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ABSTRACT

The overall objective of this research is to identify a physics-based method to characterize compressor performance in refrigeration systems with limited experimental data. The focus of this project is on positive displacement compressors, i.e., reciprocating, rotary, scroll, and screw types, configured as either semi-hermetic or open drive. Compressor performance data for these types of compressors with different sizes were obtained from various manufacturers. One data set consisted of raw experimental data, while the others datasets were based on published catalog data. Mass flow estimates are based on the polytropic compression process with a clearance volume that leads to a volumetric efficiency expression. The overall performance of the model was acceptable with maximum average mean weighted errors of 3.7%, 2.3%, and 0.6% for reciprocating, scroll, and screw compressors, respectively. Furthermore, it was found that the mass flow rate model with parameters estimated using data for one refrigerant accurately predicted data for a different refrigerant. The compressor power requirement is also based on the polytropic model with the introduction of a combined efficiency to account for frictional effects, leakage, and motor performance in hermetic units. Comparisons of the predicted power requirements with data showed that the model fell short of predicting compressor power performance within acceptable accuracy. Average mean weighted errors over a range of operating conditions were 8% for the screw, 7.6% for the scroll, 6.4% for the open-drive reciprocating, and 5% for the semi-hermetic reciprocating compressors. Errors of as much as 40% were observed for some operating conditions.

NOMENCLATURE

C: clearance volume fraction
d, e, f: parameters in Eqn (5)
k: isentropic index
$\dot{m}$: refrigerant mass flow rate
n: polytropic coefficient
N: number of data points
OF: objective function defined in Eqn (1)
$P_{evaporation}$: evaporator pressure
$P_{discharge}$: discharge pressure
$P_{suction}$: suction pressure
RPM: compressor speed
$v_{suction}$: specific volume at suction conditions
V: compressor displacement volume
w: compressor power per unit mass flow
Δp: parameter defined in Eqn (3)
$\eta_{combined}$: efficiency factor defined in Eqn (4)

INTRODUCTION

Characterization of compressor performance is necessary in order to provide the manufacturer and the customer with refrigerant mass flow rate and compressor power requirements as a function of operating conditions. ARI Standard 540 (1999) currently provides a means of characterizing the capacity (or refrigerant mass flow rate) and power for a specific compressor operating with a specific refrigerant. The ARI standard is based on a bi-quadratic linear regression that requires a minimum of 10 calorimeter tests. The primary advantage of the ARI method is that application is relatively simple and straightforward. There are, however, a number of significant disadvantages with ARI 540. First, conducting calorimeter tests are time-consuming and expensive. Second, the ARI method is completely empirical. As a result, it cannot reliably provide estimates of compressor performance for conditions outside the range of the test data used in the development of the regression. Finally, separate regressions (and calorimeter tests) are required for each individual refrigerant used by the compressor being tested. The ARI method cannot be used for estimating compressor performance operating with different refrigerants.

The overall objective of this research was to develop a semi-empirical methodology for characterizing compressor performance that incorporates some of the physical processes occurring in the compressor, rather than by relying on a totally empirical formulation as is currently done with the ARI Standard 540. Ideally, the resulting methodology will a) reduce the number of calorimeter tests needed for characterizing the performance of a compressor operating with a given refrigerant; b) allow more accurate extrapolation of compressor performance to conditions beyond the range for which tests are available; and 3) leverage the calorimeter tests with one refrigerant for use to predict compressor performance with a different refrigerant.

COMPRESSOR DATA

Compressor performance data for reciprocating, rotary, scroll, and screw compressors of different sizes were collected from manufacturers. One data set represents raw experimental data, while the others datasets are based on published catalog data from various manufacturers. Table 1 list all compressor data sets used in this study. Each data set is assigned with a identification data set number and an upper case letter. This data set number is used to identify the compressor in the Table 1 also identifies the compressor manufacturer, the refrigerant type and the number of provided data points.

