Evaluation of Positive Displacement Compressor Testing Techniques, Variation, and Uncertainty

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ABSTRACT

The design and performance modeling of a compressor relies heavily on thermodynamic and physical models, but many times compressor models are validated and calibrated empirically. Representations of compressor performance using polynomial fits such, as defined by AHRI 540, as well as more complicated mid-fidelity semi-empirical models rely on the method of test defined in ASHRAE standard 23 to produce performance data. Even with an empirical representation of compressor performance, one must understand the sources of error in this process to best understand the impact of design related changes, or how this error may propagate to system level predictions.

This study will compare different compressor test facility types and evaluate the pros and cons of each cycle architecture as well as sources of measurement variation and uncertainty. Compressor performance testing facility cycles generally fall into 3 categories, (fully condensing, partial condensing, and non-condensing) which use similar components and measurement devices, but each have unique characteristics which will be highlighted. Characterization of testing variation and uncertainty will be evaluated considering short-term tests, long term tests, as well as testing on multiple different compressor test facilities. Variation in compressor performance metrics will be evaluated given normal variation in setpoint stability, instrument uncertainty, refrigerant composition, and other common sources of compressor testing variation.

1. INTRODUCTION

This paper is covering compressors, where the compressor is a refrigerating compressor, which means that the compressor is compressing some type of refrigerant. The following examples fit within this category, air conditioning compressors (commercial and industrial, automotive and bus AC), heat pump compressors, household refrigeration compressors, commercial refrigeration compressors, transport refrigeration compressors, industrial refrigeration compressors and cryogenic compressors. Typical compressor product applications include both heating and cooling for comfort, process, or the conditioned cold chain. The typical situation is that the product is to be chilled or maintained to a lower temperature. Typical cooling temperature groups include box/zone temperatures of comfort cooling about 70 F (21.1 C), fresh about 35 F (1.7 C), frozen about 0 F (-17.8 C), deep frozen about -20 F (-28.9 C). There are also cryogenics, that achieve cooling temperatures of a few degrees above zero Kelvin (-459.67 F, -273.15 C). The type of positive displacement compressors includes mainly reciprocating compressors, screw compressors (twin screw and single screw with gate rotors) and scroll compressors as well as different rotary compressors. Compressors can be driven through an open shaft from either an external electric motor or engine, or directly driven from an internal motor. The electric motor can also be integrated into the compressor and in this case, we talk about semi-hermetic compressors (can be taken apart with the use of mechanical joints) or fully hermetic compressors (typically a welded shell around the compressor and the electric motor). The compressors can be divided by refrigeration capacity. The compressors can also be applied as single stage or in multiple stages. Besides the compressor application, the temperatures, the refrigeration capacity, and compressor input drive, as well as the following attributes play a role in compressor test stand selection: displacement, instruments for measuring mass flow, power consumption measurements, speed measurements, torque measurements, heat measurements, pressure, and temperature measurements. Controlling at steady state plays a role as well. For fixed volume ratio compressors, mainly screw and scroll compressors, the built-in volume ratio is important. For some parts of an application window, vapor and/or liquid injection or multiple stages may have to be considered.

2. COMMON COMPRESSOR TEST CYCLES

Several different types for compressor test facilities exist and use different cycles to accomplish the main objective of conditioning the inlet and outlet states of the compression process. Testing of a compressor using a dedicated compressor test facility allows compressor operating conditions to be controlled independently of the complete system they are intended to be used in. This enables the ability to decouple and characterize compressor performance independently. Although many combinations of cycles can be used, three major types are common and these include: full condensing, partial condensing, and non-condensing.

2.1 Full Condensing

Full condensing facilities are traditionally known as compressor calorimeters and use a traditional reverse Rankine cycle where gas is compressed, condensed, expanded, and evaporated before once again entering the compressor. Before the widespread use of high precision Coriolis flow meters, these types of compressor test facilities use a secondary refrigerant calorimeter to measure the energy input into the evaporator to calculate mass flow rate. In modern full condensing facilities use of a Coriolis flow meter can confirm evaporator flow rates.

