Performance and Operating Characteristics of a Novel Rotating Spool Compressor

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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 currently well understood to achieve high efficiency at a low manufacturing cost. A dimensionless variable, the Zsoro number, is developed which represents the ratio of the geometric configuration of the compressor relative to the potential friction components of the compressor. This number allows for rapid evaluation of the geometric features of the device. Four prototype spool compressors have been tested using R410A and R134A at standard air conditioning conditions with various Zsoro numbers. Experimental data collected have shown a strong correlation between the overall isentropic efficiency and Zsoro number. These results have allowed for rapid design iteration of the rotating spool compressor. The most current prototype compressor has operated with a 5% higher overall isentropic efficiency than a typical commercial rolling piston compressor and within 5% of a commercial scroll compressor.

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.

- 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 (typically less than 0.30mm) while never contacting the bore.
- The rotor has affixed endplates that rotate with the central hub and vane forming a rotating spool.
- The practical 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

These differences are shown in Figure 1 which presents a cutaway view of a rotating spool compressor with the key geometric features highlighted.

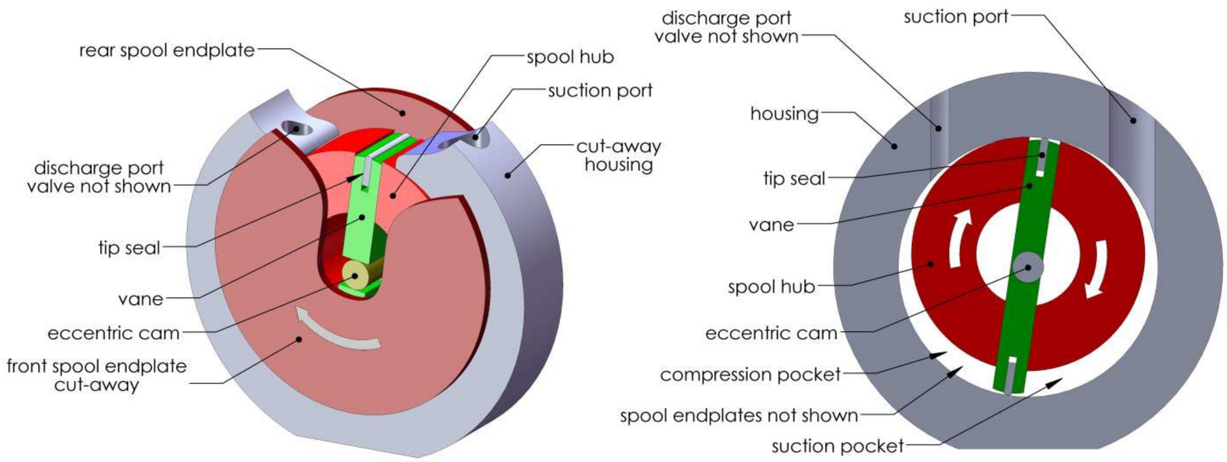

Figure 1: Cutaway view of rotating spool compressor mechanism with key components highlighted.

Due to the ever increasing need for increased performance envelopes and energy efficiency air conditioning and refrigeration system technologies has greatly evolved over the last 30 years. Shortly after the inception of vapor compression systems, the market was dominated by two types of compressors, the reciprocating compressor for the smaller cooling capacity and the centrifugal compressor for the larger cooling capacity systems. This was dictated, for the most part, from a knowledge base and manufacturing capability (Soumeri, 2010). This gave way to various compressor designs implemented over the next 30 years that capitalized on newer manufacturing technology and met the ever increasing efficiency requirements dictated by regulation. By example ASHRAE 90.1 has mandated an improvement in chiller efficiency of 61% since 1977 while employing refrigerants with lower cycle efficiency (Lord, 2009). This has produced a current technical profile in the market that encompasses at least seven fundamental device types in various configurations with various optimum technical features.

The spool compressor is a novel device that combines various design attributes of other technologies currently in production to achieve high efficiencies while maintaining low production costs. The present work characterizes the performance of the spool compressor, identifies the areas of loss and, identifies the benefits as they relate to the current market demands. This is accomplished through the iteration of prototype spool compressor designs which explore a variety of geometric configurations and operating conditions. A dimensionless number is developed to aid the design iteration which represents the frictional losses relative to the displaced volume. The prototype configurations are evaluated for trends in performance and key geometric parameters are identified. In addition, the current prototype performance is benchmarked against the performance of a rotary and scroll compressor from the current air-conditioning market.

