

# LOAD SHARING STRATEGIES IN MULTIPLE COMPRESSOR REFRIGERATION SYSTEMS

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## ABSTRACT

*Many refrigeration systems have multiple compressors that operate in parallel to meet the aggregate cooling load requirements of the system. Often, individual compressors are equipped with a means of modulating their capacity to match the instantaneous refrigeration demand. Since the efficiency of a compressor changes as it is unloaded (often efficiency decreases), the part-load characteristics of individual compressors will influence the efficiency of the entire system. For this reason, it is desirable to identify operating strategies, properly accounting for compressor unloading characteristics, that maximize the efficiency of the entire system.*

*In this paper, we show that when two identical screw compressors are operating in parallel, there exists an optimum point at which it is best to switch from each compressor equally sharing the load to one compressor operating at full load and the other unloaded to match the remaining system load. When two screw compressors of different sizes are operating, an optimal compressor control map can be developed which maximizes the efficiency of the entire system over the entire range of loads. These optimum operating maps are shown to depend on the characteristics of the individual compressor's unloading performance and the relative sizes of compressors. An optimum control strategy for systems having multiple compressors, screw and/or reciprocating, can be implemented using the concept of crossover points introduced in this paper.*

## KEY WORDS

industrial refrigeration, optimal control, screw compressor, reciprocating compressor, crossover point

## INTRODUCTION

This paper is a result of a research project that focused on modeling an ammonia-based vapor compression industrial refrigeration system serving a large two-temperature level food storage and distribution facility located near Milwaukee, WI. The system utilizes a combination of both single-screw and reciprocating compressor technologies (each operating under single-stage non-economized compression), an evaporative condenser, liquid overfeed evaporators, and direct-expansion evaporators. The system model was validated with experimental data averaged over fifteen minute intervals gathered on-site. The validated model then served as the foundation for identifying alternative designs and operating strategies that lead to optimum system performance. Space limitations prevent a complete description of the facility, field-experiments, component and system model details, and model validation; however, more information can be found in Manske [1999]. The primary thrust of this paper is directed toward the study of optimum control and sequencing of multiple compressors in systems under part-load operation.

## COMPRESSOR MODELS

The three quantities that are of interest to a refrigeration system designer or operator are the power required by the compressor(s), the amount of useful refrigeration (capacity) it provides, and the oil cooling requirements. Manufacturers of large commercial and industrial refrigeration compressors generally provide ratings of their products including power, capacity, and oil cooling loads as a function of saturated suction and saturated discharge temperatures (or pressures).

### Nominal Performance Characteristics

Polynomial correlations of the steady-state compressor power, capacity, and oil cooling load in the form given by Equations 1-3 were developed as functions of saturated suction temperature (SST) and saturated discharge temperature (SDT) from data provided by the compressor manufacturer. The correlations follow the same general form as recommended by ARI Standard 540 [1999], but fewer coefficients than used in Standard 540 were found to adequately represent the performance data. Equations 1-3 are totally empirical; consequently, caution should be exercised to avoid extrapolating compressor performance outside the range of data provided by the manufacturer.

$$POW = P_1 + P_2 \cdot SST + P_3 \cdot SST^2 + P_4 \cdot SDT + P_5 \cdot SDT^2 + P_6 \cdot SST \cdot SDT \quad (1)$$

$$CAP = C_1 + C_2 \cdot SST + C_3 \cdot SST^2 + C_4 \cdot SDT + C_5 \cdot SDT^2 + C_6 \cdot SST \cdot SDT \quad (2)$$

$$OIL = O_1 + O_2 \cdot SST + O_3 \cdot SST^2 + O_4 \cdot SDT + O_5 \cdot SDT^2 + O_6 \cdot SST \cdot SDT \quad (3)$$

where

*POW* is the compressor power

*CAP* is the compressor refrigeration capacity

*OIL* is the compressor oil cooling load (if applicable)

