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ABSTRACT

This study, which developed a laboratory prototype of a hermetic CO_2 compressor, was motivated by three primary factors. First, the CO_2 cycle is important because of global concern about ozone depletion and global warming. Second, the leak tightness, efficiency, size and weight of hermetic motor-driven compressors make them preferable to open-type compressors for conventional residential/commercial air-conditioning. Third, compact and efficient hermetic compressors are necessary to develop an equivalently sized or compact system. The performance of the CO_2 compressor was compared with that of an R-22 compressor by using the test facility developed for the transcritical CO_2 cycle. Test results show the performance potential of the CO_2 compressor which is sized to match the conventional hermetic R-22 compressor.

INTRODUCTION

Though the transcritical CO_2 cycle has gained much attention recently, most research has focused on transportation applications where an open-type compressor is used. For transportation applications, Lorentzen and Petterson (1993) and Koehler et al. (1995) published experimental data showing superior performance for CO_2 over that of R-12 at lower ambient temperatures. For conventional residential/commercial air-conditioning, a hermetic motor-driven compressor has traditionally been used. A compact, efficient hermetic compressor is necessary to develop a compact system. Recently, Kruse (1996) and Fagerli (1996a) published experimental results on open-type CO_2 compressors, and Fagerli (1996b) published experimental results on a hermetic CO_2 compressor with an isentropic efficiency of 43%. In the present study, experimental tests were carried out with hermetic motor-driven compressors for R-22 and CO_2.

EXPERIMENTAL SETUP

Test Facility

A laboratory prototype water-chiller using CO_2 as the refrigerant was built. It is a water-to-water system that allows testing of a wide range of operating conditions. Figure 1 shows the schematic diagram of this facility.

Test Conditions

The water temperatures were controlled to satisfy the conditions specified in ARI Standard 590 (1992). This standard requires a chilled water temperature entering the evaporator of 12.4 °C, leaving the evaporator of 6.7 °C, a cooling water temperature entering the gas cooler of 29.4 °C and a water flow rate in the gas cooler of 0.054 *l*/s-kW.

Compressor

Two positive displacement compressor prototypes were designed and built, as shown in Figure 2. Prototype 1 was a hermetic compressor with two pistons. The designed capacity of prototype 1 was 22 kW. Prototype 2 was a hermetic compressor with one piston and a removable lid. The designed capacity of prototype 2 was 11 kW. The compressor shell modification was conducted to enhance investigation of the inner parts and lubricant.

Design Aspects

Most parts of the prototype were utilized from the conventional reciprocating R-22 compressor. Some parts, however, were newly designed after considering characteristics of the CO_2 cycle as described below.

The compressor required redesign because of the high operating pressure. The saturation pressure of CO_2 is seven times higher than that of R-22 at 10 °C. The discharge pressure of CO_2 is 7 to 13 MPa higher than that of R-22. The suction pressure of CO_2 is 3 to 4 MPa higher. A compressor shell and discharge side parts, such as the cylinder head, muffler and discharge pipe, were redesigned with maximum allowable pressures of 19 MPa.

The pressure difference of the CO_2 cycle is five times larger than that of R-22, though the pressure ratio of the CO_2 cycle is approximately 28% lower than that of R-22 in the case of water chilling. Therefore, the sealing around the piston and the head must be considered. For this reason, piston rings were employed to prevent leakage through the gap between the cylinder and the piston. The mechanism parts' strength (connecting rod, bolts, muffler and tubing) was also increased to meet the increased pressure difference. The large pressure difference also causes a large torque. A three-phase motor of 2.2 kW output was assembled to match the increased torque.

The compressor displacement was chosen to produce the designed cooling capacity. The displacement of the CO_2 compressor is smaller than that of the R-22 compressor because the suction density of CO_2 is approximately five times higher than that of R-22, while the latent heat of CO_2 is similar to that of R-22 at similar operating conditions. Table 1 compares the displacements of the prototypes and the R-22 compressors. As shown in this table, the displacements of CO_2 compressors are only 32 ~ 41% of those of the R-22 compressors. The displacements of CO_2 compressors are slightly over sized because of the slightly larger mass flow rate requirement (approximately 5%) and the larger leak.