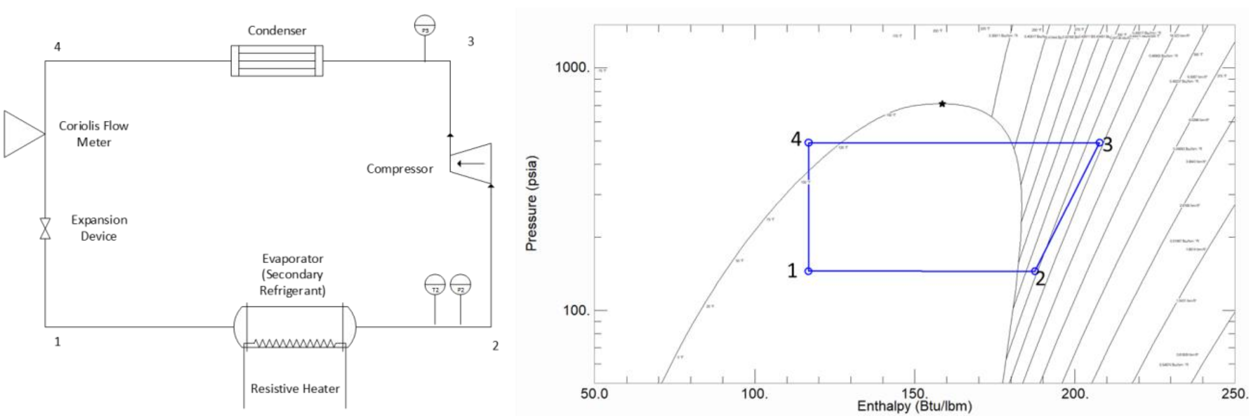

Figure 1: Simplified example of a full condensing cycle compressor test facility

Full condensing compressor test facilities, shown in Figure 1 have an ability to closely resemble a traditional vapor compression cycle. Because of this, full condensing calorimeters may be able to aid in testing and the development of cycle enhancements such as economizers, liquid injection, and potential ejectors. System sizes tend to be larger and charge sizes higher than non-full-condensing facilities, and for this reason system reaction times to setpoint changes tend to be slow. This can both be an advantage or disadvantage since this characteristic makes the system very stable but may lack the speed and productivity to test a variety of operating conditions. Accuracies of full condensing cycles can be quite good when using a Coriolis flow meters for the mass flow measurement. Flow meter measurements shall be within ±1% of measured values if flow meters are used, and most Coriolis flow meters have accuracies significantly below 1%. If a refrigerant flowmeter is not used and a secondary refrigerant calorimeter is used instead, ASHRAE Standard 23 requires a confirming flow measurement to measure within ±3%. Energy consumption of fully condensing facilities is also high because of the need to remove heat from the condenser of the entire flow stream.

2.2 Partial Condensing

Sometimes referred to as compressor gas cycle stands, partial condensing compressor test facilities utilize a hot-gas bypass circuit in order to bypass the condenser with a portion of the flow stream. Since only a portion of the flow is being condensed, a partial condensing test facility uses less energy to set suction and discharge operating conditions. Having liquid and hot gas lines in the facility also allow for expanded capability if vapor or liquid injection are needed for auxiliary injection.

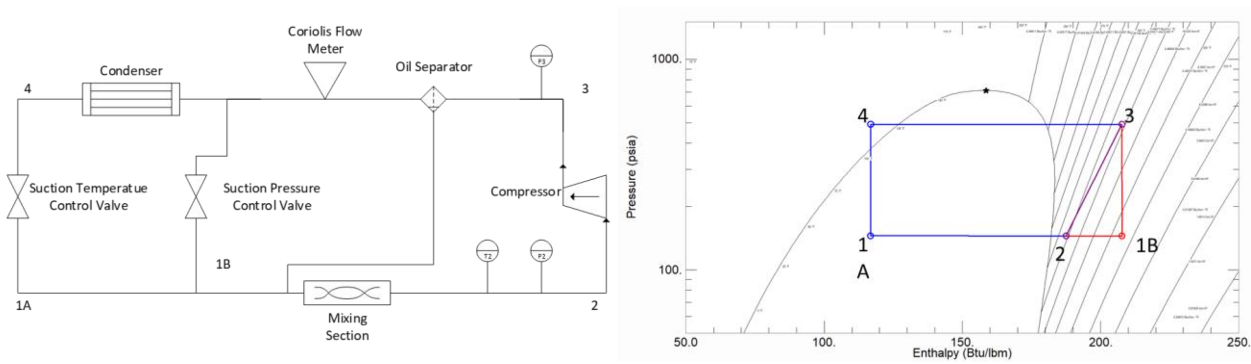

Figure 2: Simplified example of a partial condensing cycle compressor test facility

