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Performance Testing of a Dual Rotary Compressor with a Variable Speed PSC Motor

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Abstract. A new system solution has been developed and tested by the authors and has been shown to improve the Seasonal Energy Efficiency Rating (SEER) of fixed-speed condenser split-system residential heat pumps. This is achieved through the use of a low power electronic drive that allows components with PSC motors (such as fans and compressors) to operate at multiple low speeds or variable speeds when partial capacity is required. At full capacity, the electronic drive is disabled and the PSC motors are operated at fixed speeds dictated by the frequency of the power source. Previous research by the authors has shown that fixed-speed heat pumps driven by rotary compressors with PSC motors achieve significantly greater improvements in performance and in overall SEER when operated with the electronic drive than similar heat pumps driven by scroll compressors with PSC motors. The current research measures the compressor and drive performance through calorimeter testing. This paper presents the results of testing a dual rotary PSC compressor in variable speed mode in association with the new electronic drive. The purpose of this testing is to understand how an off-the-shelf compressor with a PSC motor intended for fixed-speed operation performs under variable speed operation. Preliminary results of testing a PSC scroll compressor in variable speed mode with the electronic drive are also presented for comparison with the rotary compressor. Future research will focus on testing other common HVAC compressor technologies and employing mechanistic compressor models in order to better understand the parameters which affect the performance of PSC driven compressors under variable speed operation.

1. Introduction

The market for residential air-source heat pump and air conditioning equipment in the United States is dominated primarily by fixed-speed single stage systems with low seasonal energy efficiency ratios (SEER). These systems typically utilize permanent split capacitor (PSC) motors to operate the compressor, condenser fan, and indoor blower. High efficiency systems on the other hand utilize brushless DC (BLDC) or electrically commutated (ECM) motors to operate variable speed compressors and fans which provide the ability to modulate system capacity to match the cooling or heating load. These high efficiency systems have relatively low market penetration due to the ability of manufacturers to meet standard efficiency regulations through lower cost approaches such as increasing the heat exchanger surface area [1]. However, as increased efficiency regulations take effect, these approaches will become less cost effective and manufacturers will need to take new approaches to increase the system efficiency of the traditionally fixed-speed equipment which utilize PSC motors.

A system solution to increase the efficiency of fixed-speed single stage residential air-source heat pump and air conditioning equipment was previously proposed and tested by the authors [2, 3, 4.] The proposed solution, described in detail by Chretien et al. [3, 4] uses a low power electronic drive to enable the variable speed operation of traditionally fixed-speed PSC driven components such as the compressor. SEER improvements of up to 38% have been measured in laboratory experiments when the electronic drive is applied to an off-the-shelf residential split system heat pump [4.] While previous experimental testing of the electronic drive was focused on system level seasonal efficiencies, this paper will focus on compressor performance testing.

2. Experimental Methods

A dual rotary compressor which was previously tested by the authors in an air-source split-system heat pump [4] has now been installed in a calorimeter set up for compressor performance testing. The compressor has a rated capacity of 11.8kW, a displacement of 40.8ml/rev, and is driven by a nominal 60Hz single phase PSC motor. The refrigerant used was R410A. Performance of the compressor in both the heat pump system and the calorimeter set up will be reported. In both experimental set ups, the compressor was tested at full load operation connected directly to the 60Hz single phase utility power and in variable speed mode connected to the electronic drive. Compressor power consumption, mass flow rate, and suction and discharge temperatures and pressures have been measured on both test stands in order to report overall compressor efficiency and volumetric efficiency. A high frequency power analyzer has been used on the calorimeter test stand to capture electronic drive efficiency. When connected to the electronic drive, the motor speed is controlled manually by a controller which delivers a speed signal; this will be referred to as the commanded speed. Because the PSC motor is an asynchronous motor, there will be some difference in the commanded speed and the actual speed of the rotor; this is referred to as slip. The exact rotational speed of the compressor has been determined using a high frequency pressure transducer measuring discharge pulsations, and a fast Fourier transform to convert these pulsations to a frequency.

Testing of a scroll compressor is currently underway. The scroll compressor has a rated capacity of 7.4kW, a displacement of 23.4ml/rev and is driven by a nominal 60Hz single phase PSC motor. This compressor is being tested using the same experimental approach, it was first tested in an air-source split-system heat pump [4], and now testing is taking place in the calorimeter system.