Table 1. Comparison of Compressor Displacement

Design Capacity [kW]	18	22	11	11
Compressor Displacement [cc/rev]	R-22 101	Prototype 1 32	R-22 40	Prototype 2 16

Modification from Prototype 1 to Prototype 2

The test results of prototype 1 demonstrated that the motor efficiency was not good and resulted in poor COP. The motor efficiency varies depending on the required torque, and there is an optimum torque which has a maximum motor efficiency. Therefore, the matching of motor torque and load is important to optimize the system COP. For this reason, matching the required torque and the optimum motor torque by reducing the number of pistons from two to one was attempted. Then the motor efficiency was increased from 82% to 88% (according to the motor curves), which is the optimum range.

The frequent failure of prototype 1 requires that the compressor be opened and parts needing repair be identified. Therefore, prototype 1 was modified to be equipped with a removable lid. In this paper, only the test results of prototype 2 are reported.

TEST RESULTS

Charge Optimization with R-22 Compressor

The refrigerant charge was optimized when varying the refrigerant mass flow rate with the expansion valve. Figure 3 shows the test results. The optimum charge of R-22 was determined to be 1.8 kg.

Charge Optimization with CO_2 Compressor Prototype 2

Figure 4 shows the results of the charge optimization tests. The figure indicates that the 3.5 kg and 4.0 kg charges both have higher performance than the 3.0 kg charge. The optimum charge was determined to be 3.5 kg because the lower charge is preferable since performance was similar.

Comparison of Results for R-22 Compressor and CO_2 Compressor

Table 2 summarizes all tests conducted with the R-22 compressor and CO_2 prototype 2. The capacity of CO_2 prototype 2 is 13% lower than that of R-22, while the COP of CO_2 is 40% higher than that of R-22. Table 2 shows that the pressure ratio of CO_2 prototype 2 is 27% lower than that of R-22. This comparison is not fair because the same refrigerant tube diameter is used for both cases.

Table 2. Comparison of Test Results for R-22 Compressor and CO_2 Compressor

Compressor	R-22 Compressor	CO_2 Prototype 2
Charge [kg]	1.8	3.5
Water Chilling Capacity [kW]	12.4	10.8
COP	2.5	3.5
Discharge Temperature [°C]	91.4	74.6
Suction Temperature [°C]	1.1	1.6
Evaporation Temperature [°C]	6.0	2.2
Condensing or gas cooling Pressure [kPa]	2022.4	8325.7
Evaporation Pressure [kPa]	647.4	3629.8
Pressure Ratio	3.12	2.29
Refrigerant mass flow rate [kg/s]	0.0822	0.069

Compressor Performance

Recently, Kruse (1996) published his experimental results on the open-type CO_2 compressor and Fagerli (1996b) published his experimental results on the hermetic CO_2 compressor. Specifications of these two compressors and compressor prototype 2, developed in the present study, are compared in Table 3.

Table 3. Comparison of Specifications of CO_2 Compressors

Developer	Kruse	Fagerli	Prototype 2
Shell type	Open	Hermetic	Hermetic
No. of cylinder	2	1	1
No. of piston ring	4	2	1
Swept volume [cc]	29.5 per cylinder	2.57	16.2
RPM	840	2900	3510
Displacement [cc/s]	826	124.2	947.7
Cylinder diameter [mm]	28.0	16.0	41.3
Stroke [mm]	47.9	12.8	12.1
Clearance volume ratio [%]	2.9	n/a	1.0

Table 4. Comparison of Efficiencies of CO_2 Compressors

Developer	Kruse	Fagerli	Prototype 2
Suction pressure [MPa]	4.0	3.5	3.4
Discharge pressure [MPa]	10.0	8.5	8.5
Pressure ratio	2.5	2.4	2.5
η_{ise}	0.83	0.43	0.84
η_{vol}	0.89	0.64	0.80
η_{mec}	n/a	0.81	0.95
η_{mot}	n/a	0.77	0.88

The performance of the compressors can be compared using isentropic efficiency (η_{ise}), volumetric efficiency (η_{vol}), mechanical efficiency (η_{mec}) and motor efficiency (η_{mot}). These efficiencies are obtained from the experimental data of the motor power consumption, refrigerant mass flow rate, refrigerant pressure/temperature at the suction and discharge, and the motor performance curve. Table 4 shows the comparison of these efficiencies for the three cases in Table 3. Based on this comparison, the following conclusions were obtained. Compressor prototype 2 has a similar η_{ise} to that of Kruse but a 10% lower η_{vol}. The smaller number of piston rings contributes to the larger piston leak and lower η_{vol}. Fagerli's compressor shows poor η_{ise} and η_{vol} compared to the others. This seems to be due to poor design of the motor and mechanical parts as seen from the mechanical efficiency (η_{mec}) and motor efficiency (η_{mot}). Therefore, of the three, compressor prototype 2 is the best hermetic compressor. Prototype 2 can be improved by reducing the refrigerant leak along the piston.

