The Electronic Newsletter of The Industrial Refrigeration Consortium

Vol. 6 No. 3, 2006

Single- or Two-Stage Compression?

In this issue of the **Cold Front**, we discuss the advantages and disadvantages of single- vs. two-stage compression systems. Along the way, we will also discuss the advantages of intercooling and two-stages of liquid expansion.

Introduction & History

Historically, the move toward multi-stage compression systems has its roots in the demand for lower temperature refrigeration.

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First generation industrial refrigeration compression technologies (primarily reciprocating compressors, but also rotary vane) faced two key constraints. First, both reciprocating and rotary vane compressors have physical compression ratio limits. Second, compressors and the oil used in compressors have limits on discharge gas temperature. Most reciprocating compressors are limited to compression ratios less than 8:1. The compression ratio of a compressor is simply the ratio of discharge pressure to suction pressure (both expressed in absolute terms). A saturated condensing temperature of 95°F [35°C] is a common design point and corresponds to a saturation pressure of 196 psia [13.5 bar]. At this design condition and a compression ratio limit of 8:1, a lower limit on suction pressure becomes 24.5 psia [1.69 bar], i.e. a saturation temperature of approximately -8.5°F [-22.5°C]. Clearly, reciprocating compressors operating in a single-stage of compression have a practical lower evaporating temperature limit much higher than many of the temperature demands of today's refrigeration processes.

The second factor limiting single-stage compression systems is the behavior that ammonia exhibits during the compression process. As ammonia is compressed, its pressure increases and with it, a dramatic increase in temperature. With reciprocating compression technologies, increasing compression ratios requires an external source of cooling for the compressor (water or refrigerant-cooled heads). A high discharge temperature for a reciprocating compressor would be 230°F [110°C]. High discharge temperatures increase the rate of compressor lubricating oil breakdown as well the likelihood of material fatigue (in the compressor) due to high temperatures.

Upcoming Ammonia Courses

Intermediate Ammonia Refrigeration
December 6-8, 2006 Madison, WI

Engineering Safety Relief Systems
December 11-15, 2006 (9-11 am CDT)
Anywhere via the web

Process Safety Management Audits
January 10-12, 2007 Madison, WI

Ammonia Refrigeration: Uncovering Opportunities for Energy Efficiency Improvements
February 14-16, 2007 Madison, WI

Introduction to Ammonia Refrigeration
March 7-9, 2007 Madison, WI

Ammonia Refrigeration System Safety
April 11-13, 2007 Madison, WI

Engineering Calculations for PSM & RMP
May 16-18, 2007 Madison, WI

Design of NH_3 Refrigeration Systems for Peak Performance and Efficiency
September 10-14, 2007 Madison, WI

Introduction to Ammonia Refrigeration
October 16-18, 2007 Madison, WI

See www.irc.wisc.edu/education/ for more information.

Noteworthy

- The IRC will hold its annual **Research & Technology Forum** on February 8-9, 2007 in Madison, WI. Put this event on your calendars – you won't want to miss it. The 2007 Forum will feature presentations on Sustainability, Relief Valves, and the latest on research in the area of industrial refrigeration. Details on this event to follow shortly.
- Send items of note for next newsletter to **Todd Jekel**, **tbjekel@wisc.edu**.

The combination of compression ratio and refrigerant discharge superheat 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 developed and implemented the concept of multi-stage compression systems to reduce the compression ratio on individual compressors and intercooling to remove the discharge superheat from the first stage of compression.

For screw compressors, the compression ratio limits are relaxed (generally up to 18:1 compression ratios) and the refrigerant discharge gas temperature is suppressed due to larger flow rates of oil within the compressor to provide sealing and lubrication. A high discharge temperature for a screw compressor would be 195°F [90°C]. The application of screw compressors allow for the consideration of single-stage compression in lower suction temperature cases that historically would have required two-stage compression systems.

System Configurations

Since screw compressors are capable of higher compression ratios, the need for two-stage compression becomes optional over a wider range of suction pressures. Let's consider the efficiency difference between single- and two-stage compression systems using screw compression technology.