2. DESIGN EVOLUTION

Rapid design evolution is required in the early stages of any new compressor development. Working with a new compressor concept, first the basic mechanism must pass the proof of concept, which has been presented by Kemp (2008, 2010). Then the compressor volumetric and overall isentropic efficiencies must be improved to the point of commercial viability. This involves rapid prototype manufacture as well as basic analysis to predict the potential benefits of changes in the compressor design. This section outlines the various rotating spool compressor prototype configurations explored and the design methodology used for iteration.

The rotating spool compressor platforms that were evaluated are listed in Table 1 along with their respective displacements. The evolution of these prototypes encompassed two major trends. The first was an effort to improve the stability of the rotating assembly in order to be able to operate at high speed while also improving the manufacture of the parts to reduce internal clearances and improve volumetric performance. The second trend was to change the overall geometry to improve the drag losses relative to the displacement increasing the overall isentropic efficiency.

A key element in the performance of the spool compressor is in the frictional losses in the active sealing areas relative to the compressor displacement volume. A dimensionless number was developed, called the Zsoro number, which relates the effective frictional load to the total displaced volume of the compressor.

$$\boldsymbol{Zsoro = R_{esr} \ * \ A_{ssa} / D_R} \tag{1}$$

Using the Zsoro number allowed for rapid evaluation of the potential performance improvement of the compressor based on its design geometry. To maximize compressor performance the goal was to decrease frictional losses per revolution and simultaneously increase mass flow. This tradeoff is represented by a decreasing Zsoro number. Therefore, the subsequent designs since RCP 3.1 have each had a smaller Zsoro number than the previous design platform. The Zsoro number for each rotating spool compressor design platform is given in Table 1.

Table 1: Rotating spool compressor platforms tested since 2010 with the displacement and Zsoro number of each platform.

Compressor #	Displacement	Zsoro Number
RCP2.1	61.45 cc/rev	8.79
RCP3.1	10.24 cc/rev	44.07
RCP4.2	46.54 cc/rev	1.63
RCP5.3	39.33 cc/rev	.168

As each successive prototype compressor was designed there were efforts made to improve factors, not accounted for by the Zsoro number, which would additionally improve the overall performance. Each prototype had incrementaly better bearing designs, improved intake and discharge porting, improved internal clearances allowing better management of the internal leakage paths. In addition to the design geometry accounted for by the Zsoro number key elements to the overall performance of the compressor are the active sealing elements which include the tip seals and the spool seals. A detailed discussion of those are beyond the scope of this paper but are addressed in Kemp et al. (2012) and Kemp et al. (2012a). Additional comprehensive modeling efforts, which incorporate all of the design elements, of the spool compressor are also presented in Bradshaw et al. (2012), which is also outside of the scope of the current work.

3. PERFORMANCE EVALUATION

3.1 – Test Method

Testing was conducted on a hot gas bypass type test stand, shown in Figure 2, the compressor power is supplied by an external motor and is transmitted through a non-contacting type strain gage, which is also equipped with a speed sensor. The gas flows through an external oil separator and after separation the gas mass flow is measured by a coriolis type mass flow meter. The gas stream is separated with a portion of the gas throttled in the hot gas bypass line to maintain compressor suction pressure the remainder of the gas is condensed by means of a water cooled condenser to control discharge pressure and cool the intake gas stream. Oil is returned to the compressor case via an external pump with supply from the oil separator. 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.

