IRC *TechNote*

Selection of Screw Compressors for Energy Efficient Operation

Industrial Refrigeration Consortium
College of Engineering
University of Wisconsin-Madison

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Screw Compressors: Selection Considerations for Efficient Operation

Douglas T. Reindl, Director
Todd B. Jekel, Research Scientist
Industrial Refrigeration Consortium
University of Wisconsin-Madison

Background

Compressors are a vital component in a vapor compression industrial refrigeration system. Compressors are the principle prime movers responsible for providing the motive force for circulating refrigerant throughout a system and for creating a sufficiently high refrigerant pressure differential that allows the transport of heat from spaces or products to the outside environment. Compressors are the single largest consumer of primary energy (usually electricity) in a refrigeration system. Although a number of alternative compressor technologies are available including: reciprocating and rotary vane, many refrigeration end-users are gravitating toward the specification and installation of screw compressors. For that reason, we have elected to develop this *TechNote* to provide more coverage on screw compressor selection and operation. The *TechNote* begins with a brief overview of screw compressor technologies, methods of capacity control, and volume ratio concepts. Then, energy efficiency aspects of screw compressor technologies are discussed for both fixed and variable volume ratio configurations. Finally, guidelines for screw compressor selection are outlined.

Overview

The first screw compressor was patented (#4121) in Germany by Heirich Krigar in 1878 [Cashflo 2000]. The modern day twin screw compressor was patented a half century later by Alf Lysholm in the 1930's and commercialized by the Swedish company, Ljunstroms Angturbin AB (today known as Svenska Rotor Maskiner or SRM) [Cashflo 2000]. In the time period from the issuance of the first patent on screw compressor technologies through today, screw compressors have undergone considerable advancement. Many of the advancements that underpin the success in industrial applications of this technology are the result of progress in computer-controlled machining equipment which facilitates manufacture of complex rotor geometries while maintaining close tolerances. Screw compressors and their application continue to be fertile ground for the issuance of patents. In the period from 1976-2001, over 370 patents on screw compressors and associated applications were issued by the U.S. patent office alone.

Screw compressors are available in sizes ranging from 50-3,000 BHP for application in refrigeration systems (commercial and industrial), gas compression, and air compressors. Screw compressors are the fastest growing compression technology in the industrial refrigeration marketplace today. Figure 1 shows the installation of an industrial refrigeration screw compressor in the machinery room of a plant.

Figure 1: Twin screw compressor installation.

Two screw compressor configurations are currently available for industrial refrigeration systems: single screw and twin screw. The single screw is a relatively new technology and has found good success in the industrial refrigeration marketplace considering its age. The twin screw compressor has been around since the 1930s and comprises 80% of the industrial refrigeration screw compressor market [Stosic, et al. 1997].

Screw Compressor Capacity Control

It is extremely rare for refrigeration loads in an industrial refrigeration system to remain constant. Rather, the loads on a refrigeration system vary with time. This variability may be weather dependent, product dependent, or both. The time-variability of loads could be on the order of minutes or days. The magnitude of the load can vary from full to no (or low) load during operating hours. Since the refrigeration load varies, industrial systems, and most importantly compressors, must be equipped with means of capacity control. Capacity controls will modulate the compressor's rate of gas compression to match the prevailing refrigeration load. There are multiple approaches for achieving capacity control with screw compressor technologies including: capacity control slide valve, poppet valves, twin slide valves (for both volume ratio and capacity control), and variable speed drives.

The most common approach for capacity control in screw compressors is the use of a "capacity control slide valve". Figure 2[1] shows an illustration of a single screw compressor with dual (capacity and volume ratio) slide valves. Suction gas flows from the right-hand side of the screw to fill the screw's groove or gulley. As the screw rotates, the volume of gas filling the gulley becomes trapped as the capacity control slide valve seals the trailing edge of the rotor's gulley. The volume of trapped gas is then decreased as it traverses the axis of the screw moving toward the discharge side of the compressor (left-hand side of the screw). As shown in Figure 2, the capacity control slide valve (upper slide valve) is in its maximum position, which traps the largest volume of suction gas to undergo the compression process. In this position, the compressor will have its maximum capacity.

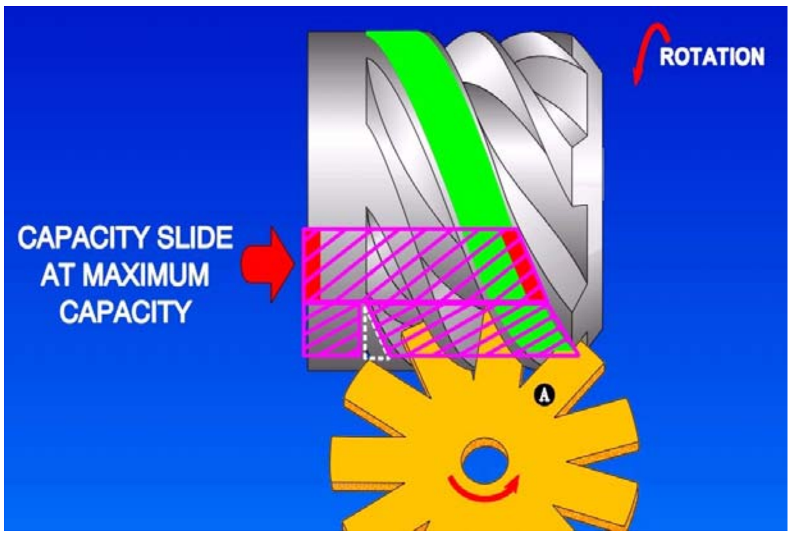

Figure 2: Illustration of a single screw compressor at full-load (source: Vilter Manufacturing).

The capacity control slide valve is capable of moving away from the suction port to unload the compressor until it reaches its minimum position. Figure 3 shows the same compressor but with the capacity control slide valve in its minimum position. By moving the capacity control slide valve toward the discharge port, the start of the compression process is delayed and the volume of suction gas captured is decreased. With the start of the compression process delayed, the volume of suction gas that can be sealed or trapped at the beginning of the compression process is reduced; thereby, reducing compressor capacity. It is important to note that the reduction in suction volume also decreases the compressor's effective volume ratio as we will see when we cover volume ratio concepts in the next section.

[1] For illustration purposes, schematics of the single screw compressor will be used because it is easier to visualize the concepts presented.