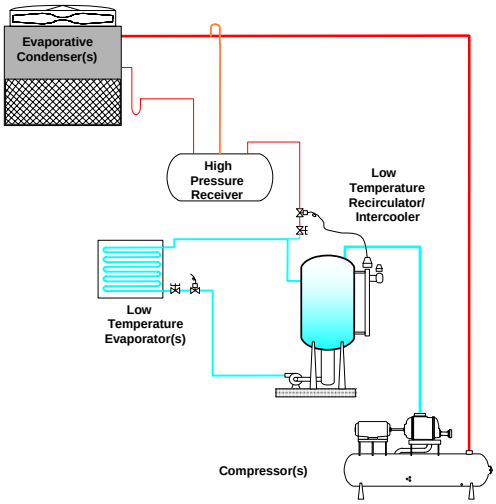
Before proceeding with coverage of the energy efficiency aspects of two-stage compression systems, we need to define terminology that we will refer to during our discussion. Multiple stages of compression are often combined with multiple stages of liquid expansion (throttling) and intercooling. The term "liquid expansion" used here refers to the number of times liquid refrigerant at the condensing pressure is expanded (reduced in pressure) to reach the lowest pressure level in the system with the resulting flash gas being recompressed to the next higher pressure level within the system at each expansion step. Two stages of liquid expansion can be implemented on both two-stage compression systems and single-stage compression systems configured with multiple operating suction pressure levels. The term "intercooling" only applies to two-stage systems and represents the process of desuperheating the booster discharge gas by direct contact with liquid refrigerant at the intermediate (intercooling) pressure. In addition to the reduced refrigerant temperature that would be sent to the high-stage, for anhydrous ammonia, intercooling also reduces the size of the high-stage compressors required.

We will consider three system configurations in the comparison. The simplest is a single-stage compression system without a higher temperature load that would dictate an intermediate pressure. The second level of complication is a single-stage compression system with the presence of a higher temperature load that fixes the intermediate pressure thus allowing for two-stages of liquid expansion. And finally, a full-blown two-stage compression system with two-stages of liquid expansion and intercooling. The three systems are shown schematically in Figure 1.

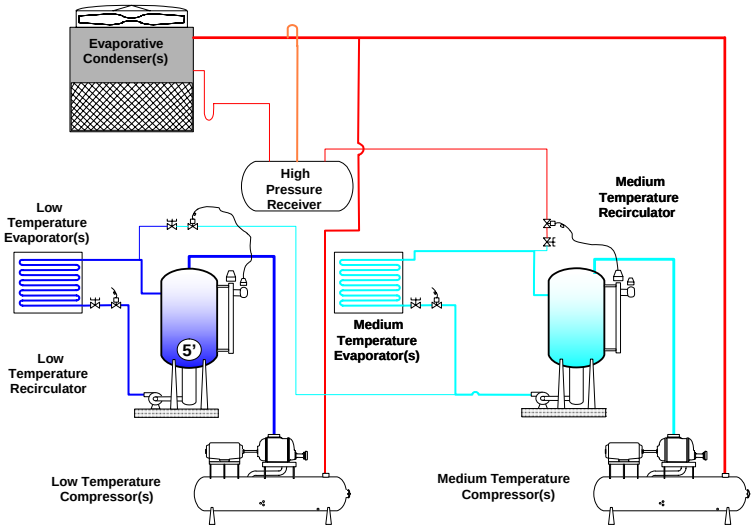
Screw compressors with external oil cooling will be used for the compressors since they are the most often specified compressor technology today. All of the compressor performance data used in this comparative analysis is taken from manufacturers' selection programs across the ranges of operating pressures considered. The performance of each compressor is optimized (i.e. variable Vi or properly chosen fixed Vi) for the conditions. The performance data (i.e. hp/ton) are for packaged compressors (i.e. including pressure losses associated with suction and discharge trim) with no suction superheat and saturated liquid from the condenser.

