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ABSTRACT

The basic mechanism of the novel rotary spool compressor has been described previously by Kemp et al. (2008, 2010). The device combines various aspects of rotary and reciprocating devices that are currently well understood. A dimensionless variable, the Zsoro number, was developed which represents the geometric configuration of the compressor and has been described previously by Orosz et al. (2012). This number in combination with extensive numerical modeling as described by Bradshaw et al. (2014, 2014a) has allowed for continued improvement in the spool compressor's efficiency and operating range. Compressors tested have nominal capacities on R410a between 1 and 5 tons of cooling capacity at standard air conditioning conditions. A new platform with improved aspect ratio based on these analyses has been constructed and tested as well as optimization of sub-components such as the tip seals, side seals and discharge porting. The most current prototype compressor is operating with an overall isentropic efficiency which exceeds 80% based on shaft power. This performance is compared against commercial scroll compressors in a similar size range.

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).

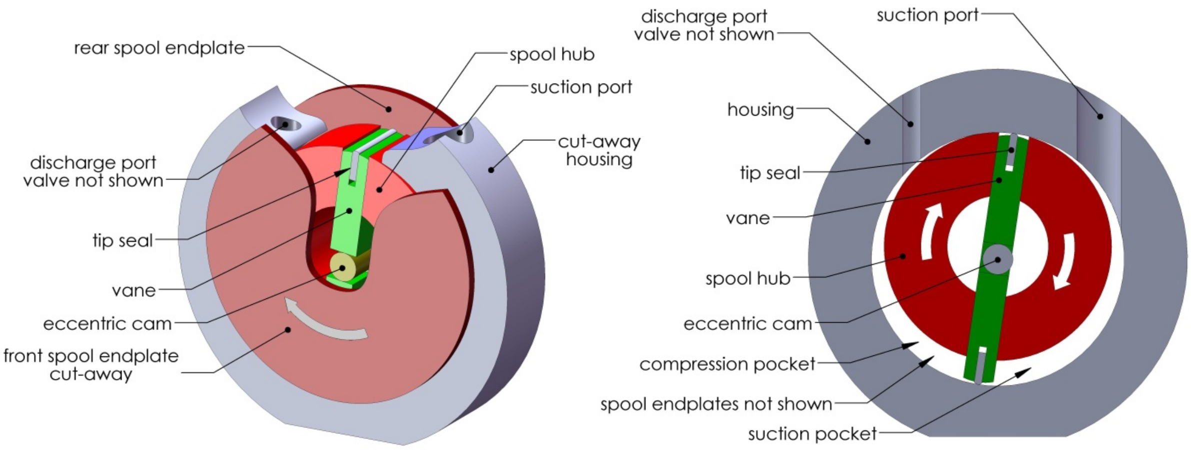

Figure 1: Cutaway view of rotating spool compressor mechanism with key components highlighted.

A traditional vane machine (either sliding vane or rotary vane) are relatively difficult platforms to scale into larger capacity ranges (larger displacement). This problem stems from the interface between the vane and the housing bore and/or the interface between the vane and the end walls of machine. The spool compressor mechanism reduces these problems by constraining the vane at the center of rotation and also rotating the endplate with the vane.

By constraining the vane as close to its center of mass the sliding velocity and kinematic forces are kept at a minimum. In addition, the vane itself is restricted from sliding on the housing bore. Instead, a secondary sealing element, the tip seal, is required to take up the gap between the end of the vane and the housing bore. The tip seal size and weight can be configured to reduce the frictional losses due to the contact between the housing bore and tip seal. A study of the tip seal design parameters was presented by Bradshaw (2013).

The rotation of the endplates of the spool compressor mechanism greatly reduces the friction generated by the vane. It is not completely eliminated because the vane must slide radially relative to the endplate by an amount equal to the eccentricity. Since the endplate is rotating relative to the compressor housing the gap between these two parts requires sealing. This is accomplished with an additional dynamic sealing element called the spool seal. An overview of the spool seal design constraints has been presented by Kemp et al. (2012). The spool seal can be designed to accommodate various machine applications, such as a seal which can handle a wide operating envelope with adequate sealing. Alternatively, the seal can be designed with lower frictional power loss for applications which have high efficiency demands and a relatively small operating envelope, such as a water-cooled chiller.