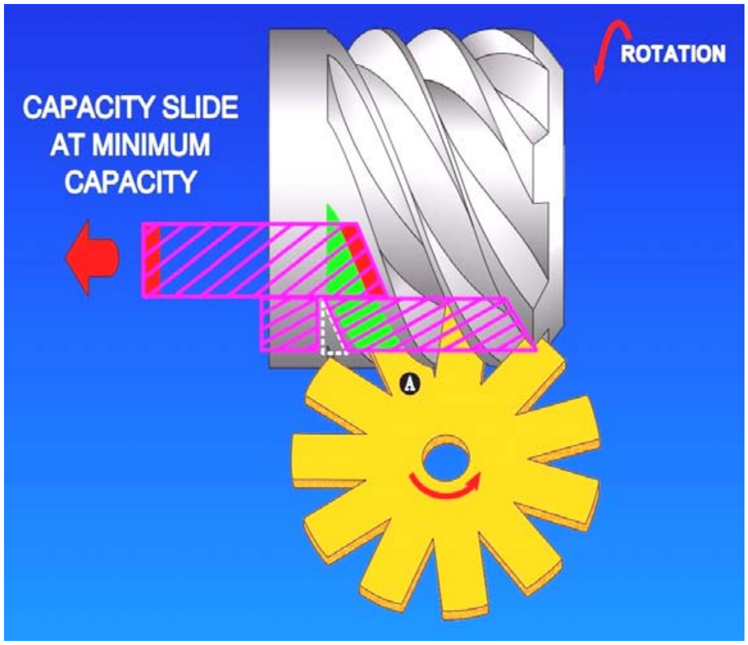

Figure 3: Illustration of a single screw compressor at part-load (source: Vilter Manufacturing)

Alternative to a capacity slide that affects the suction volume, some twin screw compressors are equipped with slide valves to decrease the discharge port area and allow some of the discharge gas to shunt back to the compressor suction (hot-gas bypass). This approach allows a single slide valve to achieve capacity control and volume ratio control; however, its operation results in an energy efficiency penalty and is not recommended for ammonia.

Volume Ratio Concepts

Inherent in their designs, screw compressors are "fixed geometric compression devices". That is, a screw compressor operates by a repetitive sequence of processes that involve:

1. drawing a fixed volume of gas through the suction port of the compressor housing and into the gulley between intermeshing screw thread threads
2. trapping a fixed volume of gas and
3. decreasing the trapped gas to a reduced fixed volume to expel it at a higher pressure on the discharge end of the screw (opposite the suction end).

An analogy to this process would be reciprocating compressor with cam-operated suction and discharge valves. In this scenario, the opening of the valves is purely a function of the piston position; therefore, the suction and discharge volumes are fixed.

The ratio of the volume of gas trapped in the thread of the screw at the start of the compression process to the volume of trapped gas when it first begins to open to the discharge port is known as the compressor's "volume ratio" or "volume index", *Vi* (ASHRAE 1996). Figure 4 illustrates the concept of a volume ratio for a screw compressor. The volume of trapped gas on the suction side internal to the compressor, $V_{suction,int}$, is larger than the volume of trapped gas on the discharge side internal to the compressor, $V_{discharge,int}$. The ratio of suction volume to discharge volume is the screw compressor's volume ratio or volume index, *Vi*.

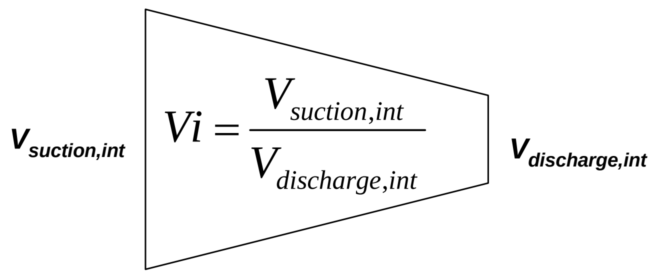

Figure 4: Volume ratio illustration for a screw compressor.

There is a relationship between the compression or pressure ratio a screw compressor is able to develop and the compressor's volume ratio as given by the following equation.

$$\frac{P_{discharge,int}}{P_{suction,int}} = (Vi)^k$$

where $P_{discharge,int}$ is the pressure of the trapped gas in the rotor gulley at a point just as the leading edge of the rotor gulley begins to uncover the discharge port and $P_{suction,int}$ is the pressure of the gas in the rotor gulley on the suction side of the compressor just prior to the rotor lobe closing off the suction port to begin the compression process, *k* is the specific heat ratio (constant pressure specific heat over the constant volume specific heat) for the refrigerant being compressed.

Historically, screw compressors were only available in fixed volume ratio designs. Today, screw compressor manufacturers have developed approaches to provide end-users of their equipment designs that vary the volume ratio for efficient performance over a wide range of operating conditions.

What is variable volume ratio screw compressor and why is it important? First, let's look at the relationship that volume ratio has with the operation of a screw compressor. Consider a screw compressor that has fixed volume ratio of 3.6. That is, the volume of trapped gas on the suction side of the compressor is 3.6 times larger than the volume of trapped gas on the discharge side of the screw compressor. In our example, the compressor is configured as a single stage operating at 0°F saturated suction temperature (30.4 psia saturation pressure). The pressure of the trapped gas just before the screw opens to the discharge port can be estimated by using the above equation assuming a ratio of specific heats of *k*=1.37 (for ammonia[2]).

$$P_{discharge,\,int} = P_{suction,\,int} \cdot (Vi)^k = 30.4 \cdot (3.6)^{1.37} = 176 \;\; psia$$

In other words, the pressure of trapped gas internal to the compressor can be expected, theoretically, to reach 176 psia just prior to the rotor lobe uncovering the discharge port opening.

In situations where the prevailing condensing pressure downstream of the compressor discharge is above 176 psia, the fixed volume ratio of the screw compressor will result in "under compression". In other words, the pressure of the trapped refrigerant gas internal to the compressor is not high enough to immediately move out into the discharge port of the compressor when the gulley of the screw begins to "see" the high-side pressure as the discharge port is uncovered leading edge of the rotor's lobe. In this case, the screw must continue to rotate further; thereby, decreasing the gas volume and raising its pressure to a level sufficient to force the compressed vapor out the discharge port. The impact of under compression is a loss in efficiency since the compressor is not raising the pressure of the gas high enough to immediately move it out the discharge port.

On the other hand, when the prevailing condensing pressure is below 176 psia, the screw compressor with a 3.6 *Vi* will "over-compress" the refrigerant vapor. If the condensing pressure is 125 psia, the screw compressor will have theoretically raised the refrigerant vapor to 176 psia (internally) just before the lobe on the rotor begins to uncover the discharge port and the over-compressed refrigerant will quickly drop in pressure as it flows to the significantly lower pressure in the discharge line. The impact of over compression is a loss in efficiency since the compressor works harder than necessary to accomplish the compression process. Another consequence of over-compression is that the discharge temperature (or oil cooling load) will be higher than if the compressor's volume ratio is matched to meet the required discharge pressure conditions.