CONCLUSIONS

A test facility for the CO_2 system including the compressor and heat exchangers was developed. The instrumentation and data acquisition system were installed. The software routines to collect and analyze the data were developed. The test facility was validated using a directly immersed heater and R-22 compressors. The performance of the CO_2 cycle was optimized with respect to the amount of refrigerant charged and the refrigerant mass flow rate. The CO_2 cycle has a higher pressure difference than R-22 between high and low pressure sides, so there is greater potential for a large internal leak. Therefore, the design of the CO_2 compressor must consider the unique characteristics of CO_2. In this study, the hermetic CO_2 compressor was developed. The volumetric efficiency was identified as 80% and the isentropic efficiency was identified as 84%.

REFERENCES

Air-Conditioning & Refrigeration Institute, 1992, Positive Displacement Compressor Water-Chilling Packages, ARI Standard 590, Arlington, VA.

Fagerli, B.E., 1996a, An Investigation of Possibilities for CO_2 Compression in a Hermetic Compressor, Proceedings of the 2nd International Conference on the Non-Artificial Substances- Applications for Natural Refrigerants '96, Aarhus, Denmark, pp. 15.6-15.11.

Fagerli, B.E., 1996b, Development and Experiences with a Hermetic CO_2 Compressor, Proceedings of the 1996 International Refrigeration Conference at Purdue, West Lafayette, IN, pp. 229-234.

Kohler, J, M. Sonnekalb, H. Kaiser and W. Kohler, 1995, Carbon Dioxide as a Refrigerant for Vehicle Air-Conditioning with Application to Bus Air-Conditioning, Proceedings of the 1995 International CFC and Halon Alternatives Conference, Washington D.C., pp. 376-385.

Kruse H., 1996, Current Status of Natural Working Fluids in Refrigeration, A/C, and Heat Pump Systems, Proceedings of The 2nd International Conference on the Non-Artificial Substances- Applications for Natural Refrigerants '96, Aarhus, Denmark, pp. C.1-C.8.

Lorentzen, G. and J. Pettersen, 1993, A New, Efficient and Environmentally Benign System for Car Air-Conditioning, International Journal of Refrigeration, Vol. 16, No.1, pp. 4-12.

