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Updated Performance and Operating Characteristics of a Novel Rotating Spool Compressor

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ABSTRACT

The novel rotary spool compressor mechanism has been described previously by Kemp *et al.* (2008, 2010). The device combines various aspects of rotary and reciprocating devices that are well understood. Increasing pressure on the global air conditioning market for adoption of ultra-low GWP refrigerants has motivated extensive modeling to develop a spool compressor design for operation on medium pressure ultra-low GWP refrigerants. Continued development of the direct drive compressor presented by Orosz *et al*. (2016) has resulted in a reliable high-performance compressor with peak overall isentropic efficiencies at 80% when operating on R134a. This makes the spool compressor well suited for ultra-low GWP medium pressure gases for use in commercial air conditioning and heat pumps. Various design modifications and changes to the spool compressor valve design have been implemented since the previous performance data was presented. Valve modifications were necessary to meet the reliability targets while maintaining high overall performance. Spool compressors with new valve designs were tested over a range of operating conditions typical for commercial air-conditioning and heat pump applications. The current compressor performance is superior to today's screw compressors operating in this size range and competitive to similar sized scroll compressors operating on R410A. Spool compressor test results are reported.

1. INTRODUCTION

The rotating spool compressor is a novel rotary compressor mechanism most similar to the sliding vane compressor. Primary differences are described by Kemp *et al*. (2008, 2010) and include three key differences from a sliding vane compressor, as shown in Figure 1.

- The vane is constrained by means of an eccentric cam allowing its distal end to be held in very close proximity to the housing bore while never contacting the bore.
- The rotor has affixed endplates that rotate with the central hub and vane forming a rotating spool.
- The use of dynamic sealing elements to minimize leakage between the suction and compression pockets as well as between the process pockets and the compressor containment.

The movement of the rotor is purely rotary with only the vane and tip seals performing any oscillating movement. The eccentric cam will force the movement of the vane to oscillate by twice the eccentricity during a single rotation. The tip seals will oscillate relative to the vane two times per rotation by an amount proportional to the ratio of diameters of the rotor to the housing bore (also known as the eccentricity ratio). The tip seal movement amount is roughly an order of magnitude smaller than the eccentricity and follows a sinusoidal path. Analytical details regarding the geometry, including the mathematical expressions describing the chamber volumes, is presented in Bradshaw and Groll (2013).

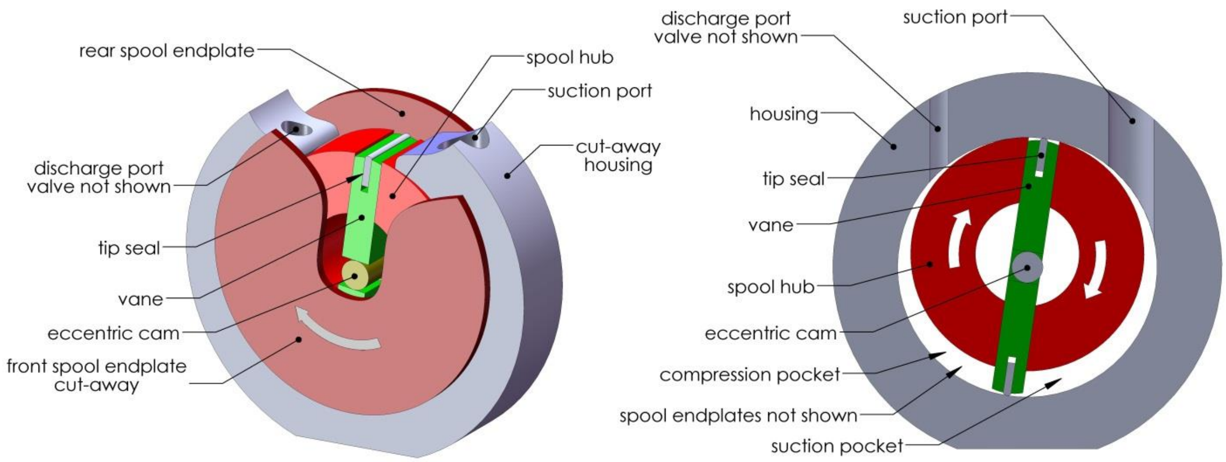

Figure 1: Cutaway view of rotating spool compressor mechanism with key components highlighted.

