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A Proposal of Graphing Methods for Improved Compressor Test Data Evaluation

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ABSTRACT

This paper provides alternative improved graphing methods for better test data evaluation of refrigeration compressors. The compressor could be of many different types of positive displacement compressors such as scroll, screw, rolling piston, and reciprocating compressors. They include both shaft driven and hermetic compressors as well as with and without vapor injection.

The conventional method is to graph volumetric and isentropic efficiencies based on test data. However, it has limitations such as defining an appropriate geometric displacement, graphing the isentropic efficiency, and evaluating the energy balance. These issues are even more complicated when considering built-in volume ratio, vapor injection, large speed range, and refrigerants with large glide.

The proposed method consists of test data that are directly or indirectly graphed: Suction mass flow, vapor injection mass flow, compressor power, discharge temperature and compressor energy balance. This proposed method is used as filtering of the test points prior to applying those test points in any type of compressor model. Examples of the proposed graphs are presented using actual test data. This approach minimizes the risk of overfitting certain parameters and drives a more harmonized approach of all parameters.

1. INTRODUCTION

A refrigeration compressor needs to be rated regarding at least refrigeration capacity and power consumption. Those are mainly dependent on condensing temperature, evaporating temperature, refrigerant, and speed for a positive displacement compressor. When explaining a positive displacement compressor, most thermodynamic handbooks start talking about volumetric and isentropic efficiency of the compressor. Based on this it is common to put those efficiencies on equation format relative to above variables. This method has a few shortcomings which is especially true when the compressor has a built-in volume ratio and is combined with vapor injection. The volumetric efficiency is normally plotted versus pressure ratio for a given discharge pressure and a given speed. The volumetric efficiency is very dependent on the definition of theoretical displacement. An example of the volumetric efficiency over 100 per cent, which is not expected is shown in the previous study (figure 16, Sjoholm, 1986a). The volumetric efficiency is also somewhat dependent on the built-in volume ratio (figure 1, Sjoholm, 1988). The isentropic efficiency is normally plotted with pressure ratio for a given discharge pressure and a given speed. It has a very different curvature at low pressure ratios; however, it is very dependent on built-in volume ratio (figure 2, Sjoholm, 1988). The volumetric and isentropic efficiencies become even more complicated when adding vapor injection, because the theoretical process must be defined along with a built-in volume ratio as shown in the previous study (figure 3 and 7, Sjoholm, 1986b).

The purpose of graphing test results is to look for trends and to see what points do not adhere to those trends. The graphing methods that are presented in this paper corresponds to very simple calculations of raw test data and the shown graphs can easily be represented with simple equations.

2. CONVENTIONAL GRAPHING METHODS

Two of the conventional methods of characterizing a compressor's performance have been to volumetric and isentropic efficiency. Volumetric efficiency demonstrates the ability of a compressor at a given displacement to move a fluid through a system.

2.1 Volumetric Efficiency

Since the compressor is the driving component, pushing the refrigerant through the cycle, the mass flow rate is determined by the compressor (ANSI/ASHRAE Standard 23.1, 2019). By calculating the compressor volumetric efficiency at a variety of conditions one may understand how effective the compressor is at pushing refrigerant through the cycle in relation to its ideal potential. The volumetric efficiency of a compressor can be computed as:

$$\eta_{vol} = \frac{(\dot{m}_{actual})(v)}{(V)\,(N)} \cdot 100 \qquad (1)$$

where:

$\dot{m}_{actual}$ = actual mass flow rate (lbm/min)
N = rotational speed of the compressor (rev/min)
V = compressor displacement (in^3/rev)
v = specific volume of refrigerant entering compressor (in^3/lbm)

Complications can arise while attempting to calculate volumetric efficiency such as: some types of compressors have complicated geometries and defining a compressor displacement may not be well defined; addition of vapor injection and its effects on compression process; and compressor characteristics over a large speed range may change drastically.