2.1. System Solution Description

A detailed description of the system solution and its benefits has been outlined previously by Chretien et al. [3, 4.] A brief description of the electronic drive operation will be provided here. The electronic drive has been designed to enable a fixed-speed PSC compressor to operate at a range of variable speeds. At nominal full speed operation (3600RPM) the PSC compressor is connected directly to the line power source (60Hz) at which point no power is being passed through the electronic drive. Under part load operating conditions the compressor can be switched to operate on the electronic drive, at which point line power is passed through the electronic drive and to the compressor. A block diagram depicting the separation of line operation circuitry and electronic drive circuitry is shown in Figure 1. The electronic drive engages and enables variable speed operation of the compressor when frequencies of 45 Hz and below are commanded by the controller, as shown in the control scheme in Figure 2. During laboratory testing, the frequency delivered by the drive can be controlled manually in increments of 1% or 0.6 Hz. It is important to note the difference between line frequency, synchronous motor speed, and actual motor speed. The synchronous speed of a motor, n, is the speed at which the magnetic field rotates in RPM. The relationship between line power frequency, f, and the synchronous speed of the motor is

$$n = f * (2/p) * 60$$

where p is the number of poles of the motor. Because the PSC induction motor is an asynchronous machine, the actual speed of the rotor will always lag behind that of the magnetic field.

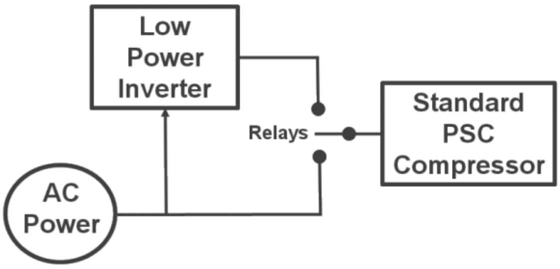

Figure 1. Separation of line circuitry and electronic drive circuitry using a relay. [Chretien et al.]

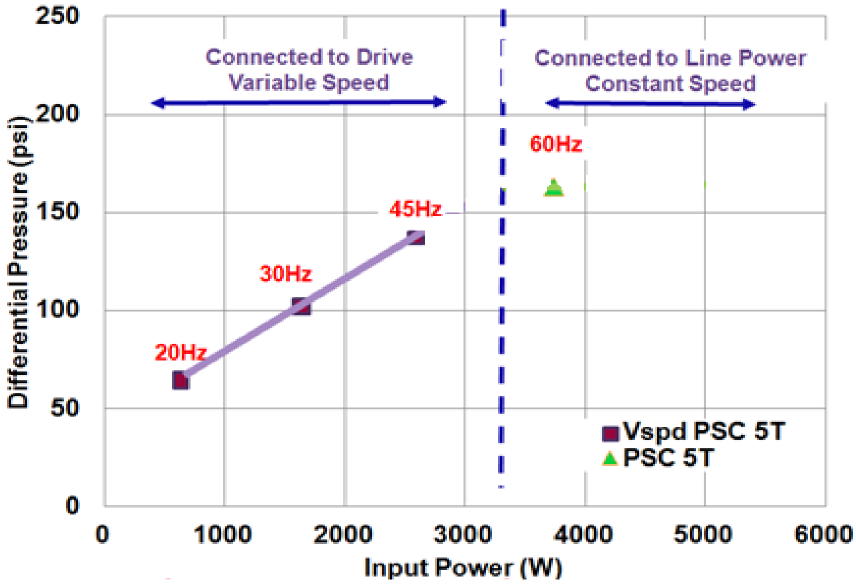

Figure 2. Electronic Drive Control [Chretien et al.]

2.2. *Psychrometric Rooms*

The air-source heat pumps used for testing both the rotary and scroll compressors were installed in a pair of psychrometric rooms with one room simulating indoor conditions and the other simulating outdoor conditions. The heat pumps were installed and tested in cooling mode according to the AHRI 210/240-2008 standard for the rating of unitary air source heat pump equipment. The data gathered from this heat pump system used to calculate compressor performance includes refrigerant temperature and pressure at compressor suction and discharge, refrigerant mass flow rate delivered by the compressor, and electrical power consumption of the compressor. Temperature and pressure were measured at the inlet and outlet of the compressor while mass flow rate was measured in the liquid line of the heat pump. Power consumption of the compressor was measured using a watt transducer.

2.3. *Calorimeter Experimental Setup*

The compressor was tested using a calorimeter to better capture the effects that the electronic drive has on the compressor. The calorimeter allows for increased testing speed, a wider range of operating conditions, and direct control of the refrigerant conditions at the inlet and outlet of the compressor. The compressor is placed inside a temperature-controlled chamber and connected to a vapor-compression cycle (condenser, expansion valve, and evaporator). Suction pressure is regulated through a compressed air powered pneumatic expansion valve, and suction temperature is regulated through the heat input capacity of the evaporator. The evaporator is a tank with a secondary working fluid in a two-phase state.

Electrical heat input into the tank leads to a constant temperature heat input into the refrigerant entering the compressor. The heat capacity input into the evaporator can be modulated by turning on or off the electrical heaters. Discharge pressure is regulated through the condensing temperature. The condenser rejects heat into a water loop where the water temperature can change based on the opening of a bypass valve that rejects circulated water and intakes chilled water. Figure 3 displays a schematic of the calorimeter. In the refrigerant line, temperature and pressure are sampled at various state points in the cycle. The mass flow is measured using a Coriolis Effect mass flow meter located after the condenser. Outside of the refrigerant line air temperature and velocity are measured in the compressor chamber, and power draw from the compressor as well as the electrical heating elements are recorded.