Efficiency Comparison

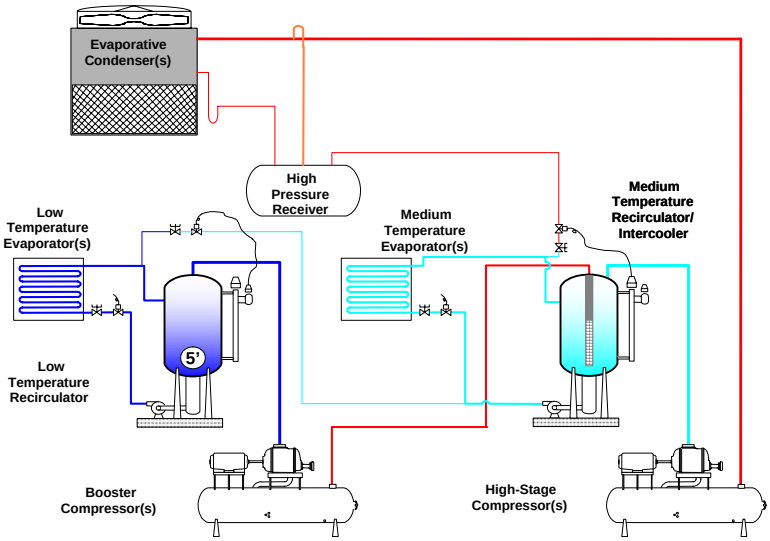
The efficiencies of each of these three different systems will be necessarily affected by the requirements of the refrigeration load (i.e. suction), the intermediate pressure (if any), and the



(a) Single-stage compression & liquid expansion



(b) Single-stage compression with Two-stage Liquid Expansion



(c) Two-Stage compression & expansion

Figure 1 Schematics of refrigeration systems considered.

ambient conditions (i.e. condensing). As such, we have a total of 8 degrees of freedom.

AS A FUNCTION OF SUCTION PRESSURE

The most obvious factor that affects the system efficiency is the low-temperature suction pressure requirement. FIGURE 2 shows the effect of suction pressure (expressed as saturated suction temperature) on the efficiencies of the three systems for a fixed saturated intermediate pressure (SIP) of 30 psig [2.1 bar(g)] (16.5°F [-8.6°C] saturated) and a saturated condensing temperature (SCT) of 85°F [29.4°C] (150 psig [10.3 bar(g)]). The condensing pressure is chosen to approximate a yearly average condensing pressure for a refrigeration system and 30 psig [2.1 bar(g)] intermediate as a typical high-stage pressure set point. The figure shows, expectedly, that as the compression ratio increases, the advantages of two-stages of compression and liquid expansion increases. Two points are shown on the figure specifically, ① indicates the point (-18.5°F [-28°C]) where the hp/ton of the two-stage system (FIGURE 1(C)) is 10% **lower** than a single-stage system (FIGURE 1(A)) and ② indicates the point (-32°F [-35.6°C]) where th hp/ton for the two-stage system is 10% **lower** than the single-stage system with two-stages of liquid expansion (FIGURE 1(B)). Another point to note is that at approximately -12°F [24.4°C] saturated suction (8 psig [0.55 bar(g)]) the two-stage and single-stage with two-stages of liquid expansion systems have the same efficiency.