2.2 Isentropic Efficiency

The compressor requires power to compress the refrigerant from low pressure to high pressure. This power can be either electrical power supplied to an internal motor or mechanical power supplied to an external shaft. The isentropic efficiency is a typical way to express how efficient a compressor is compared to ideal isentropic compression (ANSI/ASHRAE Standard 23.1, 2019). This can be computed as:

$$\eta_{isentropic} = \frac{W_{isentropic}}{W_{actual}} \cdot 100 = \frac{\dot{m}(h_2 - h_1)}{P_{actual}} \cdot 100 \qquad (2)$$

where:

$W_{isentropic}$ = isentropic power to compress refrigerant from state 1 to state 2 (Btu/hr)
W_{actual} = actual power supplied to compressor (Btu/hr)
$\dot{m}$ = mass flow rate (lbm/hr)
h_1 = specific enthalpy at compressor inlet (Btu/lbm)
h_2 = specific enthalpy of refrigerant vapor at discharge pressure following an isentropic compression of the refrigerant from compressor suction pressure and temperature (Btu/lbm)

The use of modern refrigerant blends with larger glides tend to complicate isentropic measurements over a wide range of applications. The isentropic efficiency including vapor injection may be calculated several different ways leads to further confusion.

3. TEST POINT DISTRIBUTION

This task is more complicated than it appears, especially when it is for a new refrigerant, a new or modified compressor, a compressor designed for a different refrigerant or refrigerant lubricant combination, or the compressor may be designed for a different application. Even the compressor test stand may have been designed for a different situation. In cases like this, it is usually best to test the compressor in as wide range as possible, where the limits of the test plan are based on compressor and test stand limitations. Those limitations may not be very well known with so many new or unknown factors.

The first step is to define the discharge pressure range and set up the test plan for at least three different discharge pressures. Secondly, chose about 4 suction pressures, corresponding to two frozen and two fresh conditions. If the test plan is being set up for a large speed range, a minimum of three speeds must be chosen. At this point we have 36 test points (= 3 discharge pressures x 4 suction pressures x 3 speeds). If the size of the economizer heat exchanger is known and we have a temporary simulation program, the balancing vapor injection pressure can be calculated. This means that the test points with and without vapor injection would be 72 test points. Sometimes we would not test all 72 test points due to compressor and test stand limitations. If the size of the economizer heat exchanger is not known, we may also have to test two or three different vapor injection pressures for each of the different suction, discharge pressure and speed points. This would give 108 test points (= 3 discharge pressures x 4 suction pressures x 3 speeds x 3 vapor injection pressures (2 with vapor injection and 1 without)).

It is recommended to spread out the pressures evenly within each range. Even if the basic points are distributed based on pressure, it is generally best to show the test points with discharge dew temperature on the y-axis and suction dew temperature on the x-axis. However, be prepared to change the conditions during each test, especially at edge or corner points.

Figures 3.1 and 3.2 show the test layout of actual performed tests where the compressor is a hermetic positive displacement, fixed built-in volume ratio compressor tested on a refrigerant in the 400 series with at least some portion of R-32 and R-1234yf. In fact, all the figures in chapter 4 are from the same test and the compressor has been tested without and with vapor injection.

There are three different methods to deal with the vapor injection pressure. The first method is to test each suction dew point and discharge dew point with a range of vapor injection dew points. For Figure 3.2, the temperature difference between vapor injection dew temperature and suction dew temperature corresponds to a range of 24 °F to 66 °F. The second method is to test with theoretical calculated balanced vapor injection pressure. The third method is to test with an actual economizer heat exchanger and an actual economizer expansion valve.

