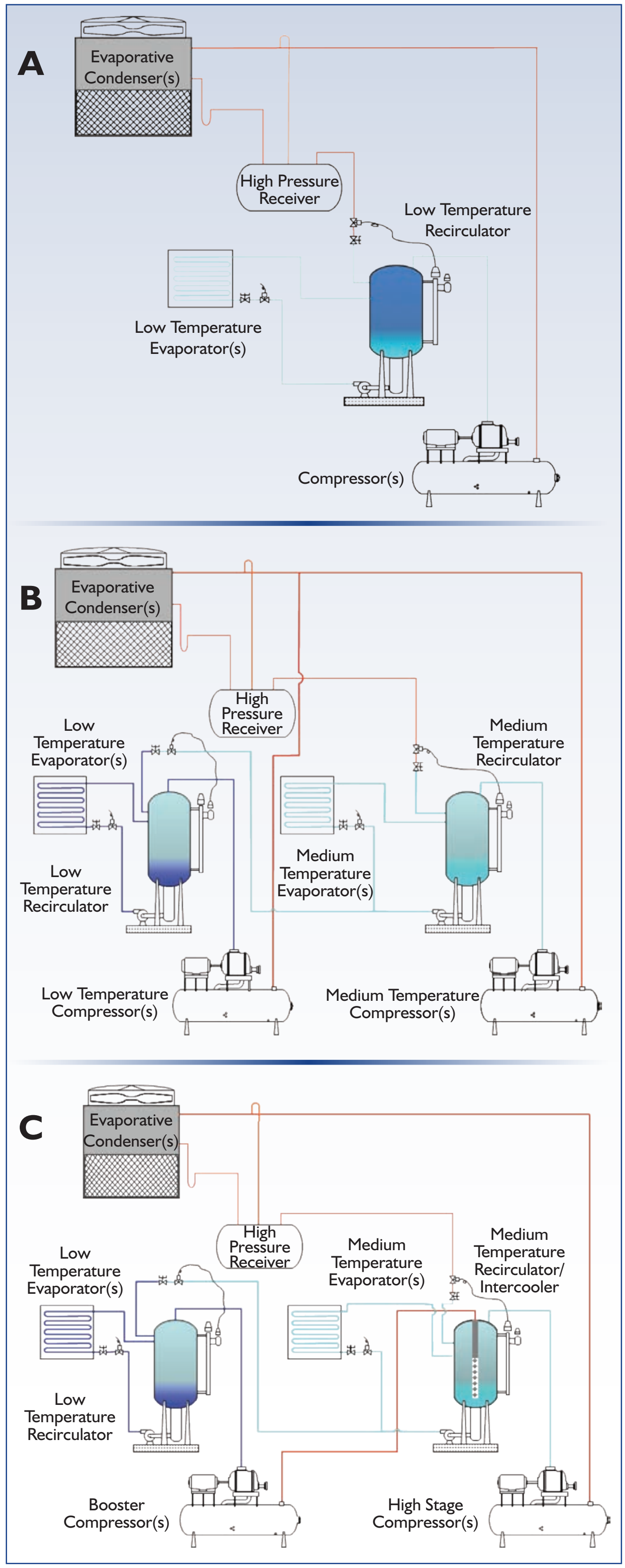

Figure 1: Schematics of refrigeration systems evaluated. 1a (top): Single-stage; 1b (center): Single-stage with two-stage liquid expansion; 1c (bottom): Two-stage.

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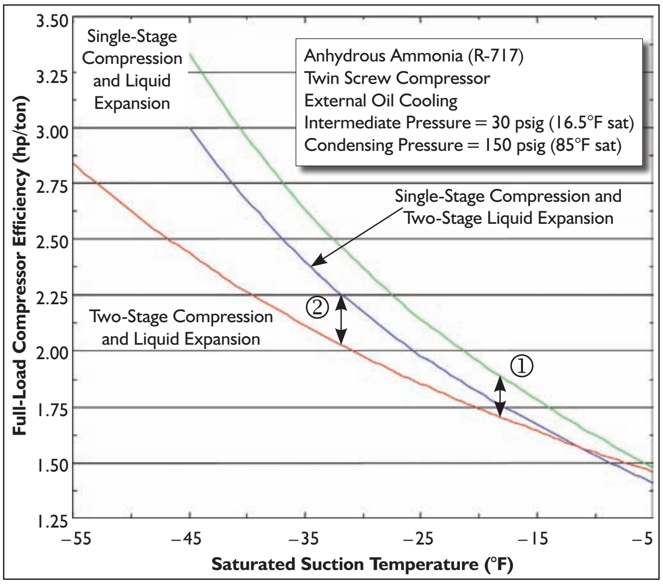

Figure 2: Refrigeration system efficiency as a function of low-temperature suction requirements.