The combination of these dynamic sealing elements give the spool compressor the flexibility to scale to sizes that have historically proven difficult for other vane machines. Additionally, this provides platform flexibility with a specific displacement that is also difficult with traditional vane machines due to the ability to modify the seal designs.

2. PERFORMANCE EVALUATION

Testing was conducted on a hot gas bypass type test stand which has been previously described in Orosz et al. (2012). Using this test stand the rotating spool compressor prototypes were tested at a variety of rotational speeds and a standard operating condition of 905 kPa, 2282 kPa, and 11 °C using R410A and 338 kPa, 950 kPa, and 11 °C using R134A for suction pressure, discharge pressure, and superheat temperature respectively. These conditions represent a typical air-conditioning condition for the respective working fluids. Additionally, for some tests, operating conditions were varied to achieve a range of pressure ratios for each working fluid.

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 (2), where the theoretical volume flow was obtained based on speed measurements and the displacement volume:

$$\eta_{vol} = \frac{\dot{m}_{act} \cdot v_1}{\dot{V}_{th}} \tag{2}$$

where $\dot{m}_{act}$ and v_1 are the measured massflow 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 (3):

$$\eta_{is,o} = \frac{\dot{m}_{act} \cdot (h_{2s} - \mathrm{h}_1)}{\dot{W}_{shaft}} \tag{3}$$

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

2.3 Shaft Power Adjustments

For an open drive prototype the true shaft power cannot be measured directly because of the power draw of the shaft seal. Similarly, for a hermetic compressor the motor inefficiencies need to be accounted for to obtain a true shaft power. Therefore, an estimate of shaft seal and motor power losses need to be estimated.

The latest spool compressor iteration, denoted CP6.1, utilizes an large shaft diameter for increased stiffness and better high speed stability. Previous versions have used shaft diameters small enough that the shaft seal torque was negligible in regards to the calculation of the overall isentropic efficiency. A compressor rotor was suspended in a RCP6.1 prototype shell with a shaft seal installed. This assembly was run at speeds from 1000 to 4000 rpm and inlet pressures from 0 to 1500 kPa. The torque measured was used to generate a map of the torque generated from the shaft seal. These data points were best fit using a 2D, 2nd order polynomial, surface fit. Figure 2 shows the data collected and the corresponding surface map.

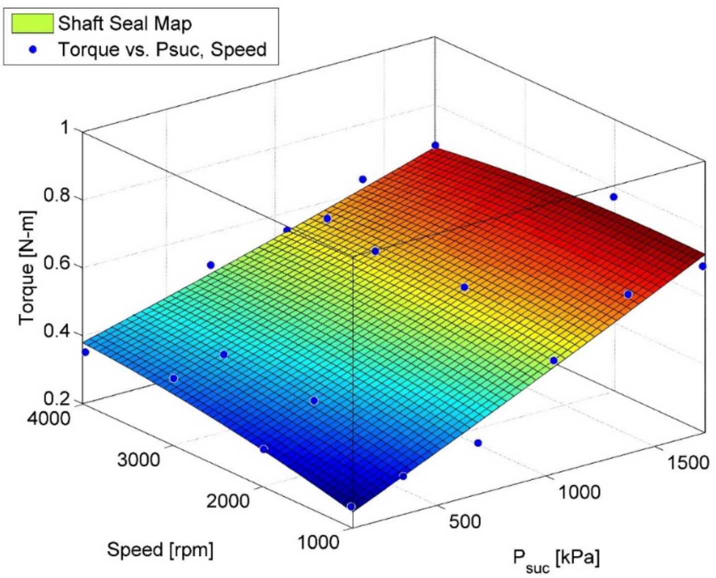

Figure 2: Shaft seal torque loss as a function of cavity pressure and shaft speed.

The map has an R-square of 0.89 and a RMS error of 0.05 N-m. To adjust hermetic compressor data to represent shaft efficiency a motor efficiency of 91% is assumed. This is chosen to represent a 4 kW, 3-phase hermetic motor used in an air-conditioning application.