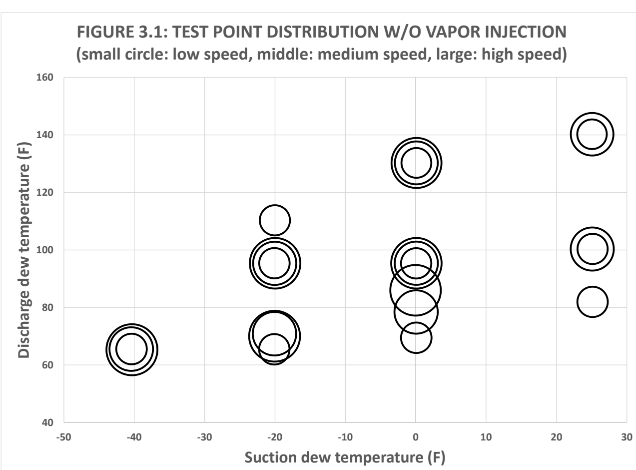


Figure 3.1: Test Point Distribution without Vapor Injection

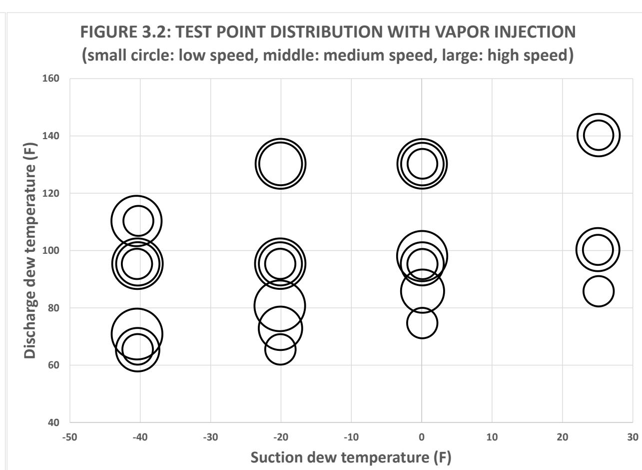


Figure 3.1: Test Point Distribution without Vapor Injection **Figure 3.2:** Test Point Distribution with Vapor Injection

4. PROPOSED GRAPHING METHODS

4.1 Suction Mass Flow

Mass flow can be measured in either liquid or gas state but in liquid sub-cooled state, increased accuracy is typical. The mass flow can be measured directly with a Coriolis type mass flow meter or can be calculated from heat measurements by calorimetric methods. The inlet or suction condition of the compressor defines the suction pressure and temperature. For each speed or frequency, simply plot the suction pressure on the x-axis and the suction mass flow on the y-axis, see Figure 4.1 and 4.2. Typically, the suction temperature corresponds to a constant super-heat.

