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Effects of Gas Injection on the Heating Performance of a Two-Stage Heat Pump Using a Twin Rotary Compressor with Refrigerant Charge Amount

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Abstract

For heat pumps used in a 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 value. In recent studies, the application of gas injection technique into a two-stage heat pump yielded noticeable heating performance improvement at low temperature conditions. In this study, the heating performance of a two-stage gas injection heat pump with a rated capacity of 3.5 kW was measured and analyzed by varying refrigerant charge amount and EEV opening at the standard heating condition. The heating performance of the two-stage gas injection heat pump was compared with that of a two-stage non-injection heat pump. The heating capacity and COP of the two-stage gas injection heat pump were improved by 2-10% at the optimal charging condition over those of the two-stage non-injection heat pump.

Key words: Gas Injection, Two-Stage Cycle, R-410A, Refrigerant Charge, Heat Pump

Nomenclature

COP : coefficient of performance [-]
comp. : compressor [-]
db : dry bulb [-]
freq. : frequency [Hz]
ID : indoor [-]
m_{actual} : actual refrigerant mass [kg]
m_{vapor} : mass at satuated vapor [kg]
m_{liquid} : mass at satuated liquid [kg]
OD : outdoor [-]
q : heating capacity [kW]
v : specific volume [m^3/kg]
wb : wet bulb [-]

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 is 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, the heating capacity at the conditions of -15°C and 90-100 Hz was 70-80% of the rated heating capacity[(1)]. An additional heat source such as an electric heater or gas heater was also introduced. However, when additional heat source was applied, the system efficiency decreased due to additional power consumption. In addition, a two-stage compression cycle, which consists of a flash tank and two compressors, can be very helpful to increase COP and heating ca-

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pacity in a heat pump. By applying these technologies, some companies in Japan reported that the heating capacity of a heat pump was more than 100% of the rated capacity at low temperature conditions(2).

In this study, a two-stage heat pump 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. Operating characteristics of the two-stage gas injection heat pump (called as "gas injection cycle") were also compared with those of the two-stage non-injection heat pump (called as "non-injection cycle") in the heating mode. The heat pump was tested by varying refrigerant charge amount and EEV opening.

2. Experimental setup and test procedure

2.1 Experimental setup

Fig. 1 shows the schematic diagram of the experimental setup designed to measure the heating performance of the gas injection cycle. 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

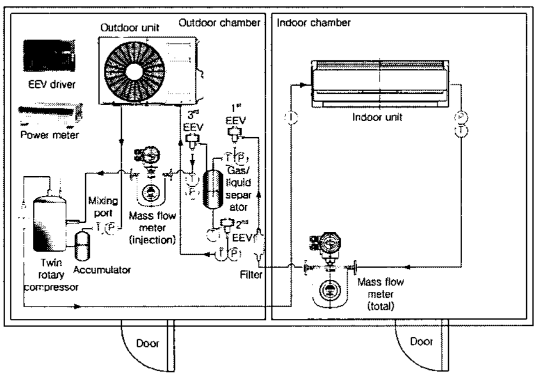

Fig. 1. Schematic diagram of the experimental setup.

indoor and outdoor units used finned-tube heat exchangers with the rated capacity of 3.6 kW. The indoor heat exchanger consisted of 15 steps, 2 rows, and 4 paths, whose tube outer diameter was 5 mm. The outdoor heat exchanger consisted of 22 steps, 2 rows, and 3 paths, whose tube outer 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(3).

2.2 Test procedure

Table 1 shows test conditions of the gas injection cycle. The heating performance was measured by varying refrigerant charge amount from 900 g to 1,200 g and EEV opening from 40% to 80%. Indoor and outdoor room temperatures were maintained at the heating standard conditions of the KS Standard(3), and the compressor frequency was fixed at 60 Hz. The openings of the first- and second-EEV were controlled to be the same value. The injection EEV was used to shift operating mode from the injection to 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 ±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 uncertainly of ±0.13%. Refrigerant and air temperatures were measured by using T-type thermocouples with an uncertainty of ±0.2°C. Airside heating capacity was determined by utilizing the air enthalpy method based on both air flow rate and enthalpy difference across the condenser according to ASHRAE Standard 116(4).