*SDT* is the compressor saturated discharge temperature

*SST* is the compressor saturated suction temperature

$P_1$ - $P_6$ ,  $C_1$ - $C_6$ ,  $O_1$ - $O_6$ , are empirical coefficients

### Actual Performance

When manufacturers rate their compression machines, the pressure and corresponding saturation temperature is measured at the inlet and outlet flanges of the compressor. The compressor ratings do not include pressure losses and the associated saturation temperature change due to trim (valve trains or oil separators) that surrounds their compressors even though such trim is required to make the compressor functional and many of the components are commonly included with the purchase of the compressor package. Also, some manufacturers list saturated discharge temperature (SDT) as “saturated condensing temperature (SCT)” even though their measurements are at the discharge flange of the compressor and not literally at the condenser. These factors create some uncertainty in the application and optimal control of this important system component. The investigation of optimal control of multi-compressor installations presented in this paper only considers non-economized compressor arrangements. The results presented here should not be assumed to apply to economized systems without further analysis.

To account for refrigerant pressure drop and its impact on compressor performance, our refrigeration system models rely on the Darcy [Crane, 1988] and Colebrook equations [Avallone, 1996] for each piping element connected to the compressor(s). The most critical piping sections in refrigeration systems are the suction lines because they operate at the lowest pressures and carry vapor refrigerant with a highest specific volume. These factors directly influence the operating capacity, volumetric efficiency, and power demanded by the compressors.

The capacity information provided by the compressor manufacturer represented in Equation 2 is based on a specified amount of superheat and a specified amount of liquid subcooling. Corrections to the manufacturer’s compressor capacity catalog data are required if the actual amount of subcooling or superheat in application differs from the nominal values assumed by the manufacturer in the process of cataloging their product’s performance. In this study, the compressor capacity was adjusted using Equation 4.

$$CAP_{actual} = CAP_{mfr} \cdot \frac{v_{mfr}}{v_{actual}} \cdot \frac{\Delta h_{actual}}{\Delta h_{mfr}} \quad (4)$$

where

$CAP_{actual}$  is the refrigerating capacity corrected for actual subcooling and superheat conditions in application

$CAP_{mfr}$  is the manufacturer’s cataloged refrigerating capacity at the specified subcooling and superheat conditions

$v_{mfr}$  is the specific volume of suction gas based on manufacturer’s-specified conditions

$v_{actual}$  is the actual specific volume of suction gas in application which depends upon the pressure and temperature at the compressor inlet and suction line losses (pressure drop)

$\Delta h_{mfr}$  is the difference in specific enthalpies of refrigerant between manufacturer's-rated compressor at suction and rated evaporator at inlet

$\Delta h_{actual}$  is the actual difference in specific enthalpies of refrigerant between refrigerant at the compressor suction and the evaporator inlet

If the compression process can be represented as a polytropic process, the compressor power will only depend on the saturated discharge temperature (SDT) and the saturated suction temperature (SST), independent of the level of superheat. This conclusion is reached from the following analysis.

The work per unit mass required to compress refrigerant from the suction conditions (state 1) to the discharge conditions (state 2) in a polytropic process (i.e.,  $P v^n = \text{constant}$ ) is given by Equation 5 (Kuehn et al., 1998).

$$W_{12} = \frac{n}{n-1} P_1 v_1 \left[ \left( \frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] \quad (5)$$

where

$W_{12}$  is the compressor work per unit mass

$n$  is the polytropic coefficient

$P_1$  and  $P_2$  are the suction and discharge pressures (corresponding to the saturated suction temperature and saturated discharge temperatures, respectively)

$v_1$  is suction specific volume of the suction gas (which is affected by superheat)

The power input to the compressor is

$$POW = \frac{W_{12} \dot{m}}{h_{motor}} \quad (6)$$

where

$\dot{m}$  is the refrigerant mass flow rate

$h_{motor}$  is the motor efficiency

The mass flowrate is related to the compressor volume displacement rate by the volumetric efficiency, which can be expressed as (Kuehn et al., 1998)

$$h_{vol} = \frac{\dot{m} v_1}{Disp. Rate} = 1 + C - C \left( \frac{P_2}{P_1} \right)^{\frac{1}{n}} \quad (7)$$

where

$h_{vol}$  is the compressor volumetric efficiency

$C$  is the clearance volume ratio

$Disp. Rate$  is the compressor displacement rate

Equation 7 neglects the pressure loss across the intake valves. When  $W_{12}$  from Equation 5 and  $\dot{m}$  from Equation 7 are substituted into Equation 6, the suction specific volume ( $v_1$ ) cancels indicating that compressor power is,

theoretically, independent of superheat condition for a specified SST. Actual compressor behavior does differ from the theoretical analysis but the dependence of compressor power on the level of suction superheat is relatively small.

