Purdue University
Purdue e-Pubs

International Compressor Engineering Conference

School of Mechanical Engineering

2014

Performance and Operating Characteristics of a Novel Positive-Displacement Oil-Free CO2 Compressor

Orkan Kurtulus
Mechanical Engineering, Purdue University, West Lafayette, IN, orkan@purdue.edu

Bin Yang
Mechanical Engineering, Purdue University, West Lafayette, IN, yang62@purdue.edu

Domenique Lumpkin
Mechanical Engineering, Purdue University, West Lafayette, IN, dlumpkin@purdue.edu

Eckhard A. Groll
Mechanical Engineering, Purdue University, West Lafayette, IN, groll@purdue.edu

Lee Jestings
S-Ram Dynamics, Franklin, TN, lee@S-RAM.com

See next page for additional authors

Authors
Orkan Kurtulus, Bin Yang, Domenique Lumpkin, Eckhard A. Groll, Lee Jestings, and Ricardo Conde

Performance and Operating Characteristics of a Novel Positive-Displacement Oil-Free Carbon Dioxide Compressor

Orkan KURTULUS[1*], Domenique LUMPKIN[1], Bin YANG[1], Eckhard A. GROLL[1], Lee JESTINGS[2], Ricardo CONDE[3]

[1]Ray W. Herrick Laboratories, Purdue University
West Lafayette, IN USA
[1]*Email:orkan@purdue.edu

[2]S-RAM Dynamics
Franklin, TN USA
Email:lee@s-ram.com

[3]ReGen Power Systems
New Salem, MA USA
Email: rconde@rgpsystems.com

ABSTRACT

In this paper, a prototype positive-displacement oil-free carbon dioxide (CO_2) compressor with a novel mechanical linkage system is introduced and tested. Preliminary compressor test results of the volumetric efficiency and overall isentropic efficiency are presented. The novel compressor design introduces a new low-friction, variable-displacement drive mechanism. The displacement of the compressor can be varied mechanically while maintaining a minor constant head clearance, eliminating the need for variable speed motors and variable frequency drives. The compressor was designed and manufactured to provide cooling capacities from 10 kW to 100 kW by changing the displacement of the piston. A test stand was constructed to map the compressor isentropic efficiency, volumetric efficiency, mass flow rate, power consumption and discharge temperature. The test stand is based on a hot gas bypass design, in which a part of the discharged refrigerant flow bypasses the condenser, whereas the other part of the flow changes phase as it flows through the condenser. The two streams are mixed to obtain the desired compressor superheat at the suction side of the compressor. The bypass valve enables control of the suction pressure as the discharge valve controls the discharge pressure. The prototype compressor was tested at pressure ratios (PR) of 1.5, 2, 2.5, and 3 at 25 Hz (≈750 rpm), and additionally at PRs of 1.5, 2, and 3 at 20 Hz (≈600 rpm). Based on the test results, the maximum isentropic efficiency is 76% at 20 Hz (592 rpm) and a PR of 2, while the volumetric efficiency is 88%.

1 INTRODUCTION

Climate change is an ever-evolving concern that over the past several decades has affected developed and developing countries alike. It has been determined that refrigerant leakage from various heating, ventilation, air conditioning and refrigeration (HVAC&R) systems over the years has significantly contributed to this environmental calamity. Government policies and regulations worldwide are forcing the HVAC&R industry to derive refrigerants with minimal ozone depletion effects and low global warming potential (GWP) to replace existing hydrofluorocarbons (HFCs) with significant global warming potential, which once replaced chlorofluorocarbons (CFCs) with high ozone depletion effects. Therefore, the industry has moved towards the development of hydrofluoroolefins (HFOs) and is exploring natural refrigerants of times past. Natural refrigerants such as air, carbon dioxide and ammonia have insignificant GWPs and no ozone depletion effects compared to their artificial counterparts. However, system performance in traditional HVAC&R systems tends to suffer with the use of natural refrigerants. As a result, novel equipment, such as the carbon dioxide (CO_2) compressor characterized in this paper, have been developed to yield high-performance, energy-efficient systems using natural refrigerants.

Research activities towards developing CO_2 compressors have increased drastically over the last two decades. Since the transcritical CO_2 cycle operates at much higher absolute pressures as compared to the conventional vapor compression cycles, it is necessary to develop new compressors or modify existing ones. The design of a new compressor for automotive air conditioning was carried out within the European RACE project. Based on the experiences with high pressure in hydraulic machinery, a wobble-plate compressor was developed by Jorgensen (1998) with a capacity modulation that uses a self-adjusting angle for the wobble plate depending on suction pressure. At the same time, a two-stage compressor based on the swash-plate design was developed by Inagaki et al. (1997) in Japan. Kruse and Suess (1996) and (1998) thoroughly investigated the wobble-plate compressor that was developed within the RACE project with respect to optimum design for capacity modulation. These investigations illustrated that due to the low pressure ratio, a high energetic performance can be obtained. However, because of the high absolute pressure differences, leakage has a strong influence on the CO_2 compressor performance and thus, compressors with a good sealing and small leakage areas are best suited for this application. Suess (1998) also recommended that reciprocating compressors with a small number of cylinders are more suitable for this application than rotating compressors, and that CO_2 compressors should have a larger stroke-to-bore ratio than the conventional design used for HFC-134a refrigerants. This work also revealed that the piston pin bearing is critical to achieve a high reliability of the driving mechanism as it experiences large forces.