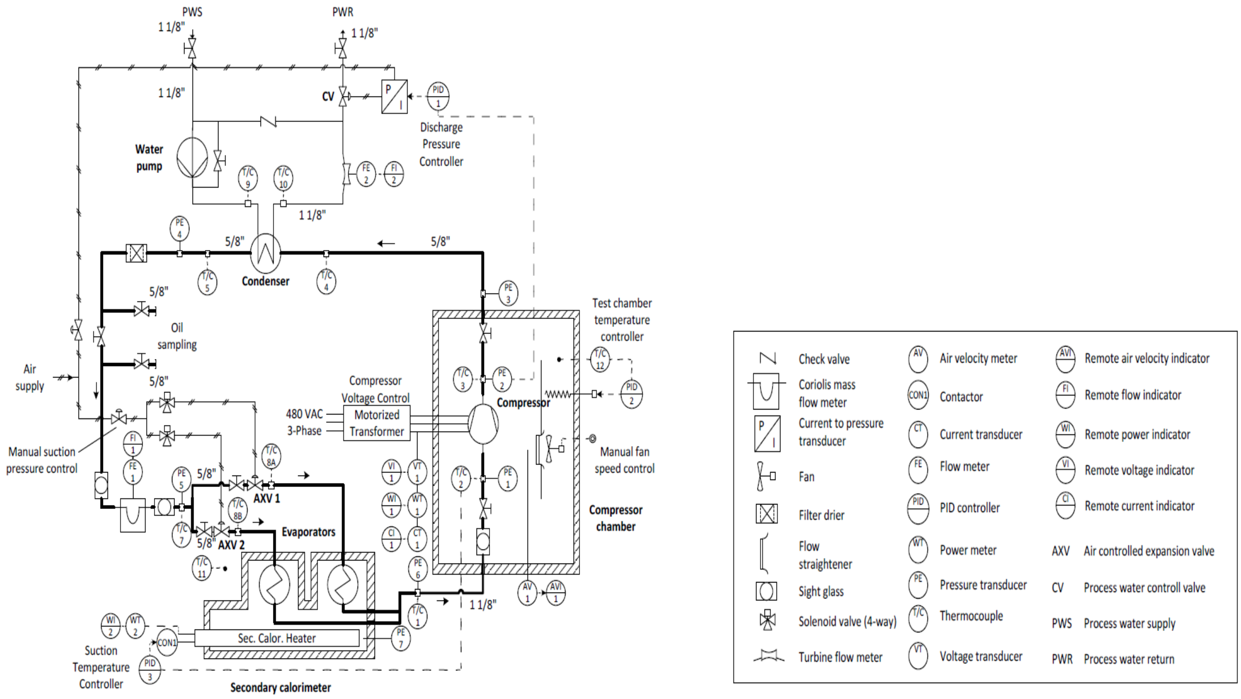

Figure 3: Schematic of the calorimeter [Mösch, T.]

2.4. Discharge Pressure Fluctuation Measurements

A high frequency pressure transducer was placed in the condensing line immediately after the compressor discharge port. This pressure transducer was sampled at a rate of 1000 Hz, and enabled the investigation of pressure fluctuation at the discharge of the compressor. The absolute pressure was sampled, and a fast Fourier transform (FFT) implemented to plot the data in the frequency domain. Pressure fluctuation was then isolated in the frequency domain, and this fluctuation was used to identify the discharge rate of the compressor and therefore the exact rotational speed of the compressor at every operating point. Because a dual rotary compressor discharges twice per rotation, the discharge rate determined will be twice that of the actual rotational speed of the compressor.

The software MATLAB [6] was used to perform the FFT on the high frequency discharge pressure measurement. MATLAB performs the FFT using an algorithm created by Frigo and Johnson [7]. Figure 4 shows the pressure fluctuation data in both the time domain and the frequency domain that is calculated using FFT. In Figure 4, the compressor is operated at full speed from line power having a frequency of 60 Hz. As seen the in the frequency domain, a pressure fluctuation is recorded at approximately 116 Hz, which means while the magnetic field of the motor rotates at 60 Hz, the actual rotational speed is 58 Hz. By identifying the actual discharge rate of the compressor, the volumetric efficiency of the compressor can be calculated more accurately at the different commanded speeds.