Data Set	Compressor Type	Manufacturer	Refrigerant	Number of Data Points
A-1	Semi-Hermetic Reciprocating	Copeland Corp.	R134a	96
A-2	Semi-Hermetic Reciprocating	Copeland Corp.	R134a	96
A-3	Semi-Hermetic Reciprocating	Copeland Corp.	R134a	96
A-4	Semi-Hermetic Reciprocating	Copeland Corp.	R134a	96
A-5	Semi-Hermetic Reciprocating	Copeland Corp.	R22	56
A-6	Semi-Hermetic Reciprocating	Copeland Corp.	R22	64
A-7	Semi-Hermetic Reciprocating	Copeland Corp.	R22	64
A-8	Semi-Hermetic Reciprocating	Copeland Corp.	R22	64
A-9	Semi-Hermetic Reciprocating	Copeland Corp.	R22	64
B-1	Open-Drive Reciprocating	Vilter	R22	134
B-2	Open-Drive Reciprocating	Vilter	R22	134
B-3	Open-Drive Reciprocating	Vilter	R22	134
B-4	Open-Drive Reciprocating	Vilter	R717	78
B-5	Open-Drive Reciprocating	Vilter	R717	78
B-6	Open-Drive Reciprocating	Vilter	R717	78
C-1	Rotary	/	R22	71
D-1	Scroll	/	R22	16
D-2	Scroll	Copeland Corp.	R22	53
D-3	Scroll	Copeland Corp.	R22	53
D-4	Scroll	Copeland Corp.	R22	53
D-5	Scroll	Copeland Corp.	R22	53
E-1	Single-Screw	Vilter	R22	36
E-2	Single-Screw	Vilter	R22	36
E-3	Single-Screw	Vilter	R717	36
E-4	Single-Screw	Vilter	R717	36

Table 1: Summary of Compressor Data Sets used in the study

COMPRESSOR PERFORMANCE MODEL

The performance of positive-displacement compressors have been well-studied and many performance models of varying detail can be found in the literature, e.g., Prakash and Singh (1974), Röttger and Kruse (1976), Brok et al. (1980), Sjöholm (1988), Todescat et al. (1992), Cavallini et al. (1996) and Chen et al (1998). However, most of these models require information that is not readily available and the detail in the models, although useful for design, is not appropriate for the characterization that is of interest in this study. For this reason, the present study focused on the simple polytropic model, as described by Kuehn et al. (1998) and recently used in studies by Haberschill et al. (1994), Popovic and Shapiro (1995), Browne and Bansal (1998), Jaehnig (1999) and Kim and Bullard (2001).

Our efforts here are aimed at extending the work of Jaehnig (1999) to larger scale reciprocating compressors, to other compressor technologies (screw, scroll), and to other compressor configurations (open-drive). Refrigerant mass flow rate and compressor power were separately modeled using a semi-empirical form based on a polytropic model. The empirical parameters in the model were determined using non-linear regression with a commercial software application (Klein and Alvarado, 2001) to minimize the objective function in Eqn (1) by altering the values of the parameters within specified bounds. Normalizing the error with the average of all measured values, as is done in Eqn (1), ensures that all data points are weighted equally.

$$OF = \sqrt{\frac{\sum_{i=1}^{N}\left(\frac{X_{meas} - X_{calc}}{X_{mean}}\right)^2}{N}} \tag{1}$$

where

OF	objective function
N	number of data points
X_{meas}	measured mass flow rate or specific power (power per unit mass flow rate)
X_{calc}	calculated mass flow rate or specific power (power per unit mass flow rate)
X_{mean}	average of all measured mass flow rate or specific power (power per unit mass flow rate) data

Mass Flow Model

The mass flow model is based on the concepts of a polytropic process and volumetric efficiency. Volumetric efficiency is defined as the as the ratio of the volume of suction gas actually entering the compressor (at the prevailing suction and pressure) to the maximum volume of gas that could be drawn into the cylinder (i.e. the compressor displacement volume). Kuehn et al. (1998) show that for a reciprocating compressor, the refrigerant mass flow rate, $\dot{m}$, can be expressed as:

$$\dot{m} = \left(1 + C - C \cdot \left(\frac{p_{discharge}}{p_{suction}}\right)^{\frac{1}{n}}\right)\frac{V \cdot RPM}{v_{suction}} \tag{2}$$

where

C	clearance volume ratio
$p_{discharge}$	discharge pressure
$p_{suction}$	suction pressure
n	polytropic exponent, i.e., the value for n for which $p\,v^n$ is constant
V	compressor displacement volume
RPM	compressor speed
$v_{suction}$	specific volume of the refrigerant under intake conditions