Some early investigations of CO_2 compressors focused on the design issues associated with the use of CO_2 in hermetic-type compressors (Fagerli, 1996a; Fagerli, 1997). Others focused on the modification of existing HCFC-22 compressors for use with carbon dioxide (Adolph, 1995; Fagerli, 1996b; Koehler et al., 1997; Hwang and Radermacher, 1998). In other studies, prototype designs of hermetic compressors for use with carbon dioxide have been built and analyzed. Tadano et al. (2000) developed a prototype hermetic two-stage rolling piston compressor with a 750 W cooling capacity and studied the motor frequency, superheat, and pressure ratio as well as the durability over 1000 hours. This compressor was considered a “first cut” device and was used as the basis to develop larger compressors for heat pump water heating, refrigerating, or air-conditioning applications. The compressor was designed to operate between a low pressure of 3 to 4 MPa and a high pressure of 10 MPa. The compressor shell diameter was 117.2 mm, the height was 244.3 mm, and the displacement was 2.633 cm^3. A two-stage compression with two rolling pistons was chosen to maintain small pressure differences across each compression stage. The intermediate pressure was selected to be 5 to 6 MPa. The inside of the hermetic shell was at intermediate pressure to minimize the gas leakage between the compressing chambers and the inner space of the shell. The authors reported isentropic efficiencies of up to 88% disregarding motor and shell losses. Hasegawa et al. (2000) constructed and analyzed a prototype hermetic CO_2 scroll compressor with a cooling capacity of 4.3 kW with a compressor displacement of 7.23 cm^3. The compressor was fabricated by reducing the wrap height of a medium capacity R-410A scroll compressor. The compressor performance was evaluated at a broad range of air-conditioning operating conditions. The compressor had a volumetric efficiency of up to 86% and an overall compressor efficiency of up to 47%. The volumetric efficiency of the CO_2 scroll compressor was expected to be substantially lower than the original R-410A scroll compressor due to the large absolute pressure difference between suction and discharge pressure; however, the measured difference in volumetric efficiency between the CO_2 and R-410A compressor was negligible.

The objective of this paper is to introduce the performance of a prototype positive-displacement, open-drive reciprocating CO_2 compressor with a cooling capacity of 10 to 100 kW for use in power generation, heat pumping, and marine applications.

2 NOVEL DRIVE MECHANISM

The prototype compressor is using the Sanderson Rocker Arm Motion mechanism (S-RAM), which is a simple new drive mechanism. The mechanism converts reciprocating to rotary motion, producing high efficiency in both directions without the energy-robbing side forces on the pistons or crossheads common to crankshaft, swash plate or wobble plate drive mechanisms. The drive mechanism can vary the piston stroke while maintaining a fixed head clearance, which is not possible with other drive mechanisms, and is critical for high-performance compression. The S-RAM mechanism can be configured either as single-side variable or with opposed pistons. Figure 1 shows a simple kinematic diagram of the S-RAM compressor in an opposed piston configuration. The S-RAM near straight-line piston motion and piston sealing method enables the elimination of oil lubrication in the case and piston area, providing significant compressor and CO_2 system benefits.

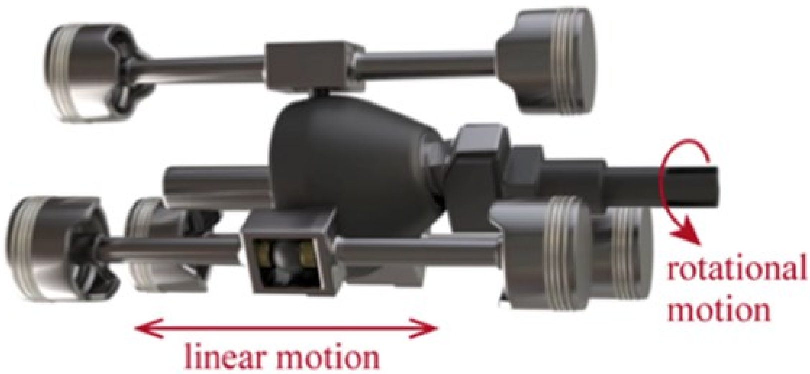

Figure 1: S-RAM Opposed configuration (S-Ram.com, 2011)