Since its introduction in 2008 the spool compressor has been improved and better understood. A comprehensive model was developed to deepen the understanding of the device holistically, as a system (Bradshaw and Groll, 2013). This holistic approach has allowed for better prediction of non-intuitive behavior of the spool compressor especially applied to applications and working fluids without experimental experience. The model includes 10 leak paths, 8 heat flow paths, port geometries, as well as oil solubility and frictional component sub-models.

Additionally, sub-components of the spool compressor have been studied in detail. Bradshaw (2013) presented a study on the analytical model developed to represent the tip seal in the spool compressor. The model was developed based on the tip seal dynamic model coupled with hydrodynamic lubrication theory. Kemp *et al.* (2012) presented iterations on the spool seal design and described the critical nature of the design of this component.

Bradshaw *et al.* (2016) recently presented the development of a loss analysis to describe the 5th generation spool compressor. This analysis utilized high-speed pressure measurements to determine the indicated, or flow, losses within the device. Coupled with analysis of the frictional losses a complete loss Pareto was generated. This analysis has resulted in a deep understanding of the location of potential improvements as well as an improved comprehensive modeling tool which accounts, more accurately, for the various frictional losses within the device. The resulting improvements using this analysis were also presented which showed a significant improvement in performance between the 5th and 6th generation compressor. These results echo the improved performance presented by Orosz *et al.* (2014).

Subsequent work by Bradshaw *et al.* (2018) and Yarborough *et al*. (2020) which conducted high speed pressure measurements to determine the indicated work and flow losses of the previous 141 kW (40 TonR) direct drive prototype and a 106 kW (30 TonR) Semi-Hermetic prototype respectively allowed a clear analysis of the losses associated with the compression process in the size range of the subject of this paper. The discharge valve operation was identified as the major source of inefficiency accounting for shaft losses between 3% and 15% depending on compressor speed and operating condition. Utilizing Computational Fluid Dynamics various valve configurations and designs were explored with the aim of reducing these valve losses. Based on that work a new valve design was arrived at and hardware constructed is designated Valve Type 2.

This work presents an experimental characterization of a 106 kW (30 TonR) prototype spool compressor for application on commercial air-conditioning. The custom-built hot-gas bypass load stand is used which was previously described in Orosz *et el. (2016).* The experimental performance data at equivalent line voltage speeds for two valve designs is presented and discussed.

2. PERFORMANCE EVALUATION

A 106kW (30 TonR) R134a spool compressor was used to conduct testing on the various valve designs. The overall geometry of the current 106 kW (30 TonR) design is the same as the previous 141 kW (40 TonR) design, that is the eccentricity ratio and the L/D are the same with changes made to the diameter to reduce the swept volume of the compressor. This was done to better accommodate the testing of lower pressure ratio points on the test stand at the higher shaft speeds. The compressor data presented is at 1750 rpm with a range of suction pressures from 281.4 kPa (40.81 psia) to 497.5 kPa (72.2 psia) and a range of discharge pressures from 699.1 kPa (101.4 psia) to 1472.0 kPa (213.5 psia) utilizing R134a as the working fluid. This represents evaporating and condensing temperatures of -1.1 °C (30 °F) to 15.5 °C (60 °F) and 26.7 °C (80 °F) to 48.9 °C (120 °F), respectively. Intake gas superheat was fixed at 11.1 ºC (20 ºF) for all data points. These conditions represent a typical commercial air-conditioning application for both water and air cooled conditions.

2.2 Test Stand

Construction of a hot gas by-pass test stand that was necessary to test the constructed prototype was previously described in Orosz *et el. (2016)*. The test stand includes the ability to separate and meter oil back into the compressor sump as well as into the intake for better simulation of actual system conditions. In addition, the test stand was designed to be ASHRAE 23.1 compliant and, where practical, includes redundant measurements on all critical functions including pressure, temperature and mass flow rate. Mass flow sensors are installed on the discharge as well as the intake of the compressor. The discharge mass flow meter is utilized to calculate the efficiencies because the gas is entering at a higher super heat with less risk for two-phase flow. The mass balance is considered acceptable at a level below 1.5%. Steady state is determined by calculating the total statistical variance of critical measurements over a 15 minute period. The test point is considered stable when the temperature variance is less than 0.2 K (0.4 R), pressure variances below 6.8 kPa (1 psi), speed variance is below 1 rpm, and mass flow rate variance is below 0.007 kg/sec (1 lbm/min). The operating conditions of the compressor are controlled with software driven PID controllers which adjust the bypass valve, liquid line valve and condenser water flow controls. The power measurement is obtained with a rotary torque sensor and verified with the motor input power accounting for approximate motor efficiency.

