International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

2000

Load Sharing Strategies in Multiple Compressor Refrigeration Systems

K. A. Manske
University of Wisconsin-Madison

D. T. Reindl
University of Wisconsin-Madison

S. A. Klein
University of Wisconsin-Madison

LOAD SHARING STRATEGIES IN MULTIPLE COMPRESSOR REFRIGERATION SYSTEMS

K. A. Manske, D.T. Reindl, and S.A. Klein
Solar Energy Laboratory
University of Wisconsin - Madison

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 prevailing cooling demand. Since the efficiency of a compressor tends to decrease as it is unloaded, compressor part-load characteristics influence the efficiency of the entire system. For this reason, it is desirable to identify operating strategies that properly account for compressor unloading characteristics and 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 cooling demand When two compressors of different sizes are operating, an optimal control compressor operating map can be developed which maximizes the efficiency of the entire system over a complete range of cooling 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 can be implemented using the concept of cross-over points introduced in this paper.

INTRODUCTION

This paper is a result of a research project which focused on modeling an ammonia-based industrial vapor compression 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 operating under single-stage 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 basis for identifying alternative designs and operating strategies that lead to optimum system performance. Space limitations prevent a complete description of the experimental facility, the model details, and the validation studies; however, more information can be found in Mankse [2000]. The thrust of this paper is directed toward the study of optimum control and sequencing of multiple compressor systems under part-load operation.

COMPRESSOR MODELS

The three quantities that are of most 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/pressures.

Nominal Performance Characteristics

Polynomial correlations (of the form given in equations 1-3) of the steady-state compressor power, capacity, and oil cooling load were developed as functions of saturated suction temperature (SST) and saturated discharge temperature (SDT) from data provided by the manufacturer. The correlations are in the same general form as that recommended by ARI Standard 540 [1999], but fewer coefficients than used in Standard 540 were found to adequately represent the performance information. 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:

CAP = compressor refrigeration capacity
OIL = compressor oil cooling load (if applicable)
POW = compressor power
SDT = compressor saturated discharge temperature
SST = compressor saturated suction temperature
C_1-C_6, O_1-O_6, and P_1-P_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 valve trains or oil separators even though both 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.

In our refrigeration system simulation, pressure losses in the system piping were modeled using the Darcy [Crane, 1988] and Colebrook equations [Avallone, 1996] for each piping element. The most critical piping sections in refrigeration systems are the suction lines because they operate at the coldest temperatures, carry vapor refrigerant with a high specific volume, and directly influence the operating capacity/volumetric efficiency of the compressors.

The capacity information provided by the compressor manufacturer (represented in Equation 2) is based on a specified amount of superheat (which governs the mass flow and specific enthalpy of the refrigerant at the evaporator outlet) and a specified amount of subcooling (which governs the specific enthalpy of the refrigerant to the evaporator inlet). Corrections to the manufacturer's compressor capacity 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).

$$CAP_{actual} = CAP_{mfr} \cdot \frac{v_{mfr}}{v_{actual}} \cdot \frac{\Delta h_{actual}}{\Delta h_{mfr}} \tag{4}$$

where

v_{mfr} = specific volume of suction gas based on manufacturer's-specified conditions.

v_{actual} = 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).

Δh_{mfr} = difference in specific enthalpies of refrigerant between manufacturer's-rated compressor at suction and rated evaporator at inlet.

Δh_{actual} = actual difference in specific enthalpies of refrigerant between refrigerant at the compressor suction and the evaporator inlet.

Compressor power and oil cooling load are dependent only on the saturated discharge temperature (SDT) and the saturated suction temperature (SST). Adjustments for different levels of subcooling and superheat are not required.

Part-load Compressor Operating Characteristics

Large screw and reciprocating compressors typically have the capability for reducing their capacity to match the required refrigeration demand by the system. Screw compressors accomplish this task by the use of a slide valve that, effectively, changes the point where the compression process begins along the axis of the screw. Most screw compressors have the ability to continuously modulate capacity between 10 to 100% of its available full load capacity. Reciprocating compressors can be equipped with unloaders. Unloaders consist of hydraulically or electrically-actuated push rods that hold open suction valves on individual or groups of cylinders. By holding open the suction valves, the number of cylinders that are providing active gas compression is reduced; thereby, reducing the compressor's capacity.

As screw compressors are unloaded, their power and oil cooling requirements decrease, but not necessarily in direct proportion to capacity. Reciprocating compressors tend to unload more linearly. Unloading curves for both the screw and multi-cylinder reciprocating compressors are shown in Figure 1. These curves give the fraction of full load power (%FLP) the compressor will use when operated at a specific percent of its full load capacity (%FLC) or part-load ratio.

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 (5). The part-load characteristics for both compressors were obtained from the compressor manufacturer (Fisher [1999]). These characteristics are expected to be representative, but the characteristics of compressors from different manufacturers may differ from those in Figure 1 due to different compressor designs and manufacturing processes.