3 EXPERIMENTAL SETUP

The performance of the prototype positive-displacement reciprocating CO_2 compressor was tested using a hot-gas bypass test stand. A schematic of the test stand is presented in Figure 2. The test stand consists mainly of the compressor, condenser, and several control valves. The idea behind the test stand concept is to anchor the intermediate pressure below the critical pressure in the two-phase region by condensing a fraction of the refrigerant flow. Using this stable anchoring pressure, the suction and discharge pressures are controlled by using appropriate metering valves in the discharge line and bypass line.
Figure **3** illustrates the state points of the ideal process cycle in a logarithmic enthalpy-pressure diagram. The compressor discharges high-pressure, high-temperature CO_2 at state point 1, which is throttled to the intermediate pressure at state point 2. After passing through the flow meter, the CO_2 flow is split. Most of the flow goes through the bypass loop, while the remaining flow enters the primary loop. The bypass loop includes the bypass metering valve, where the fluid is throttled to the suction pressure (state point 6). The primary loop condenses the CO_2 in the water-cooled condenser. Subcooled liquid at state point 3 exits the condenser and is throttled through the primary metering valve to the suction pressure (state point 7). The two fluid streams are then combined just before the mixing chamber and exit the mixing chamber at state point 5. After the mixing chamber, the fluid stream flows into the compressor body to pick up heat from the body and then flows to the suction distributor at state point 4.

A schematic of the load stand indicating all relevant components is shown in Figure 2. All control valves are based on manually operated metering valves. In order to obtain the desired operating conditions, such as suction pressure, superheat, and discharge pressure, three control valves are used. The metering valve in the discharge line is used for controlling the discharge pressure. The metering valve in the bypass line is used for controlling the suction pressure. The metering valve in the liquid line is used to control the superheat by controlling the amount of liquid refrigerant mixed with the hot gas from the bypass line. The control strategy for the load stand has been successfully implemented for several other compressor load stands of the same design (Chen et al., 2004; Sathe et al., 2008; Bradshaw et al., 2011; Bradshaw & Groll, 2013a; Yang et al., 2013; Yuanpei et al., 2013; Bradshaw & Groll 2013b; Bell et al., 2013; Holloway et al., 2010; Mathison et al., 2013; Kim & Groll, 2007).

The test stand is constructed using stainless steel piping that is designed to withstand the maximum pressure of 135 bar (2000 psia). To be able to control the rotational speed of the motor, a variable speed drive (VFD) is used. The refrigerant mass flow is determined by a Coriolis-effect mass flow meter (±0.35%). The compressor total power consumption is measured with a power meter (± 0.5 %). The temperature (T) and pressure (P) at each state point are measured using T-type thermocouples (±1 K) submerged in the flow and pressure transducers (±0.5%), respectively. A conventional data acquisition system connected to a personal computer is used for collecting data.