Figure 2: Photo of 141 kW hot gas bypass test stand

2.2 Efficiency Definitions

Test data is collected to calculate the Volumetric Efficiency and Overall Isentropic Efficiency as a function of pressure ratio and speed. The volumetric efficiency was determined using Equation (1), where the theoretical volume flow was obtained based on speed measurements and the displacement volume:

$$\eta_{vol} = \frac{\dot{m}_{act} \cdot v_1}{\dot{V}_{th}} \quad \textbf{(1)}$$

where $\dot{m}_{act}$ and v_1 are the measured mass flow rate and specific volume, respectively. The overall isentropic efficiency is a frequently used measure for the first law efficiency of compressors by using an overall control volume, i.e., an evaluation by using the thermodynamic states at the compressor inlet and outlet. The overall isentropic efficiency is obtained based on Equation (2):

$$\eta_{is,o} = \frac{\dot{m}_{act} \cdot (h_{2s} - h_1)}{\dot{W}_{shaft}} \quad \textbf{(2)}$$

where $\dot{W}_{shaft}$ is the shaft power input to the compressor mechanism only.

The uncertainties' associated with testing can be found in Orosz *et al.(2016).*

3. COMPRESSOR PERFORMANCE

3.1 Volumetric Efficiency Analysis

The 106 kW (30 TonR) rotary spool compressor was evaluated based on the above running conditions using refrigerant R134a. Figure 3 is representative of the Valve Type 1 configuration. Many valve configurations were tested with variations in the number of valves, row locations, valve weight and spring stiffness. The presentation of Valve Type 1 and Valve Type 2 represent optimized configurations each of their respective designs. Figure 4 represents the volumetric efficiency for the Valve Type 1 design that is near optimal at full load conditions for air conditioning operation. Data is plotted against the pressure ratio for a fixed saturated suction temperature of 4.44 ℃ (40 ℉). Testing has been done at various saturated suction conditions and the volumetric efficiency is insensitive to the change in saturated suction for the range of normal air conditioning application. The data shows a slope of 1.8 % loss in volumetric flow rate per pressure ratio. Having such shallow slope allows the compressor to maintain its capacity at system operating pressure ratio increases at high ambient. Figure 5 presents the same data for the Valve Type 2 design. In this case we see that the slope is approximate 1.0% loss in volumetric flow per pressure ratio. Compared to the Valve Type 1 design this is reduced substantially. As these are two distinct compressors we could suspect that the machines are not identical in construction. It is also possible that due to the large reduction in over-pressure in the compression process the Ve could be improved due to reduction internal leakage. This resulted in a Ve of 94.2% on the Valve Type 2 design verses 93.0% on the Valve Type 1 design.