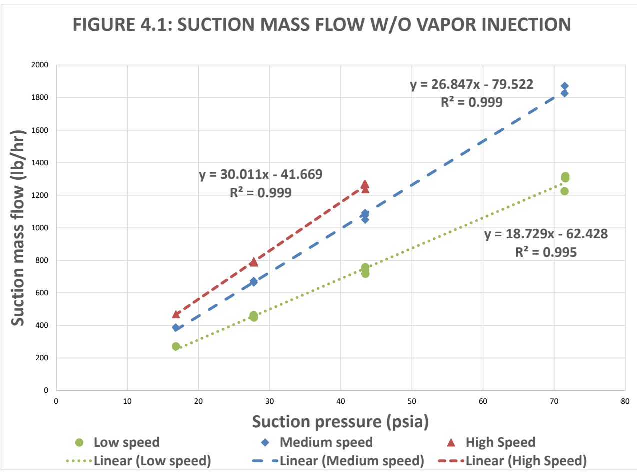


Figure 4.1: Suction mass flow without vapor injection

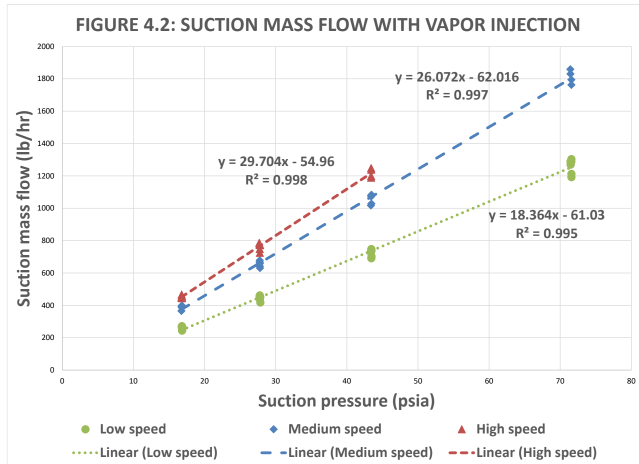
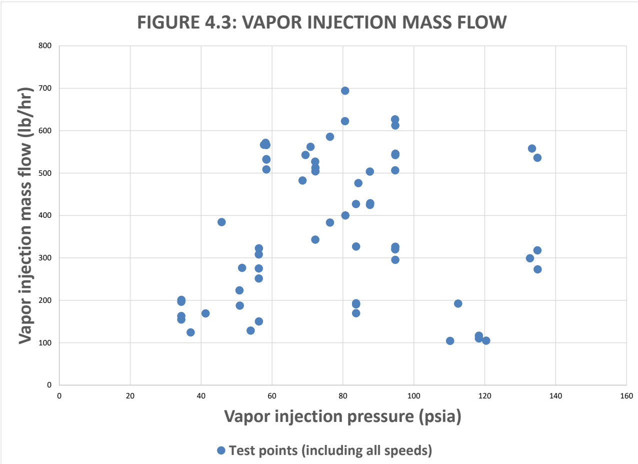


Figure 4.2: Suction mass flow with vapor injection

4.2 Vapor Injection Mass Flow

The vapor injection mass flow can also be measured on the liquid and gas side. The vapor injection mass flow is dependent on many factors: The placement and size of the economizer port, the compressor speed, the refrigerant, a balance point corresponding to a certain size economizer heat exchanger and the control of the economizer expansion valve. Figure 4.3 is created to show that the vapor injection mass flow and vapor injection pressure cannot be treated like suction mass flow and suction pressure. A relative way to deal with vapor injection mass flow and vapor injection pressure is shown in the study (Tello-Oquendo, *et al.*, 2017). Figure 4.4 shows the relative vapor injection pressure plotted versus the relative vapor injected mass flow. Typically, the vapor injection temperature corresponds to a constant super-heat.



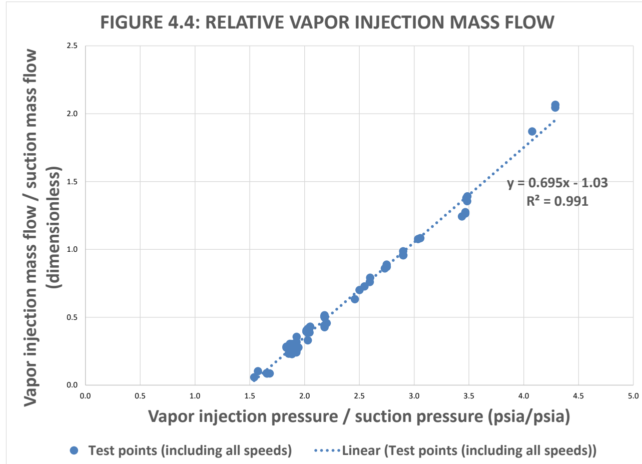


Figure 4.3: Vapor injection mass flow

Figure 4.4: Relative vapor injection mass flow

4.3 Compressor Power Number (CPN)

The compressor power is typically measured with a torque meter together with a speed sensor for an open shaft compressor. For a hermetic or semi-hermetic compressor, the electric power going into the built-in motor is measured with a watt meter. For an ideal compressor with fixed built-in volume ratio compressing an ideal gas, a non-dimensional number in the Si system, here called the Hjalmar's number, can be calculated taking the power consumption divided by the suction pressure, the compressor's displacement, and the speed. This number plotted on the y-axis and the pressure ratio on the x-axis give a straight line. Hjalmar Schibbye is an engineer that used to work at SRM, Svenska Rotor Maskiner in Sweden.

