2008

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ABSTRACT

With the increasing of the environmental concerns, carbon dioxide (CO$_2$) as an alternative refrigerant has been investigated intensively. In this study, the heating performance of a transcritical CO$_2$ heat pump system with a twin-rotary compressor was investigated. The transcritical CO$_2$ heat pump with gas injection was tested by varying gas injection ratio and outdoor temperature at the optimum refrigerant charge of 6400 g, compressor frequency of 40 Hz, and first and second EEV openings of 6% and 6%. The heating capacity ratio and COP ratio increased with the increase of the gas injection ratio at all outdoor temperatures due to the increase of the total mass flow rate. The heating performance of the transcritical CO$_2$ heat pump with gas injection was improved at low outdoor temperature. The heating capacity and COP ratio of the system with gas injection were improved by 45% and 24%, respectively, over the non-injection system at the outdoor temperature of -8 ^{o}C.

1. INTRODUCTION

Due to large expansion losses and irreversibility during the gas cooling process, the transcritical CO$_2$ system showed lower performance than conventional air-conditioning systems. Various studies have been conducted on the transcritical CO$_2$ cycle. Hwang and Radermacher [1999, 2000] studied a two-stage transcritical CO$_2$ cycle. Groll et al. [2002] studied the effect of the pressure ratios across compressors on the performance of a transcritical CO$_2$ cycle with two-stage compression and intercooling. Joseph and Pega [2005] developed transcritical CO$_2$ systems for small commercial applications. Kim [2002] studied the performance of an auto-cascade refrigeration system for a water heater using CO$_2$ as a working fluid.

In this study, the performance of a transcritical CO$_2$ heat pump with a twin-rotary compressor using gas injection was measured and analyzed. The gas injection technique was applied to increase heating capacity and improve compressor reliability at high pressure conditions. The heating performance of the transcritical CO$_2$ heat pump was measured by varying the gas injection ratio, refrigerant charge, compressor frequency, EEV opening, length of an internal heat exchanger, and outdoor temperature. In addition, a control algorithm to improve the heating performance of the transcritical CO$_2$ heat pump was developed.

2. EXPERIMENTAL SETUP AND TEST PROCEDURE

Figure 1 shows the schematic diagram of the experimental setup. The test setup consisted of a variable speed twin-rotary compressor, indoor unit, outdoor unit, flash tank, three different expansion valves, and mass flow meters. The twin-rotary compressor having two-stage cylinders in a compressor shell was driven by a BLDC motor. The outdoor unit consisted of a compressor and an outdoor coil (evaporator), and the indoor unit consisted of an indoor coil (gascooler) and an electronic expansion device (EEV). Gas refrigerant was injected from the flash tank to the inlet of the second-stage cylinder of the twin-rotary compressor by controlling an injection EEV. The injection ratio was defined as the ratio of the injected refrigerant mass flow rate to the total mass flow rate. The volume ratio of the second-stage cylinder to the first-stage cylinder in the twin-rotary compressor strongly affects intermediate pressure and compressor performance. The volume ratio of the twin-rotary compressor used in this study was 0.7. Fin tube heat exchangers were used as the gascooler and evaporator. Table 1 shows the specifications of the indoor and outdoor coils. The refrigerant flow rate was controlled by using the EEVs with an orifice diameter of 3 mm. An internal heat exchanger was installed in between the evaporator outlet and the gascooler inlet. Table 2 shows the uncertainties of the experimental parameters used in this study.

Table 3 shows test conditions used in the experiments. The heating performance of the transcritical CO_2 heat pump was measured by varying the refrigerant charge, injection EEV opening, and outdoor temperature. The optimum refrigerant charge was determined at the standard heating test condition: indoor test conditions of 20 ^{o}C DB and 15 ^{o}C WB, and outdoor test conditions of 7 ^{o}C DB and 6 ^{o}C WB.
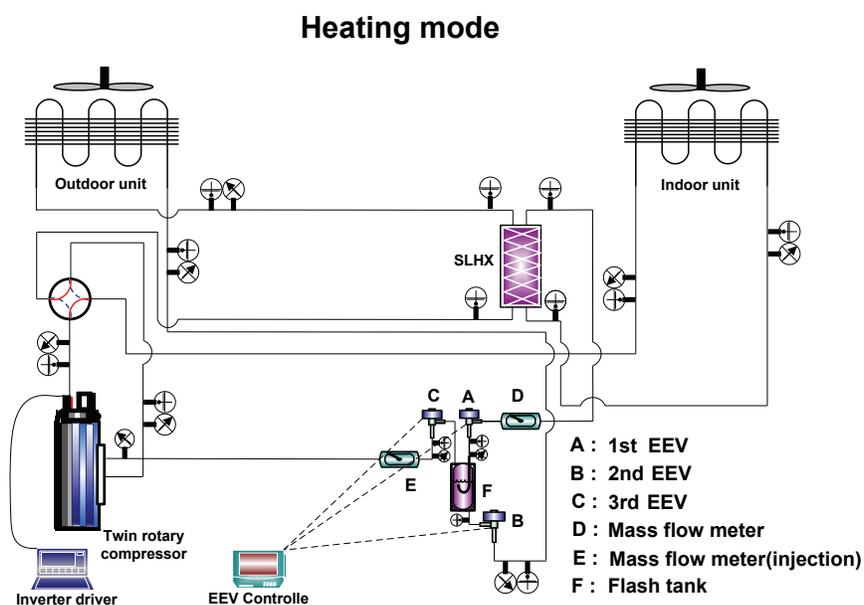
Heating mode