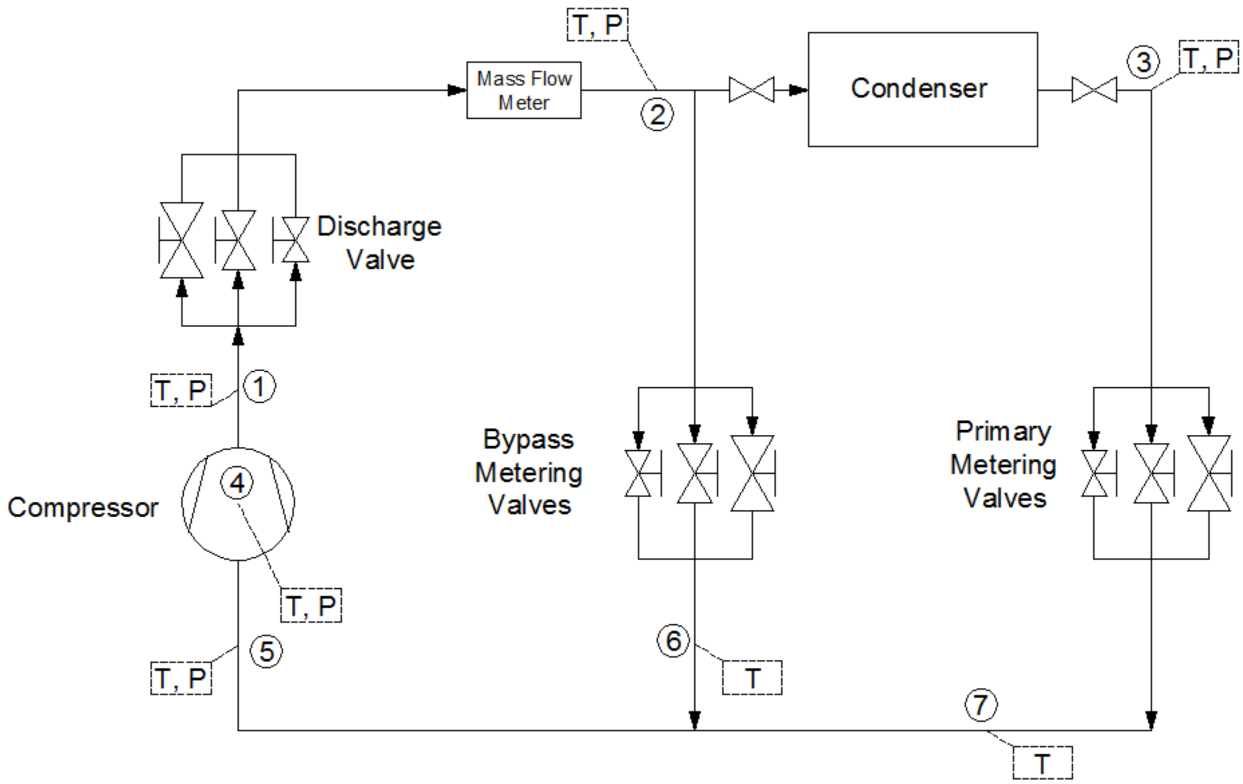

Figure 2: Schematic of the carbon dioxide compressor experimental setup

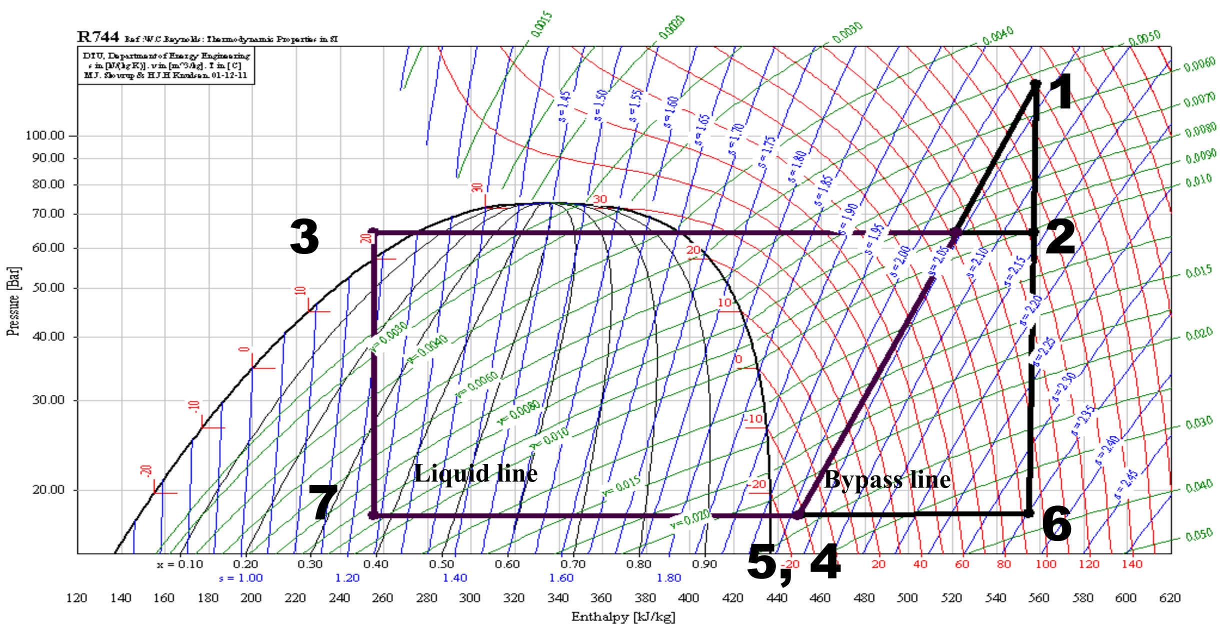

Figure 3: Load stand cycle in a P-h diagram

4 RESULTS AND ANALYSIS

4.1 Compressor specifications and test method

The prototype compressor is a single-stage, single-ended, oil-free open-drive reciprocating compressor with an estimated cooling capacity of 80 kW at 90 bars and a PR of 3, while running at 1800 rpm. The compressor is the first prototype of the single-ended piston mechanism design with 5 pistons, 345 cm^3 displacement and 37.25 m^3/hr @ 1800 rpm. The motor power is 60 HP, 460 V, 69 amps at 1800 rpm. At full load, it can generate a torque of 240.65 N-m.

The compressor body was modified to remove heat by connecting the suction line to the body and run refrigerant through the body. The superheated refrigerant was then connected to the distributor of the compressor suction port. The compressor body case is designed to hold 35 bars in the case. Because of safety concerns, a relief valve is attached to the compressor case, which was set to release CO_2 into the environment at pressures of higher than 35 bars.

The desired test matrix to measure the compressor performance has 25 test points; but, so far, only 6 tests have been completed. The test matrix can be seen in Table 1. During the operation, the manual metering valves were adjusted until the system reached the desired steady-state test conditions. Steady-state operating conditions were determined when the suction and discharge pressures were stable within ±2% and the temperature fluctuation was not more than ±1 K. When the desired test conditions were achieved, data was recorded for at least 5 minutes using the data acquisition system with a sampling rate at 2500 Hz.

Table 1: Prototype compressor test matrix

Test order	Speed [rpm]	Pressure ratio [-]	$P_{suction}$ [Bar]	$P_{discharge}$ [bar]
1	600	1.5	25	37.5
2	600	2	25	50
3	600	3	25	75
4	750	1.5	32	37.5
5	750	2	32	50
6	750	3	25	75

After each test was completed, the collected data was processed and thermodynamic properties were calculated using average steady-state data, REFPROP (Lemmon et al., 2013) and EES (EES, 2013). A program was developed in EES to calculate the compressor performance using Equation 1 and Equation 2. Equation 1 defines the compressor volumetric efficiency, which is a measure of the actual volumetric flow rate relative to the theoretical volumetric flow rate based on the compressor cylinder geometry.