When adding vapor injection, it is challenging to define the displacement for the vapor injection port because the vapor injection port is not a positive displacement port. Due to this situation, we define the Compressor Power Number (CPN):

$$CPN = \frac{W}{P_s \cdot \dot{m}_s \cdot v_s + P_{vi} \cdot \dot{m}_{vi} \cdot v_{vi}} \tag{3}$$

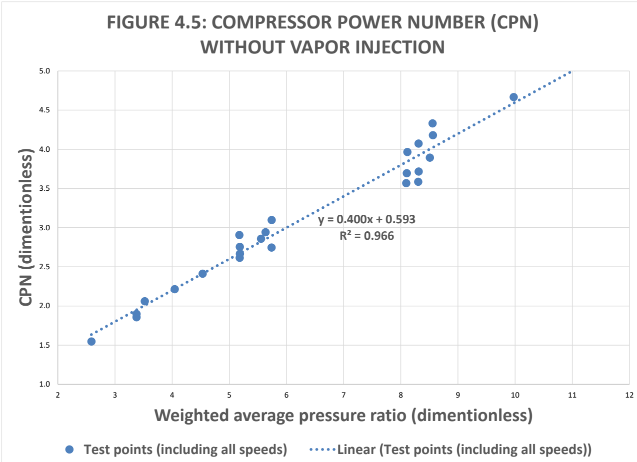
And weighted average pressure ratio:

$$Weighted\ average\ pressure\ ratio = \frac{P_d}{P_s} \cdot \frac{\dot{m}_s}{\dot{m}_s + \dot{m}_{vi}} + \frac{P_d}{P_{vi}} \cdot \frac{\dot{m}_{vi}}{\dot{m}_s + \dot{m}_{vi}} \tag{4}$$

Where:

P_d = compressor discharge pressure (Pa)
P_s = compressor suction pressure (Pa)
P_{vi} = vapor injection pressure (Pa)
W = compressor power (watt)
$\dot{m}_s$ = suction gas mass flow (kg/s)
$\dot{m}_{vi}$ = vapor injection gas mass flow (kg/s)
v_s = specific volume of gas entering compressor suction inlet (m^3/kg) using suction temperature T_s and P_s
v_{vi} = specific volume of gas entering compressor vapor injection inlet (m^3/kg) using vapor injection temperature T_{vi} and P_{vi}
CPN = compressor power number (dimensionless)

Figure 4.5 and Figure 4.6 show the weighted average pressure ratio versus the compressor power number (CPN) regarding without and with vapor injection.



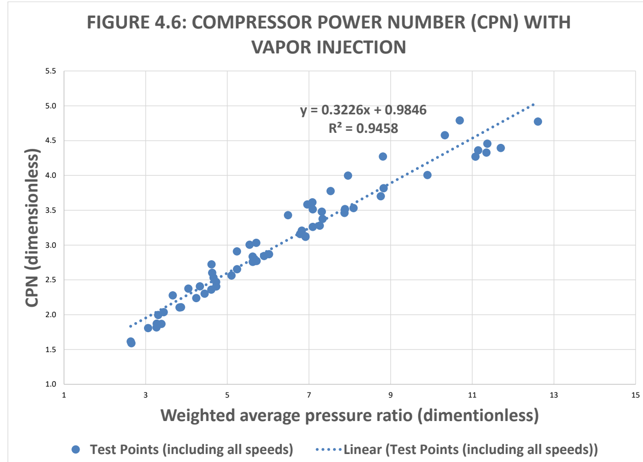


Figure 4.5: Compressor Power Number without vapor injection

Figure 4.6: Compressor Power Number with vapor injection

4.4 Discharge Temperature Index (DTI)

A similar approach used in Chapter 4.3 is applied to define DTI. We create a discharge temperature index (DTI) by replacing the numerator compressor power with discharge temperature and use the same denominator as in the compressor power number (CPN). Plot the weighted average inlet pressure on the x-axis and the discharge temperature index on the y-axis. The purpose of the Discharge Temperature Index (DTI) is to create graphs where the discharge temperature is indirectly graphed against fewer parameters as compared to the Compressor Heat and Power Ration (CHPR) in Chapter 4.5.

