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## Experimental Performance of a Prototype Carbon Dioxide Compressor

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### ABSTRACT

Implementing carbon dioxide (CO<sub>2</sub>) as a natural refrigerant in heating and cooling applications is becoming a technology of increasing importance. This report discusses the performance of a CO<sub>2</sub> prototype compressor and compares the results with its predecessor prototype and other prototype compressors.

The compressor tested was a semi-hermetic, single-stage, axial configuration with an estimated cooling capacity of 10-14 kW. Testing was performed utilizing a hot-gas bypass test apparatus. While operating at steady-state conditions, parameters such as suction pressure, suction temperature, discharge pressure, discharge temperature, refrigerant mass flow rate, and compressor power consumption were measured and recorded. Thermodynamic properties and performance measures were calculated from the experimental data using software programs such as REFPROP and Engineering Equation Solver.

The results of the testing indicated volumetric efficiencies ranging from 67-75%; isentropic efficiencies ranging from 49-73%; and overall isentropic efficiencies of 38-59%. Significant improvements were seen relative to the predecessor prototype compressor. When compared to other carbon dioxide compressors tested, this prototype also demonstrated competitive performance marks.

### 1. INTRODUCTION

The transcritical cycle technology using carbon dioxide as the refrigerant has received increased attention as a possible replacement for conventional fluorocarbon-based vapor compression cycle refrigerant technologies in use during the past decade. In particular, four applications of carbon dioxide systems can be identified that show a comparable performance and may be economically feasible compared to conventional systems. The most prominent of these applications is automotive air conditioning. The second application is environmental control units (ECU), which are packaged air-to-air air-conditioners that are used in cooling of mission critical electronics and personnel. The third application that shows great promise for transcritical carbon dioxide systems is the one of heat pump water heaters. The fourth application is vending machines and glass door coolers. A detailed review of the latest developments with respect to these applications can be found in Groll (2006). While the compressor development for automotive air conditioning applications has advanced over the last several years, relatively little information is available in the literature with respect to hermetic or semi-hermetic compressors for use in the other three applications mentioned above. Therefore, laboratory testing was conducted which specifically focused on measuring the performance of carbon dioxide compressors. Details of this study can be found in Hubacher *et al.* (2002) and Hubacher and Groll (2003).

### 2. BACKGROUND ON CO<sub>2</sub>-COMPRESSORS

Most of the early investigations on hermetic-type CO<sub>2</sub> compressors focused on design issues associated with the use of CO<sub>2</sub> (Fagerli 1996a; Fagerli 1997), or the modification of existing compressors to use with carbon dioxide

(Adolph 1995; Fagerli 1996b; Koehler *et al.* 1997 and 1998; Hwang and Radermacher 1998). In more recent 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 cooling capacity of 750 W. The authors reported isentropic efficiencies of up to 88% not including motor and shell losses. Neksa *et al.* (2000) reported on the development of a series of semi-hermetic reciprocating compressors (single- and two-stage). A volumetric efficiency of up to 80 % and an isentropic efficiency of up to 60 % were reported for the compressor. In summary, most investigations with respect to carbon dioxide compressors have focused on developing a prototype compressor or a better understanding of the fundamental concepts of carbon dioxide compression. However, much of the information associated with these studies is not necessarily available to the public and detailed performance data of CO<sub>2</sub> compressors is still difficult to obtain. This information however, is needed to be able to evaluate the performance potential of the transcritical carbon dioxide technology on a system level.

### 3. CO<sub>2</sub> COMPRESSOR LOAD STAND

For the purpose of measuring the performance of carbon dioxide compressors, a compressor load stand for cooling capacities from 5 to 20 kW was utilized. The load stand is based on a hot-gas bypass cycle concept as can be seen from the P-h diagram in Figure 1. The key idea behind this 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 using appropriate metering valves in the discharge line and bypass line. The compressor discharges supercritical, high pressure, high temperature carbon dioxide, which is throttled to the intermediate pressure equal to the condensation pressure. After passing through the main flow meter the CO<sub>2</sub> 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 expansion valve where the fluid is throttled to the suction pressure level. The primary loop condenses the CO<sub>2</sub> in the water-cooled condenser. Subcooled liquid exits the condenser and is throttled through the primary expansion valve to the suction pressure level as well. The two fluid streams are then combined in a straight mixing pipe. A schematic of the load stand indicating all significant features is shown in Figure 2. All control valves manually operated which allows for very stable operation.