3. DESIGN EVOLUTION

Some of the first prototypes were designed based on rules of thumb which applied to other technology, some intuition based on experience, and some guessing. However, during the testing of the 4th generation prototype, a useful nondimensional number was developed to aid in compressor design, as presented in Orosz et al. (2012). The number, called the Zsoro number, is given as the ratio of spool seal losses to the displaced volume.

$$Zsoro = R_{esr} \ * \ A_{ssa}/D_R \qquad (1)$$

As shown in Figure 3, the initial overall isentropic efficiencies achieved for the 2nd through 6th generation prototype compressors (RCP2.1 to RCP6.1) plotted against Zsoro number. This relationship has provided a benchmark to start from when designing a new compressor.

The Zsoro number does not account for all phenomenon, such as porting, valves, leakage, heat transfer, and tip seal friction to name a few items. Therefore, it was imperative to develop a more comprehensive compressor model to aid in designing and optimizing the spool compressor. A comprehensive compressor model was developed and presented by Bradshaw and Groll (2013). This model sets a framework which has been matured over the last three years and has been used to predict compressor performance for many other types of compressors in addition to the spool. It models the open drive spool compressor prototypes and includes all major phenomenon including valve dynamics, heat transfer, leakage, friction and sub-models for the dynamic sealing elements. The model has predicted volumetric and overall isentropic efficiency of the 5th generation prototype to within 2% and 5%, respectively. While the model is continuously evolving, it has an accuracy level that yields confidence in its predictive power. These predictions have allowed great progress in understanding the potential of the spool compressor. Using the RCP5.3 platform as a research platform extensive analysis was made of the tip seal function, side seal function and discharge valve operation (Bradshaw et al. 2014, 2014a). Based on these efforts and the Zsoro number a geometric configuration for RCP6.1 was developed. Mechanical construction of the CP6.1 is similar to the CP5.3 regarding bearing design, intake porting, vane and tip seal design.

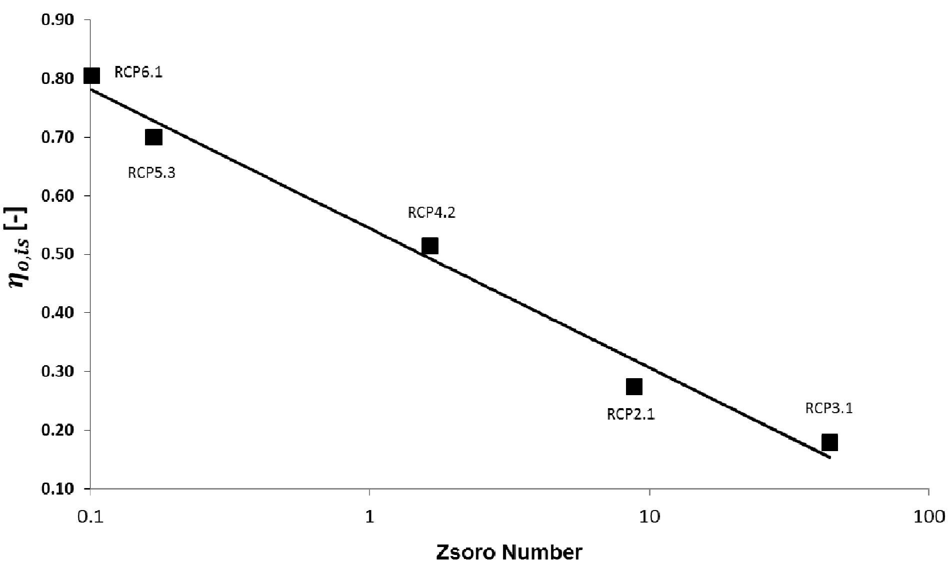

Figure 3: Isentropic efficiency vs. Zsoro number at standard operating conditions (experimental uncertainty within marker width).