Figure 1 Cycle Diagram for CO2 Facility

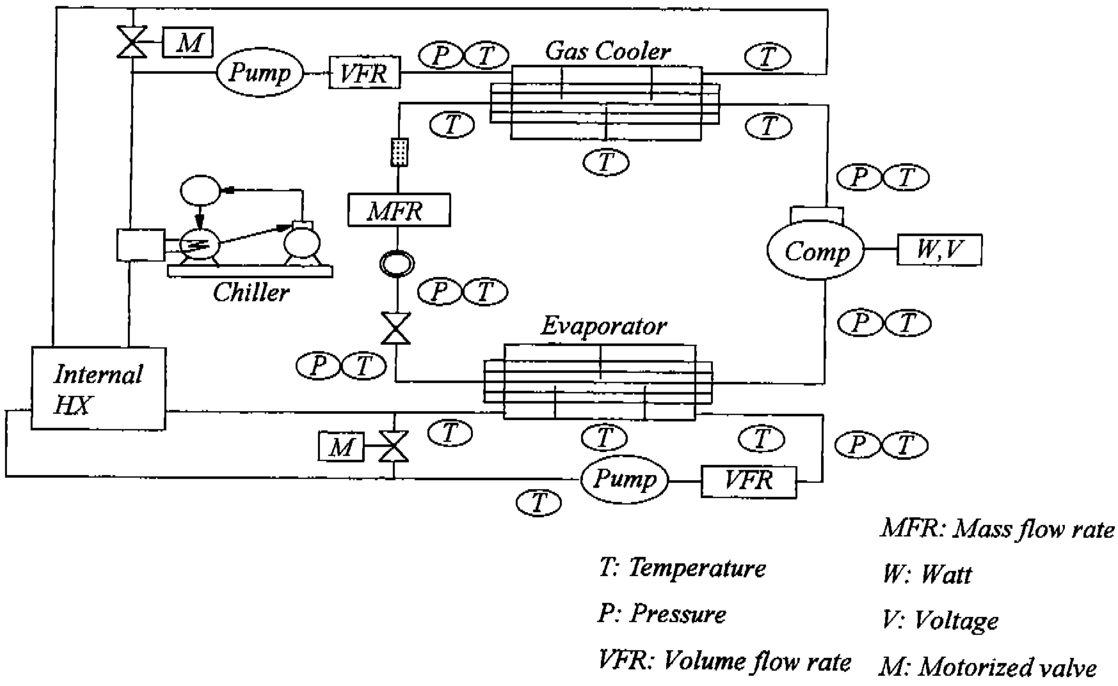


Figure 2 CO2 Compressor Prototype

(a) Hermetic Compressor (b) Hermetic Compressor with Lid

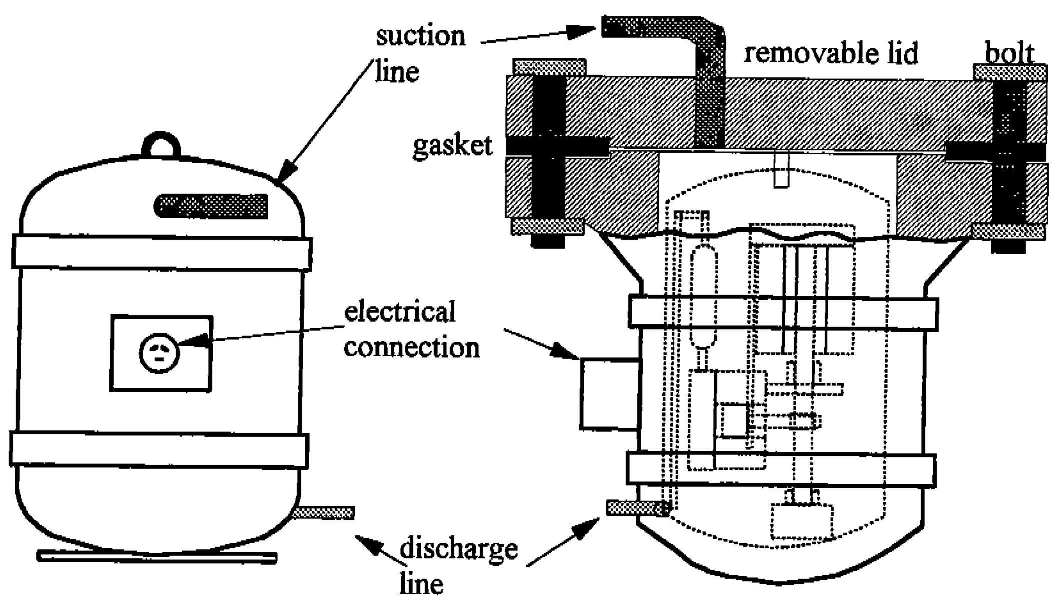


Figure 3 Charge Optimization of R-22 Compressor (Water Chilling)