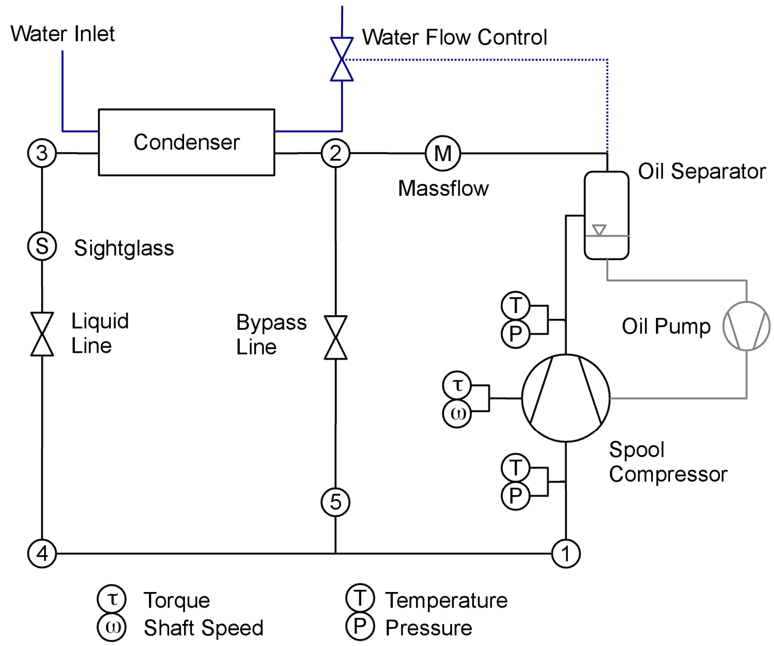

Figure 2: Schematic diagram of hot-gas bypass compressor load stand used to test rotating spool compressor prototypes.

3.2 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:

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

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):

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

3.2 Uncertainty Analysis

The uncertainties associated with the measurements collected from the load stand are presented in this section. These measurements include temperature, pressure, refrigerant mass flow rate, compressor speed, and compressor torque. The absolute uncertainty associated with each measurement is given in

Table 2. Using the absolute uncertainty the propagation of uncertainty associated with the reported efficiency values is calculated using an uncertainty propagation analysis (Fox et al., 2004). The volumetric and overall isentropic efficiencies have average uncertainty values of 0.9% and 0.5%, respectively.

Table 2: Uncertainty of Experimental Measurements.

Measurement	Sensor Type	Uncertainty
Temperature	K-type Thermocouple	1 °C
Suction Pressure	Omega PX 4202-200	3.4 kPa
Discharge Pressure	Omega PX 4202-600	10.3 kPa
Refrigerant Mass Flow	CMF 025	0.35%
Compressor Speed	Honeywell 1604	5 rpm
Shaft Torque	Honeywell 1604	0.1%

4 PERFORMANCE TRENDS

Figure 3 shows the range of volumetric efficiency obtained for each compressor prototype tested. Looking at the Figure 3 it can be seen that the spool compressor can be sealed in the range of 55% to 94% η_{vol} over a Pr range of 2.5 – 4.5. The latter two prototype compressors (RCP 4.2 and RCP 5.3) have displayed volumetric efficiencies that are comparable with commercial compressor designs, with η_{vol} levels between 67% and 97% at operating Pr between 2.5 and 4.5. A portion of this success can be attributed to the design iteration utilizing the Zsoro number. However, the main accomplishment over the prototype progression was the ability to increase the operating speed range while maintaining mechanical stability, shown in Figure 4. This effort was facilitated by an improved bearing design that allowed a high degree of rotor stability. This improvement allowed for better control of the internal clearances in the machine and resulting consistently high volumetric efficiency.