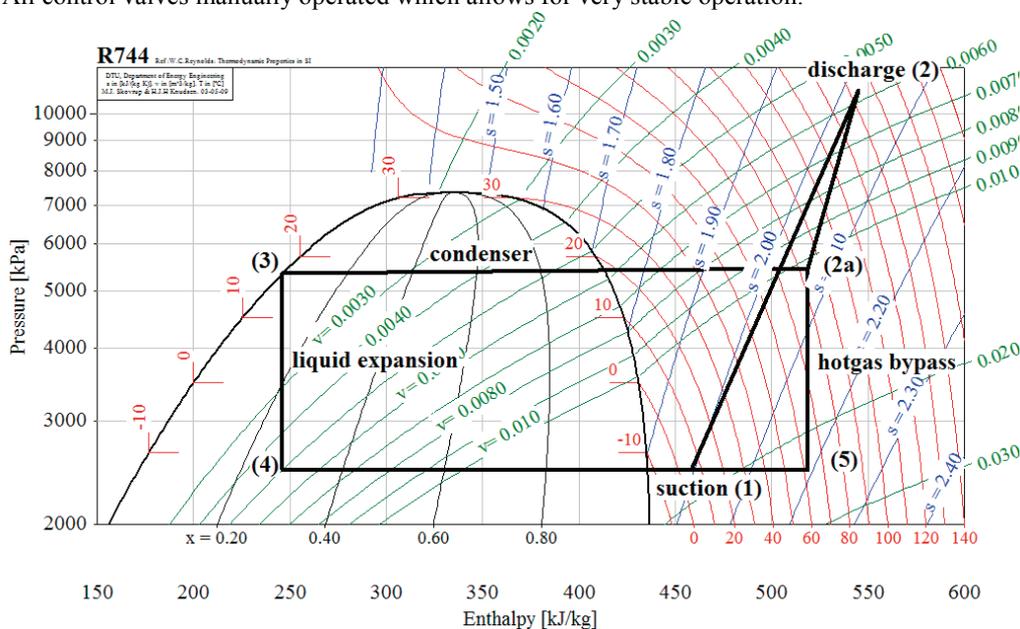
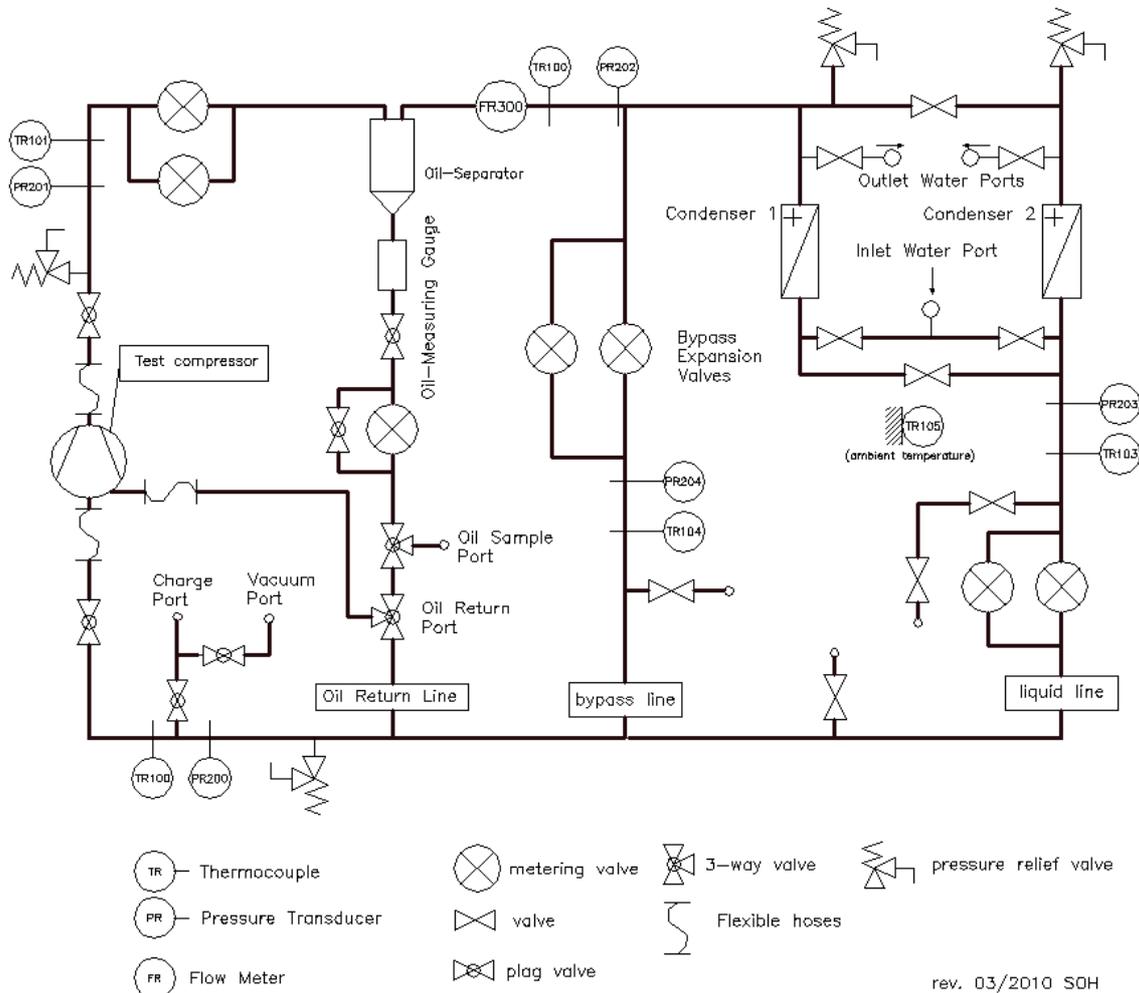