$$\eta_{vol} = \frac{\dot{m}_{measured} v_{dis}}{\dot{V}_D} \tag{1}$$

Equation 2 is the overall isentropic efficiency, which is the ratio of the power consumption needed for an adiabatic and reversible process operating between the actual inlet state (P_4, v_4) and actual outlet pressure (P_1) and with the actual mass flow rate to the actual compressor power consumption.

$$\eta_{is,o} = \frac{\dot{m}_R (h_{dis,s} - h_{suc})}{W_{elec,measured}} \tag{2}$$

4.2 Performance results and comparisons

Based on the measurement results and the calculations, the compressor volumetric efficiency and overall isentropic efficiency were plotted relative to pressure ratio. Each set of data corresponds to motor speeds of 600 rpm and 750 rpm. The overall isentropic efficiency can be seen in Figure 4. The highest efficiency of 76% was achieved at 600 rpm, a PR of 2, where the average suction pressure was 24.3 bars and the discharge pressure was 50.08 bars. The lowest overall isentropic efficiency of 55% was achieved at 750 rpm, a PR of 2.5, where the average suction pressure was 24.3 bars and the discharge pressure was 60.75 bars. The overall isentropic efficiency was calculated using the power meter reading, which includes motor losses and VFD losses in the power consumption.

The compressor volumetric efficiency as a function of pressure ratio can be seen in Figure 5. As the pressure ratio increases, the volumetric efficiency decreases almost linearly, which is typical behavior for reciprocating compressors. This decrease is due to internal superheat, pressure drop in valves and flow passages and back flow through valves. The volumetric efficiency is typically 70-90% at low pressure ratios, 60-80% at medium pressure ratio, and 40-70% at high pressure ratio. According to test data, the achieved volumetric efficiency was always above 80%, which is a promising result for the given compressor and indicates a very small head clearance.

In Figure 6 and Figure 7, the volumetric and overall isentropic efficiency performance measurements of the prototype compressor are compared with other prototype CO_2 compressors, which were tested at the Ray W. Herrick Laboratories in previous studies (Hubacher & Groll, 2003; Hubacher et al., 2003; Holloway et al., 2010). It can be seen from these figures that the S-RAM prototype compressor demonstrates a good performance compared to the other prototype compressors. It should be noted that the tested S-RAM prototype compressor is the only oil-free compressor of this group.

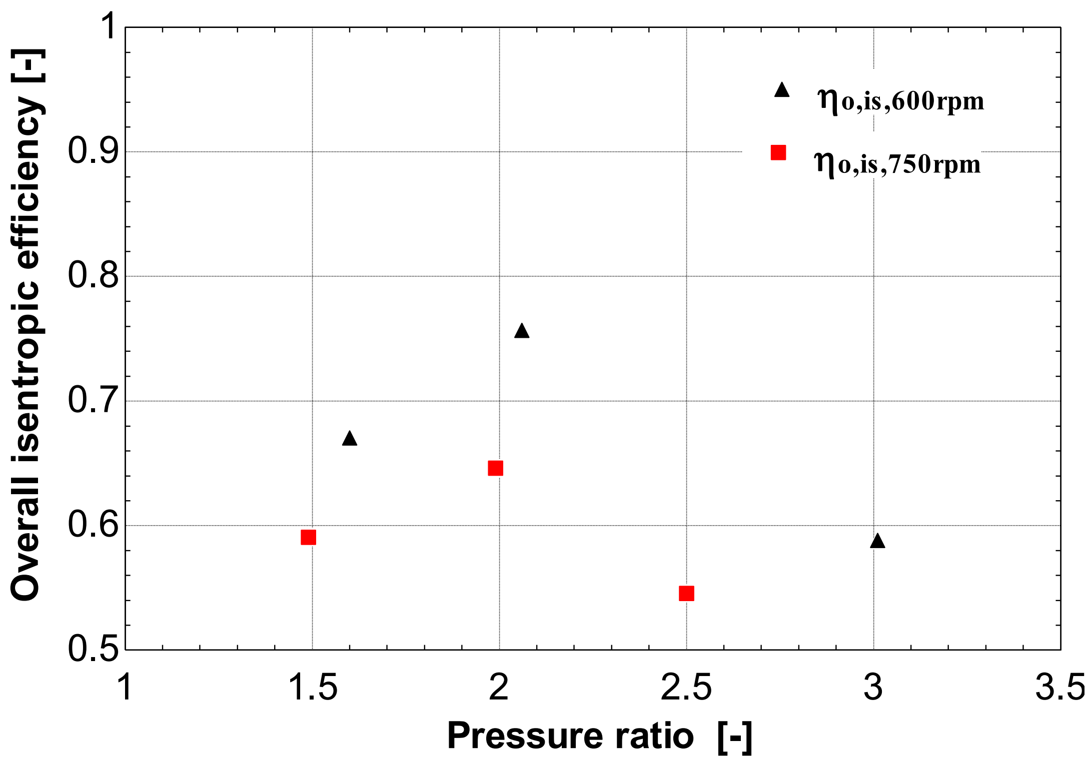

Figure 4: Compressor isentropic efficiency over varying pressure ratio at 600 rpm and 750 rpm