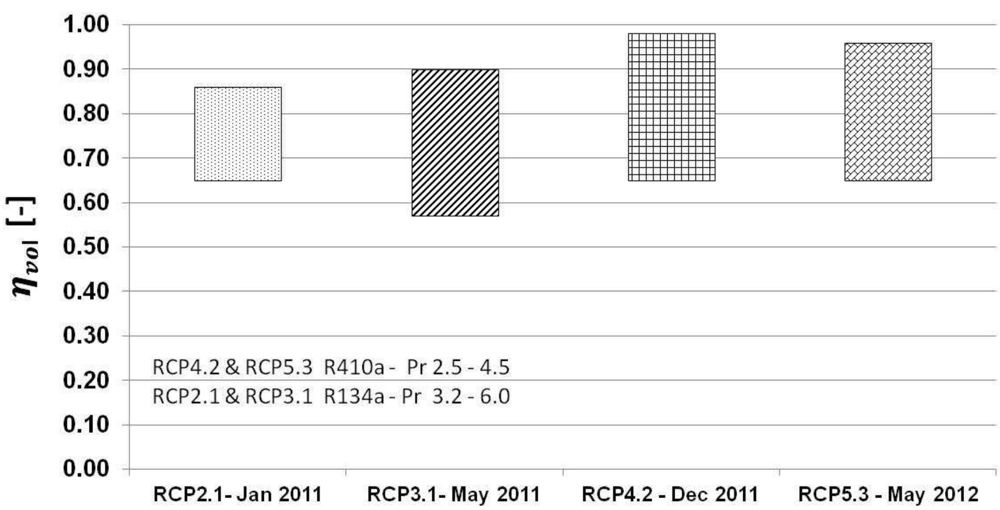

Figure 3: Volumetric efficiency evolution for each compressor prototype at various operating conditions.

In order to evaluate the effective leakage gaps and the impact on the compressor performance the compressor must function over a fairly wide speed range. Further, the volumetric efficiency is greatly affected by the speed and sealing ability of the various leakage gaps. Each gap was reduced over the different prototype designs in order to achieve the most effective clearance sealing arrangements.

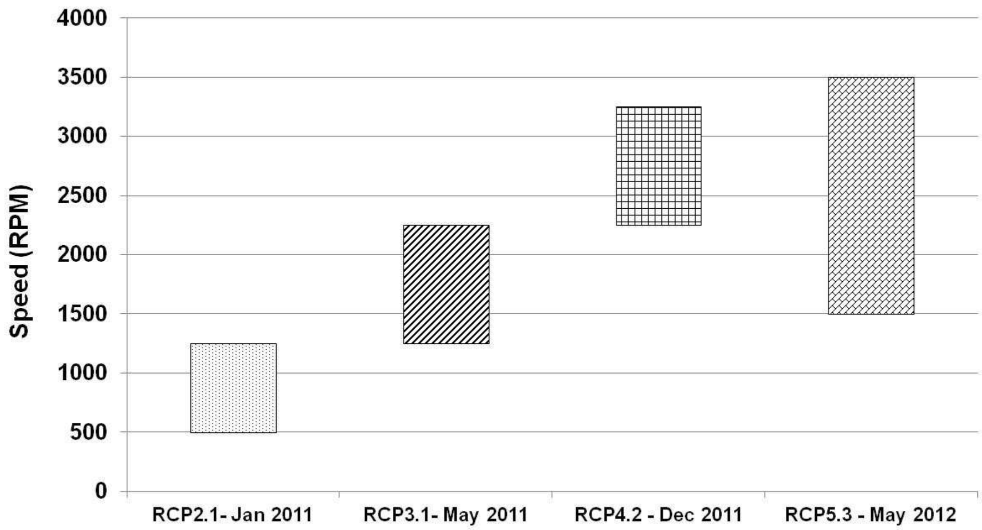

Figure 4: Compressor speed range evolution for each compressor prototype at various operating conditions.

The data in Figure 3 and Figure 4 shows the evolution of the effective operating range with volumetric efficiency in the acceptable range of the compressor for the various design prototypes. The end goal of the compressor design being the ability to operate effectively from nominal 2 pole to 4 pole speeds with good volumetric efficiency enhancing its ability to operate on variable speed applications. While RCP 2.1 displayed limited volumetric efficiency and operating speed by RCP 5.3 the prototype designs show a level of volumetric performance that is comparable with commercial compressors over a wide operating speed range.

Figure 5 shows the prototype compressors overall isentropic efficiency evaluated against the Zsoro number at standard operating conditions.