The volumetric efficiency of prototype compressor RCP6.1 was in the range of the previous two platforms. The geometry of platform RCP6.1 was achieved by increasing the eccentricity of the machine without changing either the rotor bore dimension or the L/D ratio. Are was previously asserted the volumetric efficiency is high enough at standard operating conditions that it is assumed internal leakage of the gas does not have a significant effect on the overall efficiency. The improvements in the overall isentropic efficiency correlate with the decreasing Zsoro number although as previously stated in the Orosz et al (2012) the much improved overall performance of the compressor was impacted by not only the improvement in the Zsoro number but also optimization of the various subcomponents including the tip seals, discharge valves and side seals. Continued reduction of the Zsoro number should always be considered but can be limited by other design considerations. Additionally, Table 1 lists a history of the rotating spool compressor prototype generations, their displacements, and respective Zsoro number.

Table 1: Rotating spool compressor platforms tested since 2011 with the displacement and Zsoro number of each platform.

Compressor #	**Initial Operation**	**Displacement**	**Zsoro Number**
RCP2.1	January 2011	61.45 cc/rev	8.79
RCP3.1	May 2011	10.24 cc/rev	44.07
RCP4.2	December 2011	46.54 cc/rev	1.63
RCP5.3	May 2012	39.33 cc/rev	0.168
RCP6.1	May 2014	54.64 cc/rev	0.101

Table 2 shows the performance of RCP6.1 at conditions designed to replicate those of the SEER rating points A and B. The SEER rating points represent indoor temperature of 26.7 ºC (80 ºF) and outdoor temperatures of 35 and 27.8 ºC (95 and 82 ºF). These are two of the conditions used to rate the efficiency of an air-conditioning unit and therefore useful for comparison of the spool compressor to other technologies. The overall isentropic efficiency of the A-point given in Table 2 is 83.72% which corresponds to an EER of 24.98, assuming 11K (15 ºR) subcooling.

Table 2: Performance of RCP6.1 at points which simulate the SEER A and B rating points for unitary air-conditioning equipment.

Tdis	Tsuc	Pdis	Psuc	Torque	Mass	Speed	Power	eta_vol	eta_o_is	T_sup	PR
[ºC]	[ºC]	[kPa]	[kPa]	[N-m]	[kg/min]	[rpm]	[W]	[%]	[%]	[ºC]	[-]
60.83	21.42	2270	1070	7.96	4.84	2524	2104	92.48	83.72	11.64	2.12
74.67	22.51	2760	1097	10.81	4.99	2604	2948	90.28	76.08	11.88	2.52

Figure 4 shows the range of volumetric efficiency obtained for each compressor prototype tested at various operating conditions. Looking at Figure 4 it can be seen that the spool compressor can be sealed in the range of 55% to 94% η_{vol} over a Pressure Ratio (PR) range of 2.5 – 4.5. The latest compressor design (RCP6.1) has volumetric efficiency on par with the previous two platforms. This is in the range of 93% to 85% operating at a PR between 2.5 and 4.5 respectively. All development in the past two years has been conducted at speeds predominately in the 2-pole range. However, at the time of the development of this manuscript, RCP6.1 was still being qualified and as such only data at 2500 rpm is able to be presented. The main achievement in the newer design is the significantly improved slope of the volumetric efficiency loss as a function of increasing pressure ratio.