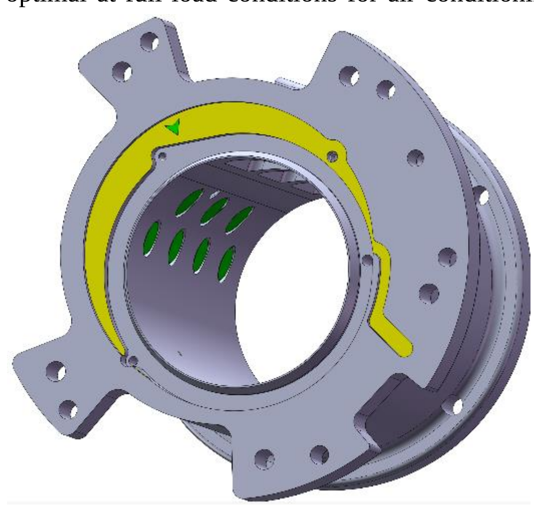

Figure 3: Typical Valve Configuration

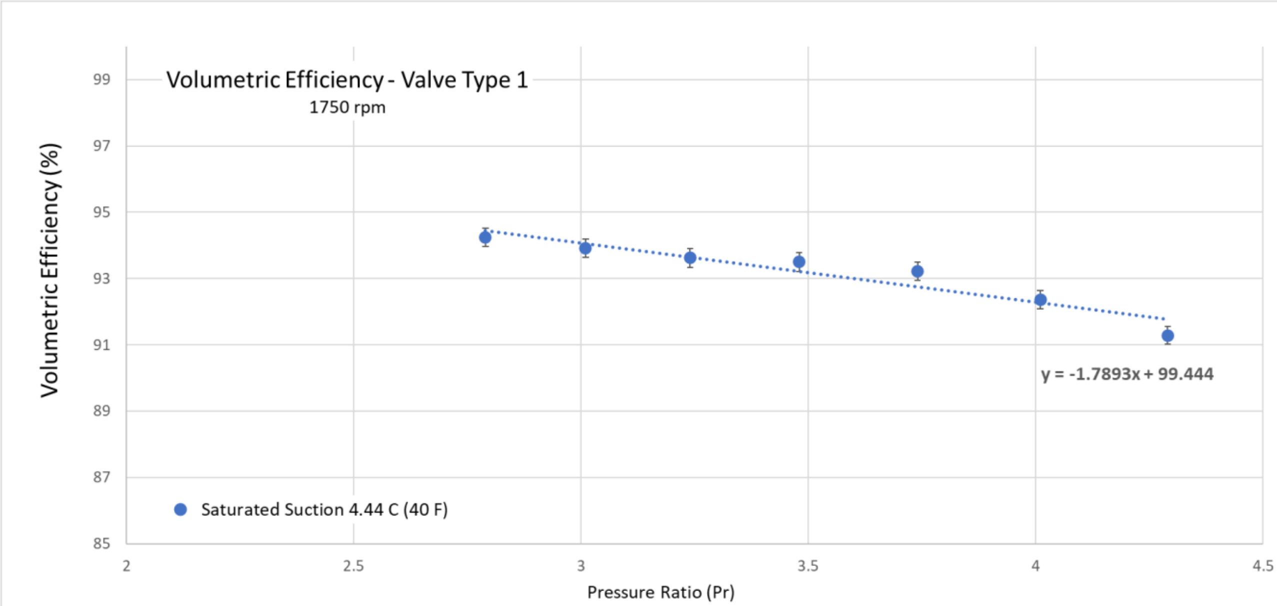

Figure 4: 106 kW spool compressor volumetric efficiency vs. pressure ratio at 1750 rpm – Valve Type 1.

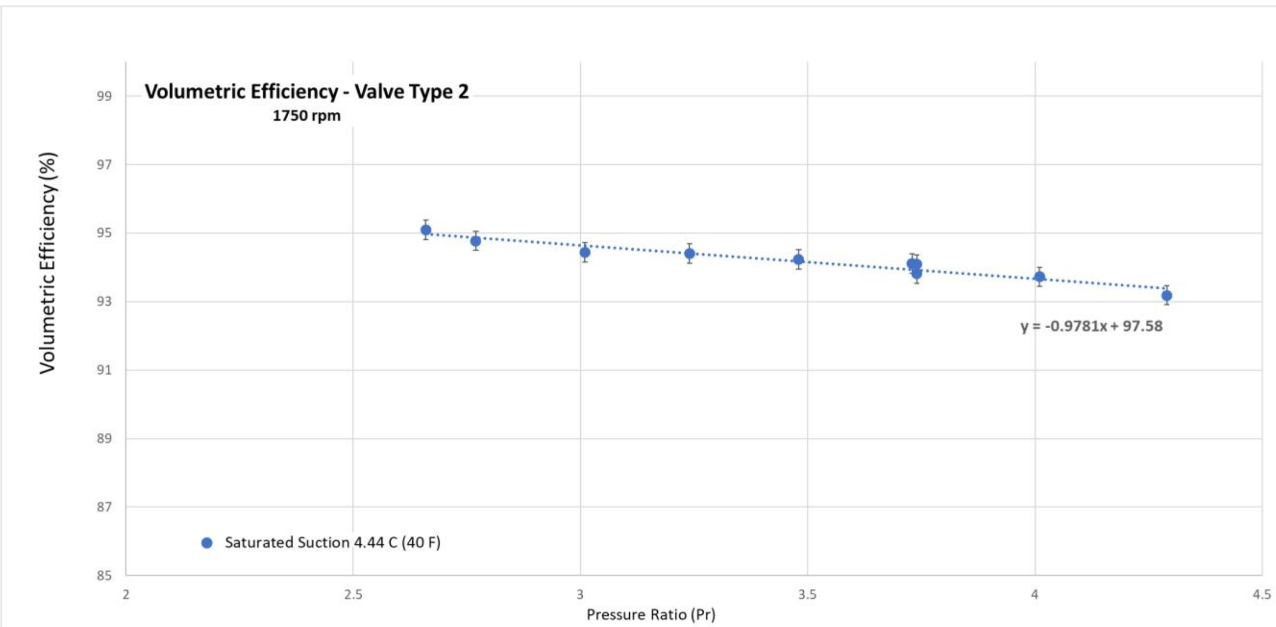

Figure 5: 106 kW spool compressor volumetric efficiency vs. pressure ratio at 1750 rpm – Valve Type 2.