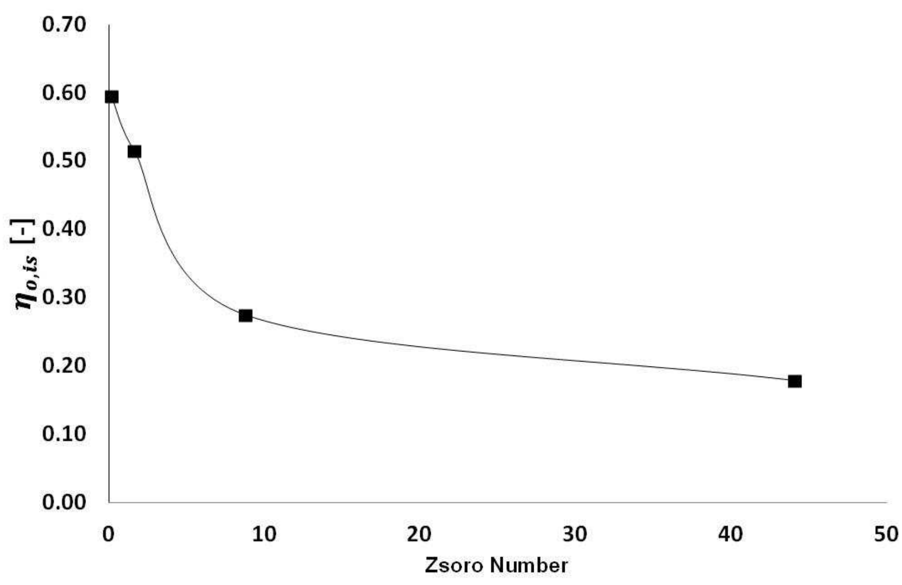

Figure 5: Isentropic efficiency vs. Zsoro number at standard operating conditions (experimental uncertainty within marker width).

The volumetric efficiency of any given design is high enough at standard operating conditions for each prototype that it is assumed that 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. Based on the very small Zsoro number of the prototype RCP 5.3 it appears that the upper limit of the efficiency improvement becomes limited at Zsoro number below .2. It is likely that other torque loses in the geometry outside of what is considered in the Zsoro calculation become dominate at values below .2 and further reduction in the number will yield ever smaller gains.

5. 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 RCP 4.2 and RCP 5.3 with a commercially available scroll and rotary compressor at various operating conditions

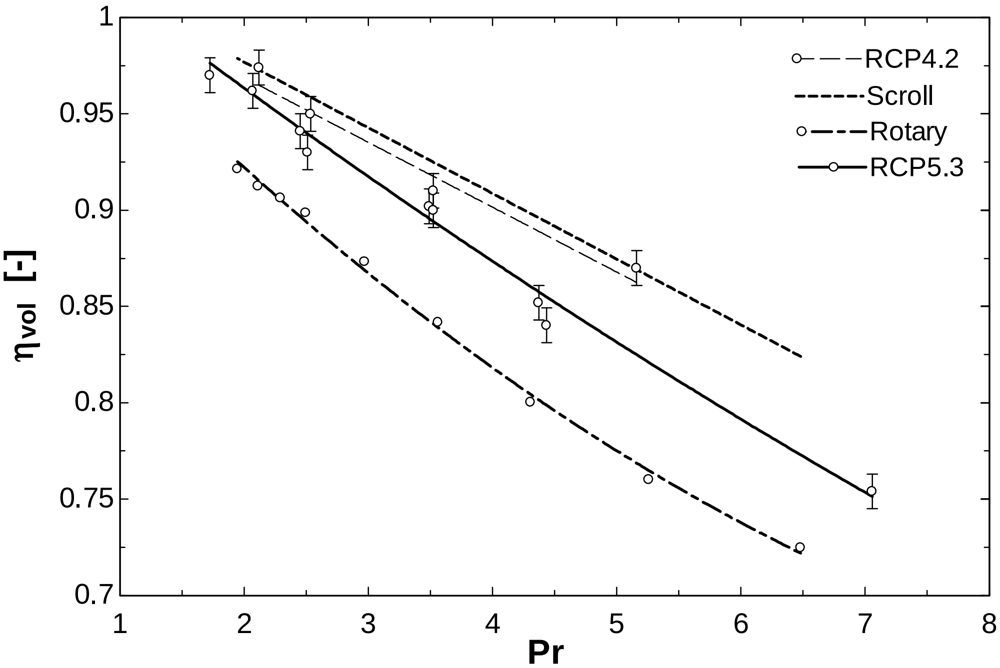

Figure 6: Rotary spool compressor prototypes volumetric efficiency vs. commercial compressor technologies.