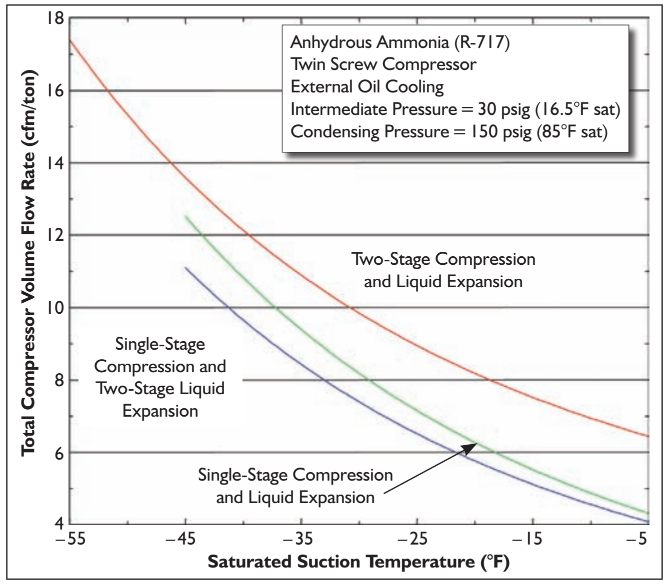

Figure 3: Compressor displacement per ton as a function of low-temperature suction requirements.

(–32°C), respectively. The condensing condition is fixed at 85°F (29°C) saturated which corresponds to 150 psig (10.3 bar). The ● symbol indicates the theoretical optimum intermediate pressure for a two-stage compression system. Notice that in this case, the actual optimum (i.e., lowest hp/ton) occurs at intermediate pressures slightly higher than the theoretical optimum. Also note that the optimum is fairly broad, that is, the intermediate pressure does not have a large effect on overall system efficiency. To illustrate that point, the results shown in *Figure 4* reveal that any intermediate pressure between 27 psig to 50 psig (1.9 bar to 3.4 bar) yields system efficiencies within 2% of the optimum.

In reality, most refrigeration systems do not have only low temperature loads. Nearly all plants will have refrigeration loads that demand higher temperature refrigerant: cooler spaces, bulk product storage tanks, production air-conditioning, post-pasteurization cooling, etc. The presence of these higher temperature loads will not significantly degrade the efficiency of meeting the low-stage loads due to the relatively broad optimum of the intermediate pressure. This is excellent news because the high-stage load temperature requirements can now dictate the intermediate pressure without compromising the efficiency of meeting the low-stage loads.

As a Function of Condensing Pressure

The last factor considered is the condensing pressure. *Figure 6* shows the effect of saturated condensing temperature (pressure) on the efficiencies of the three systems for a fixed intermediate pressure of 30 psig (2 bar) (16.5°F saturated [–8.6°C]) and –25°F (–32°C) saturated suction (1.2 psig [0.08 bar]). As the condensing pressure is reduced, the compression ratio decreases and at approximately 64°F (18°C) saturated (100 psig [6.9 bar]) the efficiency of the two-stage system is the same as for a single-stage compression system with two stages of liquid expansion.

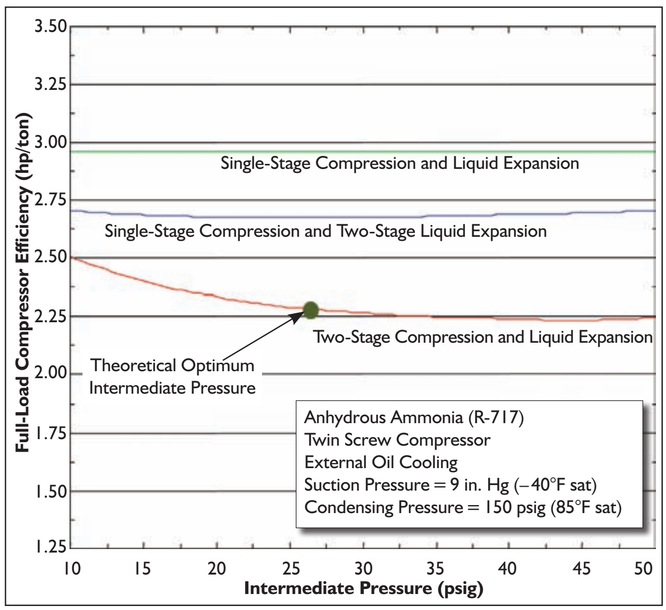

Figure 4: Refrigeration system efficiency (–40°F [–40°C] sat) as a function of intermediate pressure.