Shown in Figure 2, the flow is split after state point “3” and a portion of the flow is condensed between 3-4, and the other portion undergoes a pressure drop between 3-1B. This high temperature, high pressure vapor passes through a pressure control valve (3-1B) to reduce pressure from the discharge pressure setpoint to the desired suction pressure. Enthalpy is removed from the flow stream that is allowed to pass through the condenser (3-4) thus creating a subcooled liquid. This liquid then passes through a temperature control valve (4-1A) which has the ability to drop the pressure and expand the liquid refrigerant to a vapor state. The suction temperature control valve acts like an electronic expansion valve to set and maintain superheat entering the compressor. The low temperature, low pressure flow stream and the low-pressure/high temperature/hot gas bypass flow stream are re-introduced in the mixing section before being returned to state point 2. Mixing sections can either be a section or pipe, a series of bends, or devices such as static mixers used to create turbulence and adequately mix the flow streams. Generally, suction temperature and pressure are measured close to the compressor inlet even though their respective control valves are upstream of the mixing section.

2.3 Non-condensing

Non-condensing test stands perhaps offer the least flexibility as far as the three types discussed because there is no source for liquid in the system compared to Figure 1. These types of facilities do however have some advantages (Dirlea, 1996). The only energy load to be removed from the cycle is accomplished with a gas cooler and is only the enthalpy difference from compressor suction to discharge. Charge sizes are also reduced since components are fewer and overall system volume is less. Reduced charge sizes may be desirable when testing flammable refrigerants.

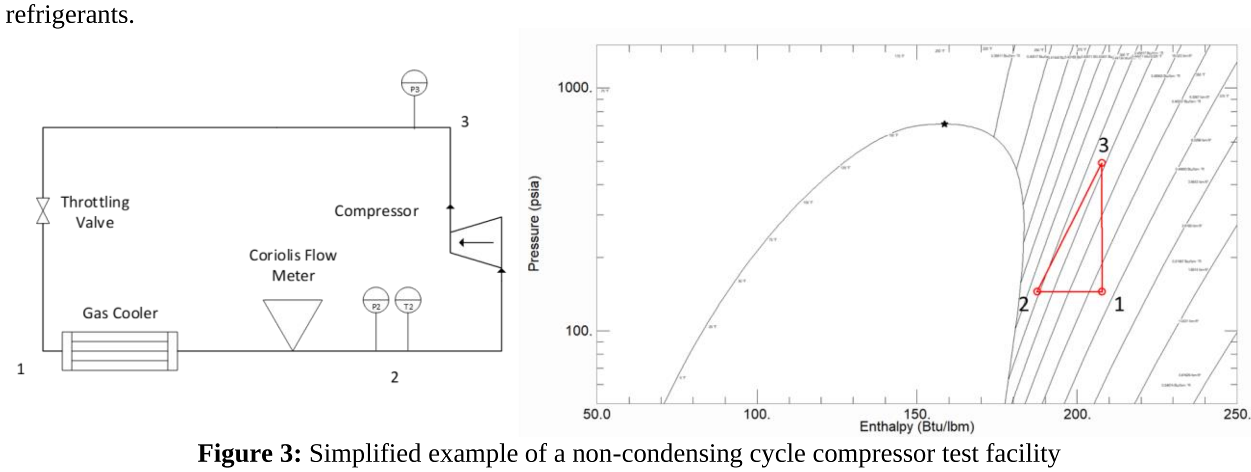

Figure 3: Simplified example of a non-condensing cycle compressor test facility

Figure 3 outlines an example of a non-condensing type of facility where vapor is compressed from 2-3, the pressure is reduced through the use of a throttling valve from 3-1, then the vapor is then cooled back to suction (1-2).