Figure 5 shows both the compression ratio and the volume ratio for a compressor operating over a range of system discharge pressures with a saturated suction temperature of 0°F. As expected, the compression ratio increases proportionally with the discharge pressure for a fixed suction pressure.

[2] *k* for anhydrous ammonia varies in a range between 1.34 – 1.51.

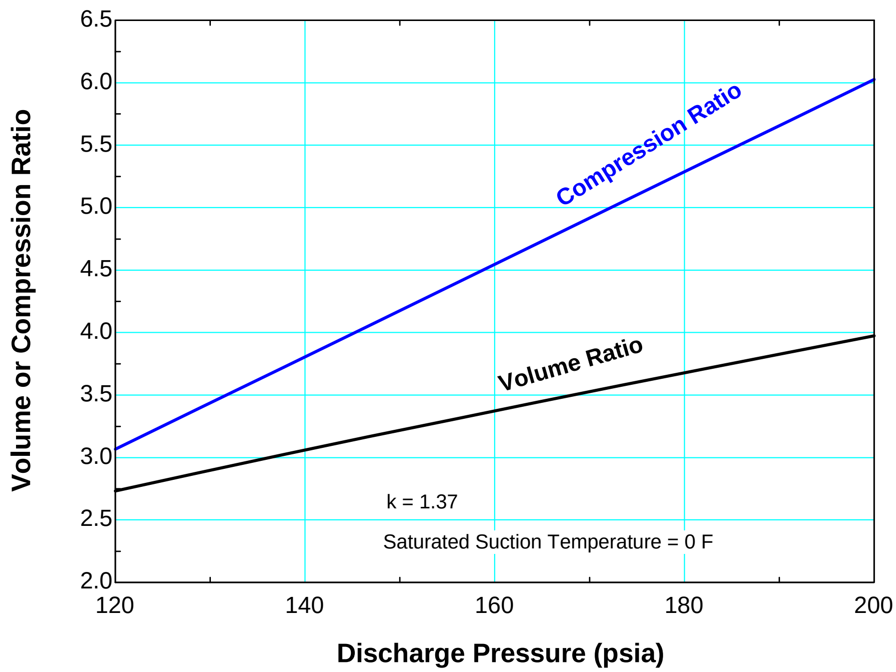

Figure 5: Compression and volume ratios for fixed suction pressure screw compressor operating over a range of system discharge pressures.

Figure 6 illustrates the volume ratio required to match condensing pressures that range from 120 psia to 195 psia (design summertime operation) over a range of suction temperatures. The variability in volume ratio is more pronounced for lower suction pressure conditions. Notice the volume ratio of a single compressor to serve the entire operating envelope shown would have to vary from 2.0 to 5.9. For a 20°F saturated suction temperature, the screw compressor's volume ratio would only need to vary from 2.0 to 2.5. At a -20°F saturated suction temperature, the compressor's span of volume ratio would increase from 4.0 to 5.8.

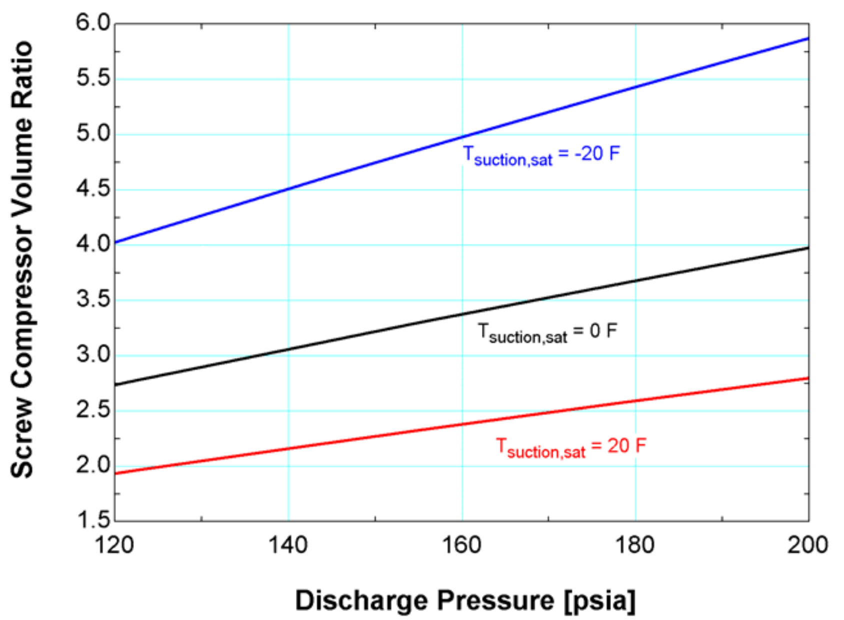

Figure 6: Variable volume ratio requirement for an ammonia compressor operating at over a range of suction temperatures.

Volume Ratio Control

Virtually all of the screw compressor manufacturers have developed the ability to vary the effective volume ratio of their machines. The general principle behind a variable volume ratio or variable *Vi* screw compressor is that the location on the screw at which the refrigerant vapor being compressed is allowed into the discharge port is movable. When a variable *Vi* compressor operates at low condensing pressure conditions, a "volume ratio slide valve" in the compressor moves in a direction toward the suction side. This allows the trapped refrigerant vapor out of the compressor earlier in the compression process before it has the opportunity to decrease the volume and overcompress the gas. During high condensing pressure conditions, the discharge of the compressed gas is delayed until further in the compression process (the volume ratio slide valve is moved away from compressor suction).

How is the volume ratio controlled in a variable volume ratio screw compressor? Compressor manufacturers have developed volume ratio control strategies that range from discrete (i.e. the volume ratio can be changed to discrete values e.g. 2.6, 3.2, 3.8, etc.) to infinitely variable. Figure 7 shows a dual slide valve configuration for a single screw compressor. The volume ratio slide valve provides "infinitely variable volume ratio control and is shown in its minimum position (minimum *Vi*). Essentially, the volume ratio slide valve in this position allows the trapped gas to leave the compression process at the lowest developed internal pressure. The volume ratio slide valve is typically found in this position during operation under low head pressure situations. The goal in varying the volume ratio slide valve position is to match the pressure of the gas trapped in the rotor groove or gulley just prior to uncovering the discharge port with the prevailing system discharge or condensing pressure. In contrast to the volume ratio slide valve that, effectively, repositions the location of the discharge port (i.e. the ending point for the compression process), the capacity control slide valve functions to reposition the location of the suction port (i.e. the starting point for the compression process).