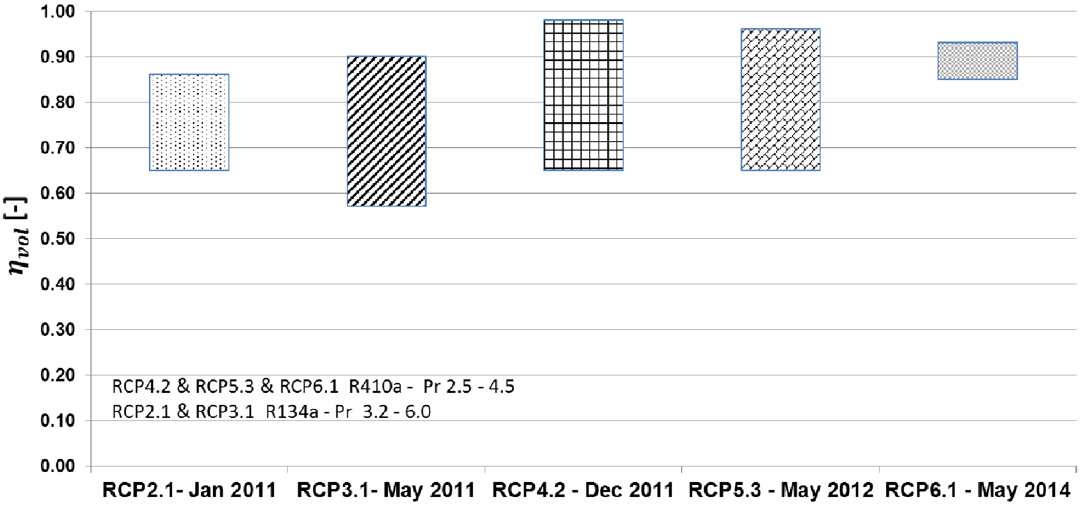

Figure 4: Volumetric efficiency evolution for each compressor prototype at various operating conditions.

Significant effort has been made to improve the manufacture of the components in order to ensure consistent management of the leakage paths in the compressor. Looking at Figure 4 it can be seen that the range of volumetric efficiency is reduced significantly on the current platform across the same operating conditions.

Figure 5 shows the evolution of the overall isentropic efficiency across the various platforms that have been constructed since 2011. These platforms represent a variety of displacements, as listed in Table 1, operating on different refrigerants and at different speeds. The last three platforms RCP4.2 – RCP6.1 have been of similar displacement, operating on R410a in approximately the same speed range and so are directly comparable.

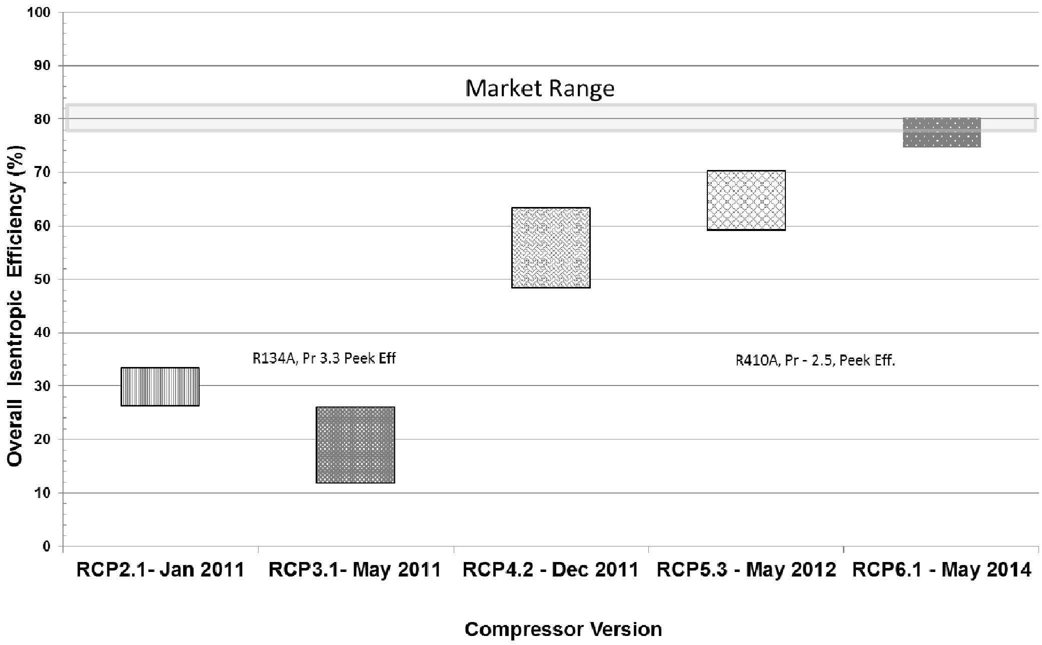

Figure 5: Compressor overall isentropic efficiency evolution by design platform.