3. COMPRESSOR TESTING VARIATION

There are numerous sources of uncertainty while testing refrigerating compressors. Some examples of uncertainty include accuracy of instrumentation, deviation from desired setpoint, environmental conditions, and refrigerant composition. When performing a compressor performance test, certain inputs are controlled to characterize the rating condition. These inputs are voltage (for hermetic compressors), input frequency, suction dew point temperature, discharge dew point temperature, suction superheat, and other ambient environmental conditions. These inputs include both measurement inaccuracies as well as instability in control strategy (example PID loop). Other factors effecting test and measurement variation may include calibration coefficient differences or instrument drift, refrigerant composition, test setup differences from removing and re-installing the compressor, rating point approach strategy (such as approaching the suction or discharge pressure/temperature from a higher or lower pressure/temperature), and individual human factors. Another important part of compressor performance variation is the manufacturing and break-in effects (AHRI,2017). In order to limit this study to compressor test and measurement variation, the testing was limited to a single compressor specimen designed and tested with R-410A. Since there are many factors impacting compressor test variation, two different methods will be used to capture testing variation which can be compared to analytical studies. These include long-term variation, short term variation, and analytical measurement uncertainty which can be outlined by Monte Carlo analysis (Aute, 2016).

3.1 Long-Term Testing Variation

A sample of test data from a 15 ton commercially available scroll compressor that was acquired over the course of 7 years was used to show long-term compressor test and measurement variation. This data came from four different partial condensing compressor testing facilities. Instruments for setpoint and measurements were recalibrated up to 6 times throughout this timeline, compressors were removed and reinstalled, refrigerant composition of R-410A was allowed to drift within AHRI 700 (AHRI, 2019) specifications and spot checked regularly, test facility maintenance was regularly performed, and multiple test technicians operated the facilities. Given the duration and sources of uncertainty, variation was expected to be the highest given these circumstances.