3.2 Overall Isentropic Efficiency Analysis

Figures 6 and 7 present the overall isentropic efficiency at various pressure ratios at a constant saturated suction temperature of 4.4 ℃ (40 ℉) at a shaft speed of 1750 rpm, representative of a 4-pole motor application. Looking at Figure 6 the peak efficiency of 76.8% occurs at 3.5 pressure ratios which, at 4.4 ℃ (40 ℉), saturated suction has an equivalent saturated discharge of 46.1 ℃ (115 ℉). The design of Valve Type 1 was reached as a compromise between good full load efficiency, which would occur at approximately 3.8 Pr and reasonable IPLV (Integrated Part Load

Value) for a chiller application. The valve design can be adjusted for peak efficiency to occur at a lower pressure ratio favoring an increased IPLV value. From Figure 7 we see that the Valve Type 2 design offers a substantial improvement over the Valve Type 1 design. At the full load operating condition of 3.8 pressure ratio the overall isentropic efficiency for Valve Type 1 design is 76.5%, for the Valve Type 2 design it is 79.2%, an improvement of 3.5%. The Valve Type 2 design continues to improve with decreasing pressure ratio to reach a maximum tested value of 80.2% at 3.0 pressure ratios. Test stand flow limitations prohibit testing pressure ratios lower than 2.8. Based on these measurements the peak efficiency of the Valve Type 2 design is 4.7% better the than the Valve Type 1 design even if they occur at different pressure ratios. Further, in the range of measurement there is no case where the Valve Type 2 design is not better than the Valve Type 1 design even at the higher pressure ratios. Figure 8 data shows 3 distinct condensing temperatures for various saturated suction conditions. As the pressure ratio is reduced the overall isentropic efficiency converges to a value between 79% - 80%. This is due to a convergence of the volumetric efficiency. The 43.3 ℃ (130 °F) condensing line flattens out as the saturated suction is reduced. This is due to a nonlinear reduction in the torque at lower mass flows for a proportional loss in volumetric efficiency.