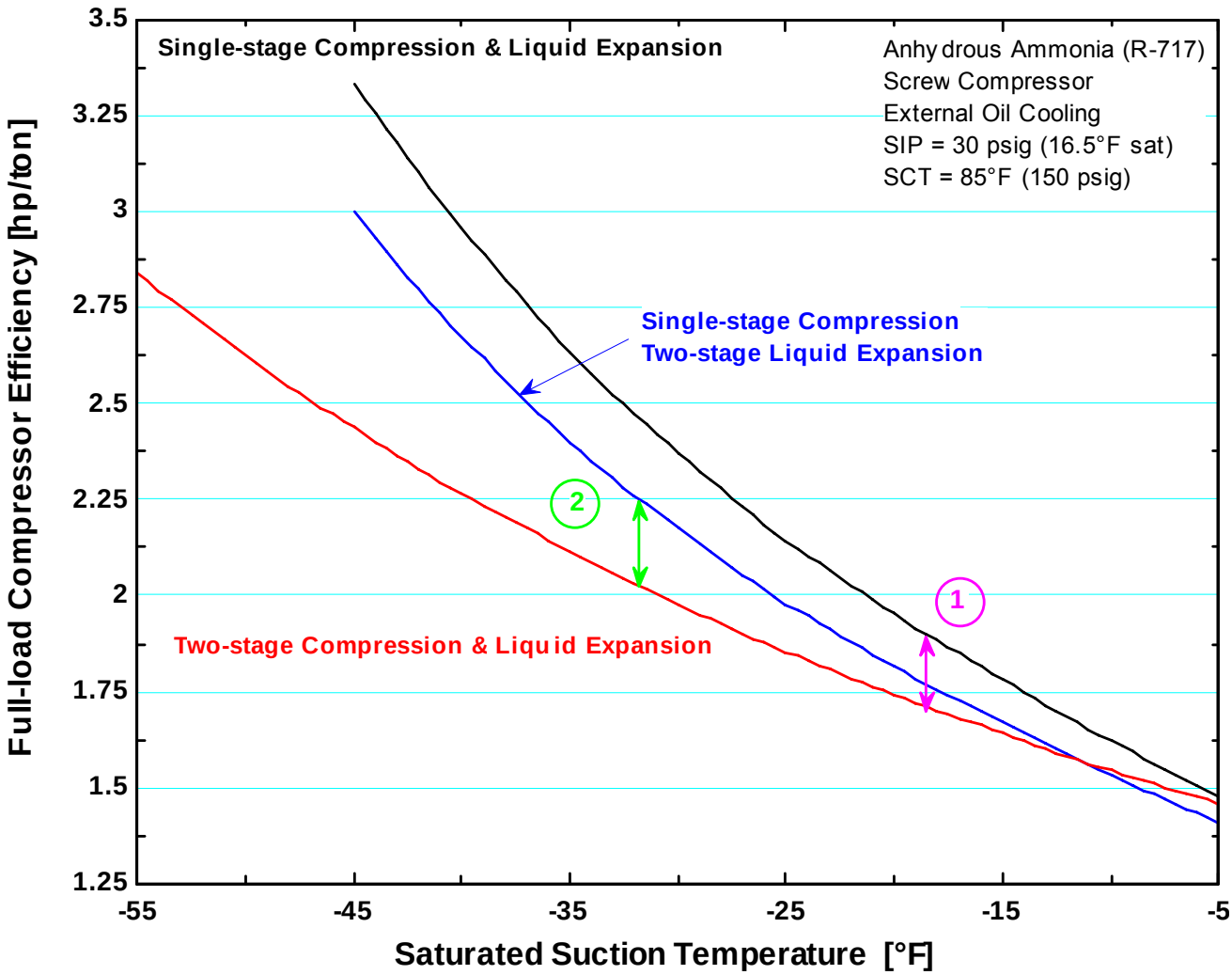


FIGURE 2 Refrigeration system efficiency as a function of low-temperature suction requirements.

The advantage of a two-stage compression system is increased system efficiency, especially for lower temperature process requirements, but that increased efficiency comes with a price: more compressors. More compressors mean more maintenance, more machinery room floor space, etc. How these factors play into the total life cycle cost of the system is beyond the scope of this article; however, you may have internal figures that allow you to account for these factors. **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 (SST) from -45 to -5°F [-42.8 to -20.6°C], the two-stage system has an incremental compressor size of 1.5-2 cfm/ton [0.75 to 1 m^3/h-kW_t] compared to a single-stage system and 2.5 cfm/ton [1.2 m^3/h-kW_t] compared to a single-stage system with two-stages of liquid expansion.

As a function of Intermediate pressure

Undoubtedly, you have heard of the presence of an "optimum" inter-stage pressure for a two-stage compression system. This "optimum" is a function of the total compression ratio on the system and theoretically (i.e. mathematically) is the pressure that results in equal compression ratio across each of the two stages. It is theoretical though, which means that it is unencumbered by the realities of the equipment used. **Figures 4 & 5** show the effect of intermediate pressure on the efficiencies of the three systems for a fixed saturated suction