For higher head pressure conditions, the compressor needs to raise the refrigerant to a higher pressure to match the system discharge or condensing pressure. To accomplish this, the variable *Vi* compressor operates with a larger volume ratio by moving the volume ratio slide valve to its maximum position as shown in Figure 8. In this case, the compressor has delayed the discharge of gas out into the system until later in the compression process; thereby, allowing the refrigerant to be compressed to a higher pressure.

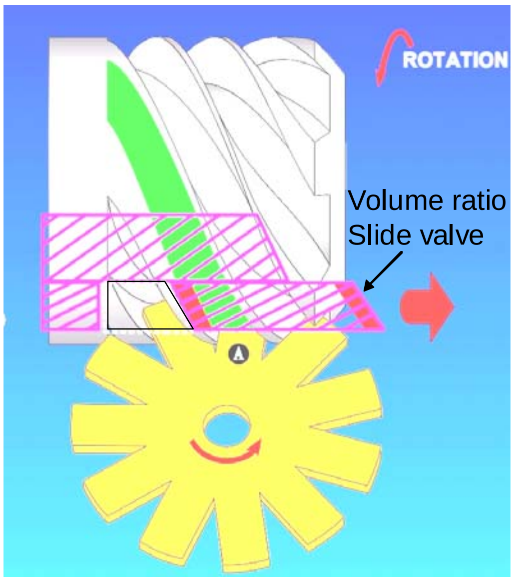

Figure 7: Single screw compressor volume ratio slide valve at its minimum position (source: Vilter Manufacturing).

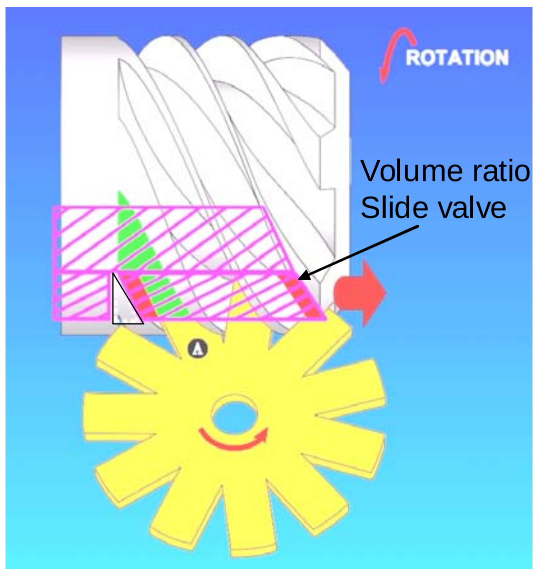

Figure 8: Single screw compressor volume ratio slide valve at its maximum position (source: Vilter Manufacturing).

It is interesting to note that if the compressor unloads (i.e. moving the capacity control slide) and the volume ratio slide valve position remains unchanged, the compressor's volume index would decrease since the volume of trapped gas at the suction is decreasing while the volume of trapped gas at discharge remains the same. In many situations, this is undesirable and would lead to under compression if left unchecked. To compensate, compressors equipped with dual slide valves are configured work in concert with one another. As the compressor unloads using its capacity control slide valve, on-board controls also decrease the volume ratio slide valve to establish and maintain the required volume ratio.

By better matching the compressor discharge pressure with the system discharge pressure, variable volume ratio screw compressors eliminate the inefficiency caused by over-compressing or under-compressing the refrigerant. Variable volume ratio screw compressors are often recommended for applications where the discharge (and suction) pressures will vary over the operational life of the machine. Good applications for variable volume ratio screw compressors include: swing compressors and compressors serving loads that have highly varying loads (pull-down or process). Recognize however, that variable volume ratio screw compressors will have higher maintenance costs and potential for greater reliability problems due to the operation of the volume ratio control slide valve, hydraulic circuitry, and controls.

Energy Efficiency Considerations

Fixed or variable volume? The volume ratio of a screw compressor should be one of the factors considered when selecting a new compressor or modifying the operation of an existing screw compressor installation. Fixed volume ratio compressors offer the advantage of lower capital cost, lower maintenance costs, and greater reliability; however, the chief disadvantage is diminished operational flexibility. In this section, we will evaluate the operational disadvantage more closely to determine the extent it constrains or limits the energy efficient operation of an industrial refrigeration system. We will look at both full-load and part-load performance of fixed volume ratio screw compressors and assess the impact of volume ratio mismatch that arises under both floating discharge and suction pressure operation. We conclude the section by comparing the energy performance of the fixed volume ratio machines with variable volume ratio designs. Note, the results of the analysis presented in this section are based on data as published in compressor manufacturer's selection programs.

Since fixed volume ratio compressors are available in a finite number of increments, designers must be judicious in the process of compressor selection to achieve efficient operation upon integration into a system. To explore the influence of volume ratio on compressor efficiency under floating head pressure conditions, the full-load compressor package efficiency for five unique volume ratios (1.7, 2.2, 2.8, 3.0, 3.5, and 5.0) is compared for three separate saturated discharge temperatures (95°F, 85°F, and 75°F) while operating at three separate saturated suction temperatures: 0°F, 20°F and -20°F. Figure 9 below shows results of analyzing manufacturer's performance data for a screw compressor operating at a saturated suction temperature of 0°F.

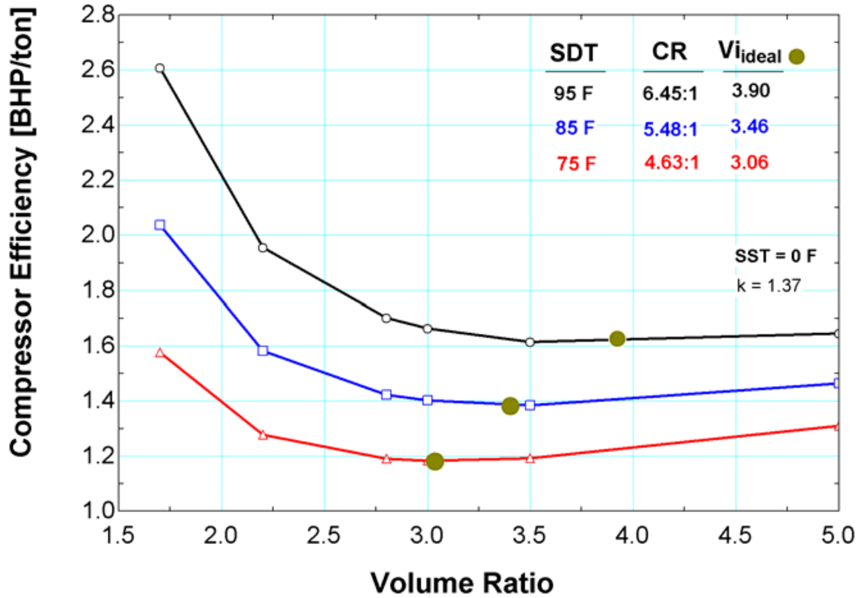

Figure 9: Influence of volume ratio on full-load efficiency for a fixed *Vi* compressor operating at 0°F saturated suction temperature.

