 doi:10.20944/preprints201712.0063.v1

Article

Experimental study on the cycle performance characteristics of the CO_2 heat pump system under the cooling condition

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Abstract: Developing high performance HVAC system using natural refrigerants including carbon dioxide (CO_2) has been important in respect of environmental preservation and associated technologies. Thus studies to optimize the HVAC (heating ventilation air conditioner) system using natural refrigerants through clarifying the cycle performance characteristics are necessary. The CO_2 heat pump system using air and water sources was consisted to examine its performance characteristics, and by varying conditions of several factors that affect or characterize the system performance like the amount of refrigerant charge, EEV (electronic expansion valve) opening, and internal heat exchanger under cooling mode. The performance characteristics of CO_2 heat pump system were tested by using an air enthalpy calorimeter. In the case of the CO_2 heat pump system without internal heat exchanger, the opening of #3 EEV and #4 EEV was 60% and refrigerant charge amount was 5,600g. However, in the case of that with internal heat exchanger, the best performance was obtained when the opening of #2 EEV is 20%. From the present studies, it was observed that the performance variation and operational characteristics of the CO_2 heat pump system were affected by design factors like refrigerant charge amount, EEV opening, and internal heat exchanger and thereby, the configuration on an optimal operation conditions of the system was enabled.

Keywords: Capacity, Cooling and heating, CO_2, Cycle performance, EEV, Heat pump, Internal heat exchanger

1. Introduction

CFC and HCFC refrigerants, which are used as refrigerant for the heating and cooling system, are already regulated for production and use due to ozone depletion problem. The optimal design of the heat pump system using natural refrigerant is important for this solution, and it is necessary to understand the cycle characteristics of the heat pump and the performance change of the internal heat exchanger. Natural refrigerants are mostly innoxious and uninfluential to ozone layer destruction or global warming. And these materials are chemically stable and are mixed with mineral oil easily exhibiting their excellent thermodynamic properties. Studies on the application of HVAC system employed natural refrigerants have been actively conducted in developed countries; and among them, studies examining technologies to apply carbon dioxide refrigerant (R-744, CO_2) to heat pumps for motor vehicles or for hot water supplying system are representative ones.

Ways to improve the performance of HVAC system using CO_2 have been explored by studies broadly carried out since it was reported by Lorentzen and Pettersen [1] for the first time. In Japan,

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CO_2 refrigerant has been applied to domestic hot water supplying system. However, the performance of HVAC system or hot water supplying system employed CO_2 refrigerant are greatly vulnerable to types and conditions of varied heat sources to be evaporated or to be condensed. Thus such systems employed CO_2 refrigerant generally show comparatively poor cyclic performance comparing to systems using conventional refrigerant (R-22) due to the great performance variation characteristics induced by varied types or conditions of heat sources. Therefore, studies exploring ways to improve the cyclic efficiency of systems employed CO_2 as refrigerant have been carried out broadly since 1990's. The development of high performance HVAC system employed natural refrigerants is especially important in respect of environmental and technological aspects and accordingly, the optimization studies intended for the development of systems employing natural refrigerant like CO_2 and investigation of cycle performance characteristics thereof are needed.