The polytropic index, *n*, should in theory depend on the extent of heat transfer occurring during the compression process. Limiting values for the polytropic index are unity for isothermal compression and the isentropic index for adiabatic compression. Compressors are often assumed to operate adiabatically, although heat transfer can be particularly significant for small compressors and for larger screw compressors due to oil cooling. Our study attempted to find a best value of polytropic index, both as a constant and as a function of suction and discharge conditions. We found that the objective function (Eqn 1) to be only slightly affected by varying the polytropic exponent and so to simplify the method, n is set equal to the isentropic exponent, k, for the respective refrigerant being used in the compressor. The isentropic exponent was set to a constant value determined at 65°F (18.3°C) and the evaporator pressure.

The pressure of the refrigerant at the compressor suction is lower than the evaporating pressure. This effect is considered in the model by introducing parameter, Δp, defined in Eqn (3). The specific volume, $v_{suction}$, is then determined for the specified refrigerant at the compressor suction temperature and pressure ($p_{suction}$).

$$p_{suction} = p_{evaporation}\left(1-\Delta p\right) \tag{3}$$

There are two parameters in the mass flow model the clearance volume, C, and the pressure drop, Δp. These parameters are selected to minimize the objective function in Eqn (1). However, in some cases, the displacement rate of the compressor, $V \cdot RPM$, was not known and it too had to be estimated from manufacturer's data. In this case, the product ($V \cdot RPM$) was estimated as the ratio the mass flow rate to inlet density at conditions that provided high volumetric efficiency, i.e., high refrigerant suction pressures and low refrigerant discharge pressures. Although the foundation of this mass flow model is based on reciprocating compressors, it was applied to all of the compressors. Screw compressors equipped with variable volume ratio control approximate the behavior of a reciprocating compressor.

Power Model

The polytropic model can be extended to provide an estimate of the specific compressor power, w, defined as the ratio of the power to mass flow rate, as described in Kuehn et al. (1998). We found it necessary to modify the polytropic model expression to include an efficiency factor, $\eta_{combined}$ so that the specific power is as represented in Eqn (4). $\eta_{combined}$ is called the combined efficiency since it represents the combined efficiency of the electric motor (in hermetic units) and inefficiencies in the compressor operation such as friction and leakage. All other parameters in Eqn (4) were taken as defined in the mass flow analysis.

$$w \cdot \eta_{combined} = \frac{n}{n-1} \cdot p_{suction} \cdot v_{suction}\left[\left(\frac{p_{discharge}}{p_{suction}}\right)^{\frac{n-1}{n}} - 1\right] \tag{4}$$

By comparing manufacturer's data with w determined from Eqn (4), we concluded that that $\eta_{combined}$ can not be considered to be a constant. The investigation focused on identifying a suitable relationship to predict the combined efficiency using measured variables such as discharge and/or evaporating pressure. Jaehnig (1999) suggested two different relationships for the combined efficiency that depend only on the evaporating pressure. One of the early conclusions for this project was that the Jaehnig relationships for the combined efficiency did not perform well for larger compressors (including both hermetic and open-drive configurations). A number of alternative relationships were proposed involving both the evaporating and discharge pressures. The relationship shown in Eqn (5) was found to perform as well or better than any of those investigated.

$$\eta_{combined} = d + e \cdot p_{suction} + f \cdot p_{discharge} \tag{5}$$

where d, e and f are curve fit parameters fit by regression to manufacturer's data.