Table 1. Test conditions.

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

Table 2. Uncertainties of measured and reduced parameters.

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

The air flow rate in the condenser was measured by using an airflow chamber according to ANSI/AMCA 210[5]. Uncertainties of COP and heating capacity estimated by using the single-sample analysis with ASHRAE Guideline 2[6] were ±2.2% and ±3.5%, respectively. Uncertainties of the experimental parameters are summarized in Table 2.

3. Results and discussion

3.1 Heating performance with refrigerant charge

The operating range of the gas injection cycle is much wider than that of a typical single-stage heat pump by using the inter-cooling effects and variable speed compressor. In addition, the gas injection cycle shows significantly different operating characteristics from the non-injection cycle. In this study, the performances of the injection and the non-injection cycles were mesured 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[7]. 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

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

From pretest results with the variation of first- and second-EEV opening, the highest COP was observed at the EEV opening of 50% and 70% in the injection and non-injection cycles, respectively. The increase of first- and second-EEV openings from 50% to 70% in the injection cycle leaded to the increase in system mass flow rate. In addition, the highest performance of the injection cycle was observed at the injection valve opening of 20%.

Fig. 2 shows the variation of COP with normalized charge and cycle option. 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 varied with the cycle option. The highest heating COP of the non-injection cycle was observed at the normalized charge between 0.256 and 0.289, while the highest COP of the injection cycle was observed at the normalized charge between 0.224 and 0.256. The refrigerant flow rate and heat transfer efficiency increased due to the inter-cooling effects in the gas injection cycle. Therefore, the system performance and efficiency were optimized at relatively lower refrigerant charge than those of the non-injection cycle.

Fig. 3 shows the variation of heating capacity with normalized charge and cycle option. The highest heating capacity of the non-injection cycle was observed at the normalized charge between 0.256 and 0.289, while 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.

Fig. 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

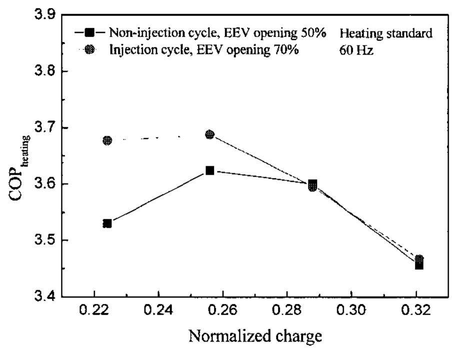

Fig. 2. Variation of COP with normalized charge and cycle option.

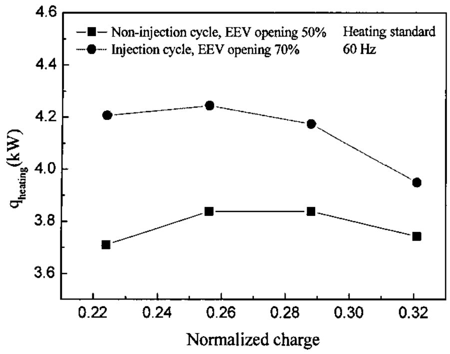

Fig. 3. Variations of heating capacity with normalized charge and cycle option.

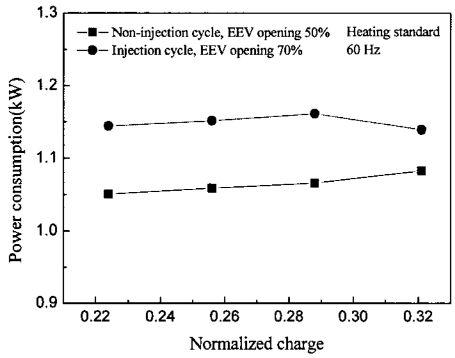

Fig. 4. Variation of power consumption with normalized charge and cycle option.

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.

3.2 System characteristics with refrigerant charge

Fig. 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.

Fig. 6 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.

