Rating Equations for Positive Displacement Compressors with Auxiliary Suction Ports

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ABSTRACT

A method is described for generating performance rating equations for positive displacement compressors with fixed auxiliary suction ports from test data. Considerations are made to minimize the test data required to adequately describe the performance given the four independent variables of suction pressure, auxiliary suction pressure, discharge pressure, and input shaft speed. The equations are formulated to facilitate estimation of compressor performance with alternate refrigerants.

INTRODUCTION

This paper deals primarily with presenting performance data of twin screw oil flooded refrigeration compressors equipped with an auxiliary suction port (ref 1). In this case the compressor is an open shaft drive that can be applied at several different speeds. The auxiliary suction port can be used to support an additional evaporator or enhance system performance in an economizer cycle. Previous work has shown a method for presenting performance with a specific auxiliary pressure for each pressure ratio (ref 2). The method presented is more general in that it provides for a range of auxiliary suction port pressures.

It is customary to present refrigeration performance data as capacity and input power curves as a function of saturated suction temperature and saturated discharge temperature. The mathematical representation of this is usually a generalized cubic polynomial equation. However, the addition of two variables (speed and saturated economizer temperature) makes the generalized polynomial method quite cumbersome.

VARIABLE TRANSFORMATION

It has been shown in previous work that if volumetric efficiency and isentropic efficiency are shown as functions of pressure ratio and discharge pressure, considerable simplification of the equations is achieved (ref 3). In general that method is used here.

A variable ϕ_{ax} is added to represent the various auxiliary port pressures:

$$\phi_{ax} = \left(\frac{P_{ax}}{P_s}\right)^{\frac{1}{k_{ax}}} - \frac{V}{V_{ax}}$$

The main pressure ratio is represented by the variable ϕ_{mn} :

$$\phi_{mn} = \left(\frac{P_d}{P_s}\right)^{\frac{1}{k_{mn}}}$$

Where:

P_{ax} = Auxiliary port pressure (absolute)
P_s = Suction pressure (absolute)
V = Displaced volume
V_{ax} = Displaced volume after auxiliary port closing
k_{ax} = Polytropic constant for an isentropic process between P_s and P_{ax}
P_d = Discharge pressure (absolute)

P_s = Suction pressure (absolute)
k_{mn} = Polytropic constant for an isentropic process between P_s and P_d

The remaining independent variables are the absolute discharge pressure P_d and the speed n.

The dependent variables are:

η_{mn} = Main suction volumetric efficiency

$$\eta_{mn} = \frac{V_{mn}}{V} \times 100\%$$

η_{ax} = Auxiliary suction volumetric efficiency

$$\eta_{ax} = \frac{V_{ax}}{V_t} \times 100\%$$

η_{is} = Isentropic efficiency

$$\eta_{is} = \frac{Q_{mn} + Q_{ax}}{Q_{total}} \times 100\%$$

Where:

V_{mn} = Volume flow into suction port
V = Displaced volume rate
V_{ax} = Volume flow into auxiliary suction port
V_t = Displaced volume rate
Q_{mn} = Ideal power to compress main vapor from P_{mn} to P_d
Q_{ax} = Ideal power to compress auxiliary vapor from P_{ax} to P_d
Q_{total} = Input shaft power

EQUATIONS

When the appropriate interaction and curvature terms are added the performance curves are as follows:

$$\eta_{mn} = A_0 + A_1 \cdot n + A_2 \cdot n^2 + A_3 \cdot \phi_{mn} + A_5 \cdot n \cdot \phi_{ax} + A_6 \cdot \phi_{ax} + A_7 \cdot \phi_{mn} \cdot \phi_{ax} + A_8 \cdot n \cdot \phi_{ax} + A_{11} \cdot P_d$$