Lorentzen and Pettersen [2] have compared the performance of HVAC system employed CO_2 with that of conventional system using R-12 and identified the competing performance of the system employed natural refrigerant of CO_2 and, Gentner [3] also demonstrated the competing performance of HVAC system of CO_2 to that of the HVAC system employed conventional R-134a and, they all have reported the systems adopted the refrigerant of CO_2 as an alternative environmentally friendly HAVC systems that could replace conventional HVAC systems. Experimental studies conducted by Yin et al. [4] and McEnaney et al. [5] have compared the conventional HVAC system with another system employed CO_2 as a refrigerant and have also exhibited the high capacity and COP of HVAC systems of CO_2 in an environment of low temperature despite the low level of heat transfer and COP of the HVAC system of CO_2 situated in an environment of high ambient temperature. Nekså et al. [6] have applied the refrigerant of CO_2 to hot water supplying system of which maximum pressure in high pressure area reached 11MPa through experiment; and Hwang and Radermacher [7] conducted a theoretical study that applied CO_2 to hot & chilled water supplying system as a refrigerant and demonstrated the performance enhancement of about 10% of the system compared to conventional system employed the R-22 refrigerant. Kauf [8] have identified the presence of discharge pressure enabling the maximum performance coefficient in the system employed CO_2 as a refrigerant and derived an experimental formula of optimized discharge pressure. Baek et al. [9] performed the study simultaneously explored the cyclic performance variation corresponded to the varied compression ratios of 1- and 2-stage compression in a two-stage compression intercooling cycle and the cyclic performance enhancement through the employment of expander. Cho et al. [10, 11] and Lee [12] et al. examined internal heat exchangers to improve the performance thereof and investigated performance characteristics of the system in accordance with varied conditions of the operation of respective systems. Chen [13] et al. have carried out an analytical study on cyclic pressure changes and resulting performance of gas cooler employed an internal heat exchanger; and Boewe [14] have suggested that an optimal operation of HVAC system of CO_2 can be enabled solely by the control of opening of electronic expansion valves.

Thus, more comprehensive and systematic studies on cycle characteristics of CO_2 heat pump system are required to develop the performance of HVAC system using CO_2. The study on the optimal operation conditions of CO_2 heat pump system is necessary to develop a compact and high performance HVAC system. In the present study, the performance characteristics in a cooling cycle of CO_2 heat pump system that employed heat sources of air and water are investigated experimentally. To investigate optimal operation conditions of CO_2 heat pump under a cooling condition, a CO2 cycle loop is constructed. And factors of refrigerant charge amount, EEV opening and performance of CO_2 heat pump by internal heat exchanger are examined and measured through an air enthalpy calorimeter. These results can be utilized in the design of a compact HVAC system using a CO_2 refrigerant.

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2. Experimental equipment and method

2.1. CO_2 heat pump system

The CO_2 heat pump system consisted of the inverter rotary compressor of 13.0 cc/rev BLDC type, the outdoor unit equipped with 3-columns fin tube heat exchanger, the indoor unit of cassette ducted type equipped with 1-column fin tube heat exchanger, the 4-way valve, and EEVs (electronic expansion valves) to evaluate the performance characteristics of the system.

To measure the temperature and pressure in the CO_2 cycle, the thermocouple of t-type and pressure gauge were installed; and to measure the power consumption of the whole system including the compressor, the digital power-meter was used. Measurements of pressure gauges and thermometers installed in each interval were obtained through the connected data acquisition system. To supply fluids of different conditions to the gas cooler and evaporator, the constant temperature water bath and the constant temperature - humidity chamber with 2 rooms were employed. Besides, the multi-nozzles enabled the measurement of air flow rate, dry bulb temperature, wet bulb temperature, and differential pressure was used to measure the air flow rate and enthalpy.

2.2. Experimental method

The CO_2 heat pump system employed in the performance test was the refrigerant-to- water system. Under the condition of cooling operation, the refrigerant charge amount, opening of EEV, and performance factors of internal heat exchanger were tested. For the acquisition of data to be used for the analysis and evaluation of system performance, the data corresponded to conditions of temperature variation range of ±0.1℃, pressure variation range of ±5kPa, and flow rate variation range of ±0.2g/s lasted for over 15 minutes were obtained through the data acquisition system. Table 1 represents the conditions applied to the performance test.

Table 1. Test condition of air and water source heat pump

		Test condition	
Water source	Gas cooler	Temperature [℃]	25
		Flow rate [ℓ/min]	20
	Evaporator	Temperature [℃]	12
		Flow rate [ℓ/min]	17
Air source	Indoor	Dry bulb [℃]	27
		Wet bulb [℃]	19.5
	Outdoor	Dry bulb [℃]	35
		Wet bulb [℃]	24

The cooling capacity of CO_2 heat pump system can be obtained from the flow rate and difference in temperatures between inlet and outlet of air side of evaporator. The heat balance of the heat transfer rate measured from air side of and water side was remaining within ±4%. The heat transfer rate for the cooling of indoor side is defined as in the following equation (1).

