

EFFECTS OF GAS INJECTION ON THE HEATING PERFORMANCE OF A TWO-STAGE HEAT PUMP WITH REFRIGERANT CHARGE AMOUNT

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Abstract: For heat pumps used in the cold region, it is very important to obtain appropriate heating capacity. Several studies using a variable speed compressor and an additional heater have been performed to enhance heating capacity at low ambient temperatures. However, for outdoor temperature conditions below -15°C , it is still difficult to obtain enough heating capacity above the rated heating capacity. In recent studies, a two-stage cycle using gas injection showed noticeable heating performance improvement at low temperature conditions. In this study, a two-stage heat pump system with a rated capacity of 3.5 kW was tested. The heating performance of the gas injection cycle was measured and analyzed by varying refrigerant charge amount and EEV opening at the standard heating condition. The heating performance of the gas injection cycle was compared with that of the non-injection cycle with a variation of refrigerant charge amount. The heating capacity and COP of the injection cycle were improved by 2-10% at the optimal charging condition.

Key Words: *Gas Injection, Two-stage cycle, R-410A, Refrigerant charge, Heat pump*

1 INTRODUCTION

When an air source heat pump is operated at low ambient temperatures below 0°C , the heating capacity significantly decreases due to the increase of irreversibility in the compression process and the reduction in mass flow rate. In a typical heat pump, the heating performance was decreased by approximately 40% at the ambient temperature of -15°C . In the cold climate region, it is very important to obtain appropriate heating capacity at low temperature conditions. Several methods to enhance the heating performance of a heat pump have been studied. The heating performance was significantly improved by application of a variable speed compressor. When the variable speed type compressor was applied to the heat pump system, the heating capacity at the conditions of -15°C and 90-100 Hz was 70-80% of the rated heating capacity (Kim, 2001). An additional heat source such as an electric heater or gas heater was also introduced. The effects of an intermediate heat exchanger were also studied to increase the superheat at the evaporator exit. In addition, a 2-stage cycle, which consists of a flash tank and two compressors, could be helpful to increase COP and heating capacity of the heat pump. By applying these technologies, some industries of Japan reported that the heating capacity of a heat pump was more than 100% of the rated capacity at low temperature conditions (Hwang, 1998). However, when additional heat source was applied, the system efficiency decreased due to additional power consumption. A two-stage cycle consisted of a flash tank and two compressors is appropriate to increase COP and ensure reliability by using inter-cooling effects between the exit of the first-stage compressor and the inlet of the second-stage compressor. However, due to high

cost and large space requirements, the application of the two-stage cycle was limited to advanced systems.

In this study, a 2-stage heat pump system with gas injection was tested to improve heating performance and efficiency at the standard temperature condition. The rated heating capacity of the heat pump was 3.5 kW. A twin rotary compressor, which consisted of two separate cylinders with one motor, was used in the heat pump. The inter-cooling was made by the heat exchange between the exit of the first-cylinder and the inlet of the second-cylinder. R-410A was used as a working fluid. The objective of this study is to investigate operating characteristics of the gas injection cycle in the heating mode. The two-stage heat pump was tested by varying refrigerant charge amount and EEV opening.

2 EXPERIMENTAL FACILITY AND TEST CONDITIONS

2.1 Experimental Facility

Figure 1 shows a schematic diagram of the experimental setup designed to measure the heating performance of the heat pump with gas injection. The test setup consists of a twin rotary compressor, an indoor unit, an outdoor unit, a gas/liquid separator, two expansion devices (1st & 2nd EEV) for controlling intermediate and low pressure, and an expansion device (3rd EEV) for adjusting injection mass flow rate. The refrigerant gas from the gas/liquid separator was injected into a mixing port between two cylinders of the compressor. Operating frequency of the compressor was controlled by a commercial BLDC motor driver. Both indoor and outdoor units used finned-tube heat exchangers (3.6 kW). The indoor heat exchanger consisted of 15 steps, 2 rows, and 4 paths, whose out diameter was 5 mm. The outdoor heat exchanger consisted of 22 steps, 2 rows, and 3 paths, whose out diameter was 6 mm. The EEV was driven by a stepping motor using the 1-2 excitation control method. All experiments were performed in a psychrometric calorimeter based on the KS Standard (KS C 9306).