Figure 1: Compressor Load Stand Cycle in P-h Diagram

The refrigerant oil is removed from the refrigerant by the oil-separator and returned to the suction line or compressor oil port. The oil-separator is extended with an oil-gauge, which contains a large sight glass to observe the oil level. During normal steady state operation, a constant oil level is maintained in the sight glass using a fine metering valve. In order to measure the oil volume flow rate, a shut-off valve, located downstream of the oil-gauge, is closed and the

separated oil is collected in the oil-gauge for a certain time. The measurement of the collected oil together with the time duration is used to determine the oil volume flow rate. Advantages of this setup are simplicity and permanent observation of the lubricant level. The method has an estimated measurement error of  $\pm 15\%$ .



**Figure 2: Load Stand Schematic with Instrumentation**

The load stand is equipped with Type-T Thermocouples ( $\pm 1$  K) to measure temperature, and pressure transducers ( $\pm 0.25\%$ ) to measure the pressure at all five state points indicated in Figure 2. The refrigerant mass flow is determined by a Coriolis-based mass flow meter ( $\pm 0.5\%$ ) directly after the oil-separator (FR300). All of the sensors are connected to a data acquisition system, which records the measurements in four-second intervals.

## 4. APPROACH AND ANALYSIS

### 4.1 Compressor Specifications

The prototype CO<sub>2</sub> compressor was a semi-hermetic, single-stage, axial configuration with an estimated cooling capacity of 10-14 kW. Presently, there exist two types of axial displacement compressors; the wobble plate and swash plate. The swash plate design permits higher displacement volume in a more compact design, exceptional performance at high speeds, and relatively low noise levels during operation (Gerhard, 2009).

## 4.2 Testing Method

The test matrix included 18 tests at varying discharge and suction pressures and while keeping the superheat and motor frequency constant. During operation, the metering valves shown previously in Figure 2 were manually adjusted until steady-conditions were achieved. The definition of steady-state is the point at which temperature measurements did not fluctuate more than  $\pm 1\text{K}$  and a 0.5 standard deviation of the associated data, and pressure measurements were recorded within the range of  $\pm 0.25\%$ . When this was achieved, the data was recorded in the specified four second interval over a ten minute period.