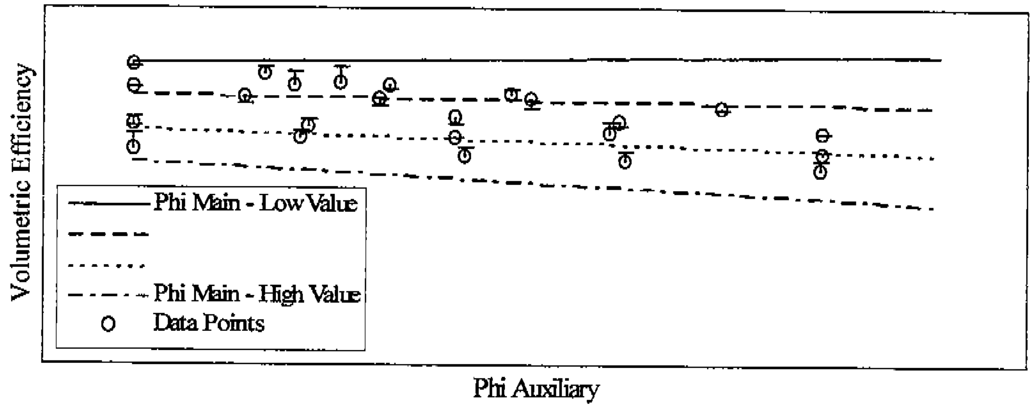
$$\eta_{ax} = B_6 \cdot \phi_{ax} + B_7 \cdot \phi_{mn} \cdot \phi_{ax} + B_8 \cdot n \cdot \phi_{ax} + B_9 \cdot n \cdot \phi_{ax}^2 + B_{10} \cdot n^2 \cdot \phi_{ax}$$

$$\eta_{is} = C_0 + C_1 \cdot n + C_2 \cdot n^2 + C_3 \cdot \phi_{mn} + C_4 \cdot \phi_{mn}^2 + C_5 \cdot n \cdot \phi_{mn} + C_6 \cdot \phi_{ax} + C_7 \cdot \phi_{mn} \cdot \phi_{ax} + C_8 \cdot n \cdot \phi_{ax} + C_{11} \cdot P_d$$

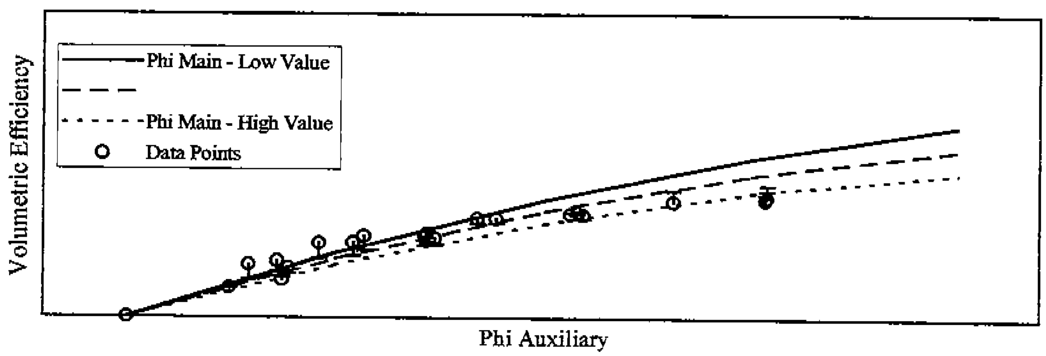
RESULTS

The following graphs show the results for an example compressor. The lines represent calculated data from the fitted equations. The circles are the test data. The bars at the end of the data point tails are the calculated data for that data point.