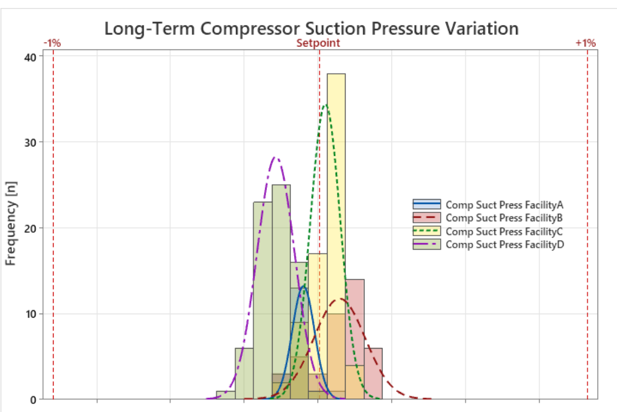

Figure 4: Long-Term Compressor Suction Pressure Variation

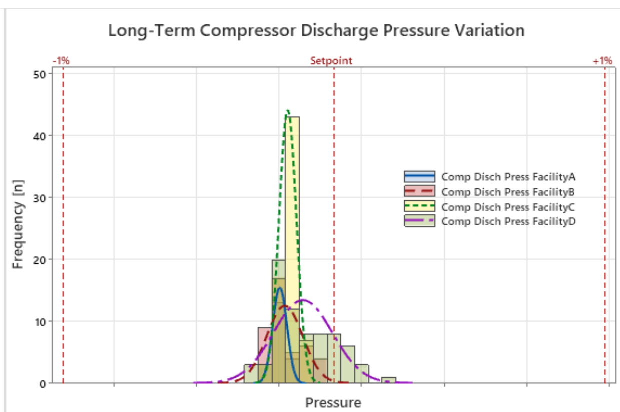

Figure 5: Long-Term Compressor Discharge Pressure Variation

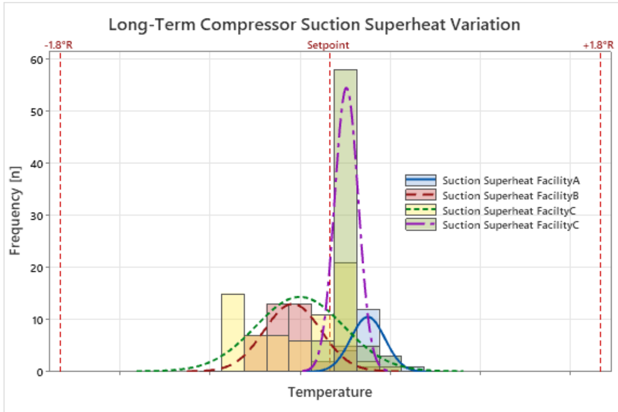

Figure 6: Long-Term Compressor Suction Superheat Variation

Given the long-term compressor testing criteria, variation of suction pressure for each unique testing facility can be noted in Figure 4. Mean values of each data set vary slightly from the setpoint pressure of 144.81 psia, but all data points lie well within the ±1% of desired suction pressure specified in section 7.5.6 of ASHRAE Standard 23 (ASHRAE, 2022). Differences in standard deviation are noted between each unique facility and are likely due to differences in instrumentation and control strategies. The distribution of discharge pressure for the tests performed on each facility is shown in Figure 5. Mean values and distributions all lie well within the ±1% of desired discharge pressure specified in section 7.5.9 of ASHRAE Standard 23. Suction superheat is to be controlled to ±1.8°R of the specified setpoint as specified by section 7.5.7 of ASHRAE Standard 23. Figure 6 shows the distribution of suction superheat given each uniqiue test facility.

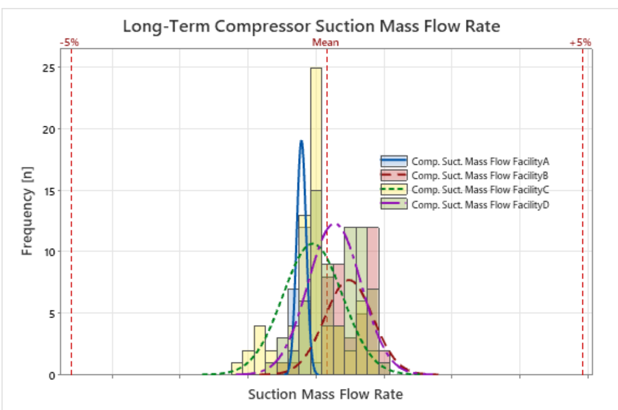

Figure 7: Long-Term Compressor Suction Mass Flow Rate Variation

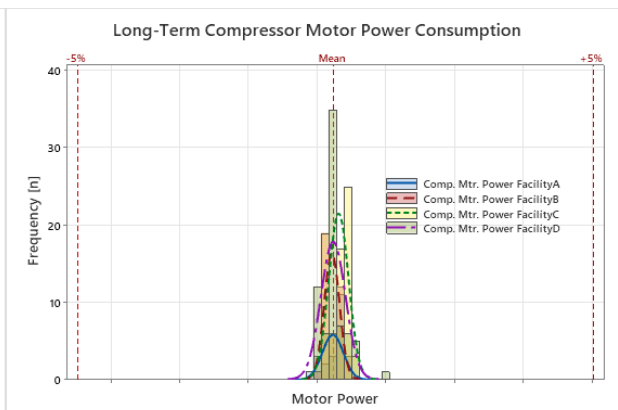

Figure 8: Long-Term Compressor Motor Power Variation

Given the test article is the very same in all cases, the resulting suction mass flow rate distribution shown in Figure 7 is impacted by all factors outlined in the long-term testing variation. Setpoint control variation, intrumentation calibrations, measurement uncertainty, and refrigerant blend variations all contribute to differences in mass flow rates. The unit under test underwent the manufacturer reccommended break in procedure prior to testing, and compressor run time was limited to shown test data. Changes to compressor performance as a function of run time in this study are considered negligible. The reference rating condition used for this test corresponds to the 45°F/130°F condition shown in region 3 of AHRI Standard 540 (AHRI, 2020). Rating uncertainty limits in this region allow for 95% of published rating for mass flow rate, and 105% of published rating for power input. These uncertainty limits are represented by the limits shown in Figure 7 and Figure 8.