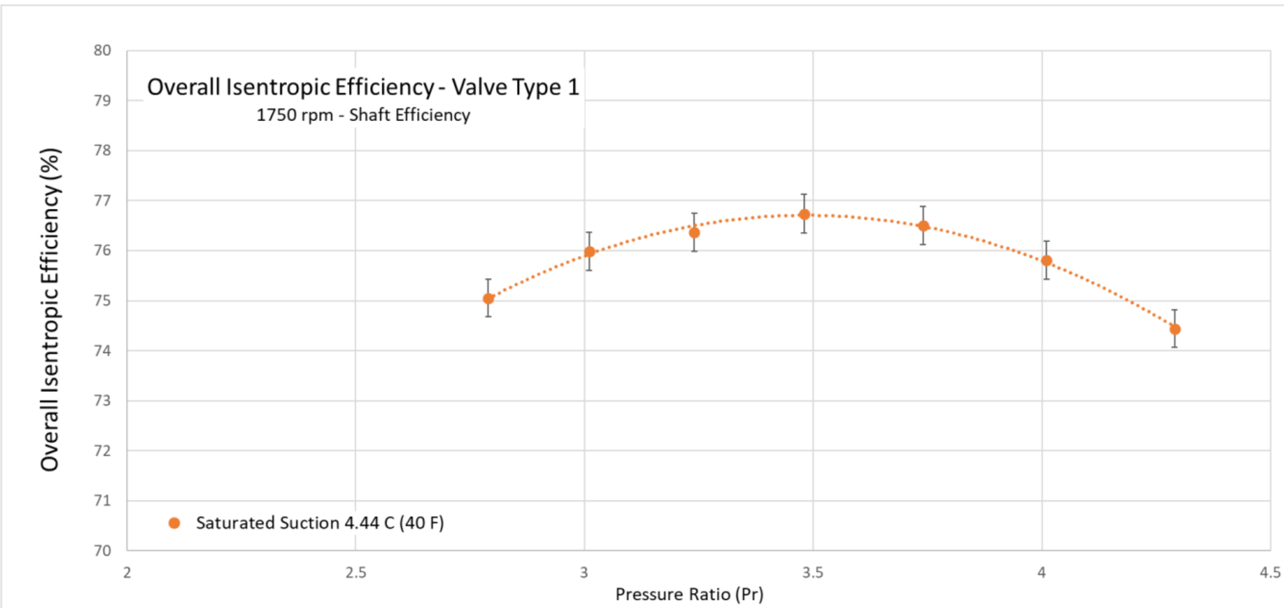

Figure 6: 106 kW spool compressor overall isentropic efficiency ($\eta_{is,0}$) vs. pressure ratio at 1750 rpm – Valve Type 1

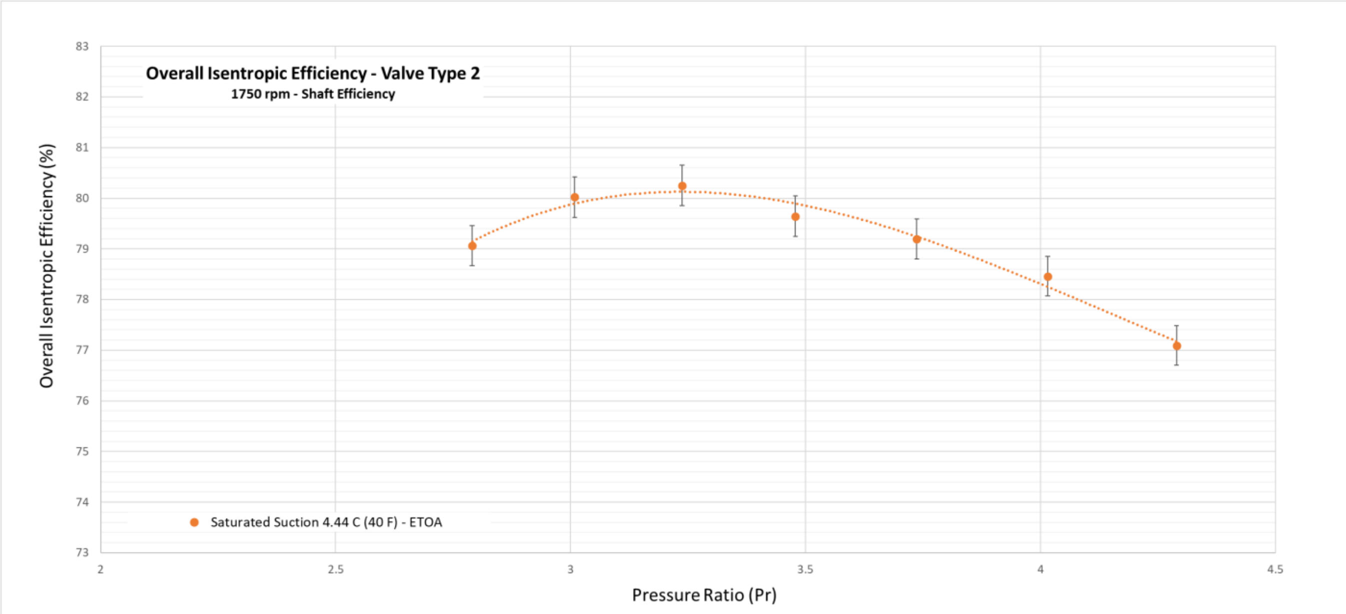

Figure 7: 106 kW spool compressor overall isentropic efficiency ($\eta_{is,0}$) vs. pressure ratio at 1750 rpm – Valve Type 2