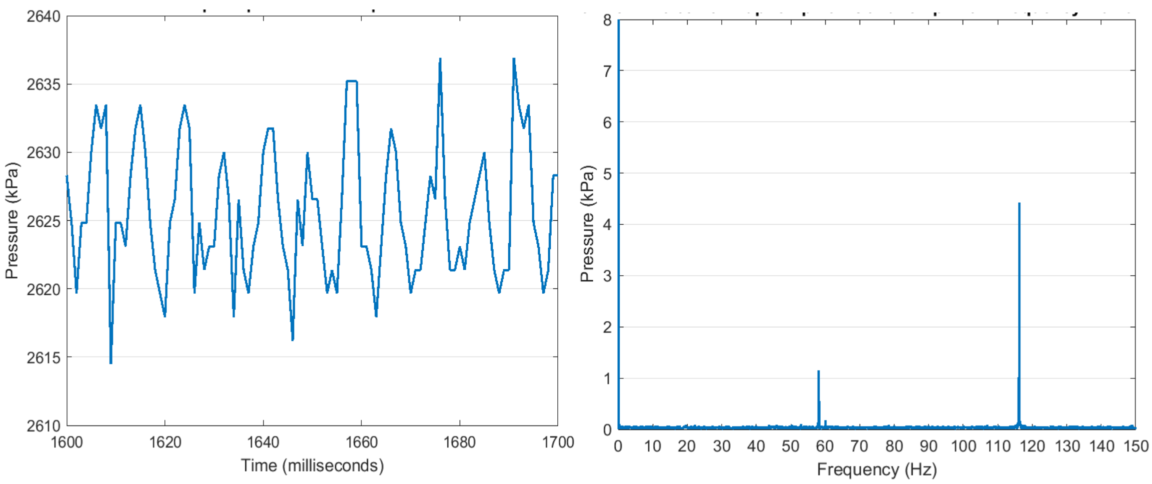

Figure 4: Absolute pressure measurement in the time domain (left), and the pressure amplitudes of the signal in the frequency domain (right).

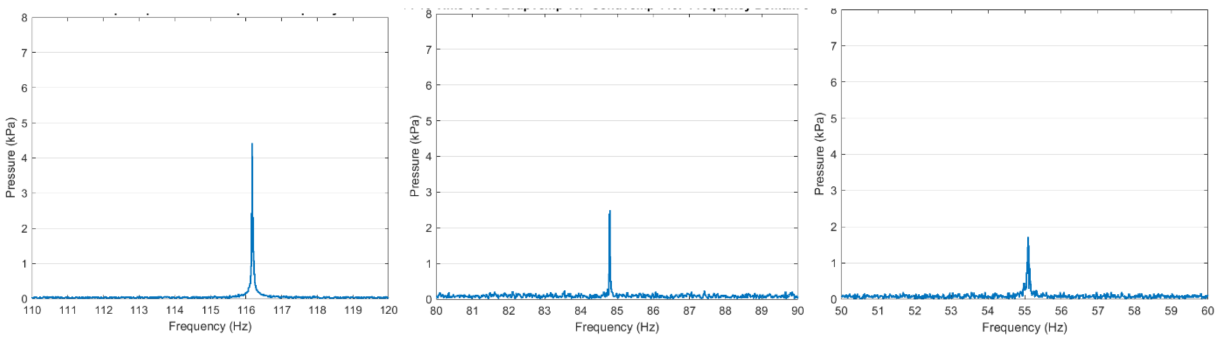

Figure 5: Discharge rate indicated by pressure fluctuation at different speeds. Discharge rate decreases as commanded frequency decreases from 60 Hz (left), 45 Hz (center), and 30 Hz (right).

3. Results

The results of testing the PSC rotary compressor under variable speed operation are presented for the heat pump system in Section 3.1 and for the calorimeter testing in Section 3.2. Preliminary results of testing the PSC scroll compressor in the calorimeter are presented in Section 3.3 and compared against the rotary compressor test results. The overall isentropic efficiency of the compressor is reported from heat pump system testing. Results from the calorimeter testing of the rotary compressor include overall isentropic efficiency, volumetric efficiency, rotational speed, and motor slip.

3.1. Heat Pump Results of Rotary Compressor

Results of testing the PSC rotary compressor in a heat pump system are shown in Figure 6. The pressure ratios in this system ranged from 1.2 to 2.5 with evaporating temperatures ranging from 5 to 10 C. The compressor was operated at its design speed of 3600 RPM at 60 Hz for AHRI tests A_2 and B_2, and was operated using the electronic drive at frequencies from 45 Hz down to 25 Hz for variable speed tests E_v, B_1, and F_1. The results show that even when operating at low speeds, the PSC rotary compressor can achieve similar efficiencies to its design speed of 3600 RPM. A peak in compressor efficiency was found when operating at low speeds at a pressure ratio of approximately 1.4. Below this pressure ratio, the compressor efficiency dropped significantly. One possible explanation for this is increased recompression of refrigerant in the compression chamber as the pressure ratio approaches 1. Lower pressure ratios are typically associated with reduced leakage between high pressure and low pressure chambers of the compressor and therefore higher volumetric efficiencies. However, it has also been observed that as the pressure ratio approaches unity, a larger percentage of the refrigerant mass remains in the compression chamber to be recompressed during the next revolution. A second explanation could be a lack of proper oil circulation in the compressor shell at lower speeds.