For a fixed suction pressure, the compression ratio (and ideal volume ratio) changes as the discharge pressure varies. At 0°F saturated suction temperature and a typical design condensing or head pressure of 181 psig (95°F saturation temperature), the compressor needs to develop a compression ratio of 6.45:1. At this condition, an ideal[3] volume ratio for the screw compressor would be 3.90. The dots in Figure 9 denote the calculated ideal volume ratios for each condensing temperature case. A compressor selected with a lower or higher volume ratio than the ideal will operate at lower efficiency; however, the impact of the penalty for this suction pressure does not start to become significant until the volume ratio drops below 2.8.

If the compressor shown in Figure 9 operates at 95°F saturated condensing temperature with a fixed volume ratio of 1.7, it would require 2.60 BHP/ton. Increasing the volume ratio to 3.5 would decrease the required BHP/ton to 1.61 – a 38% improvement in efficiency! As the condensing temperature drops to 75°F, the ideal volume ratio will also decrease. At a condensing temperature of 75°F, a 3.5 *Vi* compressor would operate at 1.19 BHP/ton while the 1.7 *Vi* compressor efficiency would degrade to 1.58 BHP/ton. In this particular case, the penalty for the low *Vi* selection drops from 38% to 25% as one would expect since the compressor's compression ratio is decreasing which decreases the required volume ratio. At 75°F condensing temperature, a 3.0 *Vi* compressor is near optimal and its performance would be 1.18 BHP/ton which is less than a 1% improvement over the next highest available *Vi* selection at 3.5. In this case, selecting a screw compressor with volume

[3] The "ideal" volume ratio is estimated by applying the following: *Pressure Ratio* = Vi^k where *k* is the ratio of specific heats.

ratio of 3.0 would yield good performance over the entire range of condensing temperatures.

Figure 10 shows the influence of volume ratio on compressor efficiency for a fixed saturated suction temperature of 20°F. As expected, the volume ratio for peak efficiency is less than that for the 0°F suction case. Although the shape of the curves for the 20°F suction temperature differs somewhat from the 0°F case, both illustrate the importance of matching the compressor volume ratio with the anticipated operating conditions. The full-load performance at high suction temperatures is fairly insensitive to volume ratio over a broad range of fixed volume ratio choices. In this case, selecting a screw compressor with volume ratio of 2.8 would yield good performance over the entire range of condensing temperatures.

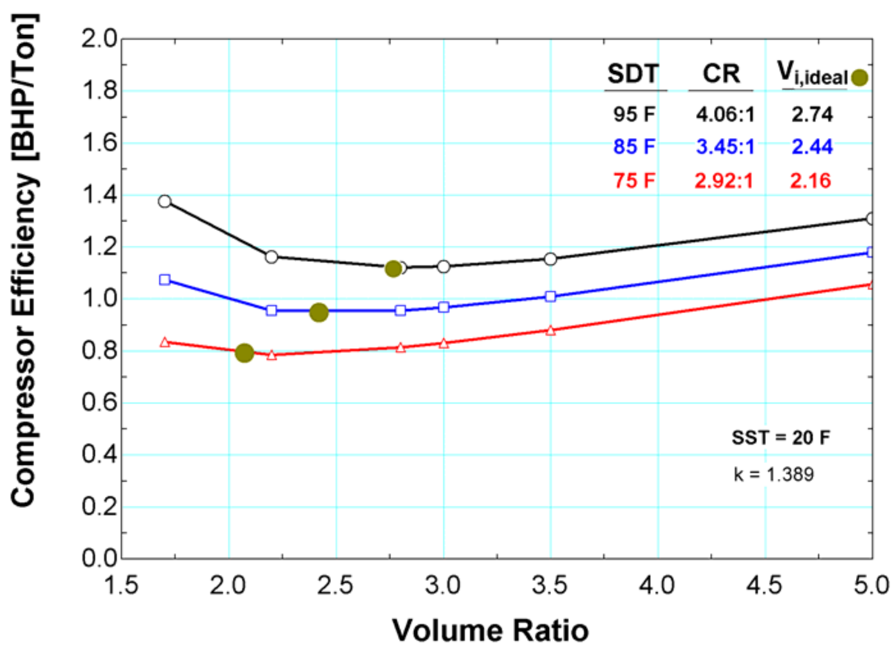

Figure 10: Influence of volume ratio on full-load efficiency for a fixed *Vi* compressor operating at 20°F saturated suction temperature.

Figure 11 shows the full-load performance of fixed volume ratio machines operating in a single stage at a fixed suction temperature of -20°F over a range of condensing temperatures. At condensing temperatures above 85°F, the ideal volume ratio exceeds the largest available volume ratio in this compressor series. Operating low volume ratio compressors under these conditions leads to significant performance penalties.

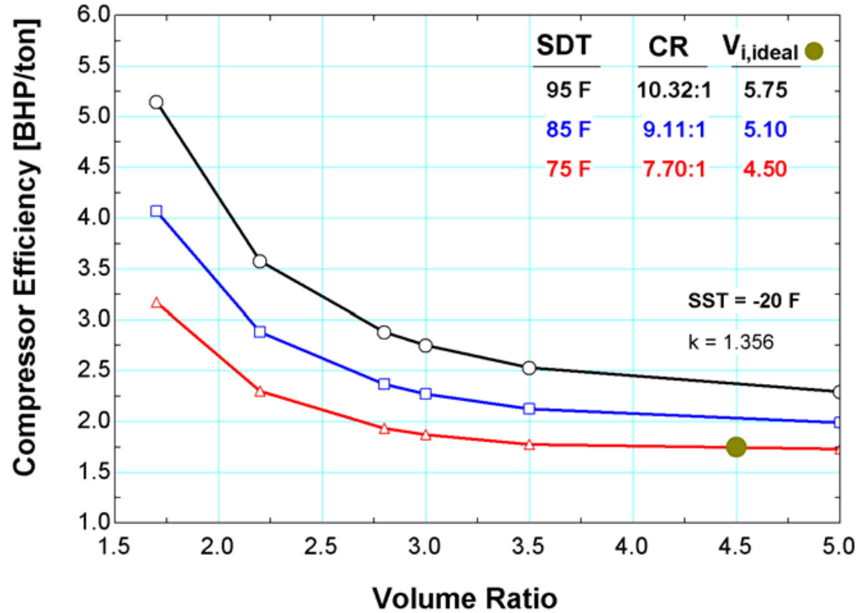

Figure 11: Influence of volume ratio on full-load efficiency for a fixed *Vi* compressor operating at -20°F saturated suction temperature.