Conclusions

The use of two stages of compression is common in low-temperature industrial refrigeration systems. The efficiency benefit of a two-stage system increases as the temperature requirements are lowered and is weakly dependent on the intermediate pressure. The decision on whether to configure the system for multiple-stages of compression is one that should weigh both the advantages and disadvantages at the suction pressures required by the loads. As a general rule, two-stage compression systems should always be considered when loads demand low suction temperatures, particularly those lower than –25°F (–32°C) (1.2 psig [0.08 bar]).

Regardless of the number of stages of compression, configuring the liquid side for two-stages of expansion will nearly always in-

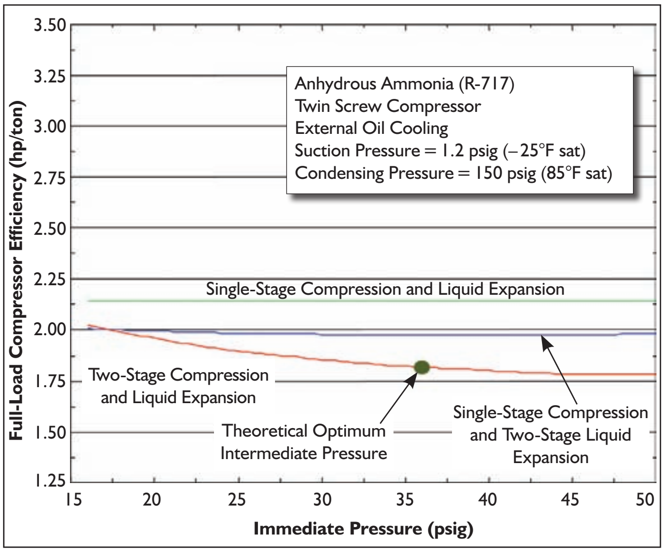

Figure 5: Refrigeration system efficiency (–25°F [–32°C] sat) as a function of intermediate pressure.

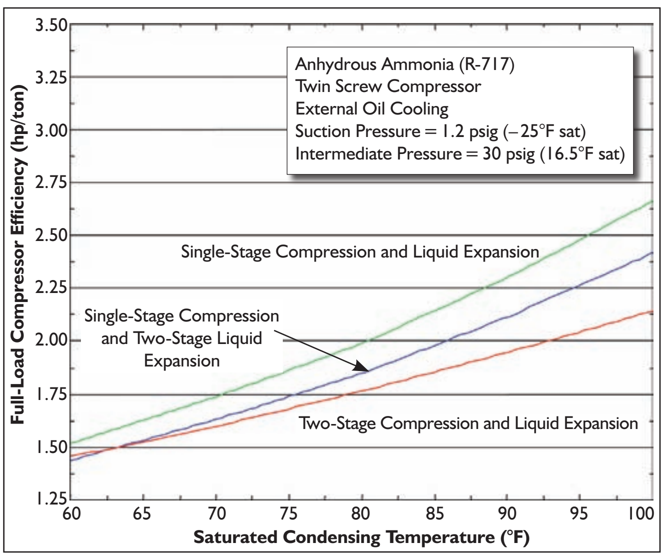

Figure 6: Refrigeration system efficiency (–25°F [–32°C] sat) as a function of condensing pressure.

crease the efficiency of the refrigeration system. The only possible exception would be a single-stage system with two temperature levels during low condensing pressure operation with very small coincident intermediate pressure loads (e.g., winter operation in cold climates). The condition results in a small amount of flash gas from liquid makeup to the low-temperature loads and often results in low part-load efficiency of the intermediate pressure compressor(s). In this case, it would be an advantage to be able to shut off the intermediate pressure compressors and provide liquid makeup to the low-temperature recirculator package directly from the high pressure receiver during winter.

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