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$$Q_{cr} = \frac{Q_{vr}(h_{a1}-h_{a2})}{v_n(1+x_n)} + Q_{t1} \quad (1)$$

Here, Q_{cr}, Q_{vr}, h_{a1}, h_{a2}, v_n, x_n, and Q_{t1} respectively denote the whole heat quantity determined for the cooling of indoor side (W), measurement of indoor side air flow rate of the system (m^3/s), intake enthalpy of indoor side (J/kg), discharge enthalpy of indoor side (J/kg), specific volume at the point of air flow rate measurement (m^3/kg), absolute humidity at the point of air flow rate measurement (kg/kg), and heat penetration (W) of the measurement device.

4 EEVs were employed to optimize the cooling cycle of CO_2 system, and the configuration of EEVs enables an independent control of each valve. EEVs of 1, 3, and 4 were designed for the function of optimization of cooling cycle; and EEV 2 was designed for the function to optimize the internal heat exchanger. Specifications applied to the CO_2 heat pump system are summarized in Table 2. The optimal point of efficiency of cooling system was explored through the test of cycle optimization carried out by varying the amount of refrigerant and opening of EEVs. Under the standard temperature condition for the cooling, the test started with the compressor frequency of 45Hz that yields the highest energy efficiency and the initial amount of 2,800g of refrigerant charge that varied by each increment of 200g for the test. The configuration of cycle optimization test is illustrated in Figure 1.

Table 2. Specifications of CO2 heat pump system

	Specifications	
Compressor	Type	2 Stage rotary
	Volume [cc]	13
Indoor heat exchanger	Fin type	Louver
	Fin pitch [mm]	1.5
	Row/Column number	2, 28
	Tube type	Micro groove
	Tube outer diameter	7
Outdoor heat exchanger	Fin type	Louver
	Fin pitch [mm]	1.5
	Row/Column number	3, 66
	Tube type	Micro
	Tube outer diameter	7
Internal heat exchanger	Type	Shell-tube
	Tube length [m]	4
	Water tube outer diameter [mm]	19.05
	CO_2 tube outer diameter [mm]	3.18

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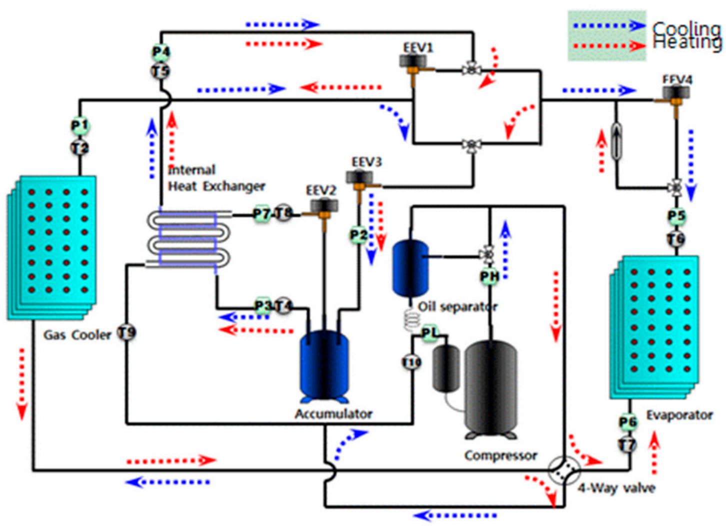