Oil cooling loads are influenced by suction superheat. As the suction superheat increases, the oil-cooling load will also increase. The rise in oil cooling load is approximately equal to the sensible heat increase attributed to the elevated level of superheat in the suction gas.

### Part-load Compressor Operating Characteristics

Since refrigeration loads are never constant, it is essential to have the ability to modulate or adjust the capacity of compressors installed in the system in order to balance the prevailing aggregate evaporator loads. A number of alternative methods for “unloading” compressors are available and vary depending on the system design, compressor technology, and compressor vintage. Unloading methods include: individual cylinder unloaders (reciprocating compressors), slide valve (screw compressors), variable speed drives (electric or engine), on/off sequencing, and hot gas bypass (not commonly used in ammonia systems). The most common method of unloading reciprocating compressors is by the use of cylinder unloaders while the most common method of unloading screw compressors is by the use of a slide valve.

The results presented in this paper assume that capacity modulation for the screw compressors is accomplished by the use of a slide valve that effectively changes the point where the compression process begins along the axis of the screw. With a slide valve, a screw compressor has the ability to continuously modulate capacity between 10 to 100% of its available full load capacity. Reciprocating compressors are assumed to be equipped with cylinder unloaders. Unloaders consist of hydraulically or electrically-actuated push rods that hold open suction valves on individual or groups of cylinders. By holding the suction valves open, the number of cylinders that are providing active gas compression is reduced; thereby, decreasing the compressor’s capacity. Refrigeration load is determined indirectly by sensing changes in suction pressure. A rise in suction pressure indicates an increasing system load and the compressor’s on-board controls will increase the capacity of the machine (by adding additional cylinders or changing slide valve position). Conversely, a drop in suction pressure indicates decreasing refrigeration load.

As screw compressors are unloaded, their power and oil cooling requirements decrease, but not necessarily in direct proportion to capacity. The screw compressor part-load performance characteristics also depend on the compression ratio. Multi-cylinder reciprocating compressors tend to unload more linearly. Unloading curves for both the screw and multi-cylinder reciprocating compressors investigated in this study (bare compressor only, i.e. no system effects) are shown in Figure 1. These curves give the fraction of full load power that the compressor will use when operated at a specific percent of its full load capacity (%FLC) or sometimes referred to as the part-load ratio. Three separate unloading curves for saturated suction temperatures of  $-20^{\circ}\text{F}$  ( $-28.9^{\circ}\text{C}$ ),  $0^{\circ}\text{F}$  ( $-17.7^{\circ}\text{C}$ ), and  $20^{\circ}\text{F}$  ( $-6.7^{\circ}\text{C}$ ) are shown for the screw compressor (a single screw equipped with variable volume ratio control) in Figure 1.

The part-load characteristic for the reciprocating compressor does not pass through the origin of Figure 1 because of an additional compressor power requirement (approx. 3%) to overcome parasitic losses associated with frictional and windage effects on the unloaded cylinders. The part-load characteristic of the single screw compressor used in the existing system is represented with the regression in Equation 8. The part-load characteristics for both compressors were obtained from a compressor manufacturer (Fisher [1998]). These characteristics are expected to be representative, but the characteristics of compressors from different manufacturers may differ from those in Figure 1.

$$\begin{aligned} \% \text{ FullLoadPower} = & 21.5733 + 0.465983 \cdot \% \text{ FLC} + 0.00544201 \cdot \% \text{ FLC}^2 \\ & - 5.55343 \cdot 10^{-6} \cdot \% \text{ FLC}^3 + 7.40075 \cdot 10^{-8} \cdot \% \text{ FLC}^4 - 2.43589 \cdot 10^{-9} \cdot \% \text{ FLC}^5 \quad (8) \end{aligned}$$