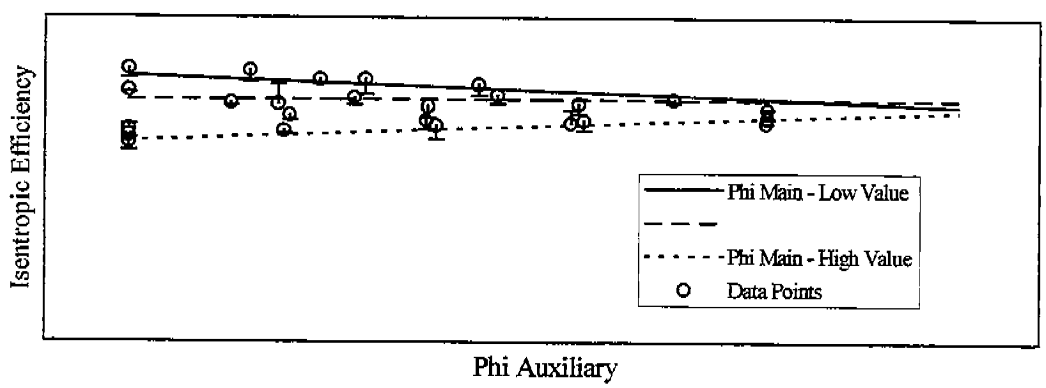
Main Port Volumetric Efficiency



Auxiliary Port Volumetric Efficiency



Isentropic Efficiency



CURVE FITTING

The data for the example was fit to curves by using multiple linear least squares regression. The interaction and higher order terms were calculated for each point and used as independent variables. The data points were selected informally so that each variable would be exercised over its entire range in combination with other variables. Intermediate points were chosen so that any curvature in the function could be uncovered. As a result, many more data points were taken then necessary. Residual error graphs for each variable was plotted to determine if any consistent errors remained. As a result of this analysis, the discharge pressure terms A_{11}, C_{11} were discarded for this example. Additional interaction terms between ϕ_{ax} and n were added.

The regression equations could be satisfied using only 12 data points. However, this number of points would not be adequate to assure that there would not be significant range and confounding errors. A good discussion of these errors can be found in reference 4. In order to avoid these errors it is suggested that a design of experiments orthogonal matrix (ref 5) be used to select data points.

ALTERNATE REFRIGERANTS

Due to environmental concerns, many refrigerants and refrigerant mixtures have been proposed in recent years. In order to facilitate performance estimates with different refrigerants, the equations have been formulated to be independent of the refrigerant properties as possible. The independent variables ϕ_{mn} and ϕ_{ax} are actually based on volumes. For example:

$$\left(\frac{P_d}{P_s}\right)^{\frac{1}{k_{mn}}} = \frac{v_s}{v_{dis}}$$

Where:

v_s = Specific volume of the suction gas.

v_{dis} = Specific volume of the discharge gas when compressed with an isentropic process.

Please note that a more practical method of calculating ϕ_{mn} is ϕ_{ax} with real properties, is to use this volume ratio and forgo the calculation of k_{mn} and k_{ax}.

Of course the equations are not completely independent of refrigerant properties. Internal pressure drop, heat transfer and oil solubility are at least three processes built into the equations that are affected by refrigerant properties. However, the equations are adequate for making rough performance estimates by using properties from a substitute refrigerant. When in doubt, test.

LIMITATIONS AND CAUTIONS

Like all empirical data, overgeneralization can lead to significantly incorrect results. The following is a list of known pitfalls:

1. The equations do not extrapolate well at high main pressure ratios. Be sure to include high pressure ratios in your test plan.

2. The performance is sensitive to wide ranges of suction temperature. If it is planned to use high superheats, which is typical when using liquid to suction heat exchangers, generate new performance curves for this application.

3. The main volumetric efficiency is not linear at low pressure ratios. Use of a second order term may be needed at pressure ratios lower than 5. Use of a reciprocal term may be needed at pressure ratios lower than 3.

4. Always calculate and plot the calculated performance over the range of expected use, especially if the range of use is extrapolated. Although the terms used here are simple, fitted curves can behave unpredictably.

5. Always plot the test data and errors over the calculated data.

6. The method is described for fixed auxiliary suction ports. A different method is required for two stage compressors.

CONCLUSIONS

A method has been shown for fitting equations to performance data from a positive displacement compressor with a fixed auxiliary suction port. The equations are generalized so that performance with various economizer heat exchangers can be calculated. The equations are formulated so that rough performance estimates using alternate refrigerants can be conveniently calculated.

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