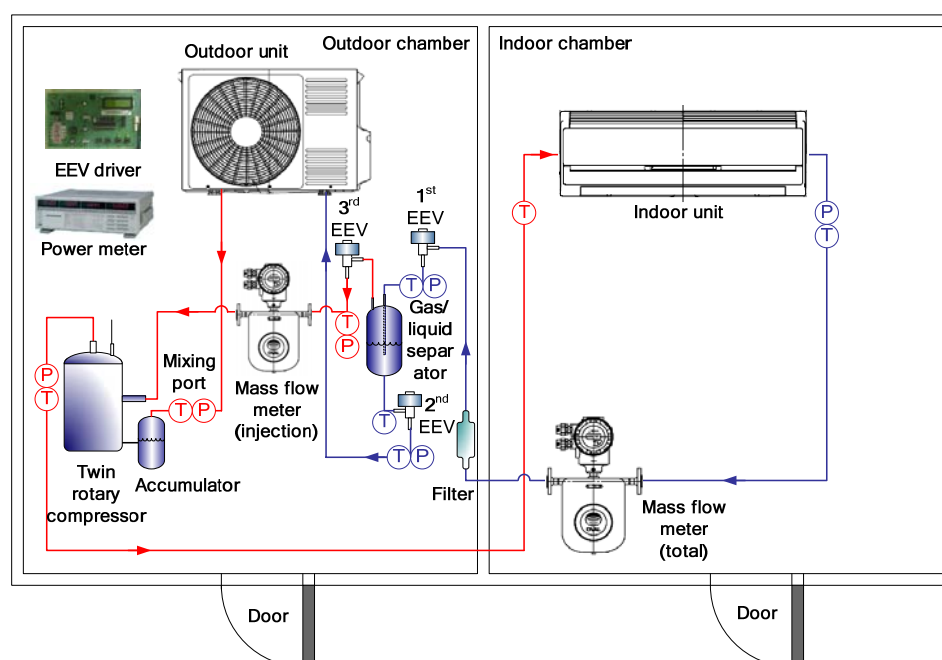


Figure 1: Schematic diagram of the experimental setup

Table 1; Test conditions

Parameters	Value
Temperature (°C)	Indoor(db/wb): 20/15, Outdoor (db/wb): 7/6
Charge amount(g)	900, 1000, 1100, 1200
Compressor frequency (Hz)	60
1 st EEV opening (%)	From 40% to 80% in a step of 5%
2 nd EEV opening (%)	From 40% to 80% in a step of 5%
3 rd EEV opening (%)	From 0% to 40% in a step of 10%

2.2 Test Conditions

Table 1 shows test conditions of the heat pump with gas injection. The heating performance was measured by varying refrigerant charge amount from 900 g to 1,200g and EEV opening from 40% to 80%. Indoor and outdoor room temperatures were maintained at the heating standard conditions of the KS Standard (KS C 9306), and the compressor frequency was fixed at 60 Hz. The 1st & 2nd EEV openings were controlled at the same value. The injection EEV was used to shift operation mode between the injection and the non-injection cycle.

During the tests, the COP, heating capacity, power consumption, refrigerant temperature and pressure at major locations of the system were measured. The compressor power input was measured by a power meter with an uncertainty of $\pm 0.01\%$. The total and injection mass flow rates were measured by Coriolis Effect flow meters with uncertainties of ± 0.2 and ± 0.1 , respectively. Refrigerant pressures were monitored by using pressure transducers with an uncertainty of $\pm 0.13\%$. Refrigerant and air temperatures were measured by using T-type thermocouples with an uncertainty of $\pm 0.2^\circ\text{C}$. Airside heating capacity was determined by utilizing the air enthalpy method (ASHRAE Standard 116, 1993) based on both air flow rate and enthalpy difference across the condenser. The air flow rate in the condenser was measured by using an airflow chamber according to ANSI/AMCA 210 (ANSI/AMCA 210, 1985). Uncertainties of COP and heating capacity estimated by using the single-sample analysis with ASHRAE Guideline 2 (ASHRAE Guideline 2, 1986) were $\pm 2.2\%$ and $\pm 3.5\%$, respectively. Uncertainties of the experimental parameters are summarized in Table 2.

3 RESULTS AND DISCUSSION

3.1 Heating Performance with Refrigerant Charge

Operating range of the gas injection cycle is much wider than that of a typical one-stage heat pump due to the effects of inter-cooling and variable speed compressor. In addition, the gas

Table 2; Uncertainties of measured and reduced parameters

Parameters	Uncertainty	Full scale
Temperature	$\pm 0.2^\circ\text{C}$	-270 - 400 °C
Power meter	$\pm 0.01\%$ of full scale	20 kW
Mass flow meter	$\pm 0.2\%$	0 - 300 kg/h
Mass flow meter	$\pm 0.1\%$	0 - 82 kg/h
Pressure transducer	$\pm 0.13\%$	0 - 1000 psig
Electronic balance weight	± 0.5 g	41 kg
Heating capacity	2.2%	-
COP _{heating}	3.5%	-