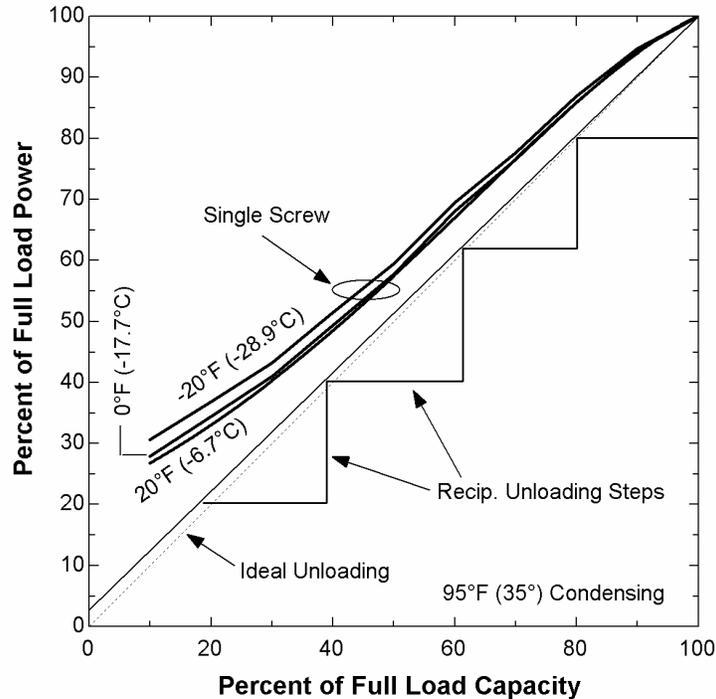


Figure 1: Part-load characteristics for the screw and reciprocating compressors (without system effects)

### MULTIPLE COMPRESSOR UNLOADING

If multiple compressors are used to meet refrigeration loads, it is desirable to operate the compressors at the lowest combined power while still meeting the system loads. In refrigeration systems with variable loads, the delivered capacity of the compressors must be modulated by unloading the compressors in order to balance the compressor(s) capacity with the refrigeration demands of the system. Each compressor, depending upon type and manufacturer, may have a different unloading characteristic. The following results are strictly valid only for the particular screw and reciprocating compressors investigated in this study; however, the general concepts can be applied to all refrigeration systems with multiple compressors.

A hypothetical refrigeration system utilizing two compressors operating in parallel formed the basis for determining operating strategies that minimize system power. We elected to characterize compressor performance in terms of a specific power, i.e., the dimensionless ratio of the compressor power to refrigeration capacity at a particular set of operating conditions (saturated suction temperature, saturated discharge temperature, and part-load ratio – including effects of system refrigerant pressure drop). The specific power is the inverse of the coefficient of performance (COP).

Figure 2 shows a performance comparison, in terms of specific power, between the screw and reciprocating compressors for several different saturated suction temperatures over a range of part-load conditions assuming a fixed saturated discharge temperature of 29.4°C (85°F). The performance maps include effects of refrigerant pressure drop in both the suction-side and discharge-side of the compressor. The figure shows that reciprocating compressors unload nearly linearly and their performance curve is nearly flat for a fixed suction temperature. The slight increasing trend (corresponding to a decrease in compressor performance) from left to right in Figure 2 for the reciprocating compressor is a result of increasing pressure drop in the dry suction line due to increasing refrigerant mass flow rate. The additional (3%) increase in total compressor power discussed above also contributes to the increasing trend in specific power. Several observations can be made about compressor operation from Figure 2.

- A single screw compressor unloaded to 25 percent of its full load capacity has nearly a 50 percent increase in specific power when compared to a reciprocating compressor.

- Screw compressors perform better than reciprocating compressors when operated near full load; the screw compressor's full load performance advantage increases as the suction pressure drops (i.e. as compression ratio increases).
- Reciprocating compressors are better suited in refrigeration systems where significant unloading, i.e. load-following, is required.
- From an energy standpoint, is more important to size screw compressors correctly as compared to multi-cylinder reciprocating compressors.