The market data reflects the range of compressor performance data that is readily available for a nominal 5 ton scroll compressor with a 3-phase motor. This data reflects the estimated shaft power based on the published compressor watt input and assuming a 91% motor efficiency. No other adjustments were made for internal losses. The range of peek overall isentropic efficiency is 77% to 82%. The CP6.1 platform has a peek performance of 80% which is competitive with current offerings.

4. COMPRESSOR PERFORMANCE COMPARED WITH CURRENT MARKET

The rotary spool compressor was evaluated against the current technologies employed in small to medium size air conditioning applications. Compressors in this market segment are considered to be the most efficient due to government regulations. Figure 6 shows the comparison the volumetric efficiency of RCP5.3 and RCP6.1 with a commercially available scroll published data at various operating conditions. Using R410A as the refrigerant the suction pressure is varied from 345 kPa (50 psia) to 1172 kPa (170 psia) at a fixed discharge pressure of 2282 kPa (331 psia), and 11 K (16 R) of superheat. This corresponds to -23.3 °C (-10 °F) to 12.8 °C (55 °F) and 37.8 °C (100 °F) evaporating and condensing temperatures, respectively. RCP5.3 data is reported at 3550 rpm while RCP6.1 is reported at 2500 rpm as this platform was still being qualified at the time this manuscript was authored. The volumetric efficiency of the scroll is represented by an upper and lower boundary as the commercially available test data shows a fairly wide range of performance.

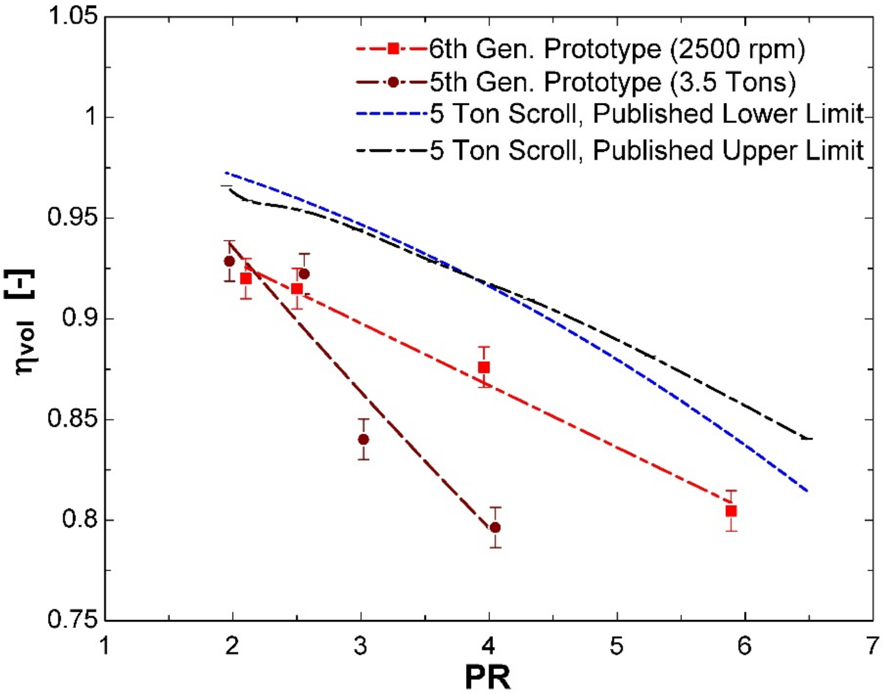

Figure 6: Rotary spool compressor prototypes volumetric efficiency vs. commercial compressor technologies.

The prototype RCP5.3 was designed to allow slightly more internal clearance in favor of reduced torque, therefore the volumetric efficiency was lower than comparable scroll compressors and the slope of the loss per PR was also higher approaching 7% which is more on par with a reciprocating machine. The prototype CP6.1 was designed with the objective of improved volumetric efficiency and loss per PR while preserving or improving the overall isentropic efficiency. Looking at Figure 6 we can see that the RCP6.1 compressor has a volumetric efficiency equal to the RCP5.3 design at 2 PR but has an improved loss per PR slope. The loss per PR is on par with a scroll compressor in the range of 3% per PR. The overall volumetric efficiency of RCP6.1 is still slightly lower than a scroll compressor due to two major effects. One is the shortness of the leak paths and the other is the fact that the spool compressor, no matter how well designed, will have some carryover volume from the discharge pocket to the suction pocket.