**Table 1: Prototype Compressor Test Matrix**

Prototype Compressor Matrix				
Test Number	Discharge Pressure (Bar)	Evaporation Temperature (°C)	Compressor Speed (Hz)	Suction Superheat (°C)
1	75.84	1.67	60	11.11
2	82.74			
3	89.63			
4	96.53			
5	103.42			
6	110.32			
7	75.84	7.22		
8	82.74			
9	89.63			
10	96.53			
11	103.42			
12	110.32			
13	75.84	10.00		
14	82.74			
15	89.63			
16	96.53			
17	103.42			
18	110.32			

After each test was complete, thermodynamic properties from the averages of the steady-state data were calculated using Microsoft Excel with REFPROP and Engineering Equation Solver (EES) software programs. Equations (1) to (3) were programmed in the software for calculating the desired performance characteristics.

Equation 1 defines the compressor volumetric efficiency, which is a measure of the actual volumetric flow rate compared to the theoretical volumetric flow rate based upon the geometry of the compressor.

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

Equation 2 is the isentropic efficiency, which is the ratio of the change in enthalpies from discharge and suction state points at isentropic conditions compared to the actual change in enthalpies from discharge and suction state points. It is also called compressor efficiency (Wang, 2000)

$$\eta_{is} = \frac{h_{2s} - h_1}{h_2 - h_1} \quad (2)$$

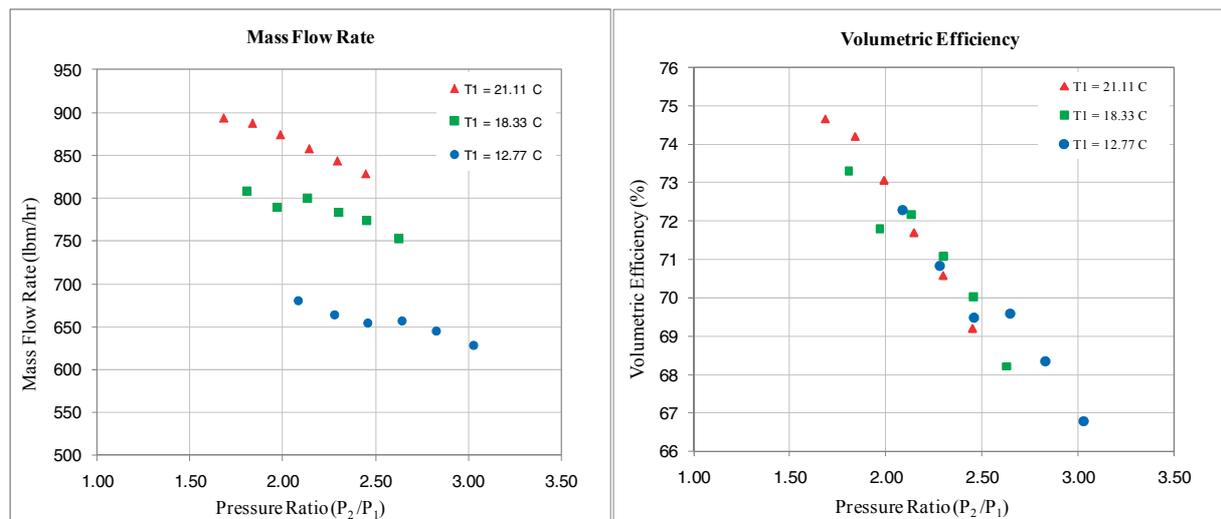
Equation 3 is the overall isentropic efficiency, which measures compressor efficiency but is a more realistic approach to compressor performance since it includes both motor and mechanical losses.