The prototype RCP4.2 was directly comparable to a scroll compressor even allowing for the presence of a discharge valve and some clearance volume on top of the vane that carries over from the discharge to suction chambers. The slope of the volumetric line or the Loss per unit of pressure is 3.4%/Pr, the same as the commercial scroll compressor. The prototype RCP5.3 was designed to allow slightly more internal clearance in favor of some reduced torque, therefore the volumetric efficiency is slightly lower. However, this compressor still compares favorable with the other current technologies employed in the market. The peak volumetric efficiency at 2.5 Pr is 2% less than the scroll compressor while the loss per unit of pressure is 3.9%. When compared to the rolling piston type the slope of the pressure loss is comparable while the volumetric efficiency is 4 points higher.

Figure 7 is a comparison of the overall isentropic efficiency of the same compressors that were compared in Figure 6 for the volumetric efficiency. The two prototype spool compressors RCP4.2 and RCP5.3 show efficiencies that are within 3% of each other with the RCP5.3 being slightly higher. The trend in the slope of the overall isentropic efficiency curves are 4.5, 4.3 and 3.7%/Pr respectively for the RCP5.3, Scroll Compressor and Rotary compressor. The absolute figures for the efficiency show the RCP5.3 at a Pr of 2.5 to be 7.5 points higher than a rotary compressor and 5 points lower than a scroll compressor. It should be noted that the spool compressor test data does not include the motor efficiency but does include an internal oil system.

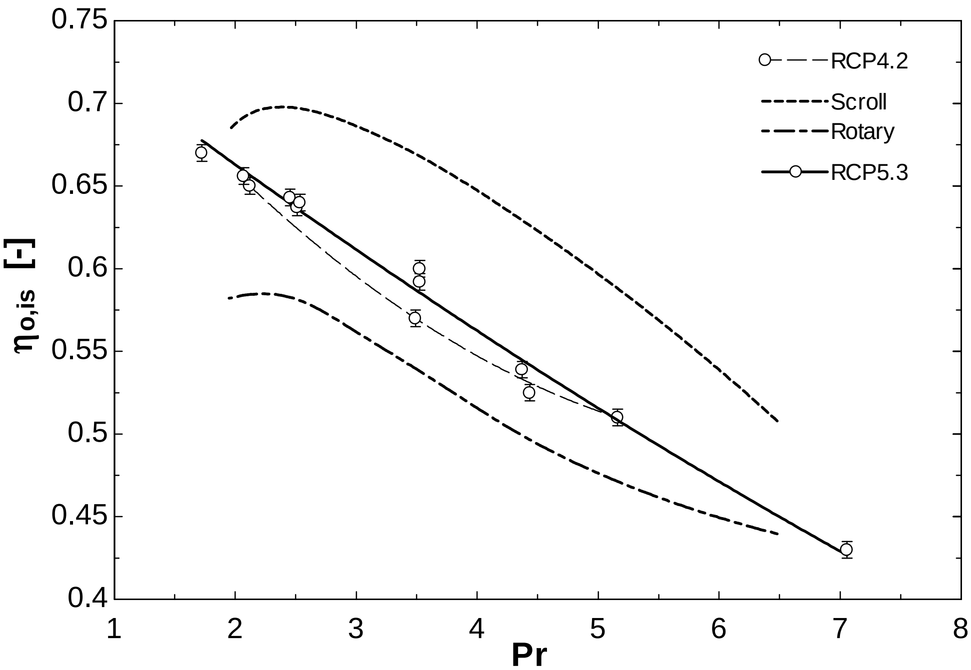

Figure 7 - Rotary spool overall isentropic efficiency vs. other technologies

6. CONCLUSIONS

The prototype design evolution of the rotating spool compressor is presented. The test method for experimental testing is also presented with the relevant experimental uncertainty. In order to be able to predict the impact of the geometric design on the performance of the spool compressor a dimensionless number, the Zsoro number, is presented. Based on the correlation between the Zsoro number and the overall isentropic efficiency we can see that it is a reasonable predictor of compressor performance. It can also be seen that a Zsoro number < 2 produces reasonable results and can serve as a design indicator for a compressor that can be further improved by optimizing the other performance related features. Achieving a Zsoro number less than two is limited by two factors, the ability to manufacture the compressor L/D larger and the ability of the longer internal tip seal to function properly. These require additional investigation as presented in Kemp et al. (2012) and Kemp et al. (2012a). For normal air conditioning operation a Zsoro number greater than two is impractical as shown by the lower performance of the spool compressor prototypes and should therefore be avoided.

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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