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

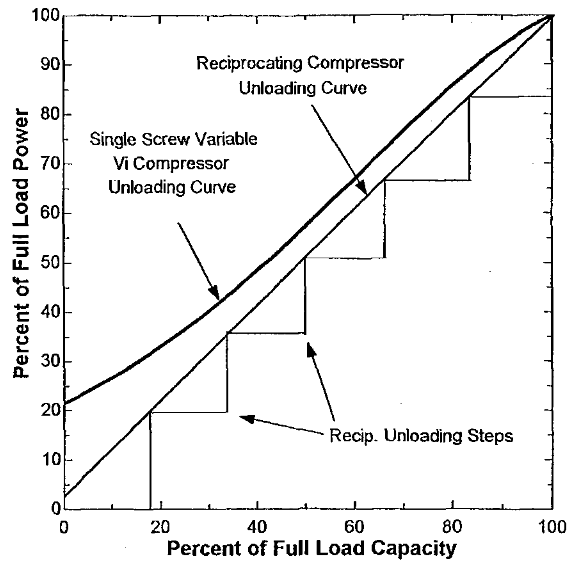

Figure 1: Part-load characteristics for the screw and reciprocating compressors

MULTIPLE COMPRESSOR UNLOADING

If multiple compressors are used to meet the refrigeration load, 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 theoretical refrigeration system utilizing two compressors operating in parallel was used to explore optimum compressor operation. Compressor performance can be characterized in terms of 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). The specific capacity 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). Reciprocating compressors unload nearly linearly and their performance curve should, theoretically, be 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 performance advantage increases as the suction pressure drops.

- 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.

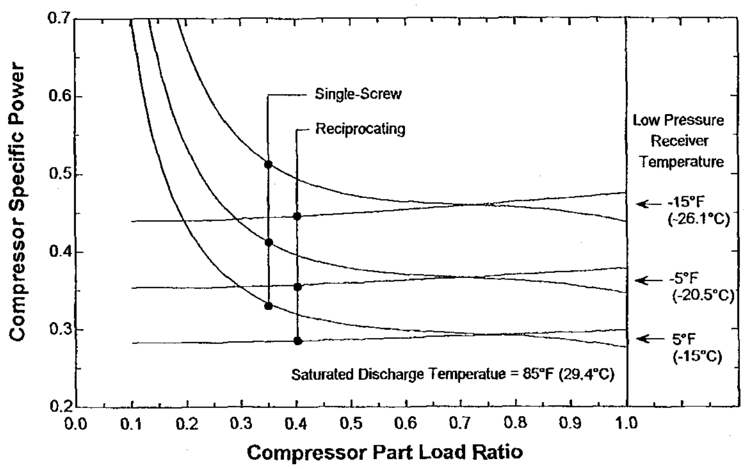

Figure 2: Comparison of the performance of the screw and reciprocating compressor

Load Sharing with Similar Compressors

When the system refrigeration load exceeds the capacity of a primary compressor, a secondary 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 reciprocating compressors are sharing a load, the load should be split so that the pressure loss in each compressor's dry suction line is equal. Reciprocating compressors have little performance degradation when unloaded so the relative distribution of a single load between reciprocating compressors has a minimal effect on system performance (from an energy standpoint). In this case, splitting the load to equalize pressure losses in the dry suction line yields optimum performance.

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) 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 system load line, the capacity of the lead compressor is reduced while that for the lag compressor is increased. Figure 3 shows that, for part load ratios below approximately 0.65, optimal operation results when the 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 part-load characteristics in Figure 1, which shows 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.

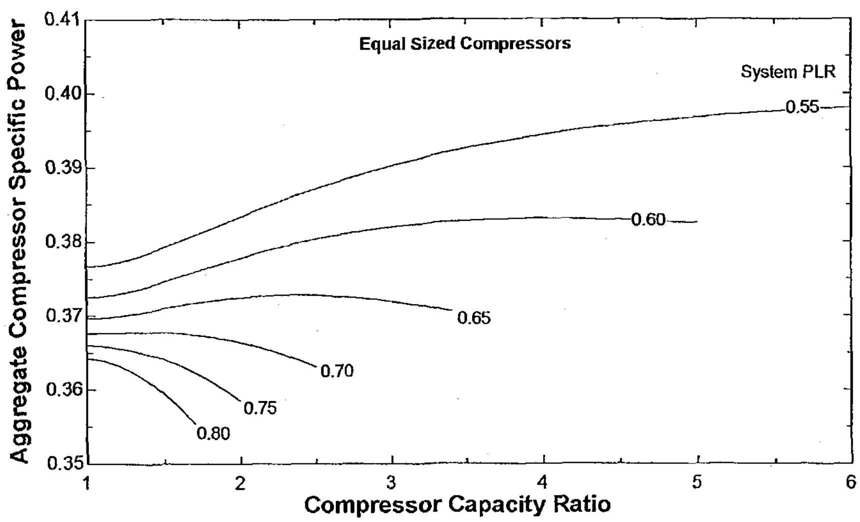

Figure 3: Equal Sized Screw Compressor Load Sharing Characteristics

Figure 4 shows the optimum compressor operation for a refrigeration system with two identical single screw compressors operating in parallel. The right y-axis (system PLR=1) represents full capacity operation of the system. For system part-load requirements (PLR<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 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. The simulation results indicate that the crossover point is nearly independent of the suction or discharge conditions. Near-optimal control can be achieved by basing the crossover point on the system PLR.