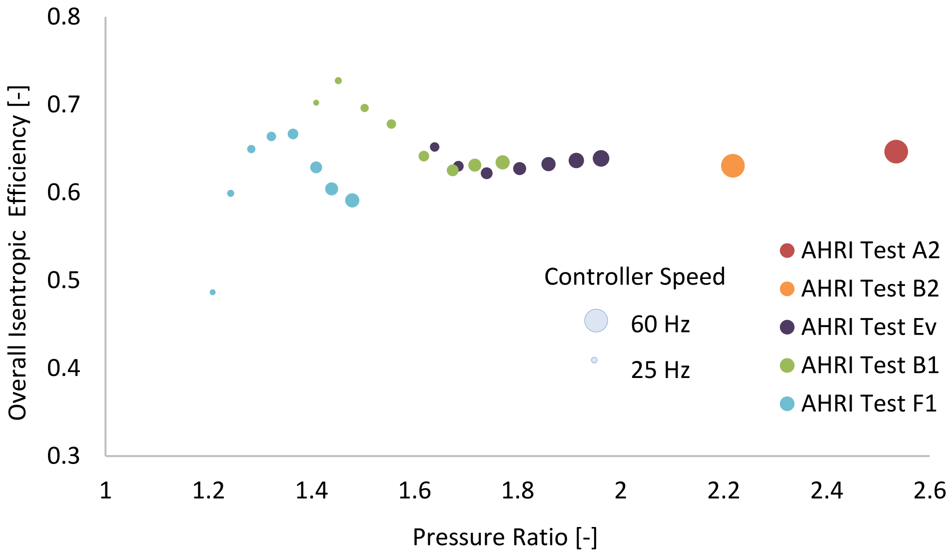

Figure 6: Overall isentropic efficiency at varying operating conditions and compressor rotational frequencies in the heat pump system.

3.2. Calorimeter Results of Rotary Compressor

Results of testing the PSC rotary compressor in the calorimeter are shown in Figures 7 to 9. The pressure ratios tested in the calorimeter were higher than those tested in the heat pump system and ranged from 2.1 to 9.5. Three different speeds were investigated during these tests, the design speed of 60 Hz, and two lower speeds enabled by the electronic drive of 45 Hz and 30 Hz. Both overall isentropic efficiency and volumetric efficiency show similar trends between efficiency and pressure ratio. It can be seen that for similar pressure ratios, the compressor was less efficient at lower speeds. Also, it can be seen that in general, motor slip increases with decreasing speed as shown in Figure 10. Motor slip also increases with increasing pressure ratio for a given compressor speed and evaporating temperature.

3.3. Scroll Results and Comparison to Rotary Performance

The performance of the scroll compressor is compared to that of the rotary compressor for heat pump system testing in Figure 11 and for calorimeter testing in Figure 12. The scroll compressor behaved very differently in heat pump testing than the rotary compressor. While full speed performance was similar, as the speed of the scroll compressor decreased, the overall isentropic efficiency continuously decreased below that of the rotary compressor. The results of calorimeter testing also show much lower efficiencies at low speeds for the scroll compressor compared to the rotary compressor. Another difference between the scroll and rotary compressor is the change in performance over time. The scroll and rotary compressors were subject to the same amount of low speed operation in the heat pump system. When the rotary was retested in the calorimeter, nearly identical performance was measured at similar pressure ratios. But when the scroll was retested, performance degraded by around 8% at full speed operation. Both the drop in efficiency at low speed operation and the degradation in performance over time could be due to insufficient lubrication of the PSC scroll compressor.