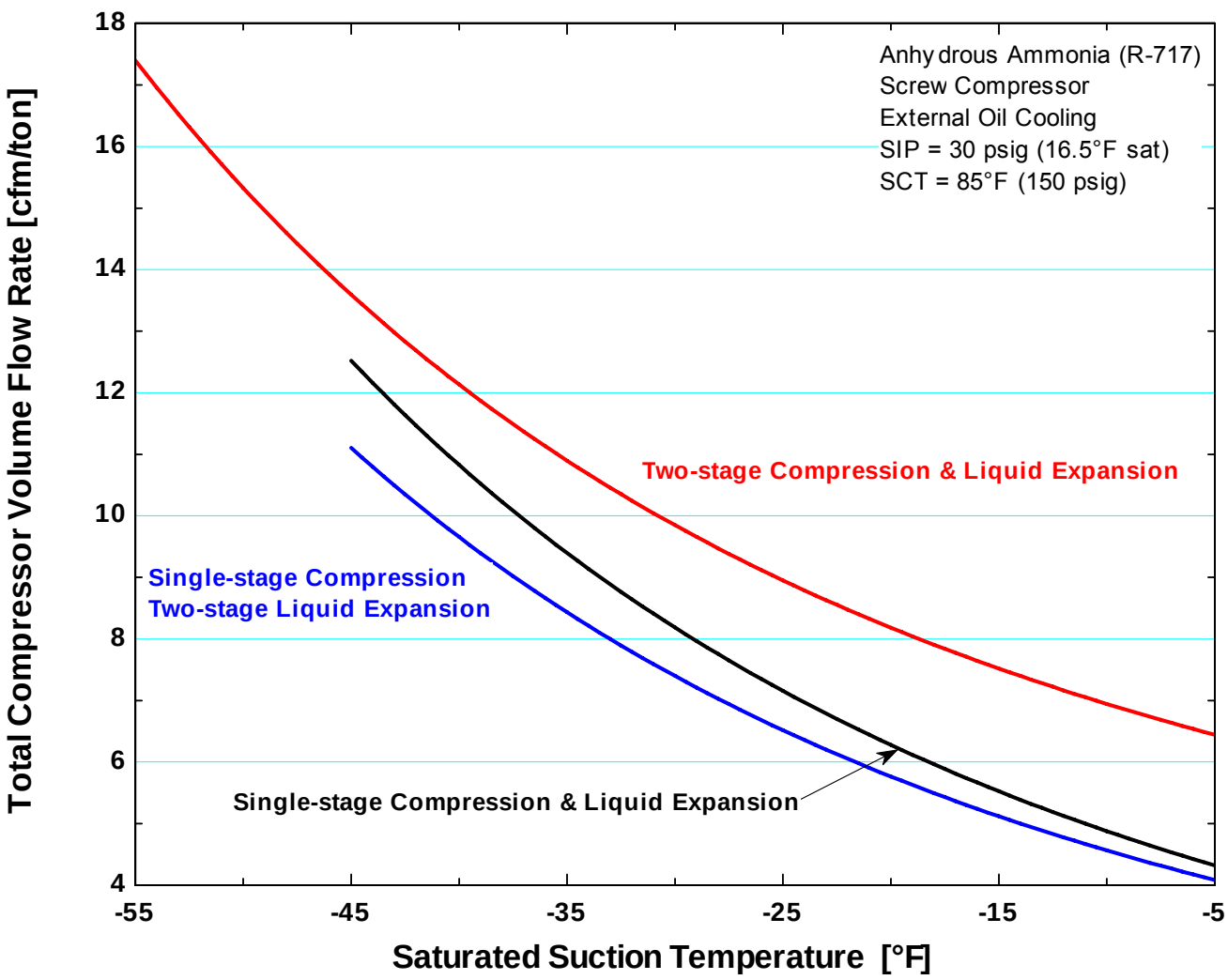


Figure 3 Compressor size (i.e. cfm) per ton as a function of low-temperature suction requirements.