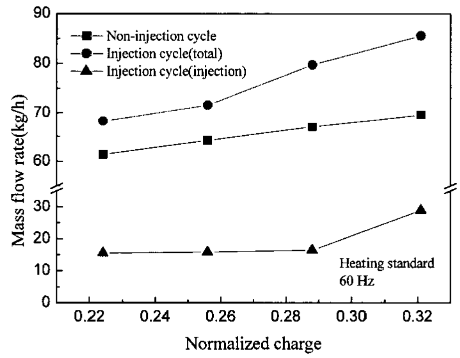

Fig. 5. Variation of mass flow rate with normalized charge and cycle option.

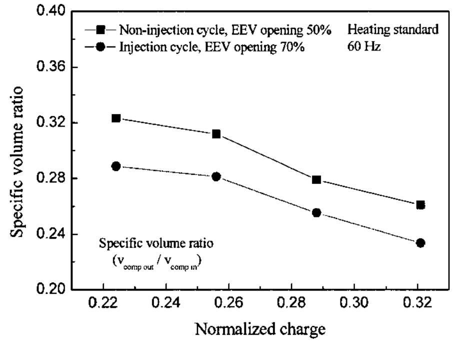

Fig. 6. Variation of specific volume ratio with normalized charge and cycle option.

Fig. 7 shows variations of high- and low-stage pressures with normalized charge and cycle option. The low-stage pressures of the injection and non-injection cycles steadily increased with normalized charge. The high-stage pressure slightly increased and then decreased in the overcharged region. The difference between high- and low-stage pressures of the injection cycle was higher than that of the non-injection cycle in the optimal charging region due to the increase of system flow rate with gas injection.

Fig. 8 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 compression ratio in the injection cycle with the increase of refrigerant charge is inevitable.

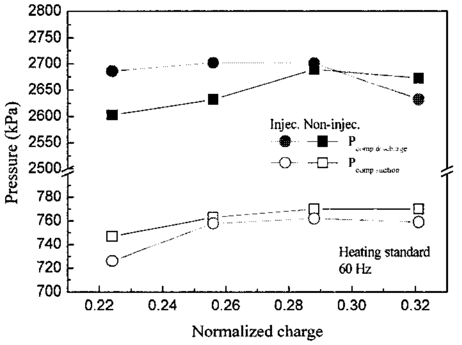

Fig. 7. Variation of high- and low-stage pressure with normalized charge and cycle option.

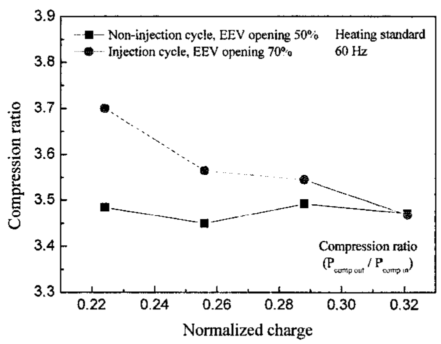

Fig. 8. Variation of compression ratio with normalized charge and cycle option.

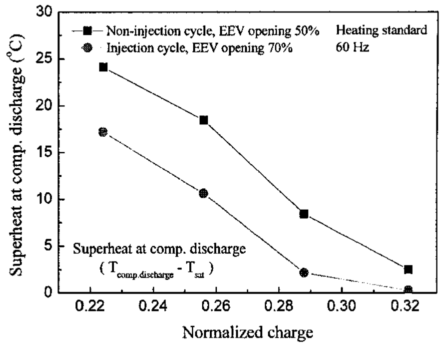

Fig. 9. Variation of comp. superheat with normalized charge and cycle option.

Fig. 9 shows the variation of compressor discharge superheat with normalized charge and cycle option. Compressor discharge superheat was linearly decreased with normalized charge in the injection and non-injection cycles. The compressor discharge superheat of the injection cycle was 6-8°C lower than that of the non-injection cycle except for the normalized charge of 0.321.

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 normalized refrigerant charges of the injection and non-injection cycles were between 0.256 and 0.289, and between 0.224 and 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. Therefore, 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. Compressor discharge superheat of the injection cycle was 6-8°C lower than that of the non-injection cycle, which can contribute to improve compressor reliability.

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