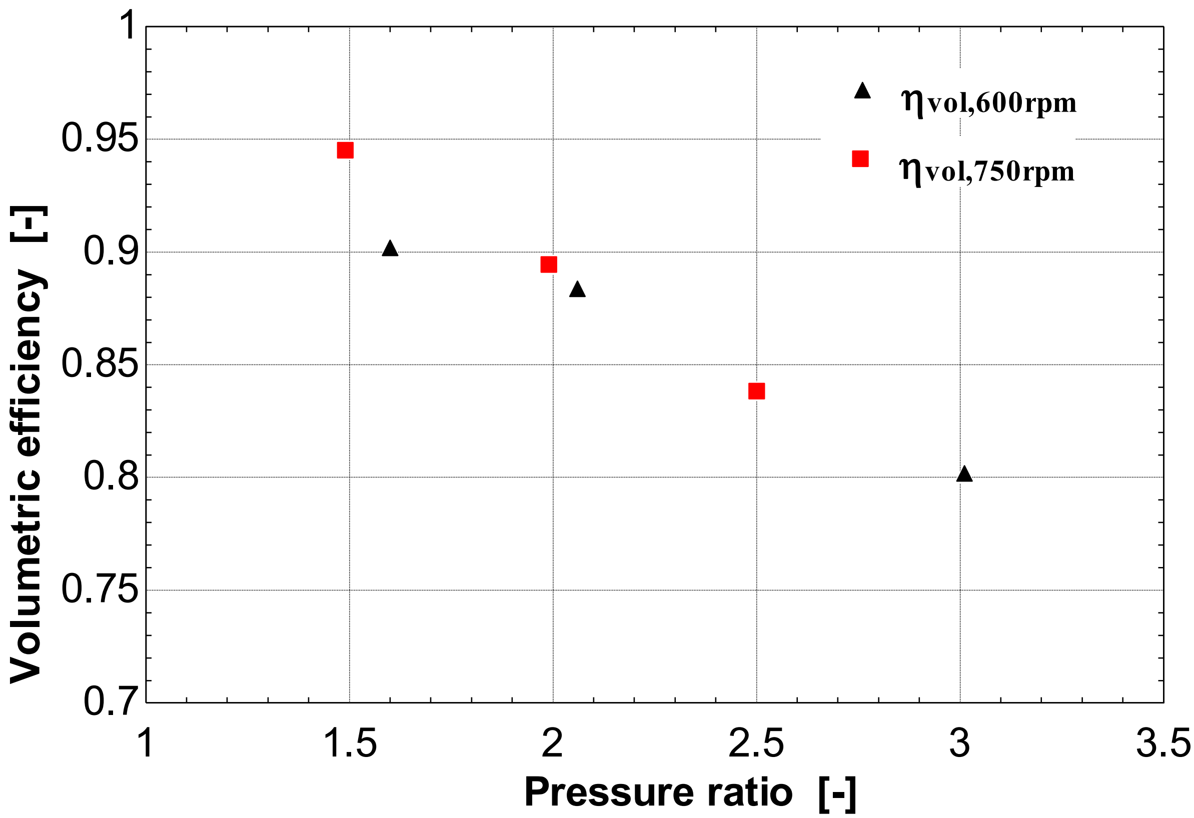

Figure 5: Compressor volumetric efficiency over varying pressure ratio at 600 rpm and 750 rpm