Figure 1. Schematic diagram of CO2 refrigerant cycle

3. Results and Discussion

3.1. Refrigerant charge amount and EEV

The performance of heat pump system depends on the amount of refrigerant charged. To observe the changes in cooling capacity corresponded to varied amount of refrigerant charge, the amount of refrigerant charge was increased by 200g from the initial charge of 2,800g to 6,800g. Figure 2 shows the changes of cooling capacity corresponded to the varied amount of refrigerant charge under the compressor frequency of 45Hz. The cooling capacity tended to increase along with the increase of the amount of refrigerant charge to the level of 5,000g～5,600g and thereafter, it tended to decrease gradually. This was owing to the decreased cooling capacity of heat pump caused by the reduction of superheating region at the extent beyond certain level of the amount of refrigerant charge, despite that the cooling capacity of heat pump increased owing to the heat transfer enhancement of the evaporator, when the amount of refrigerant charge increased. Since the temperature of refrigerant at the outlet of the expansion device(that is, at the inlet of evaporator) was increased by the increased opening of EEV, the temperature difference between refrigerant side and air side was reduced and then the cooling capacity of CO_2 heat pump system decreased. And the power consumption of compressor was increased along with the increase of refrigerant charge amount that caused the increase of mass flow rate of refrigerant flown into the compressor. The inlet and outlet pressure and temperature of compressor changed according to varied amount of refrigerant were measured and thereby the increase of inlet and outlet pressure of compressor was observed. The difference between measurements of inlet and outlet pressure of compressor was increasing along with the increased amount of refrigerant charge that brought about the increase of power consumption. At the level of 100% of the opening of EEVs 3 and 4, the outlet pressure of compressor marked the level of about 2.2 times of inlet pressure with the 6,800g of charged amount of refrigerant; and with the increase of the charged amount of refrigerant from 2,800g to 6,800g, both the measurements of inlet ($35kg_f/cm^2 \rightarrow 47kg_f/cm^2$) and outlet ($77kg_f/cm^2 \rightarrow 103kg_f/cm^2$) pressures of compressor were increased.

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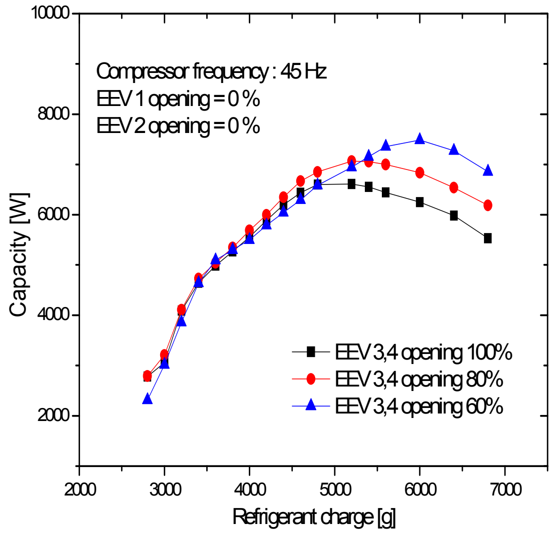

Figure 2. Variations of cooling capacity with refrigerant charge amount

Figure 3 shows the changes of COP according to varied amount of refrigerant charge at 45Hz of compressor frequency. As it was illustrated in Figure 2, the COP of heat pump increased gradually and peaked in the range of about 5,000g ~ 5,600g of charged refrigerant amount and then tended to decrease by the increase of power consumption and changes in cooling capacity. This result means that the increase of refrigerant charge amount can increase the cooling capacity of heat pump to a certain extent but, it also increases the level of power consumption thus the optimal level of refrigerant charge amount should be identified for the cycle optimal operation. COP also increased together with the decreased opening of EEVs. This result suggests the COP of system can be dependent on the varied opening of EEVs and there can be certain levels of refrigerant charge amount and opening of electronic expansion valves for the optimal system operation. In this study, the optimal levels of refrigerant charge amount and openings of EEVs of 3 and 4 were obtained to be 5,600g and 60%.

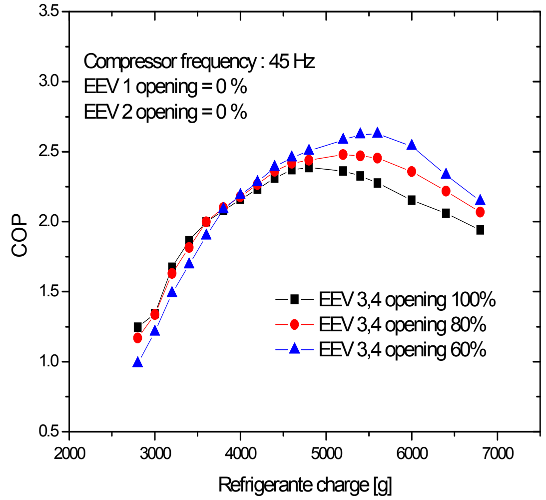

Figure 3. Variations of COP with refrigerant charge amount

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Figure 4 shows the temperature changes of outlet sides of compressor which were greater than those of inlet sides of compressor varied at each level of the opening of EEVs (opening : 100%, 80%, and 60%) and the amount of refrigerant charge. The temperature increase of inlet sides of compressor was about 16℃ along with the increased amount of refrigerant charge, and contrarily, the outlet temperature decreased greatly by about 55℃.