Load Sharing with Unequal Sized Compressors

When unequally sized screw compressors are used in parallel, an alternative set of load-sharing characteristic curves govern optimum system performance. The sequence of operating curves in Figure 5 shows that it is not advantageous to operate each compressor at equal part-load ratios 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 (system PLR>0.76) and the smaller compressor fully loaded when there is an intermediate load (0.62<system PLR<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<system PLR<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.

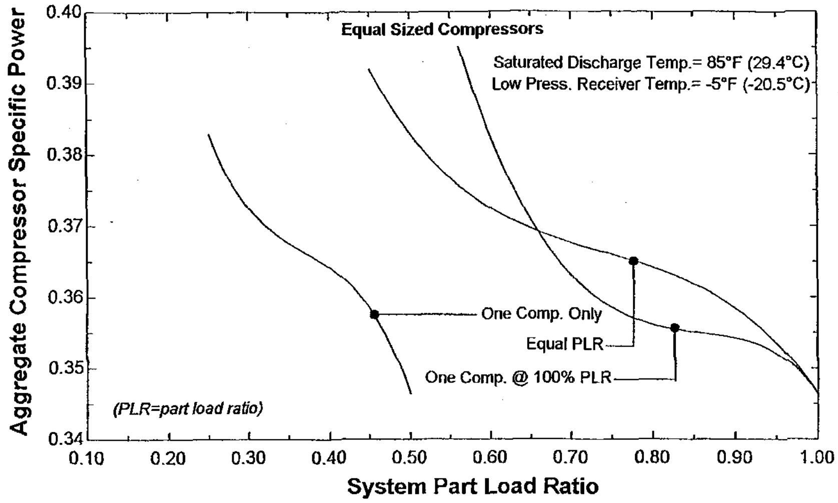

Figure 4: Optimum Performance Map for Equal-Sized Screw Compressors

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.
- When two screw compressors are sharing a load, control strategies should avoid operating any screw compressor below 50 percent of its full load capacity.
- 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.

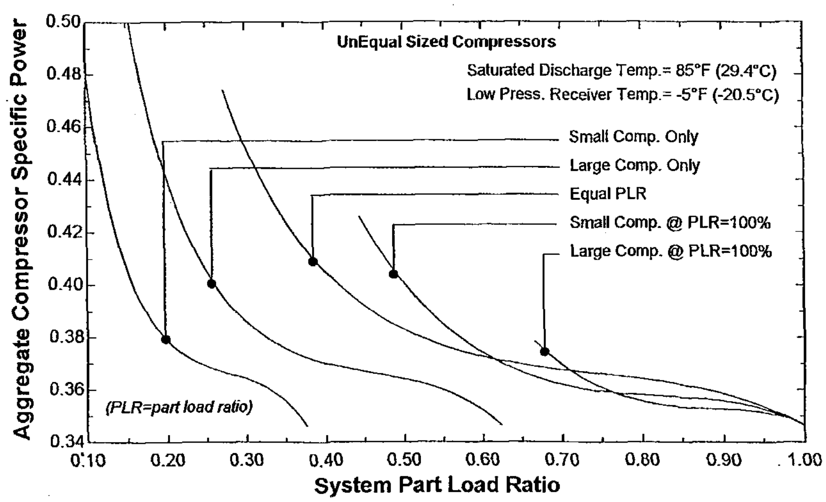

Figure 5: Optimum Performance Map for Unequal-Sized Screw Compressors

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. This is not the case however for screw compressors. 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 66 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 cross-over point occurred when the load was 76 percent of the available capacity of both compressors. Crossover points are independent of compressor suction and discharge conditions.

REFERENCES

Avallone, Eugene A (editor), Baumeister III, Theodore (editor), Marks' Standard Handbook For Mechanical Engineers, Ninth Edition, McGraw-Hill, New York, NY, 1987.

Crane Co., "Flow of Fluids Through Valves, Fittings, and Pipe." Technical Paper No. 410, Joliet, IL, 1988.

Fisher, M., Engineer, Vilter Manufacturing Company, Milwaukee, WI, private communication, 1998.

Manske, K. A., "Performance Optimization of Industrial Refrigeration Systems," M.S. Thesis, Mechanical Engineering," Solar Energy Laboratory, University of Wisconsin- Madison, 2000.

ACKNOWLEDGEMENT

This research has been sponsored by the Energy Service, Research, and Education Committee through a grant from the Energy Center of Wisconsin. The participation of Ruth Urban (Energy Center of Wisconsin) and Mike Slawny (WEPCO) in this project were appreciated.