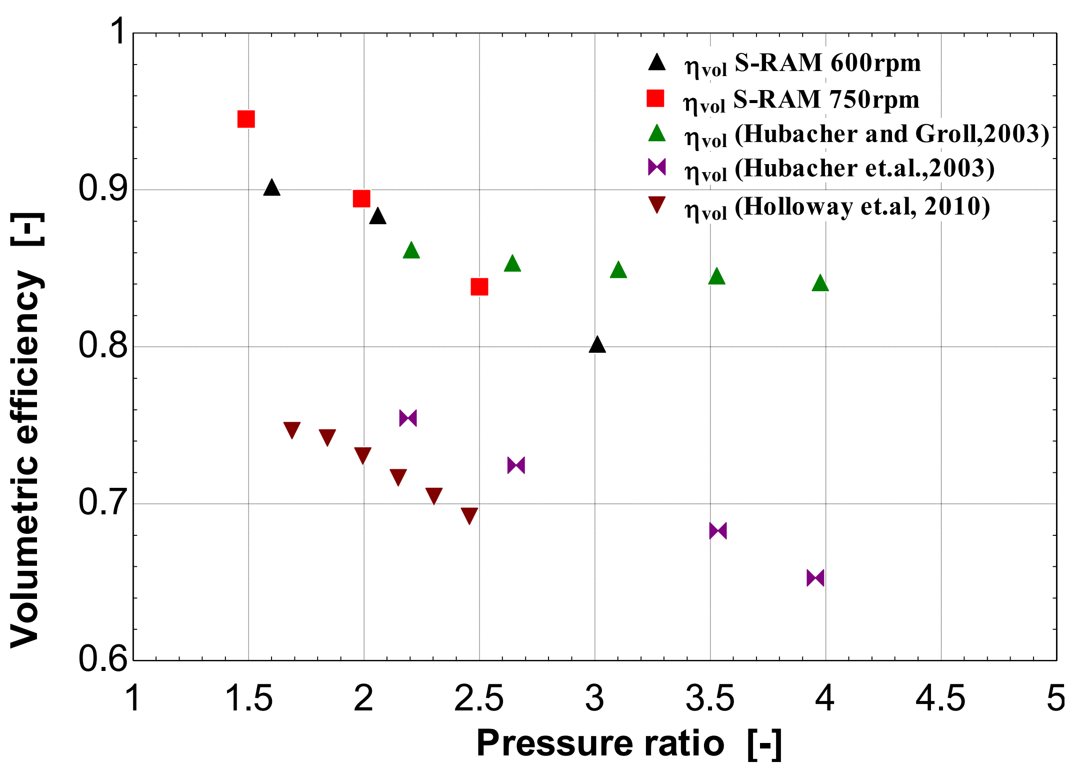

Figure 6: Volumetric efficiency comparison between predecessor and other prototypes tested

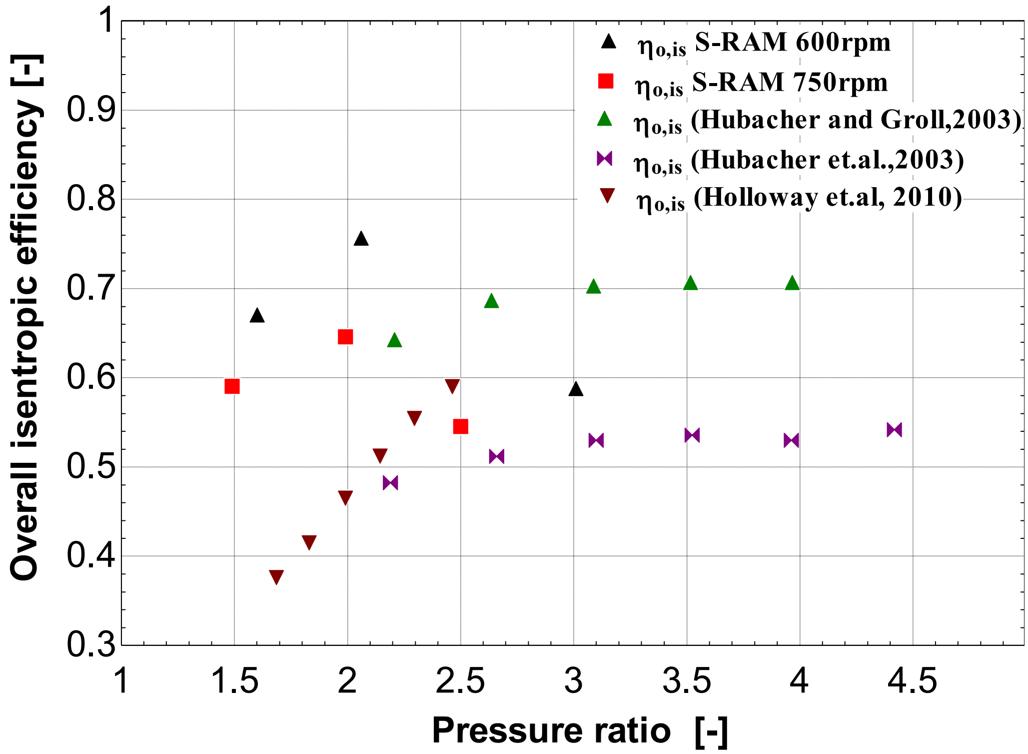

Figure 7: Overall isentropic efficiency comparison between predecessor and other prototypes tested.

5 CONCLUSIONS

A novel prototype compressor was tested using a hot-gas bypass test stand. The oil-free compressor uses the Sanderson Rocker Arm Motion mechanism (S-RAM), which converts reciprocating to rotary motion. Results for the measured volumetric and overall isentropic efficiencies are reported. According to the test data, maximum and minimum overall isentropic efficiencies are 75.66% and 54.55%, respectively. A volumetric efficiency of 94.52% to 80.18% as a function of pressure ratio was measured, which proves that the measured compressor has a small head clearance, and low piston and valve leakages. The performance tests indicate promising results for this novel compression technology, especially considering that the tested compressor was the first prototype. It is expected to achieve higher efficiencies with the next prototype.

NOMENCLATURE

η	Efficiency	[-]
m	Mass flow rate	[kg/s]
υ	Specific volume	[m^3/kg]
V	Total cylinder volume	[m^3]
H	Enthalpy	[kJ/kg-K]
W	Power consumption	[kW]

Subscript

measured	Measured data
dis,s	Discharge, isentropic
D	Determined
suc	Suction
elec,measured	Measured electric

REFERENCES

Adolph, U. (1995). Einsatz von CO2 als Kaeltemittel in Schienenfahrzeugen. *FKW Seminar XVII*. FKW GmbH, Hannover, Germany.

Chen, Y., Groll, E. A., & Braun, J. E. (2004) Modeling of hermetic scroll compressor: Model validation and application. *HVAC&R Research*, *10*(3), 307-32.