$$COV = \sigma/\mu \tag{1}$$

Table 1: Long-Term Testing Coefficient of Variance

	Suction Pressure [COV]	Discharge Pressure [COV]	Suction Superheat [COV]	Mass Flow Rate [COV]	Motor Power [COV]
Test Facility A	0.04%	0.03%	0.56%	0.09%	0.20%
Test Facility B	0.10%	0.07%	0.96%	0.47%	0.15%
Test Facility C	0.06%	0.03%	1.48%	0.58%	0.19%
Test Facility D	0.07%	0.11%	0.39%	0.52%	0.24%

The standard deviation and mean from each data set are used to calculate COV (coefficient of variance) shown in Equation 1. COV can be used as a metric to assess the precision of long-term test variation of each test facility. This is calculated for both setpoint variables as well as resultant performance characteristics. When compared to maximum COV values of 3.5% and 1.7% of mass flow rate and power computed in a Monte Carlo simulation (Aute, 2016), empirically obtained values in Table 1 show significant reduction. One possible explanation is that actual measurment device accuracies outpreformed required measurement accuracies defined by ASHRAE Standard 23.

3.2 Short-Term Testing Variation

The same compressor test article used in the long-term variation study was tested on a single test facility over the course of a 1-week timespan. The compressor was not removed from the testing facility but was shut off between each test allowing pressures and temperatures to equalize. This was repeated 10 times targeting the same operating condition.

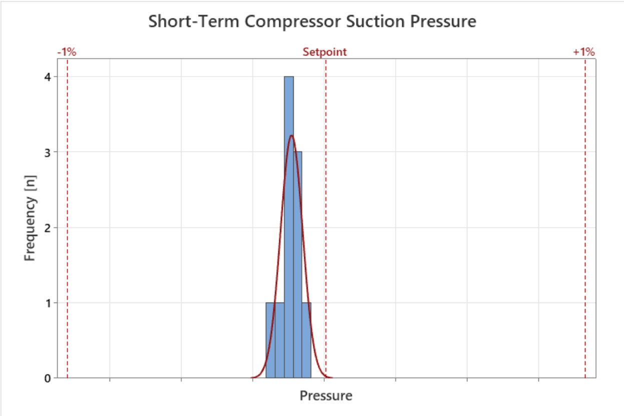

Figure 9: Short-Term Compressor Suction Pressure Variation

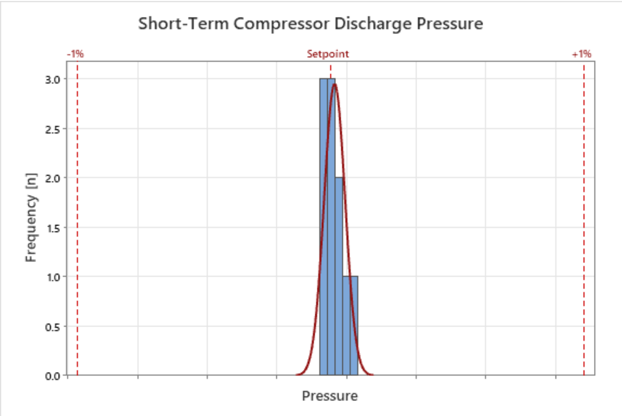

Figure 10: Short-Term Compressor Discharge Pressure Variation

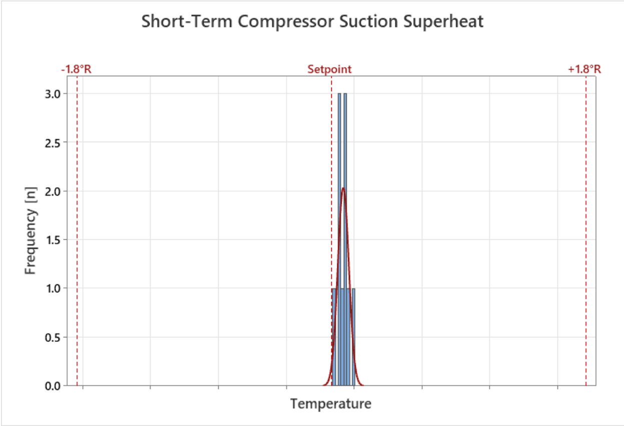

Figure 11: Short-Term Compressor Suction Superheat Variation

Similar to the long term variation results, short term variation of a single facility (Facility D) showed very small variation compared to the setpoint target and limits. Setpoint COV values in short term testing were roughly half of the COV values in long lterm testing. This represents a significant reduction in variance.