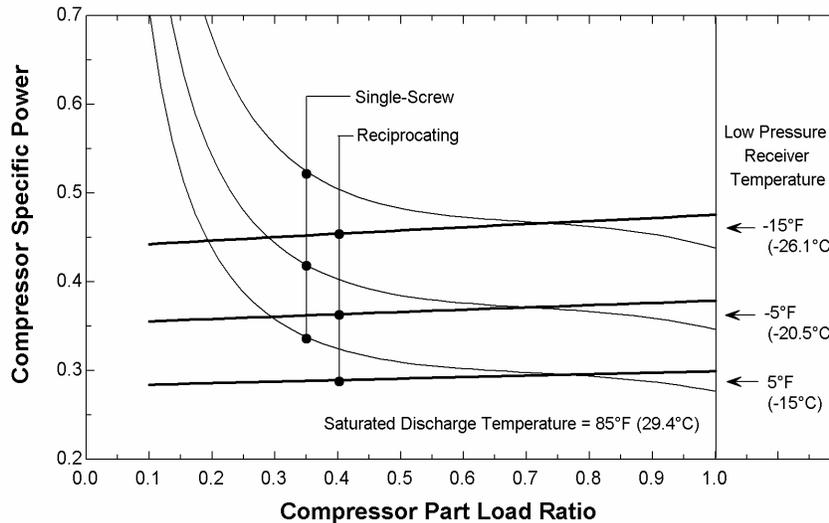


Figure 2: Comparison of the performance of single stage screw and reciprocating compressors including suction and discharge-side refrigerant pressure drop.

### Load Sharing with Similar Compressors

When the system refrigeration load exceeds the capacity of a primary (lead) compressor, a second (lag) compressor must be cycled on to augment the capacity of the primary compressor. How best to split the load between the two compressors depends upon the magnitude of the load, the type, and size of the compressors.

If two similar reciprocating compressors in parallel operation are sharing a load, the load should be split to equalize suction line pressure drop to each compressor. This operating strategy minimizes the dominant compressor performance penalty source – suction line pressure drop. This conclusion, evident from the results of our simulations, is based on several factors. First, the unloading characteristic of the reciprocating compressor shown in Figure 1 exhibits a minimal performance degradation when unloaded. Second, the pressure loss in the suction line is roughly proportional to the square of the refrigerant mass flow due to line frictional losses whereas the power per unit mass increases as the suction pressure is reduced in the manner shown in Equation 5. Since the total mass flow rate for both compressors is fixed in order to provide the required refrigeration capacity, splitting the load to equalize pressure drop yields optimum combined compressor performance. The combination of lower suction line pressure drop and minimal compressor unloading penalty lead to the improved compressor specific power as shown in Figure 2 for the reciprocating compressor.

Screw compressors unload non-linearly and their parallel operation must be treated quite differently compared to reciprocating compressors. Figure 3 shows a plot of the aggregate specific power for a system with two equally sized screw compressors operating in parallel. Each separate line on the plot represents a different total system load. The system load is expressed in terms of the system part-load ratio ( $PLR_{System}$ ) defined as the ratio of the actual delivered system capacity to the total available capacity of both compressors at full load. Starting at the far right of the plot for a given load, one compressor is fully loaded and the other compressor is operating at part load such that the combined capacity of the two compressors matches the total system load. The abscissa is the ratio of the capacity of the lead (more heavily loaded) compressor to the capacity of lag (less heavily loaded) compressor. By progressing from right to left along a constant  $PLR_{System}$  line, the capacity of the lead compressor is reduced while

that for the lag compressor is increased. The minimum compressor power for each  $PLR_{System}$  operating state is identified by a circle symbol.

Figure 3 shows that, for system part-load ratios below approximately 0.65, optimal operation results when the system load is split equally between the two compressors (a compressor capacity ratio of 1). When the system has a part load ratio above 0.65, the system performance is optimized when one compressor is fully loaded and the remaining compressor is loaded to make up the difference. This behavior can be explained by the part-load characteristics in Figure 1, which indicate that the specific power of an individual screw compressor begins to increase very rapidly if it is operated with a compressor part load ratio below 0.5. The values of the aggregate specific power in Figure 3 depend on several factors, including the compressor performance characteristics and pressure drop in the compressor suction lines. However, the conclusions relating on how to optimize the combined operation of the two compressors are independent of these factors.