RESULTS AND DISCUSSION

Figure 1a shows estimated and reported mass flow rate data for a small semi-hermetic compressor (data set A-1) using R134a. The average mean weighted error (1.9%) is the objective function, *OF*, defined in Eqn (1). The largest error was 5.8%. Figure 1b shows the results for a large open-drive reciprocating compressor (data set B-3) using R22. The average mean weighted error for this data set is 1.3% and the largest error is 3.7%. Figures 1c and d show results for a scroll compressor (data set D-5) and a screw compressor (data set E-2), respectively. The largest errors in estimated mass flow rate generally occur at the lowest and highest condensing temperatures (70°F and 130°F/140°F) Note that for some data sets, the Δp parameter determined by the non-linear regression is negative. The only possible explanation for a negative value we can offer is that the parameter, Δp, is attempting to compensate for limitations in the model or errors in the data. Nevertheless, better overall fits were observed if the constraint $\Delta p > 0$ was not enforced. The overall performance of the mass flow model was acceptable with *maximum* average mean weighted errors of 3.7%, 2.3%, 0.6%, and 0.6% for reciprocating, scroll, rotary, and screw compressors, respectively.

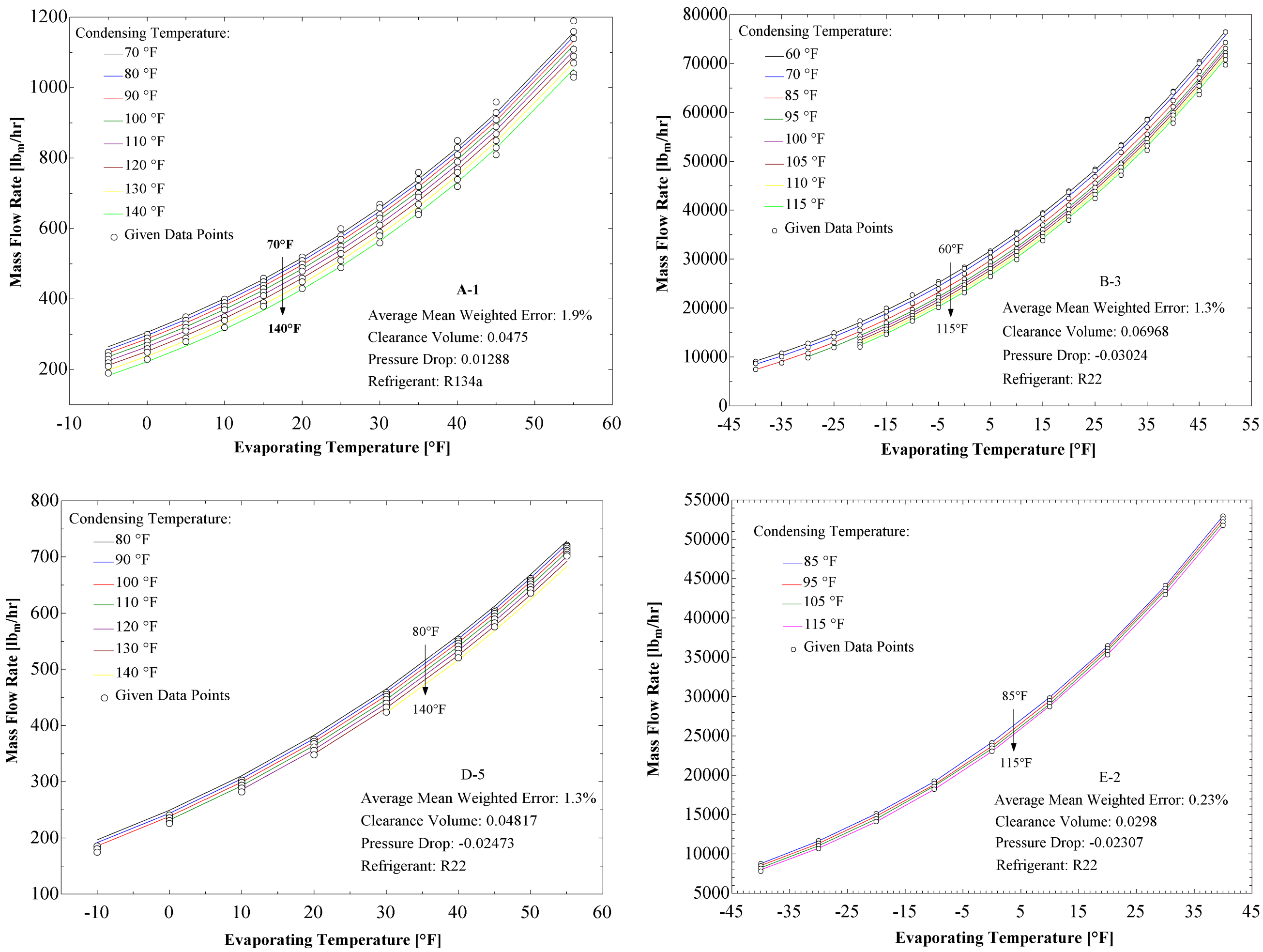

Figure 1: Comparison of mass flow model (lines) and data (symbols) for data sets A-1, B-3, D-5 and E-2.