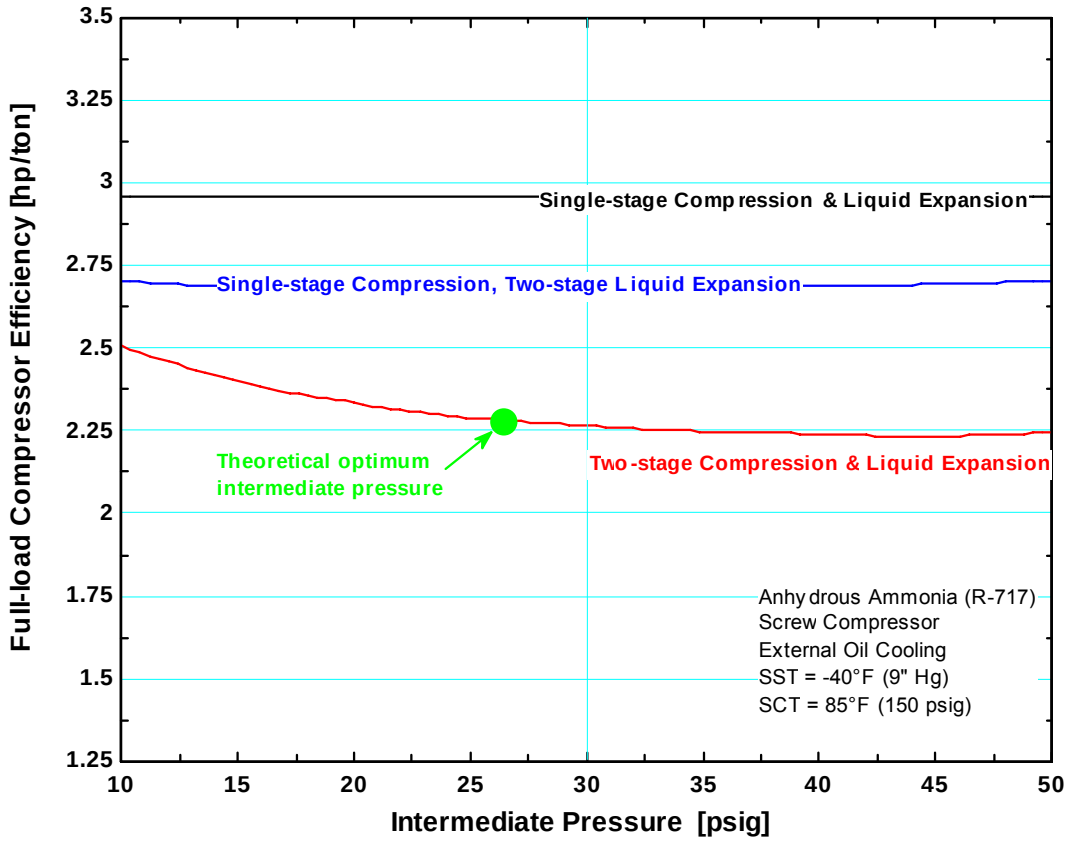


FIGURE 4 Refrigeration system efficiency (-40°F SST) as a function of intermediate pressure.