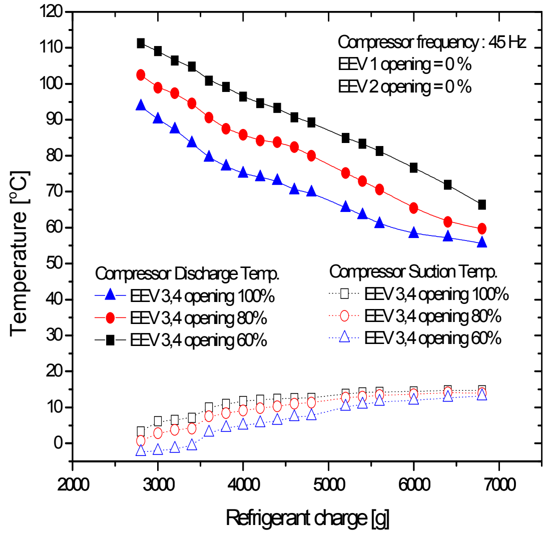

Figure 4. Variations of temperature with refrigerant charge amount and opening of EEV

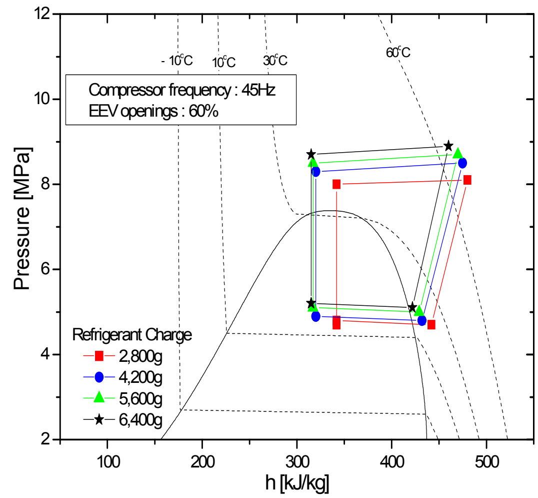

Figure 5. Variation of P-h diagram with refrigerant charge amount

Figure 5 shows the status of each part of the system plotted on the P-h diagram that represents changes of each part corresponded to each level of 2,800g, 4,200g, 5,600g, and 6,400g of refrigerant charge amount. In the case of refrigerant charge amount of 2,800g, the difference in enthalpy between outlet sides of compressor and gas cooler was small and thereby the decrease in cooling capacity was observed. On the contrary, the big increase in cooling capacity was identified with the increase of the charged amount of 4,200g of refrigerant despite the small difference in outlet pressure. However, when the amount of refrigerant charge increased above 5,600g, the degree of the increase of cooling capacity became smaller or almost constant. Also, the decrease of refrigerant quality at the inlet of evaporator was identified in accordance with the increase of refrigerant charge amount. In particular,

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the refrigerant quality decreased greatly with the increase of refrigerant charge amount from 2,800g to 4,200g. As the CO_2 heat pump system has the characteristics of great change of refrigerant quality of inlet of evaporator depending on changes of pressure of gas cooler, the optimal cooling capacity of CO_2 system can be secured with the pressure of gas cooler increased beyond certain level.