The previous results are all for compressors packages operating at full-load. As a screw compressor is unloaded, its part-load operation will also impact the fixed volume ratio compressor efficiency. Figure 12 shows the trend in part-load performance for a 0°F saturated suction and 75°F saturated discharge temperature. Under these conditions, the compressor operates at a compression ratio of 4.63:1 with an estimated ideal volume ratio of 3.34. At full load conditions, the best performing compressor, *Vi*=3.5, matches closely with the ideal volume ratio compressor; however, the efficiency of the compressor diminishes as it is unloaded. At full-load, the compressor package has an efficiency of 1.19 BHP/ton. When the compressor is unloaded to 50%, the efficiency degrades to 1.48 BHP/ton – a 25% increase in required horsepower for each ton of refrigeration. This reinforces the guidance to operate screw compressors at or near full-load as much as possible [Manske, et al. 2002].

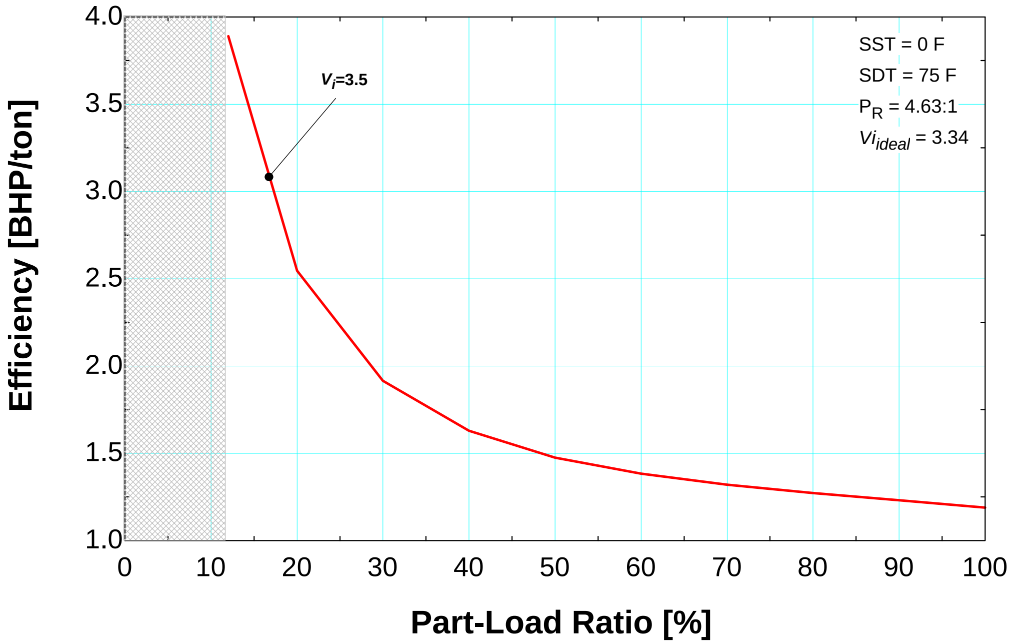

Figure 12: Compressor part-load efficiency for a range of fixed volume ratio compressors.

Figure 13 illustrates the performance of both fixed and variable volume ratio screw compressor selections operating over a range of saturated discharge temperatures for three separate saturated suction temperature (-20°F, 0°F, and 20°F). At a saturated suction temperature of -20°F, the variable volume ratio compressor operates at the compressor's maximum volume ratio (Vi=5.0 in this case). It is not until the saturated condensing temperature reaches 65°F when the variable volume ratio compressor begins to offer any performance benefits. The efficiency advantage of the variable volume ratio over the fixed Vi compressor at this full-load condition is rather small. At a 20°F suction temperature, the fixed Vi of 2.8 compares well with the variable volume ratio machine until low discharge temperatures are reached. The variable volume ratio machine then begins to exhibit an improvement in efficiency.

At a 0°F suction temperature and below 90°F discharge, the fixed volume ratio compressor (Vi=3.0) compares favorably to the variable volume ratio compressor. The differences in performance between fixed and variable volume ratio are slight because the fixed volume ratio compressor has been selected to match its anticipated operating pressure (or saturation temperature) envelope. Without a careful selection of the compressor's volume ratio, significant performance penalties will be assured. For example, Figure 14 illustrates the performance of three different twin screw compressors operating at 0°F suction over a range of discharge temperatures. The variable volume ratio and fixed volume ratio machine with Vi of 3.0 offer similar performance. Had a fixed volume ratio compressor with a Vi=2.2 been selected, the compressor would have reduced performance for a considerable range of discharge temperatures (anything above 70°F condensing).

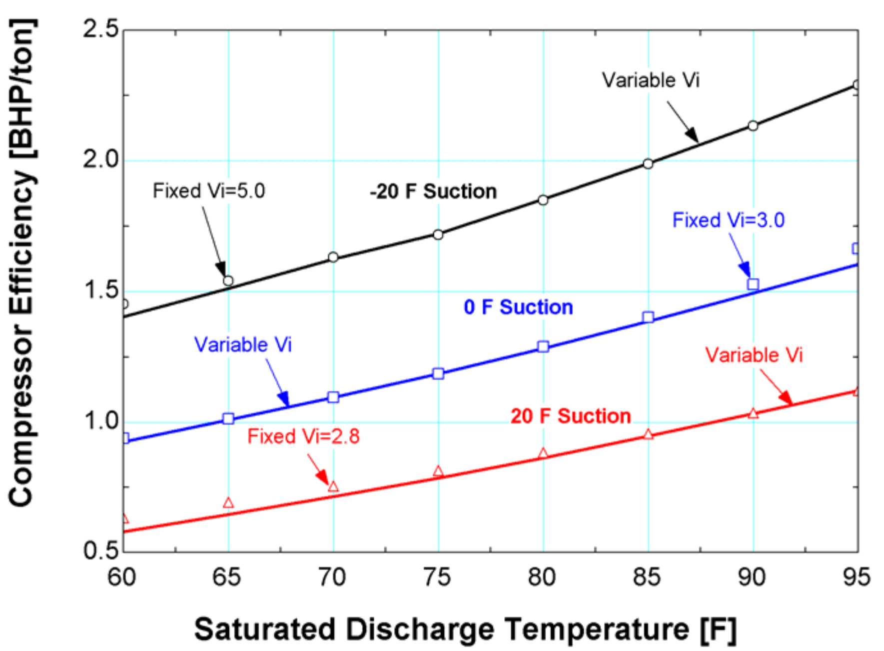

Figure 13: Fixed and variable volume ratio full-load efficiency characteristics.