An objective of this project was to determine if the performance of a compressor with a specific refrigerant could be accurately estimated using experimental data for the same compressor operating with a different refrigerant. A prerequisite for testing this possibility is the availability of performance data for a compressor with at least two different refrigerants. Data were available for three different open-drive reciprocating compressors for both R22 and R717 (data sets B-1 to B-6). The different compressors are reciprocating models of different sizes with two, six, and twelve cylinder configurations. The results summarized in Table 2 are encouraging. Although smaller errors were obtained when mass flow rate data for the refrigerant of interest were used, reasonably good accuracy was obtained using mass flow rate data for a different refrigerant.

Data Set	Refrigerant	Average Mean Weighted Error [%] (best fit to original data)	Average Mean Weighted Error [%] (using data for other refrigerant)
B-1	R22	1.3	3.3
B-2	R22	1.3	3.3
B-3	R22	1.3	3.3
B-4	R717	2.0	3.1
B-5	R717	2.0	3.1
B-6	R717	2.0	3.1

Table 2: Result of estimating mass flow with compressor data with a different refrigerant.

Figure 2 shows a comparison of the specific power estimates and manufacturer's data for the same four compressors in Figure 1.

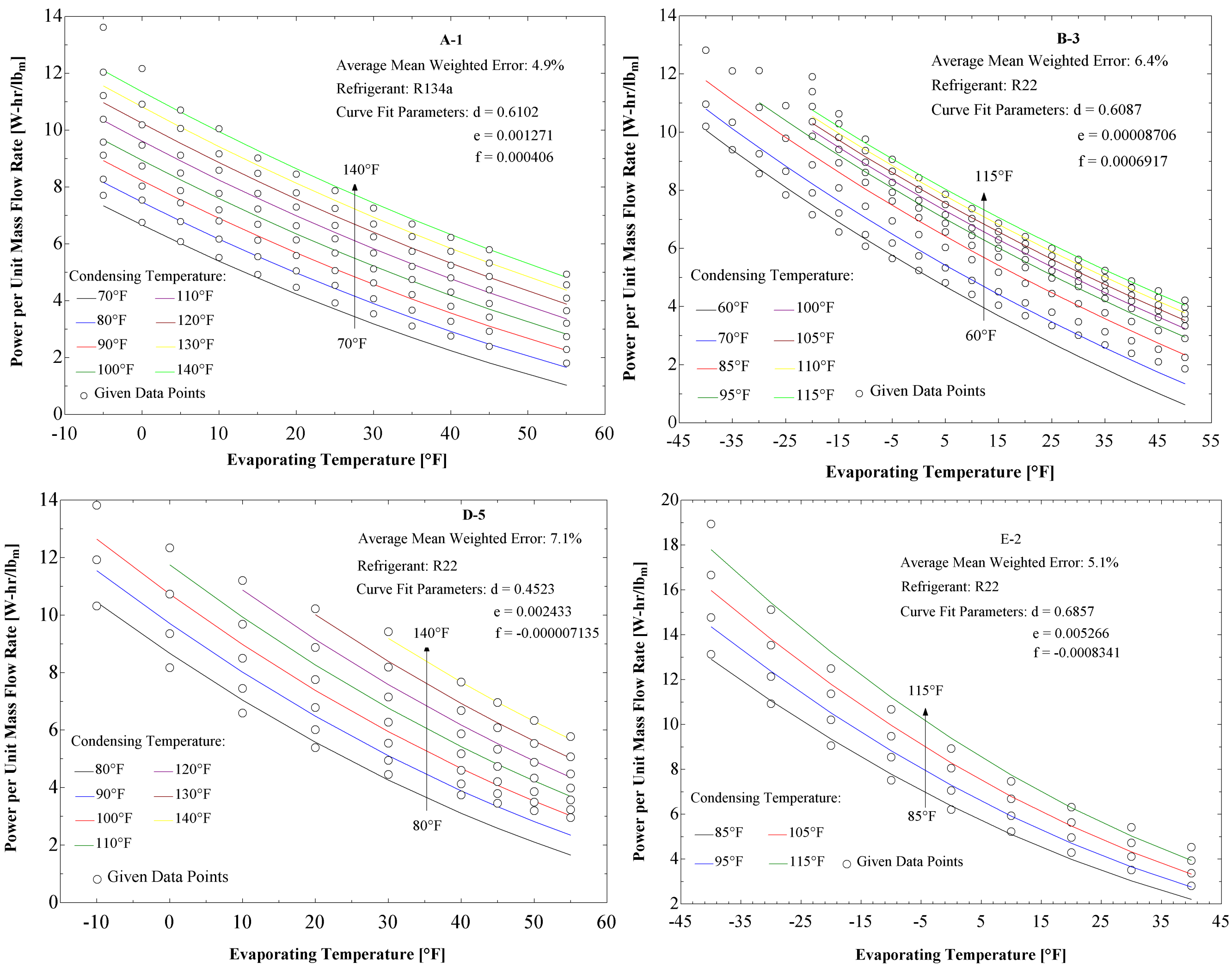

Figure 2: Comparison of specific power model (lines) and data (symbols) for data sets A-1, B-3, D-5 and E-2.

The specific power model results seen in Figure 2 are not good as evident by the average mean weighted errors above 5%. When all of the compressors in Table 1 are considered, the average mean weighted errors were 5% for the semi-hermetic reciprocating, 6.4% for the open-drive reciprocating, 7.6% for the scroll and 8% for the screw compressors. Errors of as much as 40% were observed for some operating conditions. The errors are not random. The model presented in Eqn (4) seems to be unable to fit the trends observed in the specific power data for some situations. For example, the model seems to consistently under-predict the specific power at high evaporating temperatures and low condensing temperatures. Although many alternative expressions for the combined efficiency were investigated, none was found to represent the specific power data in a more satisfactory manner.

CONCLUSIONS