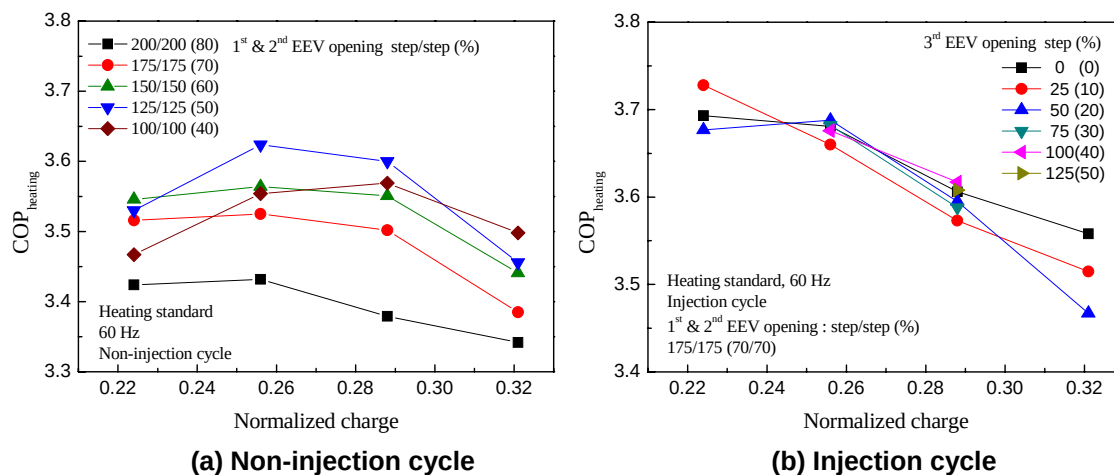


Figure 2: Variation of COP with normalized charge

injection cycle shows significantly different operating characteristics from the non-injection cycle. In this study, the performance of a 2-stage heat pump system with and without gas injection was tested by varying refrigerant charge amount, injection valve opening and EEV opening.

Optimum refrigerant charge is strongly dependent on the size of the system including each component and pipe length. To generalize these variable factors, refrigerant charge amount was normalized between 0 and 1 by the mass of the saturated vapor and liquid in the system at a room temperature of 25°C by using the method from Cho et al. (2005). When the system is totally filled with saturated vapor, the normalized charge is 0. When the system is filled with saturated liquid, the value is 1. The normalized charge was calculated by

$$\text{Normalized charge} = \frac{m_{\text{actual}} - m_{\text{vapor}}}{m_{\text{liquid}} - m_{\text{vapor}}} \quad (1)$$

Figure 2 (a) shows the variation of COP with normalized charge in the non-injection cycle. The highest COP was observed at the normalized charge from 0.256 to 0.289. At the normalized charges of 0.224 and 0.321, the COPs were optimized at EEV openings of 60% and 40%, respectively. At the normalized charges of 0.256 and 0.289, the COPs were 3.62 and 3.60, respectively, with the optimal EEV opening of 50%. Figure 2 (b) shows the variation of COP with normalized charge in the injection cycle. The third EEV was used as an injection valve, and 0% EEV opening indicates the non-injection cycle. At low and over charged regions, frost formed on heat exchanger coil due to low evaporating pressure, resulting in the reduction of system COP. The increase of first and second EEV openings from 50% to 70% in the injection cycle yielded the increase in system mass flow rate. The highest COP was observed at the normalized charge of 0.256 and the third EEV openings from 20% to 50%.

Figure 3 (a) shows the variation of COP with normalized charge. The COP of the injection cycle was 1.8~4% higher than that of the non-injection cycle at the normalized charges from 0.224 to 0.256. Based on the trends of COP with refrigerant charge amount, it was found that the optimal charge range was varied with cycle option. While the highest heating performance in the non-injection cycle was observed at the normalized charge from 0.256 to 0.289, the highest COP in the injection cycle was observed at the normalized charge from 0.224 to 0.256. Refrigerant flow rate and heat transfer efficiency increased due to the inter-