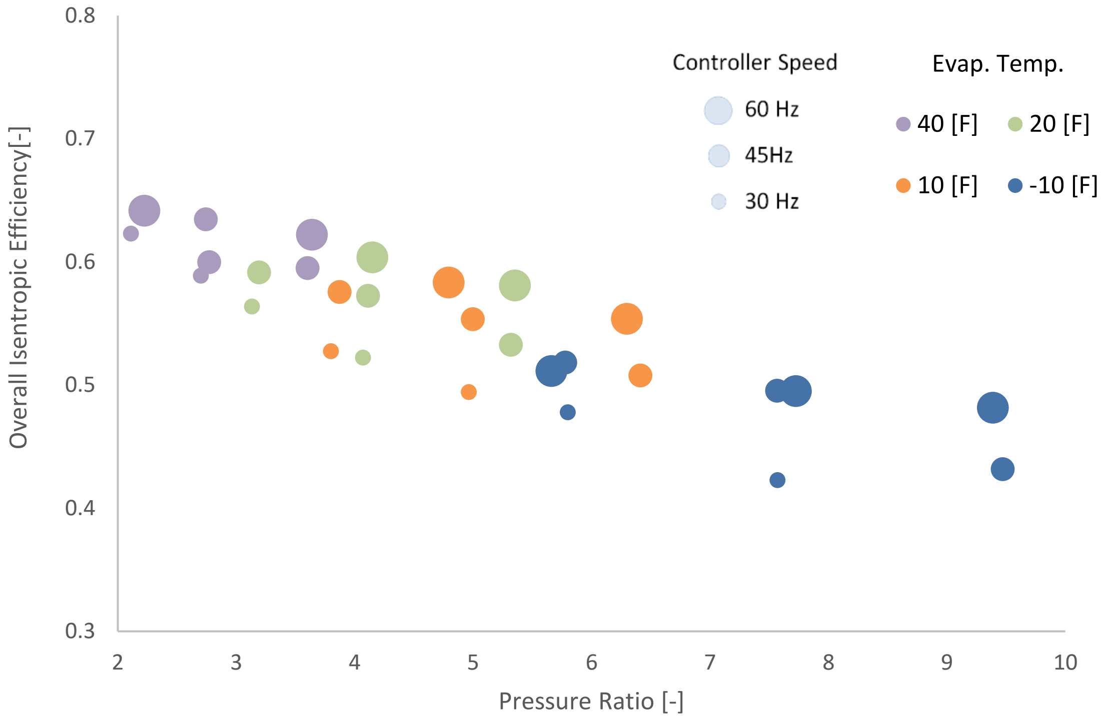

Figure 7: Overall isentropic efficiency at varying operating conditions in the calorimeter system.

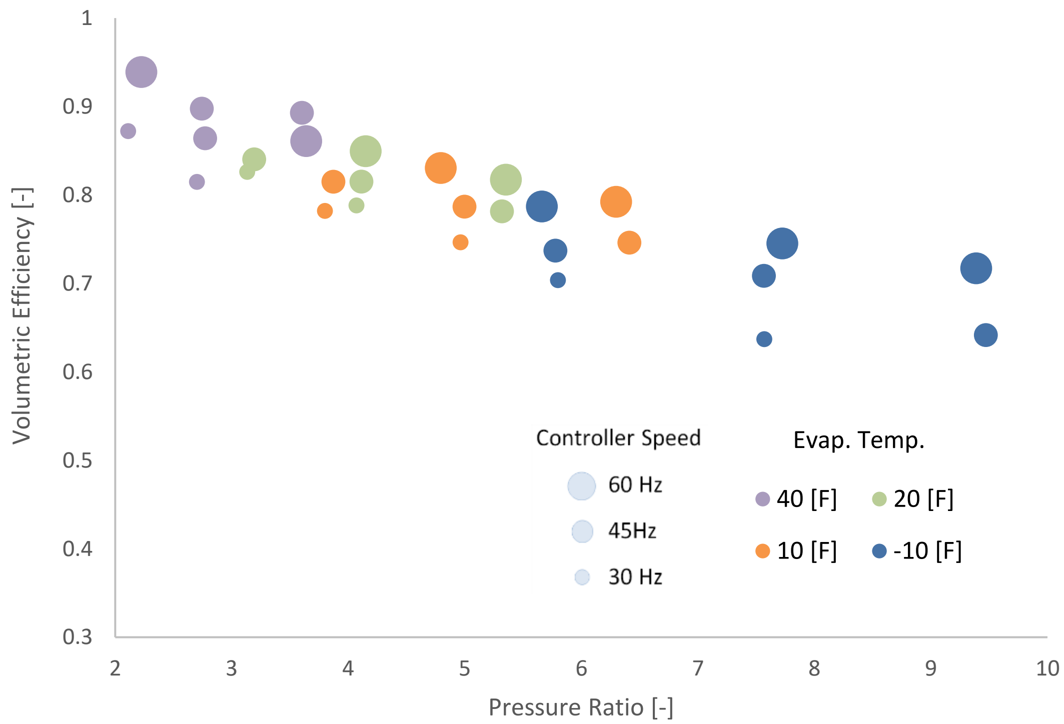

Figure 8: Volumetric efficiency at varying operating conditions in the calorimeter system.