Figure 1 : Schematic diagram of the transcritical CO_2 heat pump.

Table 1 : Specifications of indoor and outdoor coils

	Indoor coil	Outdoor coil
Row x Step x Circuits	3 x 52 x 4	3 x 32 x 8
Diameter (mm)	7	7
Fin pitch (mm)	1.2	1.8
Fin type	Louvered fin	Louvered fin

Table 2 : Experimental uncertainties

Parameters	Uncertainties
Temperature (T-type) (^{o}C)	±0.1 ^{o}C
Pressure(bar)	±0.2% of full scale
Mass flow rate(kg/h)	±0.2% of reading
Power input (kW)	±0.01% of full scale
Heating capacity(kW)	±3.71%
COP	±3.73%

Table 3 : Test conditions

Parameters	Specifications
1st/ 2nd EEV opening (%)	6/ 6
Injection EEV opening (%)	0, 8, 10, 12, 14
Compressor frequency (Hz)	40
Operating conditions (^{o}C, DB/WB)	Indoor = 20/15 Outdoor = 7/6, 2, -3, -8

3. RESULTS AND DISCUSSION

In this study, the performance of the transcritical CO_2 heat pump with gas injection was tested by varying gas injection ratio, EEV opening, and outdoor temperature at the optimum refrigerant charge of 6400 g, compressor frequency of 40 Hz, and first and second EEV openings of 6% and 6%.

Figure 2 shows the variation of mass flow rate with injection ratio. Total and injection mass flow rate increased with the increase of injection ratio at all outdoor temperatures due to the increase of the refrigerant density at the second-stage cylinder inlet in the twin-rotary compressor with intercooling effects by using the injected refrigerant.

Figure 3 shows the variation of total mass flow rate ratio with injection ratio. The total mass flow rate ratio was defined as the ratio of increased total mass rate with gas injection to total mass rate without gas injection. The total mass flow rate ratio increased with the increase of injection ratio due to the increase of the refrigerant mass flow at the second-stage cylinder inlet in the twin-rotary compressor with the increase of injection ratio. The total mass flow rate ratio at the same injection ratio increased with the decrease of the outdoor temperature, because the pressure difference between the injection EEV outlet and the first-stage cylinder outlet in the twin-rotary compressor increased with the decrease of the compressor intermediate pressure and refrigerant density.

Figure 4 shows the variation of compressor discharge temperature with injection ratio. The discharge temperature decreased with the increase of injection ratio at all outdoor temperatures because the refrigerant temperature at the second-stage cylinder inlet in the twin-rotary compressor was decreased by applying gas injection. So, a lower discharge temperature in the twin-rotary compressor increased reliability of the system.

Figure 5 shows the variation of compressor work with injection ratio. The compressor work increased with the increase of injection ratio at all outdoor temperatures because the increase in the total mass flow rate ratio was relatively higher than the decrease in the compressor discharge temperature.

Figure 6 shows the variation of heating capacity ratio with injection ratio. As the outdoor temperature decreased from 7 ^{o}C to -8 ^{o}C, the increasing slope of the heating capacity ratio increased with the increase of injection ratio due to the increase in the total mass flow rate ratio. The heating capacity ratios at the outdoor temperatures of -8 ^{o}C and 7 ^{o}C increased by 45% and 11% with the increase of injection ratio from 0 to 0.27, respectively.