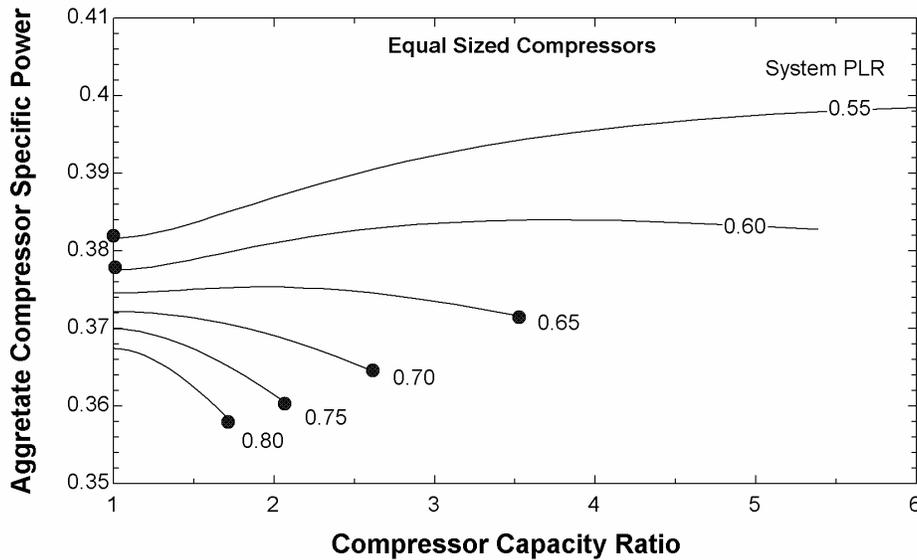


Figure 3: Equal sized screw compressor load sharing characteristics.

Figure 4 shows the optimum compressor operation for a refrigeration system with two identical single screw compressors operating in parallel. The right-hand side of the abscissa ( $PLR_{System} = 1$ ) represents full capacity operation of the system. For system part-load requirements ( $PLR_{System} < 1$ ), the alternative having the lowest aggregate compressor specific power is the optimum control strategy for that particular load. Figure 4 also demonstrates the performance penalty that a system will incur if two screw compressors are operated at part load instead of fully loading a single compressor. The crossover point, identified by the intersection of two of the curves in Figure 4, is the point at which there is no difference in performance to operating one compressor at full-load or both at equal part-load. Additional results reported by Manske (1999) and shown in Figure 5 indicate that the crossover point is nearly independent of the suction or discharge conditions for the compressor unloading characteristics used in this study. Near-optimal control can be achieved by basing the crossover point on the  $PLR_{System}$  characteristic.