(a) Capacity (b) COP

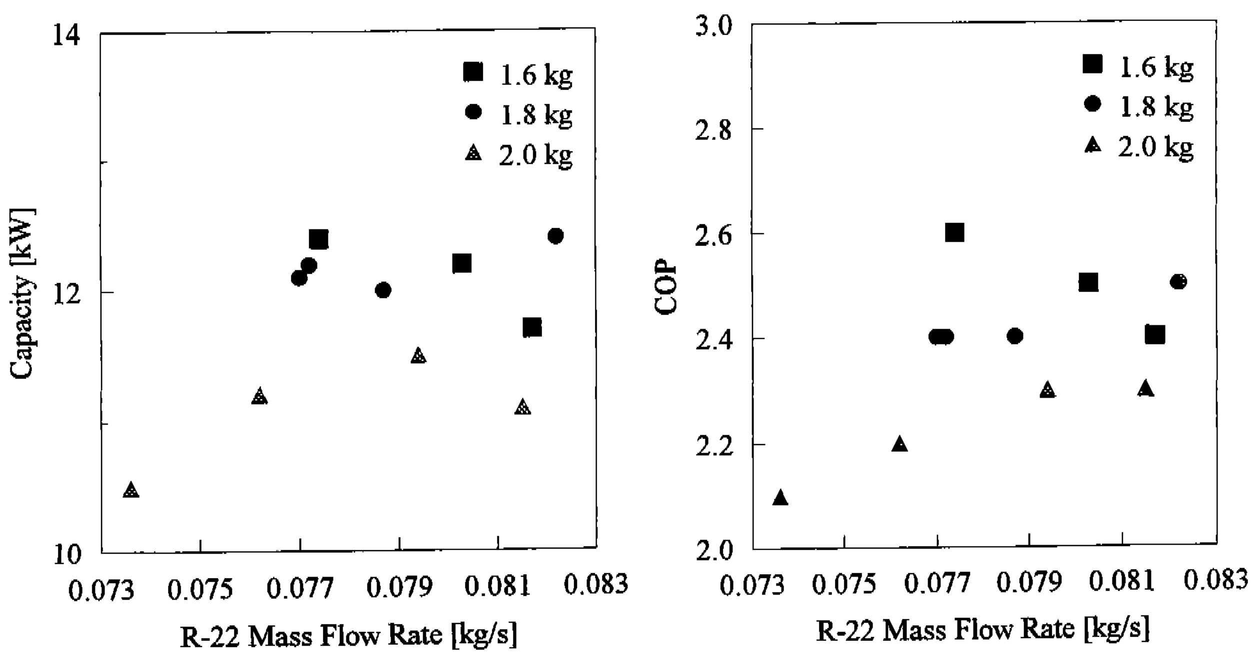
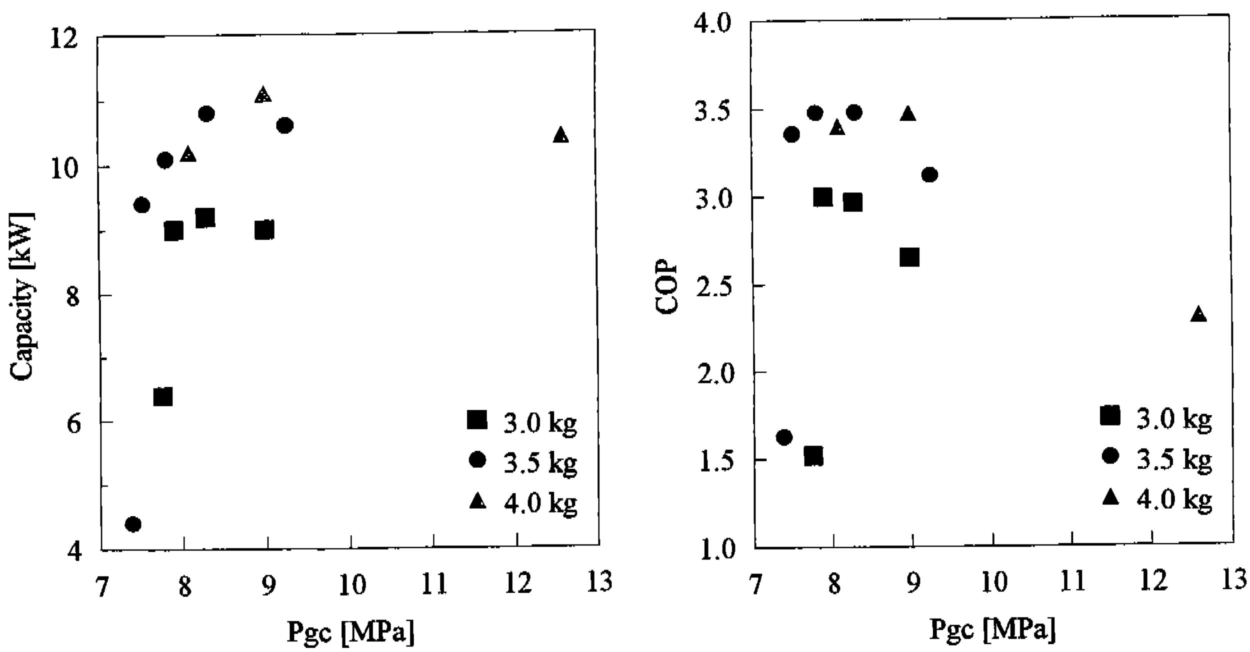


Figure 4 Charge Optimization of CO2 Prototype 2 (Water Chilling)

(a) Capacity (b) COP



VFRw,ev = 46 [l/min], VFRw,gc = 87 [l/min]