3.2. *Internal heat exchanger*

The heat exchanger applied to the heat pump system decreases the temperature of the inlet of EEVs by the heat exchange between high pressure refrigerant at the outlet of gas cooler and the refrigerant of low temperature at the outlet of evaporator. This reduces the refrigerant quality of the inlet of evaporator and thereby increases the cooling capacity of system. The power consumption of system also increases, but, as the outlet temperature of compressor is increased. In the present study, the opening of EEV 2 that influences the internal heat exchanger was varied to evaluate the performance of heat pump system employed the internal heat exchanger. By the results obtained from the previous experiment on refrigerant charge amount and EEV, the values of refrigerant charge amount and opening of EEV placed at evaporator side were determined to be 5,600g and 60% respectively.

Figures 6 and 7 show the power consumption and cooling capacity of the system varied in accordance with the increase of the opening of EEV 2 that influenced the internal heat exchanger. In the case of the increase of opening of EEV, the temperature of refrigerant at the outlet of gas cooler decreased by the heat exchange between refrigerants of the outlets of gas cooler and evaporator. This also caused the increase of cooling capacity by the decrease of inlet temperature of EEV. By the increased opening of EEV, however, the opening of EEV for the optimal COP exists because of the differential degree of increase in cooling capacity and power consumption. The opening of EEV for optimal COP was 20%, and then the increase in cooling capacity was about 4.4%. And the power consumption of system also increased by about 4.0% due to the increased temperature and pressure of compressor. Thus, the increase of COP was about 0.5%.

Figure 8 shows the pressure and temperatures of the inlet and outlet of internal heat exchanger varied in accordance with the changes in the opening of EEV 2. As showed in the figure, the pressure changes and temperatures of the inlet and outlet of internal heat exchanger decreased along with the increased opening of EEV. At the extent beyond the level of 30% of the opening of EEV, the constant temperature difference due to heat exchange was observed and the difference in pressure drop due to the internal heat exchanger was not observed.

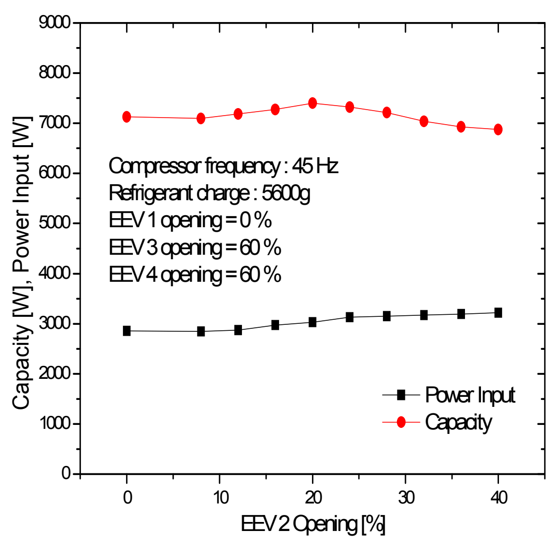

Figure 6. Variation of cooling capacity, power input with opening of EEV 2

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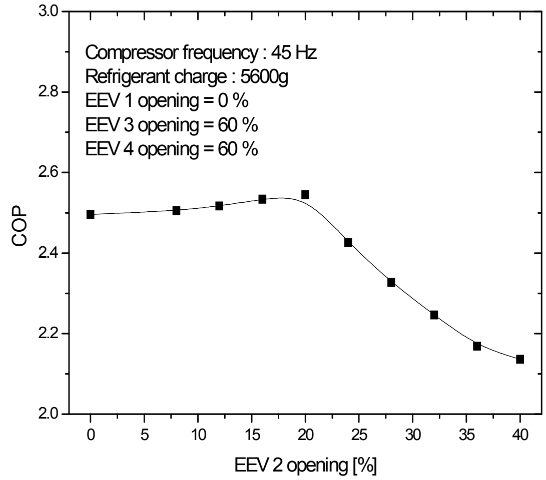