$$\eta_{is,o} = \frac{\dot{m}_R(h_{2s} - h_1)}{P_{comp}} \quad (3)$$

### 4.3 Results and Analysis

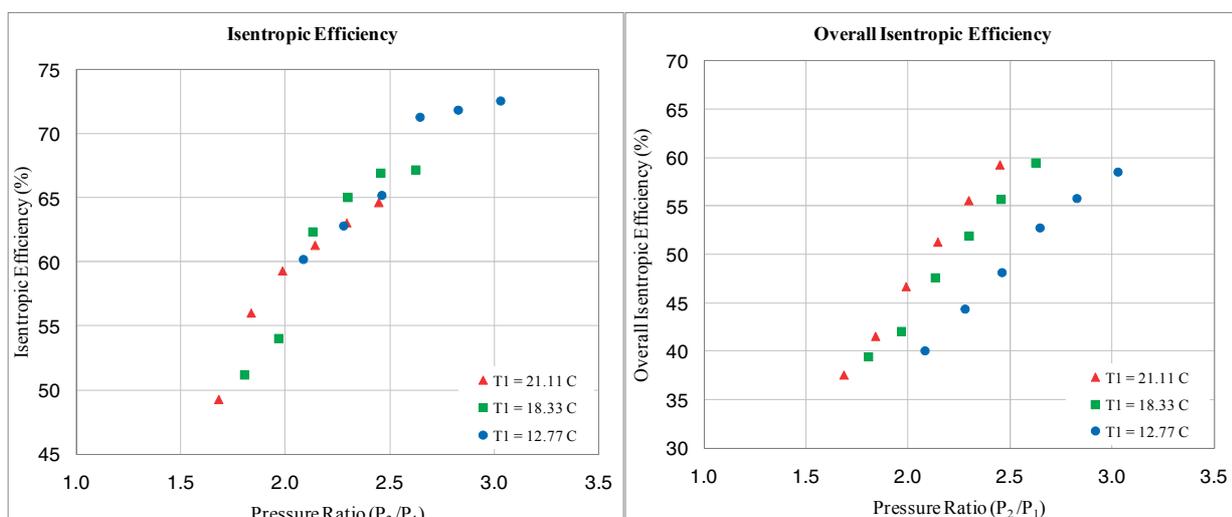
From the calculations noted, performance measures were plotted relative to the pressure ratio, which is the discharge pressure divided by the suction pressure. Each of these plots has three series, separated according to suction temperature changes in the test matrix.

The volumetric efficiency shown in Figure 3 follows the trend of decreasing efficiency as the discharge pressure increases. Although the data points are not perfectly aligned, a linear decreasing slope is observed. The highest efficiency of 74.66% occurred at test # 13. This test measured the highest suction pressure (45 Bar) and the lowest discharge pressure (75.8 Bar) of the matrix. The lowest efficiency of 66.79% occurred at test # 6. This test measured the lowest suction pressure and the highest discharge pressure.



**Figure 3: Mass Flow Rate and Volumetric Efficiency Results for the Prototype Compressor**

The results for isentropic and overall isentropic efficiencies are shown in Figure 4. As the pressure ratio increases, the compressor more closely approximates the isentropic conditions. The highest efficiency of 72.55% occurred at test # 6 and the lowest efficiency of 49.22% occurred at test #13. Contrary to the volumetric efficiency, the lowest suction pressure and highest discharge pressure had the best isentropic efficiency. Figure 5 shows the trend of the overall isentropic efficiency for each series of suction temperatures.



**Figure 4: Isentropic and Overall Isentropic Efficiency Results for the Prototype Compressor**

It is of particular interest because it describes the performance of the compressor including motor losses. It is observed that as the pressure ratio increases, the efficiency increases. Although a shift occurs at each series, the results appear well behaved and have a monotonic relationship. Matrix tests 6, 12, and 18 all have relatively the same overall isentropic efficiencies between 58.5 - 59.5%.

#### 4.4 Uncertainty Analysis

Since all experimental data has some level of uncertainty, measures were taken to quantify the degree of uncertainty for the prototype results. By using the Kline and McClintock method, relative uncertainties were calculated for each of the performance efficiencies. In Figure 5, the relative uncertainty for volumetric and overall isentropic efficiency is given. The results show that the volumetric uncertainty varies only slightly for each series. The maximum of 6.7% occurred at test #13. This appears to be an outlier relative to the other data. The minimum of 6.33% occurred at test # 1,3, and 6. Both isentropic and overall isentropic efficiency follow the same pattern of decreasing uncertainty as the discharge pressure increases. The range is from 9.64% at the minimum and 20.40% at the maximum. Additionally, higher levels of uncertainty at lower pressure ratios and higher suction pressures are common for all results.