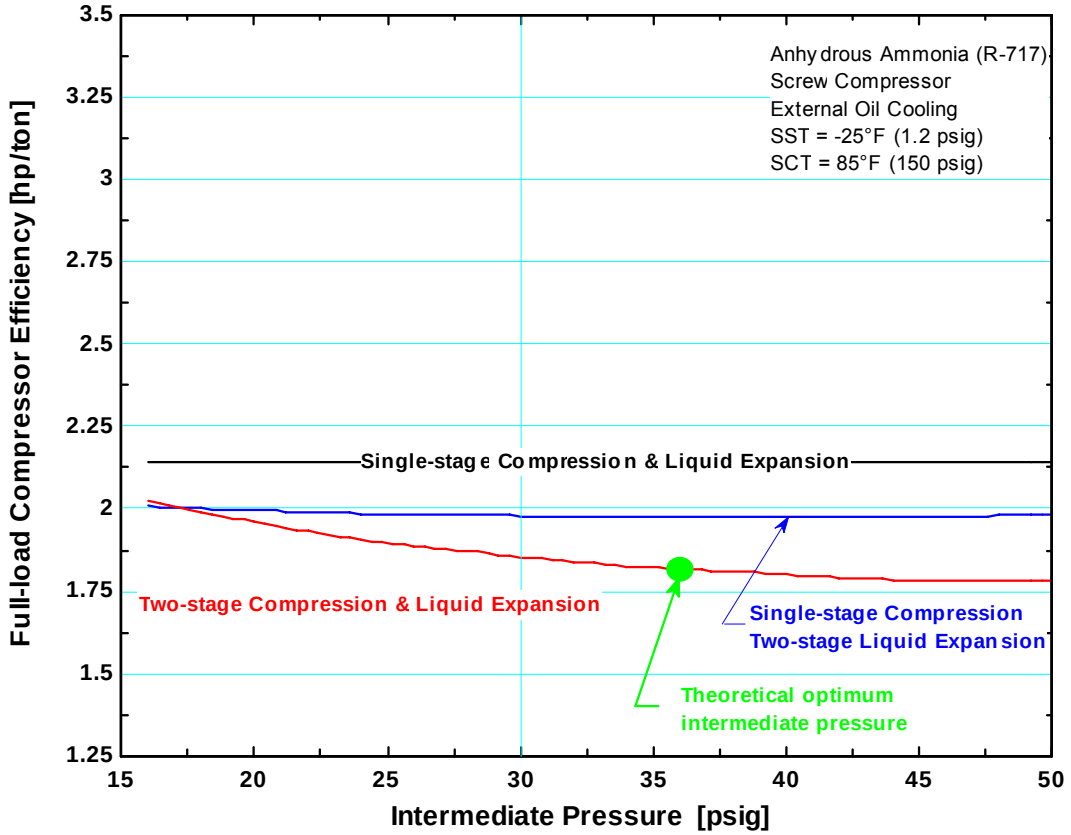


FIGURE 5 Refrigeration system efficiency (-25°F sat) as a function of intermediate pressure.

temperature of -40°F [-40°C] (9" Hg [-0.3 bar(g)]) and -25°F [-31.7°C] (1.2 psig [0.08 bar(g)]) respectively at 85°F [29.4°C] saturated condensing (150 psig [10.3 bar(g)]). The green dot (●) indicates the theoretical optimum interstage pressure for a system with two-stages of compression. Notice that the actual optimum (i.e. lowest hp/ton) occurs at intermediate pressures higher than the theoretical optimum. Also note that the optimum is wide, that is, the intermediate pressure does not have a large effect on the efficiency. To illustrate that point, in Figure 4, any intermediate pressure between 27-50 psig [1.9-3.4 bar(g)] gives system efficiencies within 2% of the optimum!

In reality, most refrigeration systems have loads at higher temperatures. Examples of higher temperature loads include coolers, bulk product storage tanks, production air-conditioning, post-pasteurization cooling, etc. These loads will also be added to the system. What we can see is that the presence of these higher temperature loads does not significantly degrade the efficiency of meeting the low-stage loads. This is excellent news because now we can let the temperature requirement of the high-stage loads dictate the intermediate pressure set point 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 [2.1 bar(g)] (16.5°F [-8.6°C] saturated) and -25°F [-31.7°C] saturated suction (1.2 psig [0.08 bar(g)]). As the condensing pressure is reduced, the compression ratio is reduced and at about 64°F saturated (100 psig [6.9 bar(g)]) the efficiency of the two-stage system is the same as for a single-stage compression system with two stages of liquid expansion.