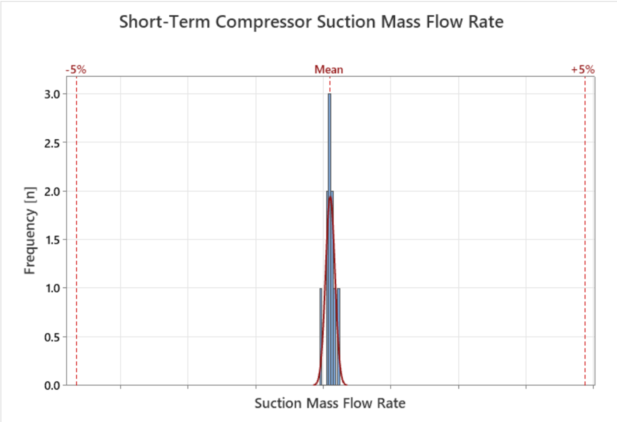

Figure 12: Short-Term Compressor Suction Mass Flow Rate Variation

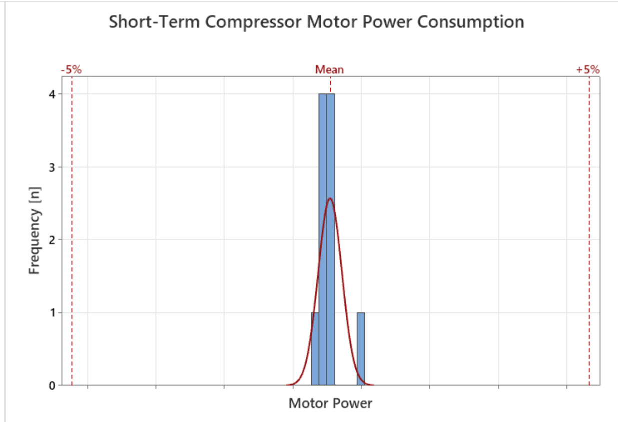

Figure 13: Short-Term Compressor Motor Power Variation

Table 1: Long-Term Testing Coefficient of Variance

	Suction Pressure [COV]	Discharge Pressure [COV]	Suction Superheat [COV]	Mass Flow Rate [COV]	Motor Power [COV]
Test Facility D	0.04%	0.04%	0.20%	0.09%	0.23%

Resultant COV values were much improved when considering only short-term variation with perhaps the largest improvement being the variation reduction in the compressor suction mass flow rate which one of the key performance metrics of refrigerating compressors.

3. CONCLUSIONS

Depending on compressor type, refrigerant, and testing needs, different types of compressor testing facilities have different pros and cons. Combinations of the three cycles could also be combined to create hybrid approaches to compressor testing and various additions to the cycle could be added to simulate an economized cycle or liquid injection. The flexibility designing dedicated compressor test stands also allows for multiple circuits to be added to increase compressor testing range or turndown.

It is important that compressors are tested in accordance with industry standards for the purpose of comparing different manufactured compressor data. However, in order to achieve more precise compressor performance measurement, significant improvement can be achieved by limiting variation factors. Using definitions outlined in this study, short-term test variation shows a COV can be improved from 0.52% to 0.09% when considering compressor mass flow rate. Compressor motor power showed a slight improvement, but to much less of an extent than compressor mass flow rate.

Although all test data in this study complied with industry test and rating standards, in order for a compressor design engineer to understand the impacts of design changes to compressor performance, a baseline of the test facility variation must be understood. If small changes to compressor performance lie within testing uncertainty, it is extremely difficult to tell if new design ideas have meaningful impact. Furthermore, the amount that testing variation can be reduced allows for better understanding of factors related to compressor design and manufacturing differences.

NOMENCLATURE

COV	Coefficient of Variance	(%)
σ	Standard Deviation	
μ	Mean	
PID	Proportional, Integral, Derivative Controller	

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