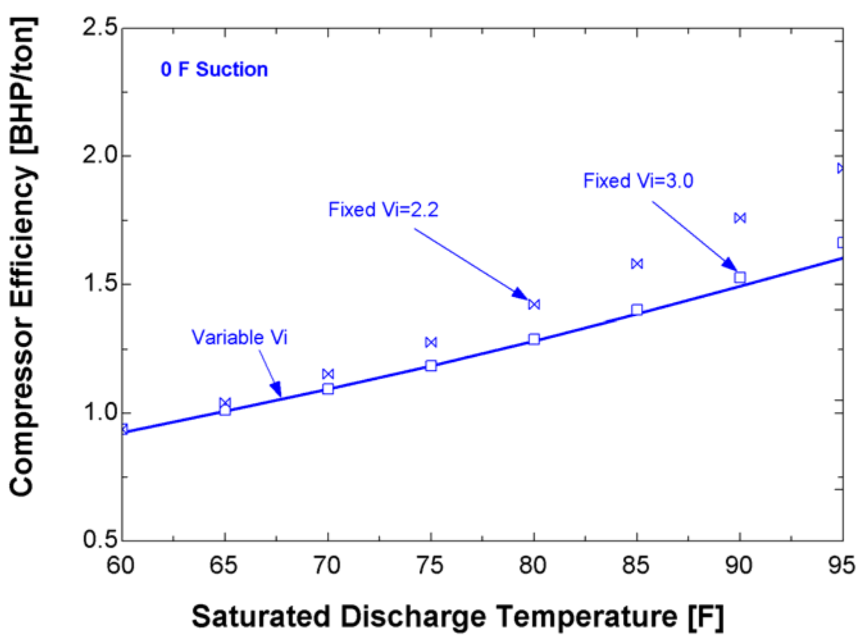

Figure 14: Fixed (*Vi* = 2.2 and 3.0) and variable volume ratio full-load efficiency characteristics at 0°F saturated suction temperature.

Selection Considerations

There are several factors that should be considered during the course of evaluating potential selections for a new screw compressor or changing the operating conditions of an existing screw compressor. Items that should be considered include:

1. Expected range of operating suction and discharge pressures
    a. single stage or two stage operation (booster or high-stage)
    b. swing duty (boosters operating as a single stage)
    c. load variability over time (large pull-down loads vs. relatively constant loads)
2. Climate type and system minimum head pressure constraints
3. Oil separator sizing/selection
4. Oil cooling methods
5. System and package losses for check valves, service valves, strainers installed around the compressor
6. Expected maintenance costs over machine's life

One of the key selection criteria is the expected operating suction and discharge pressures for the compressor. Many compressors operate with a fixed or relatively narrow compressor suction temperature; however, some compressors are designed for swing duty to serve loads at different suction levels. All compressors will operate over a range of discharge pressures and some will run over a wider range of discharge pressures due to seasonal fluctuations in condensing pressures. The greatest challenge for selecting a fixed volume ratio screw compressor are systems that operate over the widest range of condensing pressures. Table 1 below provides suggested volume ratio selections for fixed *Vi* compressors that will operate over high, medium, and low ranges in head pressure. In all cases, the data in Table 1 assumes that the maximum saturated condensing temperature is 95°F. The minimum head pressure for the "Medium" head pressure range case would be 70°F (95 – 25 = 70°F).

Table 1: Fixed volume ratio screw compressor selection ranges.

Saturated Suction Temperature [°F]	**Head Pressure Range**[1]		
	High 180 – 100 psig (95 - 65°F SCT)	**Medium** 180 – 115 psig (95 – 70°F SCT)	**Low** 180 – 135 psig (95 - 80°F SCT)
-40	5.0 or higher	5.0 or higher	5.0 or higher
-20	3.5 – 5.0	3.5 – 5.0	4.0 – 5.0
0	2.5 – 3.5	2.7 – 3.5	3.0 – 4.0
20	1.5 – 2.7	1.7 – 3.0	2.0 – 3.5
40	1.4 – 2.5	1.5 – 2.7	1.5 – 3.0

[1] The head pressure range is defined as the difference between the maximum and minimum saturated condensing temperatures.

[2] SCT is the saturated condensing temperature.

Ideal volume ratios for fixed *Vi* machines operating over a range of suction and discharge conditions were shown earlier in this *TechNote* (see Figure 6). Below, Figure 15 shows ideal volume ratios for fixed *Vi* compressors operating under booster duty in two-stage compression systems. If a compressor will operate as a swing machine, select a variable volume ratio compressor due to its ability to deliver superior performance in comparison to its fixed *Vi* counterpart. If a compressor is expected to operate over a wide range of suction pressures (due to pull-down or process variability) select a variable volume ratio screw compressor.

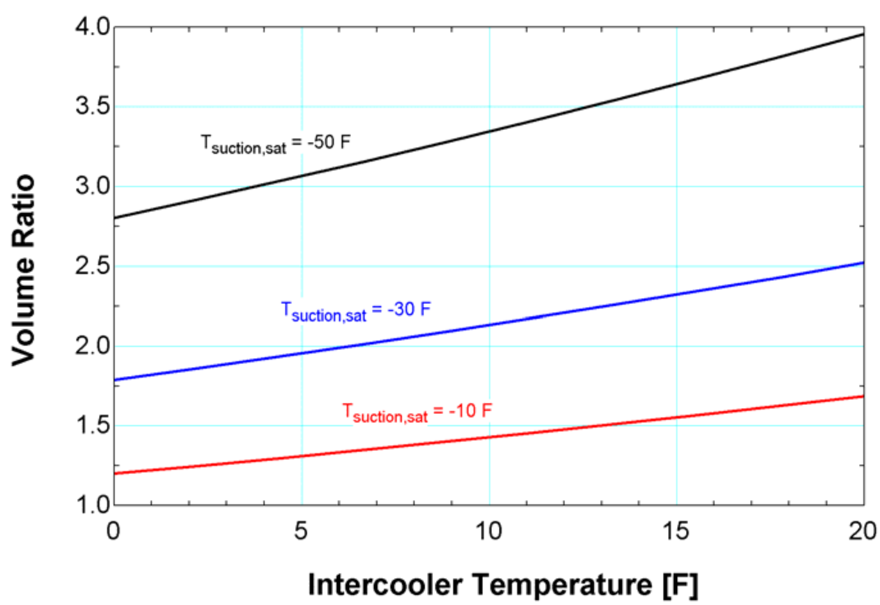

Figure 15: Ideal volume ratios for booster compressors operating in two-stage compression systems.