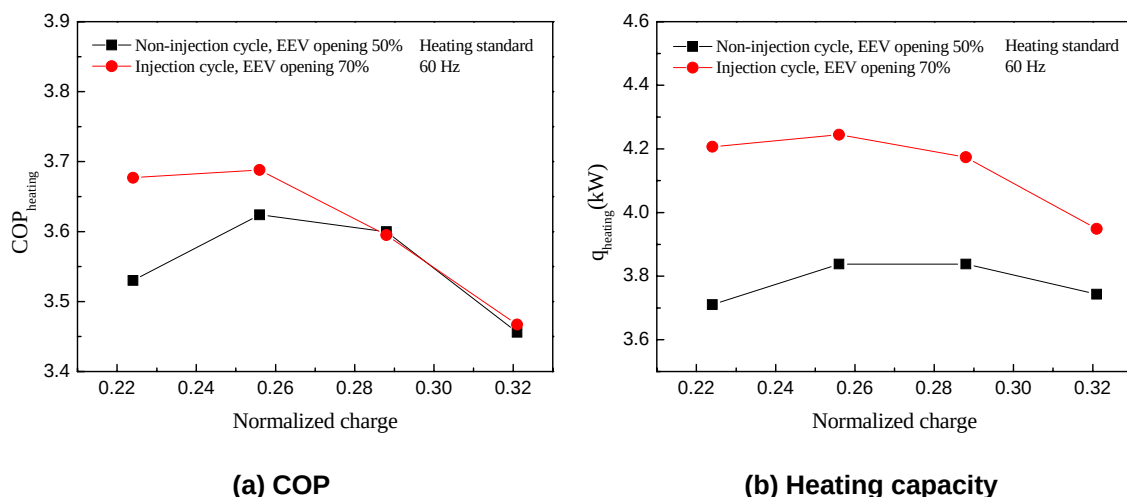


Figure 3: Variations of COP and heating capacity with normalized charge

cooling effects in the gas injection cycle. Therefore, the system performance and efficiency were optimized at relatively lower refrigerant charge than that of the non-injection cycle. Figure 3 (b) shows the variation of heating capacity with normalized charge and cycle option. While the highest heating capacity of the non-injection cycle was observed at the normalized charge from 0.256 to 0.289, that of the injection cycle was observed at the normalized charge from 0.224 to 0.256. The heating capacity of the injection cycle was 10.6% higher than that of the non-injection cycle at the normalized charge of 0.256.

Figure 4 shows the variation of power consumption with normalized charge. For the non-injection cycle, the power consumption was linearly increased with refrigerant charge amount. The power consumption of the injection cycle was 8% higher than that of the non-injection cycle due to the increase in system flow rate with the gas injection even though the inter-cooling effects slightly reduced the power consumption of the injection cycle. When the injection cycle has a similar mass flow rate as the non-injection cycle by reducing compressor frequency, the power consumption of the injection cycle may decrease due to the inter-cooling effects. For the injection cycle, the power consumption steadily increased and slightly decreased at the normalized charge of 0.321.

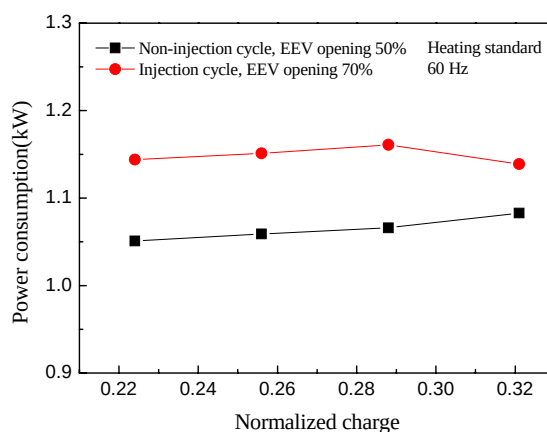


Figure 4: Variation of power consumption with normalized charge

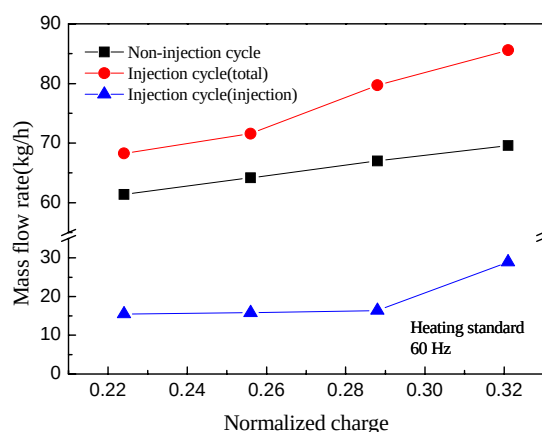


Figure 5: Variation of mass flow rate with normalized charge

3.2 System Characteristics with Refrigerant Charge

Figure 5 shows the variation of refrigerant mass flow rate with normalized charge. For the injection cycle, both total and injection mass flow rate increased with the increase of refrigerant charge amount. At the normalized charge of 0.256, the total mass flow rate of the injection cycle was 71.6 kg/h, which was 10.8% higher than that of the non-injection cycle. At the normalized charge of 0.321, both the injection mass flow rate and the injection flow rate ratio, which is defined by the ratio of the injection to the total mass flow rate, rapidly increased. This rapid change indicates the transition from gas to liquid injection due to the excessive total mass flow rate beyond the capacity limit of the separator. Therefore, the overcharge in the gas injection cycle can cause severe performance degradation due to wet compression and unstable cycle operation resulted from liquid injection.