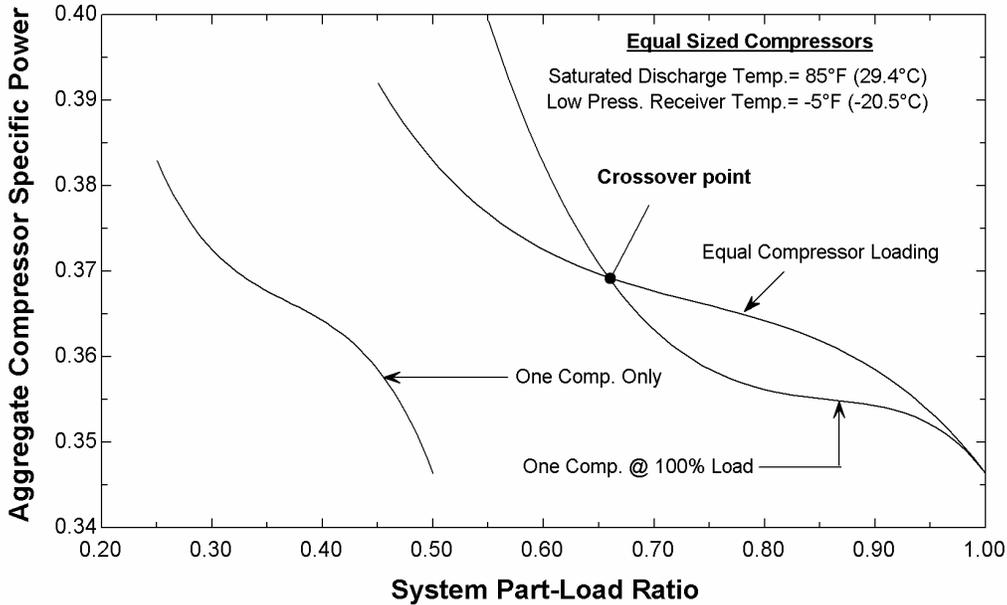


Figure 4: Optimum performance map for equal-sized screw compressors.

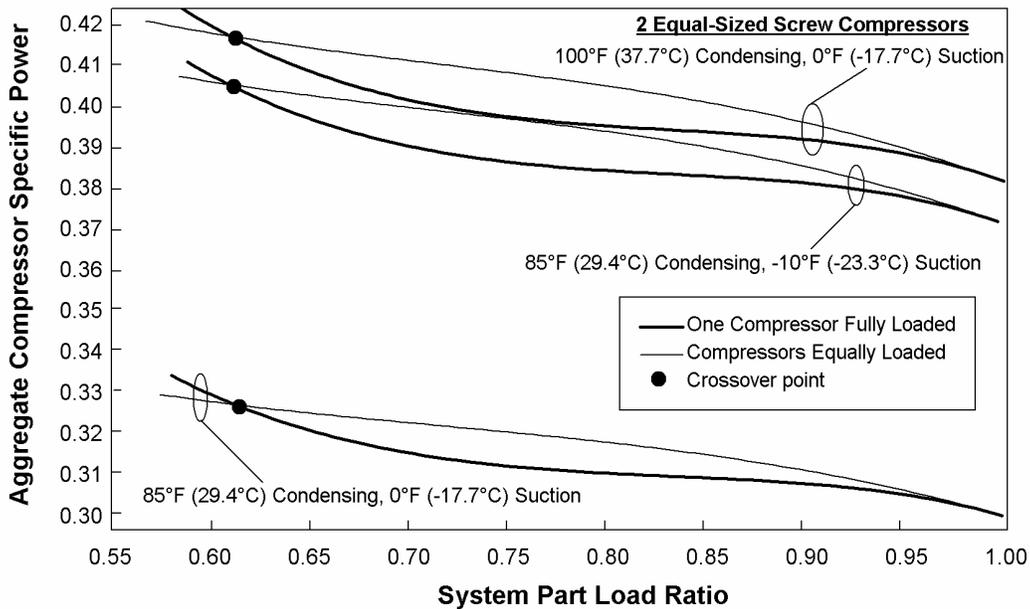


Figure 5: Location of the crossover point for different suction and discharge operating conditions.

Load Sharing with Unequal Sized Compressors

The application of unequal sized screw compressors in systems was also investigated. When unequally sized screw compressors are used in parallel, an alternative set of load-sharing characteristic curves govern optimum system performance. The results presented here show the optimum load-sharing characteristics for two compressors sized to handle 38% (small compressor) and 62% (larger compressor) of the total system peak load. The sequence of operating curves in Figure 6 shows that it is not advantageous to operate each compressor at equal loading for intermediate system loads as demonstrated with similar sized compressors. Instead, the larger compressor should be

fully loaded when the total system load is high ( $PLR_{System} > 0.76$ ) and the smaller compressor should be fully loaded when there is an intermediate load ( $0.62 < PLR_{System} < 0.76$ ). Comparing the curves generated when only a single compressor is operated to each other as well as to the curves that represent dual compressor operation also demonstrates the significant performance penalty that occurs when screw compressors are unloaded.

As the load exceeds the maximum capacity of the larger screw, the smaller screw must be started. If the larger compressor is kept fully loaded at intermediate loads ( $0.62 < PLR_{System} < 0.76$ ), the smaller compressor will be operating at low part-load ratio causing poor overall system performance. If the load is shifted to fully load the smaller compressor, the larger compressor still operates near 50 to 60 percent of its full load capacity at which it experiences a smaller performance penalty. Some conclusions can be drawn from the above analysis of screw compressors.