We define discharge temperature index (DTI):

$$DTI\ \left(\frac{Kelvin}{watt}\right) = \frac{T_d}{P_s \cdot \dot{m}_s \cdot v_s + P_{vi} \cdot \dot{m}_{vi} \cdot v_{vi}} \tag{5}$$

And weighted average inlet pressure:

$$Weighted\ average\ inlet\ pressure = P_s \cdot \frac{\dot{m}_s}{\dot{m}_s + \dot{m}_{vi}} + P_{vi} \cdot \frac{\dot{m}_{vi}}{\dot{m}_s + \dot{m}_{vi}} \tag{6}$$

Where:

T_d = discharge gas temperature (K)
P_s = compressor suction pressure (Pa)
P_{vi} = vapor injection pressure (Pa)
$\dot{m}_s$ = suction gas mass flow (kg/s)
$\dot{m}_{vi}$ = vapor injection gas mass flow (kg/s)
v_s = specific volume of gas entering compressor suction inlet (m^3/kg)
v_{vi} = specific volume of gas entering compressor vapor injection inlet (m^3/kg)
DTI= discharge temperature index (K/watt)

Figure 4.7 and Figure 4.8 show the weighted average inlet pressure plotted versus the discharge temperature index (DTI) regarding without and with vapor injection.

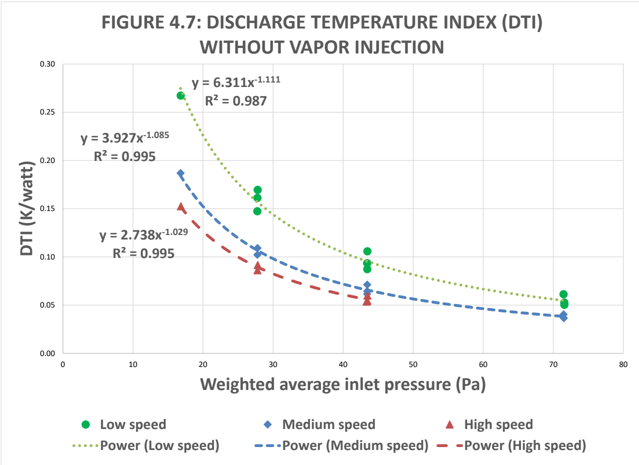


Figure 4.7: Discharge temperature index without vapor injection

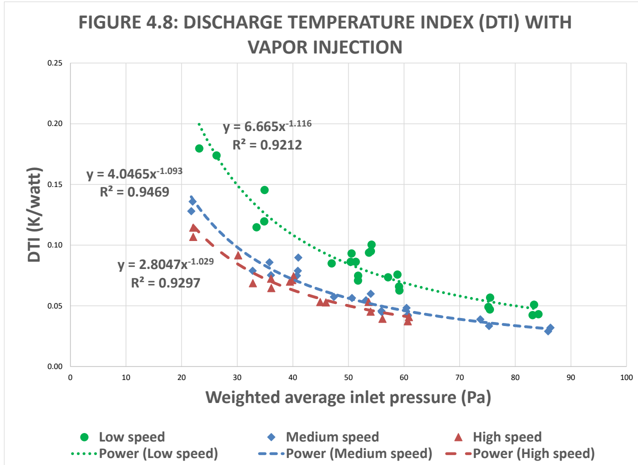


Figure 4.8: Discharge temperature index with vapor injection

4.5 Compressor Heat and Power Ratio (CHPR)

CHPR is proposed to include all measured parameters that are shown in Table 6.1.