The polytropic model for compressor performance has been extensively used in studies of compressor performance. The relationships used investigated in this paper are presented in the ASHRAE (2000) and in textbooks, e.g., Kuehn et al. (1998). The intent of this investigation was to use established compressor performance theory embodied in the polytropic model along with empirical determination of unknown parameters to establish a method of characterizing compressor performance that:

1. reduces the amount of required compressor test data
2. allows extrapolation of test data to conditions outside the range of the tests
3. allows estimates of performance using different refrigerants.

Unfortunately, the investigation has failed to be successful in achieving all of these goals for a range of compression technologies that included reciprocating (open-drive and semi-hermetic), screw (open-drive and semi-hermetic), and scroll. Mass flow rate was predicted with reasonable accuracy for most of the compressors investigated using Eqns (2) and (3) with parameters ($V\ RPM$), C and Δp determined from manufacturer's data. Even though this model is strictly applicable to reciprocating compressors, it was found to work well for other positive displacement compressors. In fact, the best agreement between the data and predictions occurred for the screw compressors. The one radial compressor investigated also showed excellent agreement. However, much larger differences between predictions and data were observed when the polytropic model was used to estimate specific power with Eqn (4). An efficiency factor that depends on compressor operating conditions was found to be a necessary part of the model. A form for this efficiency factor was proposed in Eqn (5). However, the combination of Eqns (4) and (5) did not adequately represent the available compressor data over the entire range of operation. The model failed most noticeably at low lift conditions resulting from high evaporator and low condensing temperatures. Attempts to include a term for pressure ratio in Eqn (5) did not improve the overall performance. Certainly, better agreement could be obtained by adding more parameters to the efficiency model. However, additional compressor tests would be needed to establish estimates for the additional parameters which diminishes the advantage of the semi-empirical formulation provided by the polytropic model over empirical polynomials as used in ARI-540 standard.

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