Figure 7 shows the variation of COP with injection ratio. The COP ratio of the transcritical CO_2 heat pump increased with the decrease of the outdoor temperature at all injection ratios. The COP ratio at outdoor temperatures below 2 ^{o}C increased with the increase of injection ratio because the increase of the heating capacity ratio was higher than the increase of the compressor work. However, the COP ratio at the outdoor temperature of 7 ^{o}C decreased with the increase of injection ratio because the increase of the compressor work was higher than the increase of the heating capacity ratio. The COP ratio for the outdoor temperature of -8 ^{o}C increased by 24% with the increase of injection ratio from 0 to 0.27. However, it for the outdoor temperature of 7 ^{o}C decreased by 9% with the increase of injection ratio from 0 to 0.27.
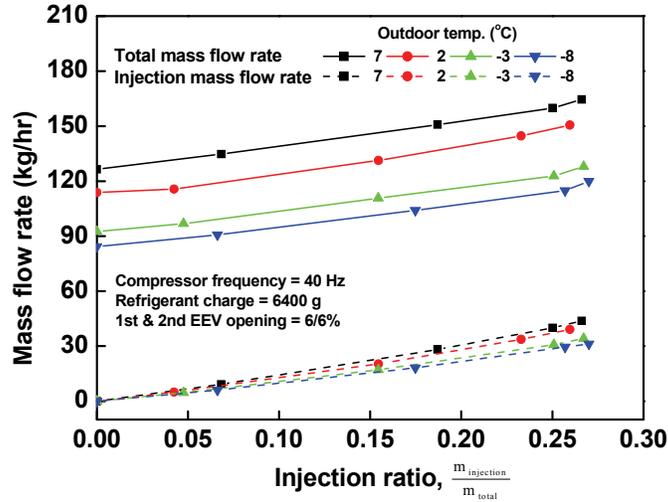
Figure 2 : Variation of mass flow rate with injection ratio.
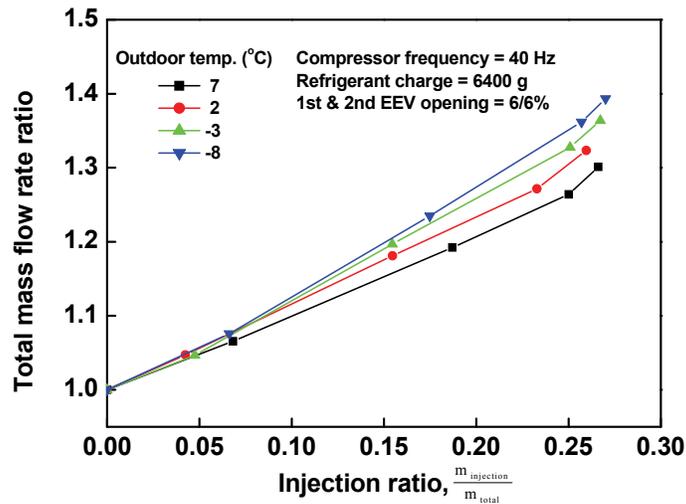

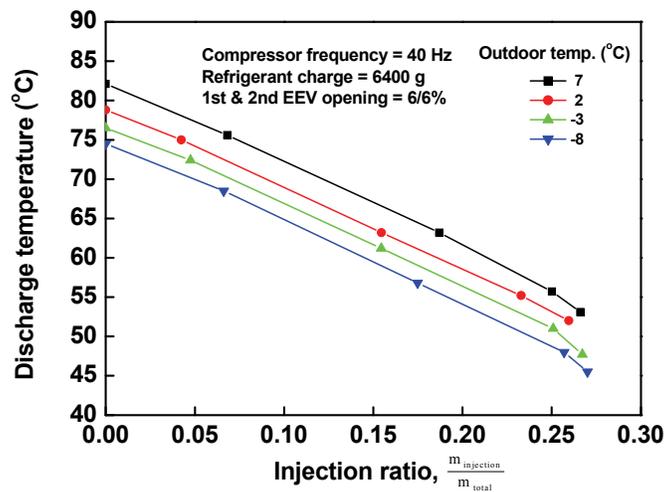Figure 3 : Variation of total mass flow rate ratio with injection ratio.