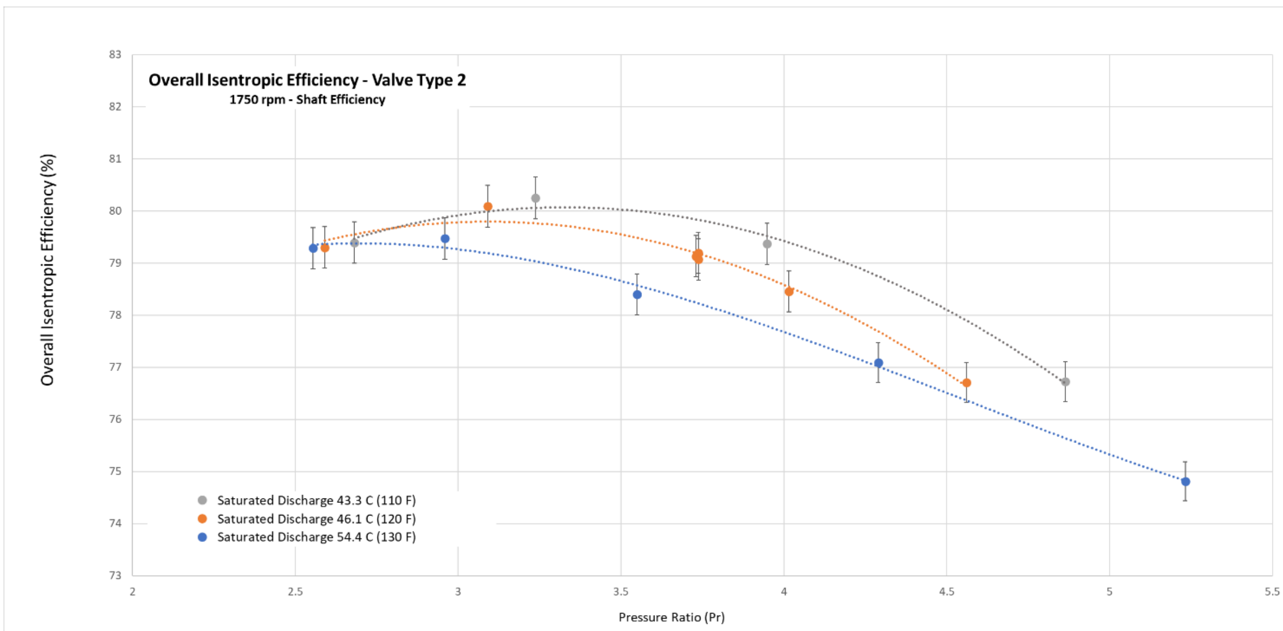

Figure 8: 106 kW spool compressor all isentropic efficiency $\left(\eta_{is,0}\right)$ Multiple condensing temperatures at 1750 rpm – Valve Type 2

4. CONCLUSIONS

An experimental characterization of a prototype 106 kW (30 TonR) R134a spool compressor is presented utilizing 2 different types of valve designs. This compressor design was based on geometry previously presented in Orosz *et el. (2016)*. Figure 9 summarizes the efficiency of each device at the nominal full load point of 3.8 Pr as applied on an air conditioning application. We can see that in both Valve Type 1 & 2 we have a substantial improvement for the earlier prototype performance due to extensive analysis of the valve losses and subsequent redesign efforts. In the case of the Valve Type 2 design the overall isentropic efficiency shows an improvement of 11.5% over the previous 141 kW design.

Compressor Type	Overall Isentropic Efficiency at 3.8 Pr
141 kW (40 TonR) - Orosz *et el. (2016)*	71.0%
106 kW (30 TonR) – Valve Type 1	76.5%
106 kW (30 TonR) – Valve Type 2	79.2%

Figure 9: Summary of Efficiencies

The current prototype compressor tested showed substantial improvement in efficiency over prior prototypes and between the Valve Type 1 and Valve Type 2 designs. Improved valve operation will allow the compressor to achieve required efficiencies when operating on R1234ze(E) with even lower capacity for same displacement. The spool compressor's ability to generate large swept volumes with good volumetric efficiency has allowed it demonstrated overall isentropic efficiencies at levels historically reserved for compressors with no valves and clearance-controlled leak paths. The spool compressor utilizing medium/low density ultra-low GWP refrigerants is particularly well suited for commercial applications where a single compressor needs to meet both heating and cooling requirements.

NOMENCLATURE

V_{Th}	Compressor Displacement, m^3/h	$\eta_{is,o}$,	Overall Isentropic Efficiency
$\dot{m}$	Mass flow, kg s-1	h	Specific Enthalpy, kJ kg-1
v	Specific Volume, m3 kg-1	P	Pressure of the gas, kPa
T	Temperature of the gas, K	$\dot{w}$	Work, kW
SSH	Intake Superheat, ºC	LSC	Liquid sub-cooling, ºC
Pr	Pressure Ratio	η_{vol}	Volumetric Efficiency
Rpm	Speed, Revolutions / min		

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