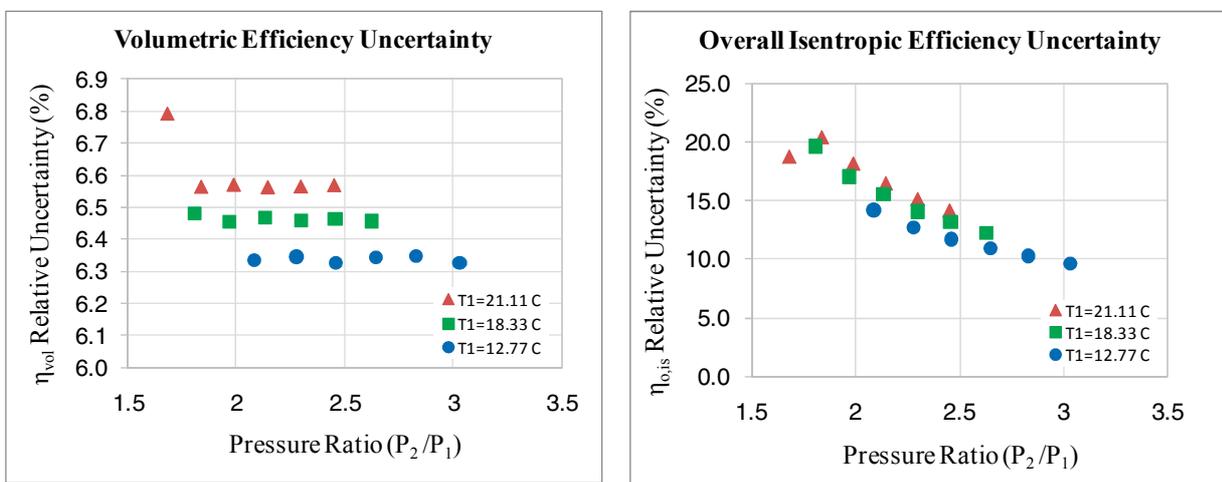


Figure 5: Volumetric and Overall Isentropic Efficiency Uncertainty Results

## 5. COMPARISON AND PERFORMANCE

In Figures 6 and 7, the volumetric and overall isentropic efficiency performance measurements of the prototype compressor are compared with its predecessor and additional carbon dioxide compressors. In each of the figures, the prototype demonstrates significant improvements as well as competitive benchmarks. Furthermore, the graph includes data collected after selected improvements were made to the prototype compressor tested. The tests were conducted at the sponsor's laboratories under similar, but not identical conditions and testing methods.