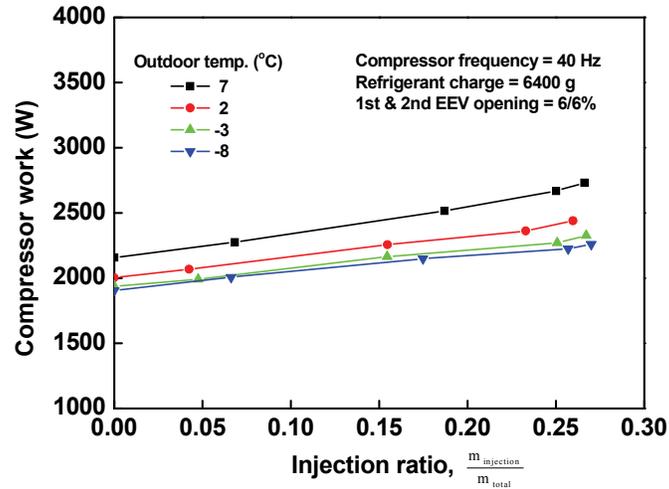Figure 4 : Variation of discharge temperature with injection ratio.

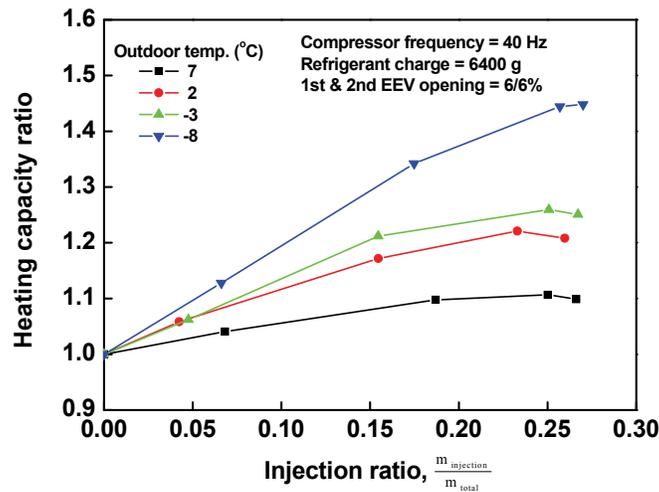Figure 5 : Variation of compressor work with injection ratio.

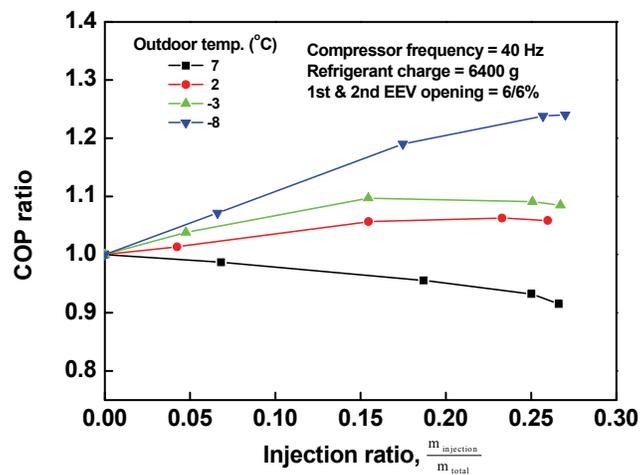Figure 6 : Variation of heating capacity ratio with injection ratio.

Figure 7 : Variation of COP ratio with injection ratio.

Figure 8 shows the variation of total mass flow rate ratio with outdoor temperature. The total mass flow rate ratio decreased with the decrease of the outdoor temperature at all injection EEV openings. The decreasing slope of the total mass flow rate ratio decreased with the increase of the injection EEV opening because the injected refrigerant flow rate increased with the increase of the injection EEV opening.

Figure 9 shows the variation of heating capacity ratio with outdoor temperature. The heating capacity ratio decreased with the decrease of the outdoor temperature at all injection EEV openings due to the decrease of the compressor discharge temperature and total mass flow rate. The decreasing slope of the heating capacity ratio decreased due to the increase of the heating capacity with the increase of the injection EEV opening.