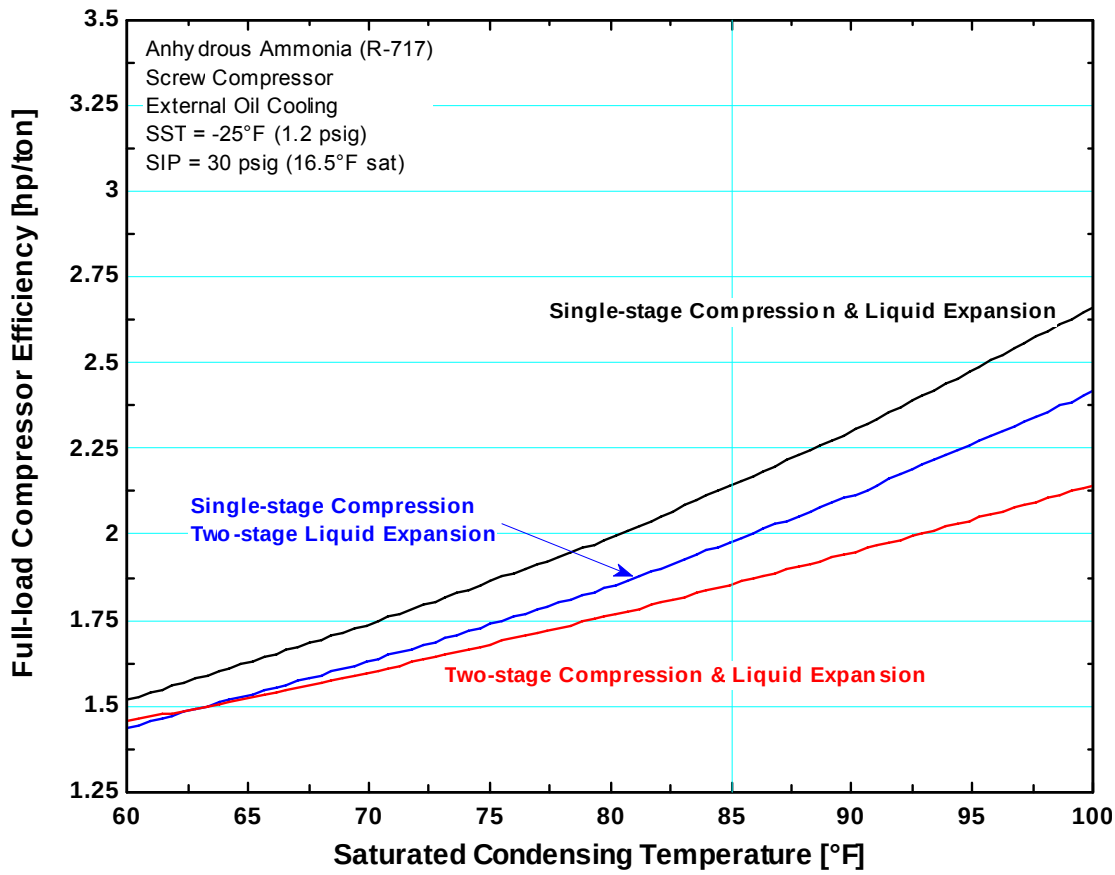


Figure 6 Refrigeration system efficiency (-25°F SST) as a function of condensing pressure.

Conclusions

The use of two-stages of compression is common in today's industrial refrigeration systems. With properly chosen compressor volume ratios, the efficiency benefit of two-stages of compression increases as the temperature requirements are lowered and is relatively independent of the intermediate pressure. The decision on whether or not to configure the system for multiple-stages of compression is one that should consider both the advantages & disadvantages at the suction pressures required by the loads. As a general rule, two-stage compression systems should be evaluated when the saturated suction temperature is lower than -25°F [-31.7°C] (1.2 psig [0.08 bar(g)]). Regardless of the number of stages of compression, two-stages of liquid expansion is an effective way to increase the efficiency of the refrigeration system.

Comments & Questions

If you have any questions or comments on the information in this article, please contact **Todd Jekel** at the **IRC (866) 635-4721**.

Other Resources

IRC, *Industrial Refrigeration Systems Energy Efficiency Guidebook*, 2006. Available for purchase by clicking here.

Stoecker, W.F., *Industrial Refrigeration Handbook*, 1998, McGraw-Hill.

Relief System Design Webcourse!

Held via the web **December 11-15, 2006** at **9-11 am Central**, this workshop is an ideal opportunity to develop or improve your understanding of engineered safety relief systems. Our primary focus is industrial refrigeration systems but many of the principles we will discuss apply equally to other applications as well.

Whether you are an end-user, equipment manufacturer, design engineer, or contractor, this course will help you build your capabilities in the area of the principles and practices of engineering safety relief systems. Participate and develop your understanding of:

- √ Codes and Standards related to safety relief systems
- √ Key aspects of engineering code-compliant relief systems
- √ Capacity determination for non-standard equipment like heat exchangers
- √ Methods for proper sizing of relief vent piping, including headered vent systems

In addition to the course, the IRC has developed a web-based safety relief systems analysis tool. This powerful tool has a high degree of flexibility to analyze, engineer, and document safety relief systems for industrial refrigeration applications. The tool features:

- √ Graphical user interface to configure relief system to be analyzed
- √ Ability to handle headered systems & multiple relief scenarios
- √ Quick and accurate algorithm to solve compressible flow equations
- √ Relief valve selection wizard
- √ Equivalent lengths for elbows & fittings included
- √ Detailed compliance checks for each system component
- √ One-click reports for easy printing

Access to the tool is provided free of charge to those completing this course. A brochure for the course is available on or website, or by clicking **here**.