The head or condensing pressure of a system is dictated, in part, by the outside air wet bulb temperature. As the outside air wet bulb temperature decreases, the condensing temperature decreases until the system's minimum is reached. The minimum condensing temperature depends on a number of system-specific constraints such as: thermostatic expansion valves, hot gas defrost (main and run-out sizing, defrost relief regulator setpoints, gas-pumping requirements, etc.), presence of liquid injection oil cooling, sizing of high pressure liquid lines, and others. Figure 16 illustrates the theoretical frequency of saturated condensing temperatures for a system with a lower limit on condensing temperature constrained at 100 psig (63°F) in Madison, WI. The system would operate at its minimum condensing temperature for 3,925 hours (45% of the time) during the year. For fixed volume ratio machines, select a volume ratio that will match the suction and discharge conditions expected during the majority of yearly operating hours but check to be sure it will meet the peak load requirements at design conditions.

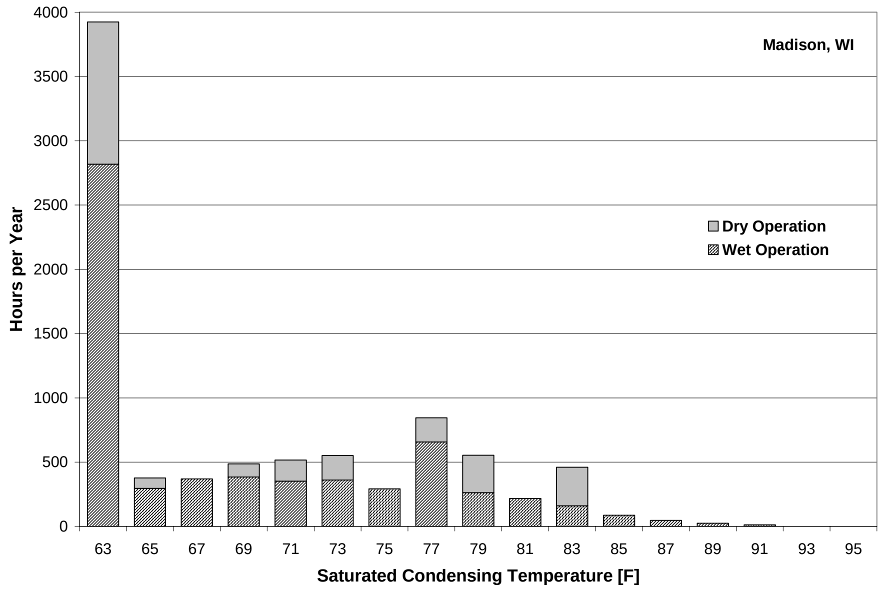

Figure 16: Frequency analysis of theoretical condensing temperatures for an evaporatively condensed industrial refrigeration system in Madison, WI.

In the course of selecting a screw compressor for peak performance during off-design conditions, oil separator sizing becomes important because the full-load volume flow rate of gas at the discharge of the compressor will increase as the head pressure decreases. The discharge volume flow rate will also increase with an increase in suction pressure because the mass flow rate of refrigerant through the compressor increases. Individually or combined, lowering condensing pressures and raising suction pressures are two widely pursued strategies for effectively improving the energy efficiency of refrigeration systems. Both have the net effect at increasing the volume flow rate of gas through the compressor and through the oil separator. If the volume flow rate of gas through the separator exceeds the rate assumed in the sizing of the separator, the efficiency of oil separation will decrease. As the oil separation efficiency decreases, the concentration of oil leaving the separator and migrating out into the system will increase. This results in the need for greater frequency of oil draining from points out in the system. Select the oil separator for full-load operation at the maximum expected suction pressure coincident with the lowest expected discharge pressure.

The choice of oil cooling methods also influences the compressor efficiency. Liquid injection oil cooling is the least first cost option; however, it results in a loss of compressor capacity and necessitates a higher minimum head pressure to maintain the required pressure differential across the oil cooling thermostatic expansion valve to maintain control authority. Maintenance costs for liquid injected oil cooled compressors will be higher than thermosiphon oil cooled. Finally, the life expectancy of a liquid injected screw is shorter than a thermosiphon oil cooled alternative. Thermosiphon oil cooling is the most efficient and lowest compressor maintenance cost option but has the largest capital cost. The payback thermosiphon oil cooling is often less than 3 years. Thermosiphon oil cooling is money well spent.

It is worthwhile to recognize that the selection of components around the compressor itself will influence its efficiency when integrated into the system. All compressor manufacturers have provisions for selecting alternative trim components including service valves, check valves, and strainers. Those options include low pressure drop components for minimum parasitic losses. Be sure to look at the difference in compressor performance with and without low pressure loss trim.

Finally, maintenance costs for compressor selections should be included with energy costs in the economic analysis of alternatives being considered. In general, maintenance costs for liquid injection oil cooled compressors are greater than thermosiphon (or water-cooled) oil cooled counterparts. Maintenance costs for variable volume ratio screw compressors are higher than fixed volume ratio machines. The increased maintenance costs for variable volume ratio screw compressors are attributed to the additional components needed for volume ratio control.

Conclusions

With a proper choice of volume ratio, fixed *Vi* compressors offer good energy efficiency performance as system head pressure floats to achieve efficient system operation. Variable volume ratio machines will deliver improved energy performance over a wide operating envelope but a price is paid for that benefit. Variable volume ratio compressors have slightly higher capital costs, increased maintenance cost, and reduced reliability when compared to their fixed *Vi* counterparts. The increased maintenance costs and reduced reliabililty are attributed to the additional components needed for volume ratio control.

If you have an opportunity, perform a life-cycle analysis for alternative compressor selections. The life-cycle cost should include capital, operating, maintenance, and replacement costs over a specified time horizon. Keep in mind that, based on evidence from the field, some ancillary equipment alternatives (such as liquid injection oil cooling) will lead to shortened compressor lifetimes when compared to others (such as thermosiphon oil cooling).

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About the IRC

The IRC is a collaborative effort between the University of Wisconsin-Madison and industry. Together we share a common goal of improving safety, efficiency, and productivity of industrial refrigeration systems and technologies. We realize this goal by conducting applied research, delivering knowledge transfer, and providing technical assistance.

The IRC offers its members a unique resource built upon professional staff that have academic qualifications, technical expertise, and practical experience with industrial refrigeration systems and technologies. We constantly strive to provide our members with high-quality objective information that is not biased by an affiliation with any particular organization.

Currently, the following industry leaders are reaping the benefits of membership in the IRC: Alliant Energy, CF Industries, General Mills, Kraft Foods, NOR-AM Cold Storage, Schoep's Ice Cream, Tropicana Products, Wells' Dairy, Xcel Energy. Complementing our end-user members in the IRC are the United States Occupational Safety & Health Administration (OSHA) and the Environmental Protection Agency (EPA) are also members.

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