Figure 7. Variation of COP with opening of EEV 2

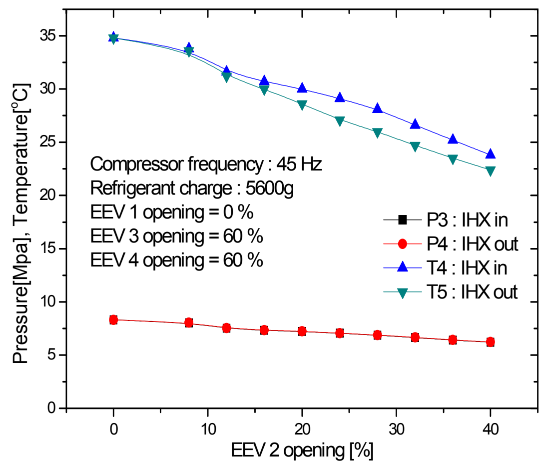

Figure 8. Variation of temperature, pressure with opening of EEV 2

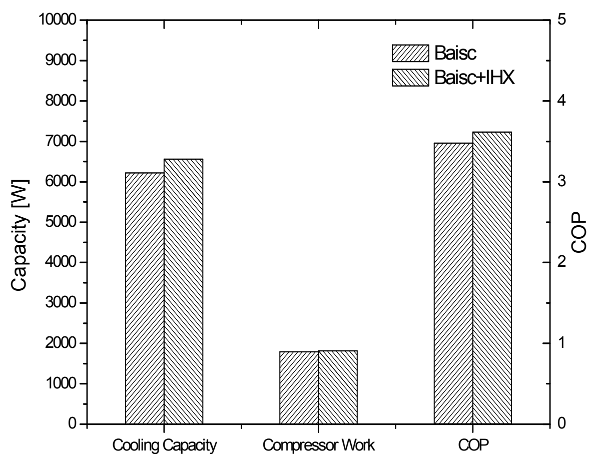

Figure 9. Variation of cool capacity, compressor work and COP according to internal heat exchanger in the cooling mode

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Figure 9 shows the results of performance experiment of CO_2 heat pump system with internal heat exchanger. The application of internal heat exchanger brought about the increase of cooling capacity of about 4%. In the case of the heating system employed the internal heat exchanger, the temperature at outlet side of gas cooler would be decreased by the heat exchange with the low-pressure stage at outlet side of evaporator. And in the case of small opening of EEV, the outlet pressure and temperature of compressor would be increasing and thereby the difference in temperature of refrigerant of gas cooler also increases. By the reduced opening of EEV, but, the flow rate of refrigerant would be decreasing.

5. Conclusions

In this study, the experiments to investigate cycle performance characteristics and the optimal operation conditions of CO_2 heat pump system employed heat sources of air and water were carried out under a cooling condition. The performance of heat pump system varied in accordance with changes in the amount of refrigerant charge, opening of EEVs, and internal heat exchanger was examined through the air enthalpy calorimeter. From the present experimental work, the following conclusions are summarized as follows:

(1) The cooling capacity tended to increase along with the increase of the amount of refrigerant charge and thereafter it tended to decrease gradually. In the range below 5,000g of the amount of refrigerant charge, the increasing trends of cooling capacity were almost similar to each other irrespective of the degree of opening of EEVs, however, in the range beyond the amount of 5,000g of refrigerant charge, the cooling capacity was changing according to changes of the opening of EEVs.

(2) The cooling capacity of heat pump system was decreasing with the increase of the opening of EEVs. As the power consumption of compressor was increased by the increase of charged amount of refrigerant, the COP performance curve appeared to be reducing greatly at the extent over 5,000g of the amount of refrigerant charge. From the experiment results, the optimal opening (of 60%) of EEV and the amount (of 5,600g) of refrigerant charge for the optimization of system operation were identified.

(3) The temperature change of the outlet side of compressor by the change of amount of refrigerant charge and opening of EEVs appeared about 3.5 times larger than that of the inlet side of compressor. Besides, the opening of EEV 2 that influences the internal heat exchanger affects the cooling capacity and power consumption of heat pump system. The cooling capacity of heat pump system would be increasing in accordance with the increased opening of EEV. However, owing to the differential increase of cooling capacity and power consumption, there will be an optimal opening of EEV which should be reflected in the system design as an operational factor. The present experiment results showed that opening of EEV identified the optimal COP was about 20% and then the cooling capacity was increased about 4.4%.

Acknowledgments: This work was supported by the Sun Moon University Research Grant of 2014.

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