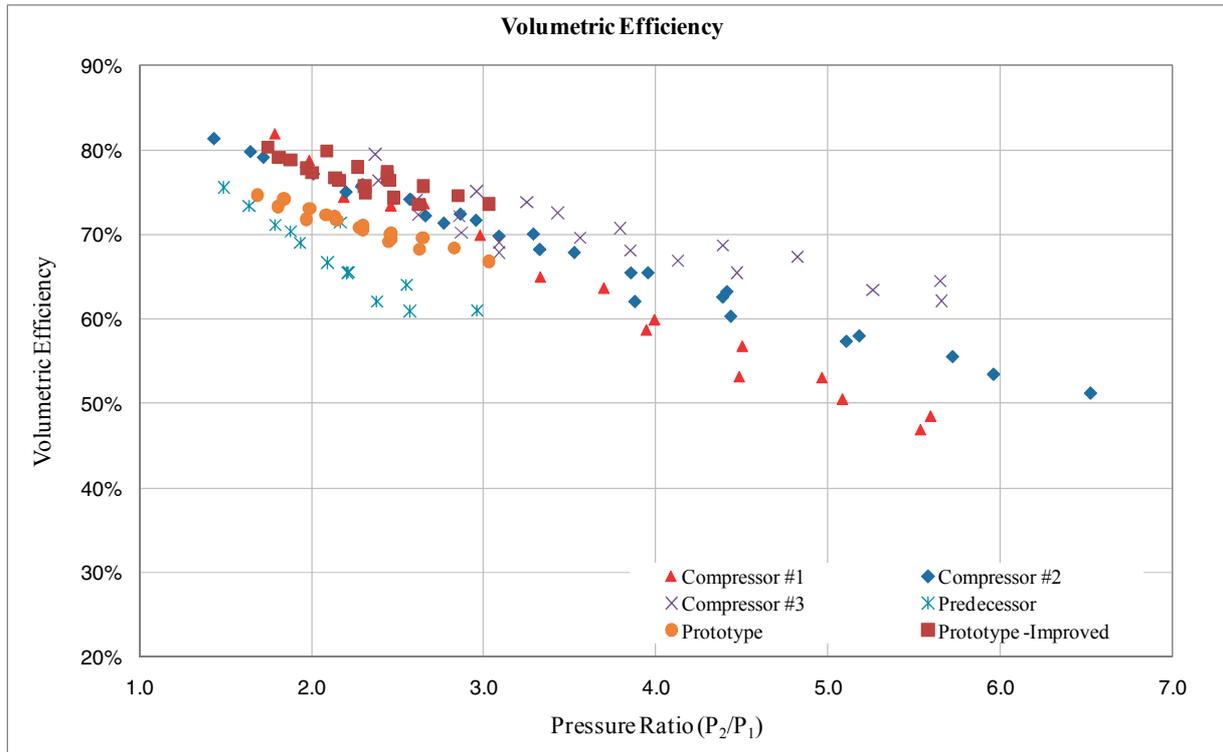


Figure 6: Volumetric Efficiency Comparison between Predecessor and other prototypes tested

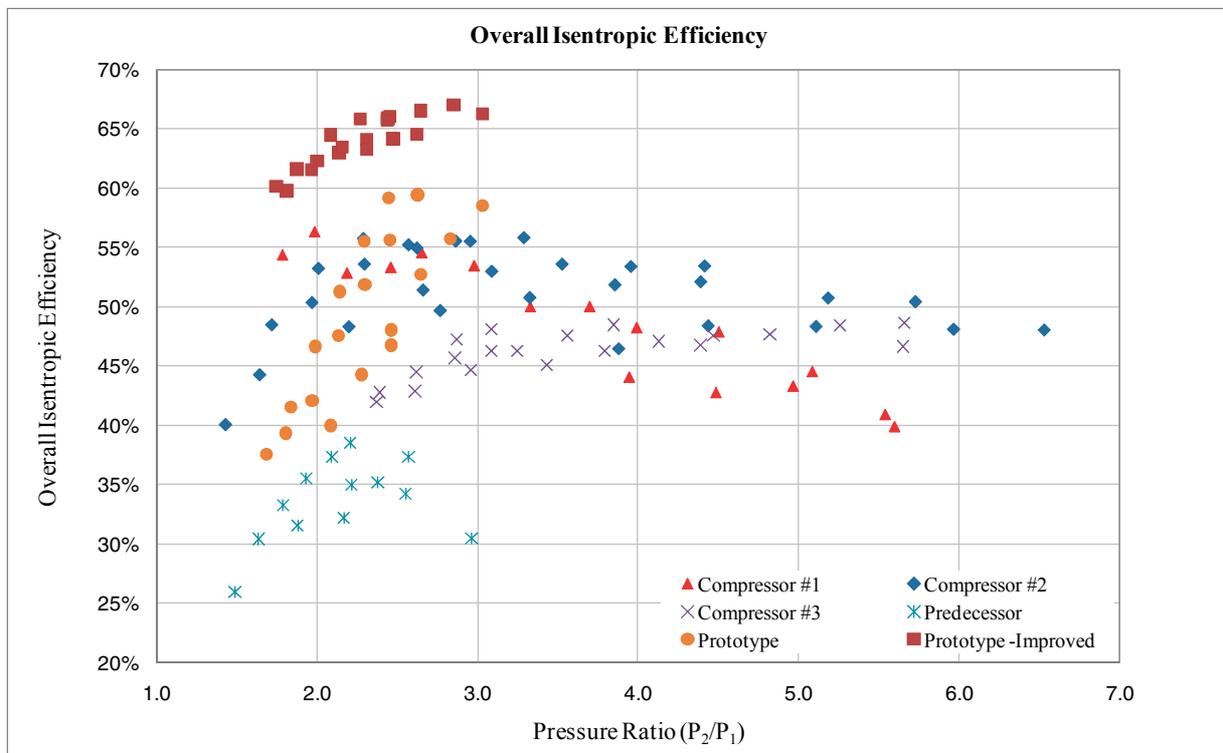


Figure 7: Overall Isentropic Efficiency Comparison between Predecessor and other prototypes tested

## 6. CONCLUSIONS

The Prototype Compressor tested on the hot-gas bypass apparatus demonstrated notable characteristics. First, the results of each calculated performance were found to be better than its predecessor compressor. When compared to additional CO<sub>2</sub> compressors that have been tested at the Ray W. Herrick Laboratories, the prototype was a competitive benchmark and can be used for further research and testing. The related uncertainty in the data was not significantly high and the performance maps created are a good representation of how well the compressor performs under a variety of situations.