Bell, I. H., Groll, E. A., Braun, J. E., & Horton, W. T. (2013) Experimental testing of an oil-flooded hermetic scroll compressor. *International Journal of Refrigeration, 36* (7), 1866-1873.

Bradshaw, C.R., Groll, E.A.,& Garimella, S.V. (2011) A comprehensive model of a miniature-scale linear compressor for electronics cooling. *International Journal of Refrigeration*, 34(1), 63-73.

Bradshaw, C. R., & Groll, E. A. (2013) A comprehensive model of a novel rotating spool compressor. *International Journal of refrigeration*, 36(7), 1974-1981.

EES version 9.434-3D, 2013, Madison, WI: F-Chart Software.

Fagerli, B.E. (1996a). An Investigation of Possibilities for CO2 Compression in a Hermetic Compressor. *Proc. IIR Conf., Applications for Natural Refrigerants*. Aarhus, Denmark, Sept. 3-6, 639-649.

Fagerli, B.E. (1996b).Development and Experiences with a Hermetic CO2 Compressor. *International Compressor Engineering Conference.* Purdue University West Lafayette, IN, July 23-26, 229-235.

Fagerli, B.E. (1997). CO2 Compressor Development. Proc. *IEA Heat Pump Centre/IIR Workshop on CO2 Technology in Refrigeration, Heat Pump & Air Conditioning Systems*. Trondheim, Norway, May 13-14.

Hasegawa, H., Ikoma, M., Nishiwaki, F., Shintaku, H. & Y. Yakumaru. (2000) Experimental and Theoretical Study of Hermetic CO2 Scroll Compressor. *4th IIR–Gustav Lorentzen Conference on Natural Working Fluids,* Purdue University.West Lafayette, IN, July 25-28, 347-353.

Holloway, S., Horton T.W., Groll, E.A., Sherman, D., & Albertin M. (2010) Experimental Performance of a Prototype Carbon Dioxide Compressor. *International Compressor Engineering Conference.* Purdue University, West Lafayette, IN 2010-01-01T08:00:00Z.

Hubacher, B., & Groll, E.A. (2003)Performance Measurements of a Hermetic, Two-Stage Carbon Dioxide Compressor. *International Congress of Refrigeration*. Washington, DC, Paper 0029.

Hubacher, B.; Groll, E. A.; & Hoffinger, C. (2002) Performance Measurements of a Semi-Hermetic Carbon Dioxide Compressor. *International Refrigeration and Air Conditioning Conference*. Purdue University, West Lafayette, IN.

Hwang, Y., & R. Radermacher. (1998) Development of Hermetic Carbon Dioxide Compressor. *Proc. International Refrigeration Conf. at Purdue.* West Lafayette, IN, July 14-17, 171-176.

Inagaki, M., Sasaya, H. & Y. Ozakli. (1997).Pointing into the Future: Two-Stage CO2 Compression. *Proc. IIR Conf., Heat Transfer Issues in Natural Refrigerants.* College Park, MD, Nov. 6-7, 131-138.

Jorgensen, S. T. (1998) Variable Automotive CO2 Compressor. *International Refrigeration Conference* Purdue University. West Lafayette, IN, July 14-17, 159-164.

Kim, J.H.,& Groll, E.A., 2007, "Feasibility study of a bowtie compressor with novel capacity modulation", *International Journal of Refrigeration*, 30(8), pp.1427-1438.

Kruse, H., & J. Suess, 1996, "Research on the Behavior of Refrigeration Compressors Using CO2 as the Refrigerant, *International Compressor Engineering Conference*, Purdue University, West Lafayette, IN, July 23-26, pp. 223-228.

Koehler, J., Kaiser, H., & B. Lauterbach. (1997) CO2 as Refrigerant for Bus Air Conditioning and Transport Refrigeration. Trondheim, Norway, May 13-14.

Lemmon, E.W., Huber, M.L., & McLinden, M.O. (2013) NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 9.1, *National Institute of Standards and Technology, Standard Reference Data Program*, Gaithersburg.

Mathison, M.M., Braun, J.E., & Groll, E.A. (2013) Modeling of a novel spool compressor with multiple vapor refrigerant injection ports. *International Journal of Refrigeration*, 36(7), 1982-1997.

Sathe, A. A., Groll, E. A., & Garimella, S. V. (2008) Experimental Evaluation of a Miniature Rotary Compressor for Application in Electronics Cooling. *CTRC Research Publications* 2008-01-01T08:00:00Z

Song, Y., Groll, E.A., & Braun, J.E. (2013) Modeling and experimental validation of a multi-port vapor injected scroll compressor. *Master thesis* 2013-01-01T08:00:00Z.

Suess, J. (1999) Untersuchung zur Konstruktion moderner Verdichter für Kohlendioxid als Kältemittel. *Deutscher Kälte und Klimatechnischer Verein* (1999), ISBN-10: 3932715624.

Yang, B,Bradshaw, C.R., & Groll, E.A. (2013) Modeling of a semi-hermetic CO2 reciprocating compressor including lubrication submodels for piston rings and bearings. *International Journal of Refrigeration*, 36(7), 1925-1937