- When a screw and reciprocating compressor are sharing a load that is below the total available capacity of the system, the screw compressor should be fully loaded and the reciprocating compressor used for load-following. This conclusion is a result of the part-load characteristics in Figure 1 which indicate that the efficiency of the screw compressor decreases as the percent of full-load capacity that it provides is reduced.
- When two screw compressors are sharing a load, control strategies should avoid operating any screw compressor below 50 percent of its full load capacity.
- When multiple reciprocating compressors are used, the system load should be divided equally among operating compressors (to minimize suction line pressure drop to each compressor).
- Unloading performance characteristics of systems with unequal sized compressors differ from systems with equally sized compressors.
- Screw compressors are best suited for base loading where they can be run at full load.

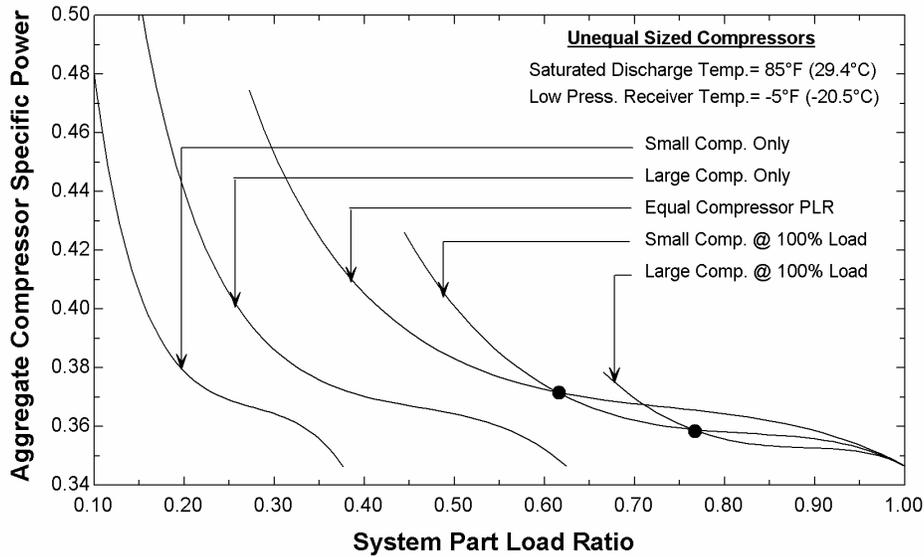


Figure 6: Optimum performance map for unequal-sized screw compressors.

## CONCLUSIONS

Compressors in industrial refrigeration systems regularly operate at part-load conditions. Part-load compressor operation is required in order to balance the refrigeration capacity of the compressors with the refrigeration demand from the system. Reciprocating compressors have near-linear unloading curves and therefore introduce relatively small performance penalties when operated at low part-load ratios. In systems utilizing multiple reciprocating compressors, optimal performance can be realized by equal compressor unloading (to minimize suction line pressure drops to each compressor).

As screw compressors are unloaded they require more power per unit of cooling capacity. It is recommended that screw compressors be sized appropriately so they can be operated at or near full capacity as much as possible. Screw compressors should be used for base loading and reciprocating compressors should be used to meet the transient portion of a varying load.

If two screw compressors are sharing a load, there exists a point where it is better to fully load one compressor rather than split the load equally. In the case of two equally sized screw compressors, the optimal situation occurs when the compressors share the load up to an identifiable crossover point which occurs when the load on the system is about 65 percent of the combined available capacity of the compressors. Beyond that point it is best to fully load one of the screws and make up the difference with the other. The crossover point is a characteristic of type and size of compressors used. When load sharing between two unequal sized screw compressors is required, it is best to first fully load the smaller of the two, then at a certain identifiable crossover point, fully load the larger of the two compressors and make up the difference with the smaller of the two. Calculations with unequal sized screw compressors (the larger compressor has 64% greater capacity compared to the smaller compressor), indicated that this crossover point occurred when the load was 76 percent of the available capacity of both compressors. Crossover points were found to be nearly independent of compressor suction and discharge conditions.

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