## NOMENCLATURE

$\eta$	Efficiency	(%)	<b>Subscripts</b>	
$f_{inv}$	Inverter Frequency	(Hz)	1,2,3	Statepoint
$h$	Enthapy	(kJ/kg)	comp	Compressor
$\dot{m}$	Mass flow rate	(kg/s)	d	discharge
$P$	Pressure	(Bar)	R	refrigerant
$T_s$	Superheat Temperature	(C)	s	suction
$v$	Specific Volume	(m <sup>3</sup> /kg)		
$\dot{V}$	Compressor Volume Flow	(m <sup>3</sup> /s)		

## REFERENCES

- Adolph, U., "Einsatz von CO<sub>2</sub> als Kaeltemittel in Schienenfahrzeugen," FKW Seminar XVII, FKW GmbH, Hannover, Germany, Dec. 6, 1995.
- Christen T., Hubacher B., Bertch, S.S. Groll E.A. "Experimental Performance of Prototype Carbon Dioxide Compressors" Proc. 2006 Int'l Ref. Conf. at Purdue, West Lafayette, IN, July 17-20
- Groll, E.A., "Recent Advances in the Transcritical CO<sub>2</sub> Cycle Technology," 18th National & 7th ISHMT-ASME Heat and Mass Transfer Conference, IIT Guwahati, India, January 4 - 6, 2006.
- Fagerli, B.E., "CO<sub>2</sub> Compressor Development," Proc. IEA Heat Pump Centre/IIR Workshop on CO<sub>2</sub> Technology in Refrigeration, Heat Pump & Air Conditioning Systems, Trondheim, Norway, May 13-14, 1997.
- Fagerli, B.E., "An Investigation of Possibilities for CO<sub>2</sub> Compression in a Hermetic Compressor," Proc. IIR Conf., Applications for Natural Refrigerants, Aarhus, Denmark, Sept. 3-6, pp. 639-649, 1996.
- Fagerli, B.E., "Development and Experiences with a Hermetic CO<sub>2</sub> Compressor," Proc. 1996 Int. Compressor Eng. Conf. at Purdue, West Lafayette, IN, July 23-26, pp. 229-235, 1996.
- Koehler, J., Sonnekalb, M., and H. Kaiser, "A Transcritical Ref. Cycle with CO<sub>2</sub> for Bus Air Conditioning and Transport Ref.," Proc. 1998 Int'l Ref. Conf. at Purdue, West Lafayette, IN, July 14-17, pp. 121-126, 1998
- Koehler, J., Kaiser, H., and B. Lauterbach, "CO<sub>2</sub> as Refrigerant for Bus Air Conditioning and Transport Refrigeration," Proc. IEA Heat Pump Centre/IIR Workshop on CO<sub>2</sub> Technology in Refrigeration, Heat Pump & Air Conditioning Systems, Trondheim, Norway, May 13-14, 1997
- Hubacher, B, Groll, E. A., and Hoffinger, C., "Performance Measurement of a Semi-Hermetic Carbon Dioxide Compressor", Proc. Int'l Refrig. Conf. at Purdue, West Lafayette, IN, July 16-19, pp. 477-485, 2002.
- Hwang, Y., and R. Radermacher, "Development of Hermetic Carbon Dioxide Compressor," Proc. 1998 Int'l Refrigeration Conf. at Purdue, West Lafayette, IN, July 14-17, pp. 171-176, 1998.
- Neksa, P., Dorin, F., Rekstad, M., and A. Bredesen, "Development of Two-Stage Semi-Hermetic CO<sub>2</sub>-Compressors" Proc. of the 4th IIR-Gustav Lorentzen Conference on Natural Working Fluids, Purdue University, West Lafayette, IN, July 25-28, pp. 355-362, 2000.
- Tadano, M., Ebara, T., Oda, A., Susai, T., Takizawa, K., Izaki, H., and T. Komatsubara, "Development of the CO<sub>2</sub> Hermetic Compressor", Proc. of the 4th IIR-Gustav Lorentzen Conference on Natural Working Fluids, Purdue University, West Lafayette, IN, July 25-28, pp. 323-330, 2000.
- Wang, Shan K. Handbook of air conditioning and refrigeration. 2. Edition. New York : McGraw-Hill, 2000. ISBN 0-07-068167-8