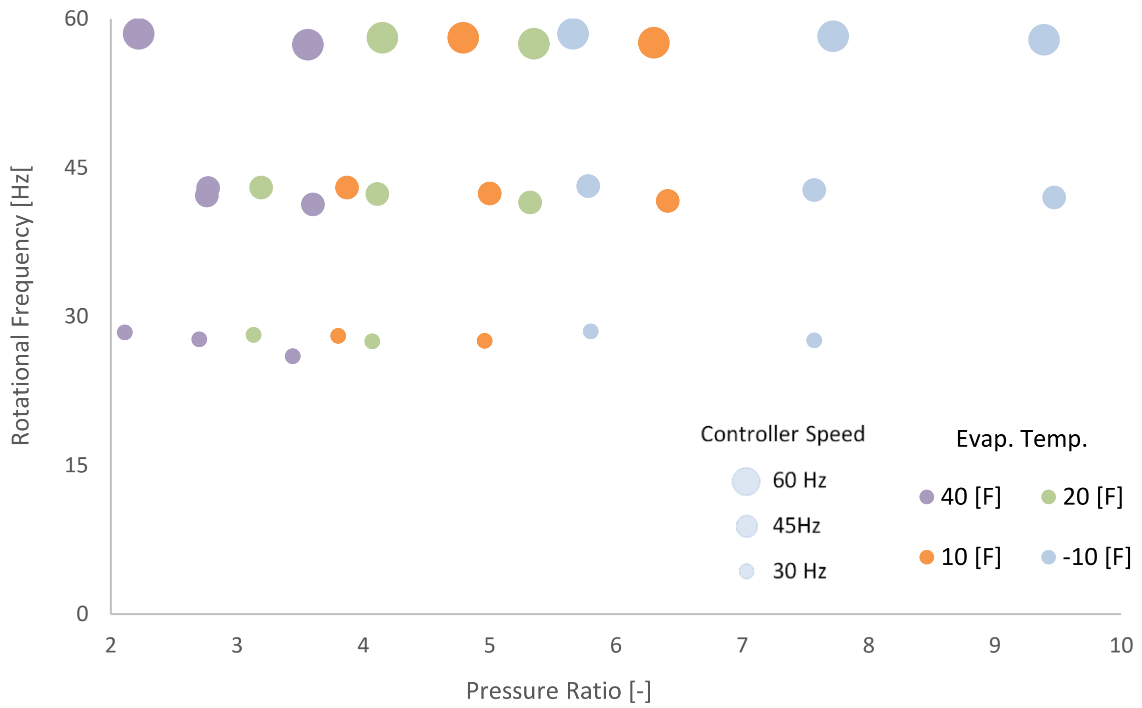

Figure 9: Compressor rotational frequency at varying operating conditions in the calorimeter system.

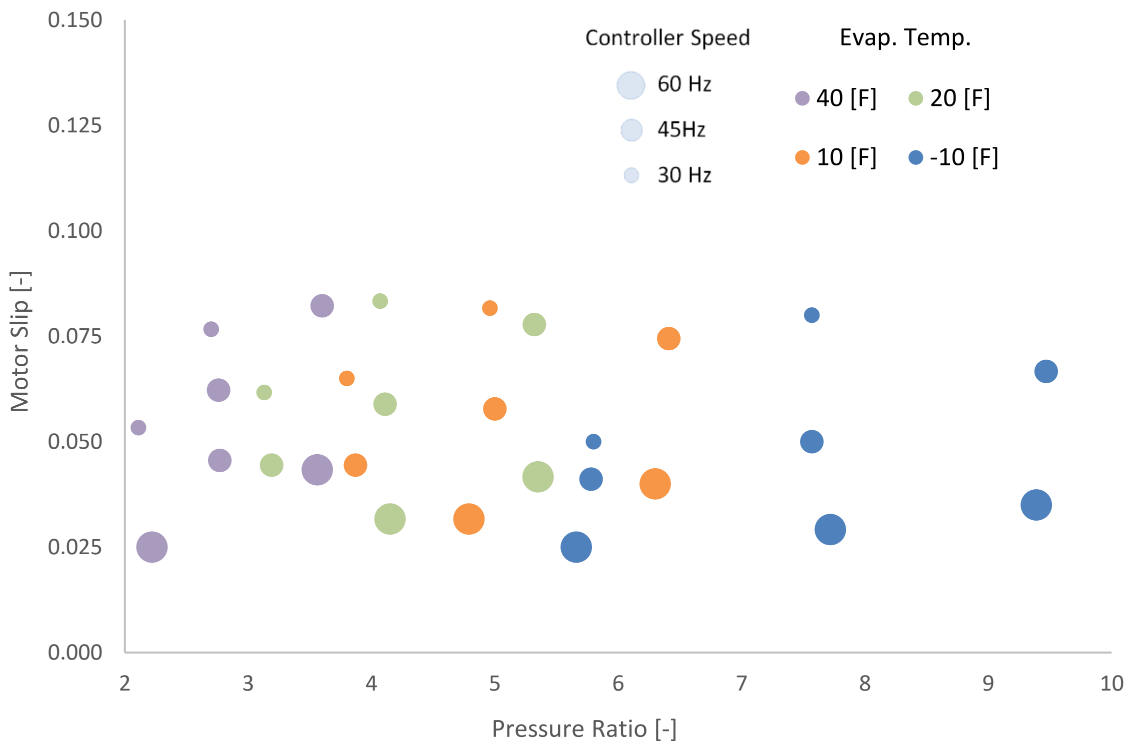

Figure 10: Motor slip at varying operating conditions in the calorimeter system.