Compressor Heat and Power Ratio (CHPR):

$$CHPR = \frac{(\dot{m}_s+\dot{m}_{vi})\cdot h_d-(\dot{m}_s\cdot h_s+\dot{m}_{vi}\cdot h_{vi})}{W} \tag{7}$$

Where:

h_d = discharge enthalpy (Btu/lb)
$\dot{m}_s$ = suction mass flow (lb/hr)
h_s = suction enthalpy (Btu/lb)
$\dot{m}_{vi}$ = vapor injection mass flow (lb/hr)
h_{vi} = vapor injection enthalpy (Btu/lb)
W = compressor power (Btu/hr)
$CHPR$ = compressor heat and power ratio (dimensionless)

The background of CHPR is to have a heat and power ratio that also indirectly gives an energy balance. The numerator describes the heat state change of the refrigerant by defining the outlet or discharge heat state of the refrigerant minus the heat state of the inlets of the refrigerants (suction heat state and vapor injection heat state of the refrigerant). The denominator is the measured power of the compressor. Without heat loss/gain from the compressor shell and no oil carry over, CHPR should be one.

Figures 4.9 and 4.10 show the weighted average pressure ratio versus the compressor heat and power ratio (CHPR) regarding without and with vapor injection. If it is assumed that the heat loss/gain is somewhat proportional to the weighted average pressure ratio, straight lines are expected when plotting CHPR versus weighted average pressure ratio. However, the R squares are relatively low because CHPR is using all test parameters shown in Table 6.1 and all CHPR values are close to one. However, CHPR is over 1 at low weighted average pressure ratios because the heat gain to the shell is noticed from the 95 °F compressor compartment temperature.