Figure 6 (a) shows the variation of compression ratio with normalized charge. For the non-injection cycle, the compression ratio was relatively constant at normalized charges from 0.224 to 0.321, but the compression ratio in the injection cycle decreased with the increase of refrigerant charge. The compression ratio is very closely related with the variation of refrigerant flow rate. The increasing rate of the refrigerant flow rate in the injection cycle was relatively higher than that in the non-injection cycle. Therefore, the reduction of the

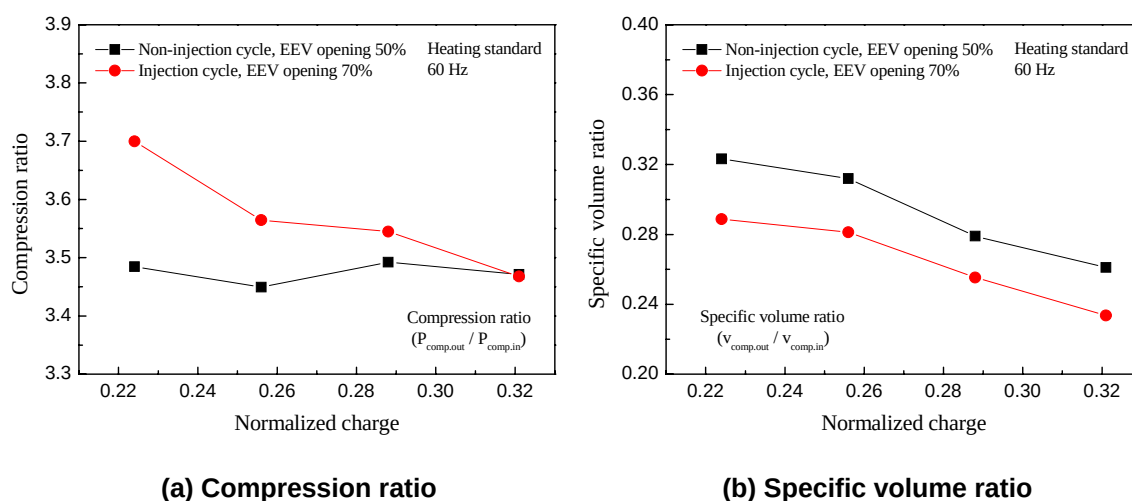


Figure 6: Variations of system characteristics with normalized charge

compression ratio in the injection cycle with the increase of refrigerant charge is inevitable. Figure 6 (b) shows the variation of specific volume ratio with normalized charge. The refrigerant specific volume in the compressor discharge port decreased after the compression process. The decrease of the refrigerant specific volume in the injection cycle was higher than that of the non-injection cycle due to the inter-cooling effects. The reduction of the specific volume can cause the increase of efficiency and total flow rate during the compression process. At the normalized charge of 0.256, the refrigerant specific volume in the injection cycle was 9.8% lower than that of the non-injection cycle.

4 CONCLUSIONS

The heating performance of the gas injection cycle was compared with that of the non-injection cycle with a variation of refrigerant charge amount. Optimal performances with and without gas injection were observed at the normalized charge from 0.256 to 0.289, and from 0.224 to 0.256, respectively. The heating capacity and COP of the injection cycle were 10.6% and 2% higher than those of the non-injection cycle, respectively. The mass flow rate of the injection cycle was 10.8% higher than that of the non-injection cycle, and the ratio of injection flow rate to total flow rate in the injection cycle was 22.1% higher than that of the non-injection cycle. The increase of mass flow rate with the rise of refrigerant charge amount was more noticeable in the injection cycle, and the system performance of the injection cycle decreased rapidly at overcharged conditions. The compression ratio of the injection cycle decreased with the rise of refrigerant charge amount due to the significant increase of mass flow rate. The specific volume ratio of the injection cycle was 9.8% lower than that of the non-injection cycle, which increased system flow rate.

5 ACKNOWLEDGEMENTS

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NOMENCLATURE

COP	coefficient of performance	(-)
q	heating capacity	(kW)
m_{actual}	actual refrigerant mass	(kg)
m_{vapor}	mass at saturated vapor	(kg)
m_{liquid}	mass at saturated liquid	(kg)

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