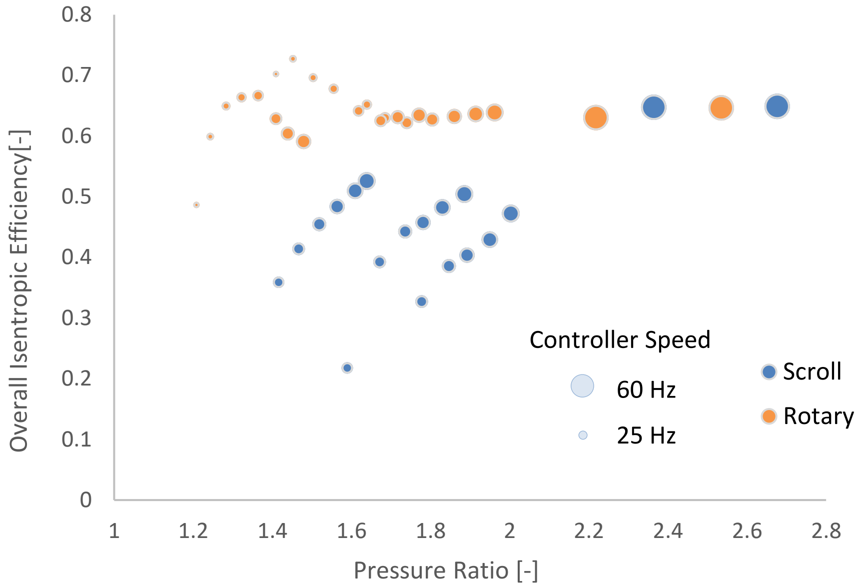

Figure 11: Overall Isentropic Efficiency of a Scroll and Rotary Compressor each tested in an Air-Source Heat Pump.

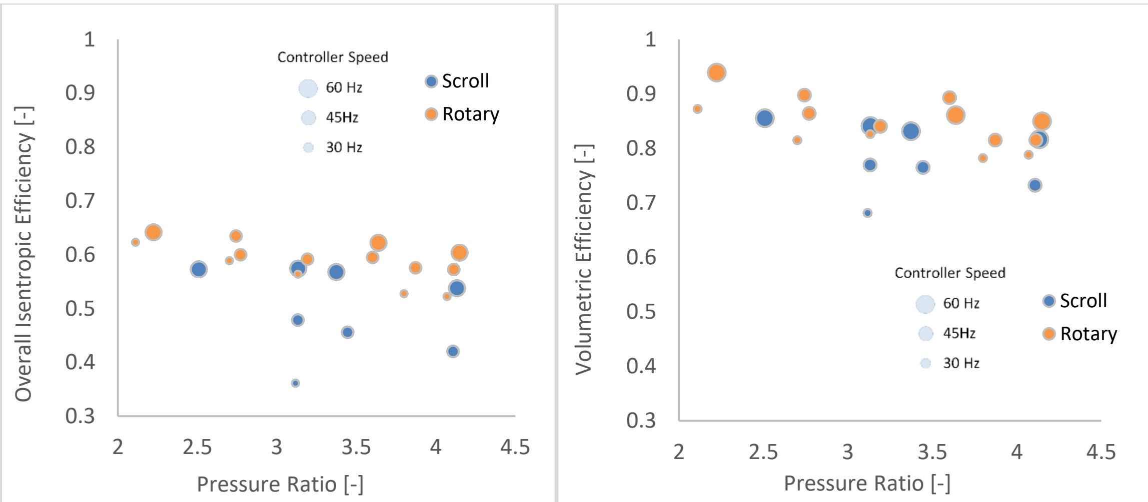

Figure 12: Overall Isentropic Efficiency (left) and Volumetric Efficiency (right) of a Scroll and Rotary Compressor each tested in the Calorimeter.

4. Conclusions

This paper presented the performance testing of a scroll compressor and a dual rotary compressor using a heat pump system and a designated compressor calorimeter. The added measurement capabilities of the calorimeter test stand allowed for additional performance data of the compressors to be captured. The results show that PSC rotary compressor performs better than its scroll counterpart when used with an electronic drive that can enable low-speed operation. More data should be collected for additional fixed-speed PSC compressors including rotary and scroll compressors. This data can be used to build models to accurately predict the performance of PSC compressors under variable speed operation.

References

[1] Goetzler, William., Sutherland, Timothy., Reis, Callie. "Energy Savings Potential and Opportunities for High-Efficiency Electric Motors in Residential and Commercial Equipment" (2013.) United States. doi:10.2172/1220812

[2] Salts, N.P. and Groll, E.A., "Inverter Drive Control and Seasonal Performance Analysis of a Single Speed Unitary Air-Source Split-System Heat Pump" (2017). 12th IEA Heat Pump Conference

[3] Chretien, L., Becerra, R., Salts, N.P., Groll, E.A. "System solution to improve energy efficiency of HVAC systems" (2017). IOP Conf. Ser.: Mater. Sci. Eng. 232 012067

[4] Chretien, Ludovic; Becerra, Roger; Salts, Nicholas; and Groll, Eckhard A., "Seasonal Energy Efficiency Rating Improvement Of Residential HVAC Systems Using A Low Power Inverter With A PSC Compressor" (2018). International Refrigeration and Air Conditioning Conference. Paper 1896.

[5] Mösch, T. "Performance tests of a scroll compressor with vapor injection using a compressor calorimeter" (2015.) Rostock, Mecklenburg-Vorpommern, Germany.

[6] Mathworks. (2018). MATLAB R2018a. *MATLAB*. Mathworks.

[7] Frigo, M., & Johnson, S. G. (2018, May). *FFTW*. Retrieved from FFTW: http://www.fftw.org/.