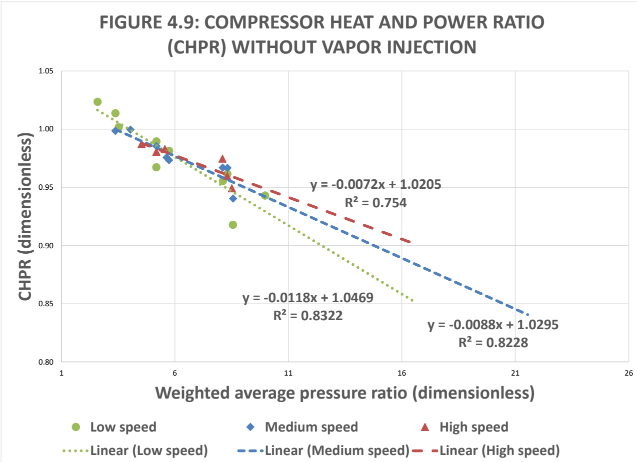


Figure 4.9: Compressor heat and power ratio without vapor injection

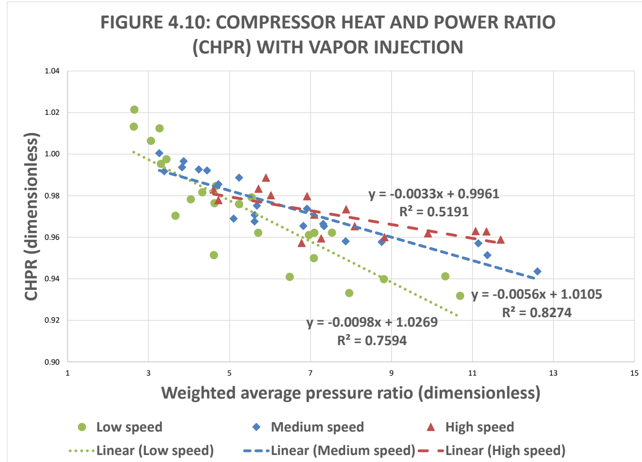


Figure 4.10: Compressor heat and power ratio with vapor injection

5. EVALUATION OF TEST POINTS USING PROPOSED GRAPHING METHODS

Once test data are generated into graphs in the Chapter 4, the graphs and data points need to evaluate the closeness to graphed equations. If there is an outlier at a single test point, the actual measured data need to be evaluated for stability, skewness, or other irregularities. Stability is not always acceptable at very low flow rates. Therefore, it is necessary to remove irregular measurement data before calculating the average value if needed. The outlier test point needs to be retested at a different PID control setting or at a different set point. By comparing the same test point in the different graphing methods, the questionable test data could be noticed if the same test point stands out in different graphing methods. After removing the questionable test point, the graphs need to be modified accordingly.

6. CORRELATION BETWEEN TEST PARAMETERS AND GRAPHING METHODS

Table 6.1 shows the correlation between type of test parameters and type of graphing methods. In traditional compressor performance evaluation, the discharge temperature is not included for basic compressor refrigeration capacity and compressor power. The measurement of compressor discharge temperature has been included as reference information only. Due to this situation limited attention has been directed towards the discharge temperature regarding probe accuracy and calibration routines. However, the discharge temperature is very important when using the compressor model in a refrigeration system model. Also, it is important when fitting a physical compressor model based on the test data as shown in the previous study (equation 13, Sjoholm *et al*., 2021). With newer low GWP refrigerants, where the heat of compression is higher, mainly due to some R-32 in the refrigerant blend, the discharge temperature is even more important to measure and predict (figure 4, Sjoholm *et al*., 2014). Studies have also been performed while making the discharge temperature with no super heat to see if isothermal compression is realistic as shown in the previous study (figure 7, Sjoholm, 1986c). The discharge temperature is needed for both the Discharge Temperature Index (DTI) and the Compressor Heat and Power Ratio (CHPR) in the Table 6.1.

Table 6.1 Correlation between test parameters and graphing methods

Measured Parameters / Graphing Methods	P_d	P_s	P_{vi}	T_s	T_{vi}	T_d	$\dot{m}_s$	$\dot{m}_{vi}$	W	rpm or Hz
Figure 4.1 Suction mass flow without vapor injection	x	x		x			x			x
Figure 4.2 Suction mass flow with vapor injection	x	x	x	x	x		x	x		x
Figure 4.3 Vapor injection mass flow	x	x	x	x	x		x	x		x
Figure 4.4 Relative vapor injection mass flow	x	x	x	x	x		x	x		x
Figure 4.5 Compressor power number (CPN), without vapor injection	x	x		x			x		x	x
Figure 4.6 Compressor power number (CPN), with vapor injection	x	x	x	x	x		x	x	x	x
Figure 4.7 Discharge temperature index (DTI), without vapor injection	x	x		x		x	x			x
Figure 4.8 Discharge temperature index (DTI), with vapor injection	x	x	x	x	x	x	x	x		x
Figure 4.9 Compressor heat and power ratio (CHPR), without vapor injection	x	x		x		x	x		x	x
Figure 4.10 Compressor heat and power ratio (CHPR), with vapor injection	x	x	x	x	x	x	x	x	x	x

7. CONCLUSION

The graphing of refrigeration compressor test data has been presented. The graphing methods are primarily used for evaluating and filtering of raw test data. This is done before the test data is used for the development of different compressor models. This approach minimizes the risk of overfitting certain parameters and drives a more harmonized approach of all parameters.

The graphing techniques can additionally be used for graphing output data from different types of compressor models.
By comparing the output of different compressor models, including the different graphing methods, it can be decided which parameters of the different models are suitable to represent the compressor.

The graphing methods also can be useful to evaluate different compressor models with different refrigerants.

The importance of measuring and modeling the discharge temperature has been presented and this is of even more important when evaluating newer low GWP refrigerants with high heat of compression.

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