Figure 7 is a comparison of the overall isentropic efficiency of the same compressors that were compared in Figure 6 for their volumetric efficiency. Looking at Figure 7 we can see that the RCP6.1 compressor design has a peak overall isentropic efficiency that is 20% better than the RCP5.3 design. A majority of the improvements are thought to be a result of an improved geometric design of the machine as well as optimization of other components which is outside the scope of the current work but detailed in Bradshaw et al. (2014, 2014a). As with Figure 6, the scroll compressor performance is represented by an upper and lower boundary based on available data. Due to the lack of a fixed internal volume ratio the spool compressors overall isentropic efficiency continues to improve with reducing PR and in fact becomes more efficient then the scroll compressor at lower PR.

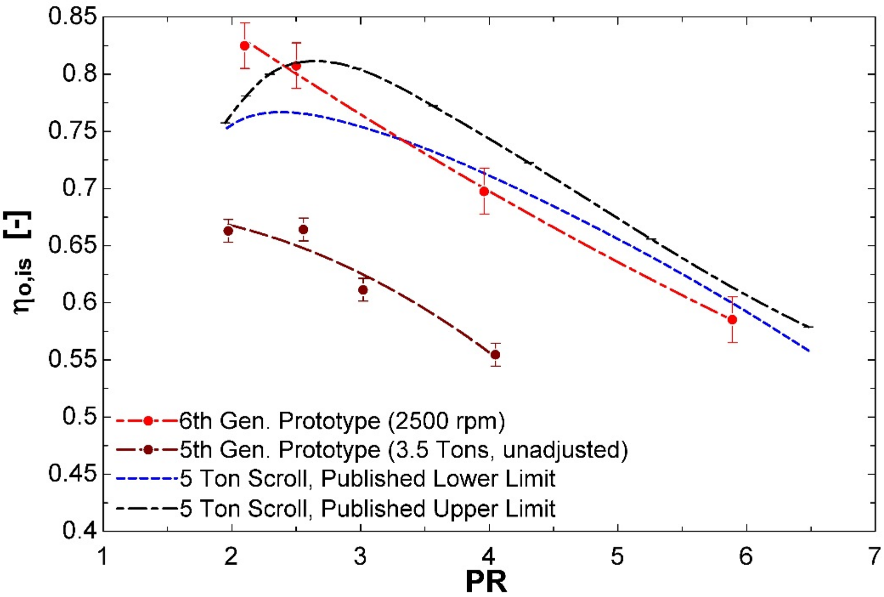

Figure 7 - Rotary spool overall isentropic efficiency vs. other technologies

5. CONCLUSIONS

The prototype design evolution from previously presented rotary spool compressor versions to the current RCP6.1 version is presented. The test method for experimental testing was presented previously by Orosz et al. (2012) and is updated to include the mapping of the shaft seal torque so the performance of the mechanism is correctly represented.

Based on previous work from Orosz et al. (2012) the correlation between the Zsoro number and the overall isentropic efficiency was seen to be a reasonable predictor of compressor performance and the RCP6.1 followed the same trend although the improvement was not linear against the Zsoro number. Continued reduction of the Zsoro number yields improvement in the overall performance but the practical limit is bounded by the geometric constants of the vane travel and the length to diameter ratio based on manufacturing limits. For the purpose of designing a spool compressor that is competitive with current product offerings a Zsoro < 0.2 would be necessary as well as sufficient detail paid to the optimization of the additional components effecting performance.

NOMENCLATURE

Zsoro	Dimensionless drag loss coefficient	R_{esr}	Radius of the effective seal surface, m
A_{ssa}	Area of the sealing surface, m^2	D_r	Compressor Displacement, m^3
$\dot{m}$	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, W
SSH	Intake Superheat, C	LSC	Liquid sub-cooling, C
Pr	Pressure Ratio	η_{vol}	Volumetric Efficiency
$\eta_{o,is}$	Overall Isentropic Efficiency		

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