Figure 10 shows the variation of COP ratio with outdoor temperature. The COP ratio of the transcritical CO_2 heat pump with gas injection decreased with the decrease of the outdoor temperature at all injection EEV openings. However, as the injection EEV opening increased, the decreasing slope of the COP ratio was reduced due to the increase of the heating capacity.
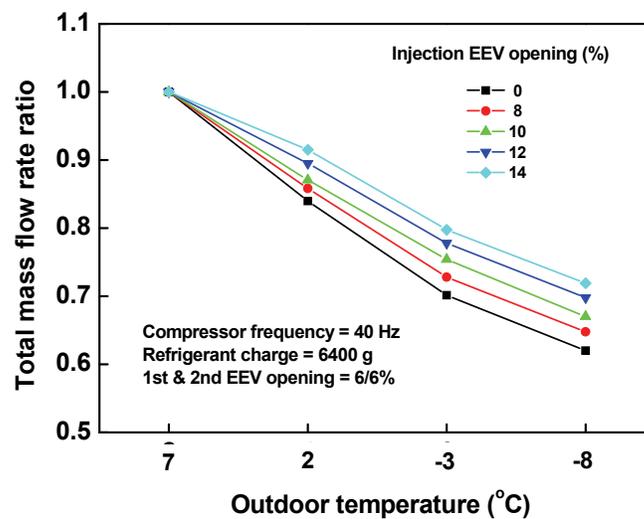
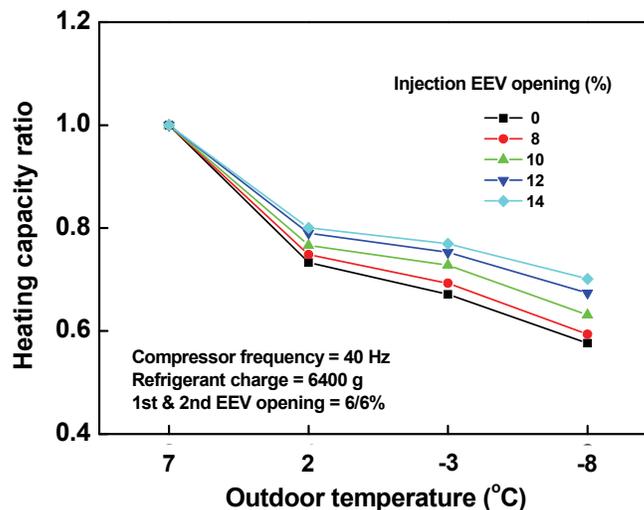Figure 8 : Variation of total mass flow rate ratio with outdoor temperature.

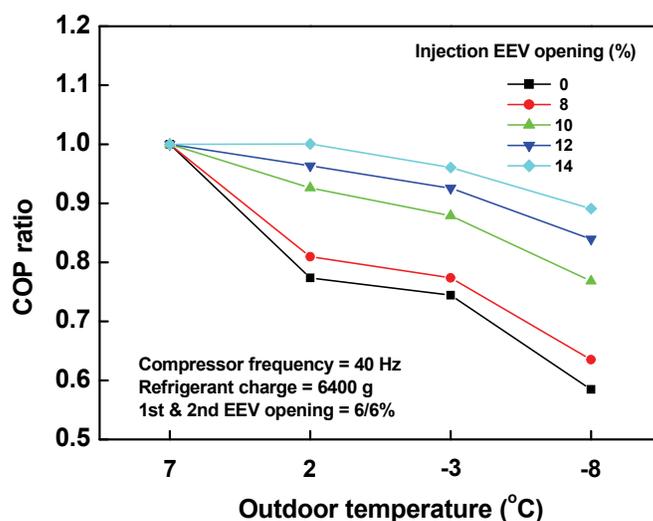Figure 9 : Variation of heating capacity ratio with outdoor temperature.

Figure 10 : Variation of COP ratio with outdoor temperature.

4. CONCLUSIONS

A transcritical CO_2 heat pump with gas injection was tested by varying the gas injection ratio and outdoor temperature. Based on the preliminary tests, the optimum refrigerant charge of the CO_2 system was 6400 g. The volume ratio of the variable speed twin-rotary compressor was 0.7. The heating capacity ratio and COP ratio increased with the increase of injection ratio due to the increase in the total mass flow rate. The heating capacity ratio and COP ratio at the same injection ratio increased with the decrease of the outdoor temperature due to the increase in the total mass flow rate ratio at low outdoor temperatures. The heating capacity ratios at the outdoor temperatures of -8 ^{o}C and 7 ^{o}C increased by 45% and 11%, respectively. The COP ratio at the outdoor temperatures of -8 ^{o}C and 7 ^{o}C increased 24% and -9%, respectively. Therefore, the transcritical CO_2 heat pump with gas injection showed better performance at low outdoor temperatures than that at normal operating conditions.

NOMENCLATURE

COP	coefficient of performance
EEV	electronic expansion valve
Temp	temperature
WB	wet bulb
DB	dry bulb

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ACKNOWLEDGEMENT

MANUSCRIPT REFERENCE NO. 2250

19th International Compressor Engineering Conference at Purdue
12th International Refrigeration and Air Conditioning Conference at Purdue
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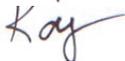
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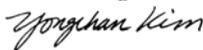|		
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