# Performance Evaluation of a Two-Stage Compression Heat Pump System With Refrigerant Charge and EEV Opening

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# PERFORMANCE EVALUATION OF A TWO-STAGE COMPRESSION HEAT PUMP SYSTEM WITH REFRIGERANT CHARGE AND EEV OPENING

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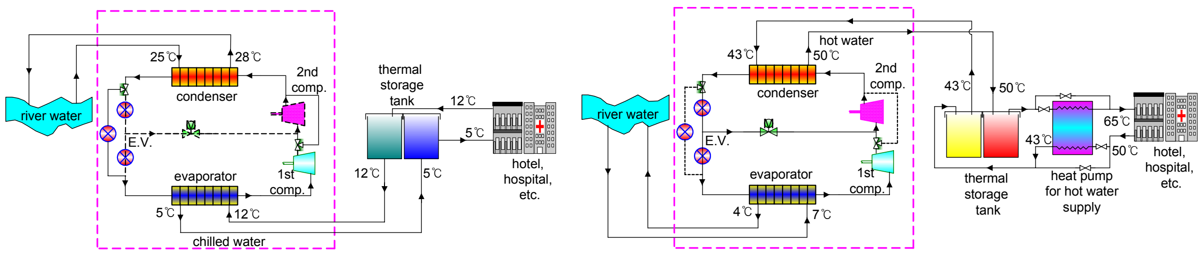
## ABSTRACT

A heat pump system using river water as a heat source operated in a single-stage cycle for the cooling and two-stage cycle for the heating to optimize system performance. The use of the two-stage compression cycle in the heating mode can improve system performance and reliability because the temperature difference between the heat source and the hot water temperature supplied for the heating is relatively large. The objective of this study is to investigate the effects of refrigerant charge on the performance of the two-stage heat pump. The experiments were carried out by varying refrigerant charge in both the cooling and heating modes. The performance of the two-stage heat pump was 23% higher than that of the conventional refrigeration system that used atmospheric air as the heat source.

## 1. INTRODUCTION

The air-to-air heat pump system has been extensively used in the cooling operation. However, the performance of the air-to-air system in the heating mode decreased with the reduction of outdoor temperature because the decrease of the temperature made the moisture contained in the air freeze and form frost on the surface of the outdoor coil. Therefore, there were some limitations on the application of the heat pump in the cold climate.

The use of river water as a heat source in winter and as a heat sink in summer yields beneficial effects on the performance of the heat pump system compared to the use of atmospheric air because the variation of river water temperature through the year is relatively small. This type of heat pump is of great interest since the use of unutilized low grade energy from various environmental sources can improve the system performance.




**< Cooling Mode >** **< Heating Mode >**

Figure 1 : Heat pump system using river water.

Figure 1 shows the heat pump system using river water for the cooling and heating. Usually, hot water supplied by the heat pump for the heating is in the range of 50-55℃. Park *et al.* (2001) illustrated applications of a two-stage compression cycle in the heat pump system with river water because the temperature difference between the heat source and the hot water temperature supplied for heating is large. Jung *et al.* (2000) analyzed multi-stage condensers that can supply hot water in much elevated temperatures as compared to conventional single-stage heat pumps by computer simulation. Baek *et al*. (2001) simulated a heat pump system for cooling and heating using sewage water as a heat source. They showed that wastewater source heat pumps could help to reduce energy consumption by 34% as compared with the conventional air-source heat pump.

In this study, the water-to-water heat pump system using river water was designed in a single-stage cycle for the cooling and two-stage cycle for the heating to optimize the system performance. The present heat pump system was tested by varying refrigerant charge and EEV opening in the cooling and heating modes. In addition, the optimum the refrigerant charge amount for the cooling and heating modes was selected.

## 2. EXPERIMENTAL SETUP AND TEST PROCEDURE

The designed cooling capacity of the present heat pump system was 3 RT using R134a as a working fluid. The cycle operated in the single-stage cycle for the cooling and two-stage cycle for the heating. Therefore, the effects of operating parameters on the performance for the cooling and heating should be considered. To make an energy efficient heat pump, the compressor should be optimized with other components. Besides, the refrigerant charge amount in the heat pump is a major parameter influencing system performance. Undercharge or overcharge of the refrigerant in the system will degrade system performance and deteriorate system reliability (Kim *et al*., 2002). The heat pump should contain an optimum amount refrigerant charge to operate with high performance over its lifetime.

Figure 2 shows the water-to-water heat pump for the cooling and heating. The test setup consisted of two compressors (low-stage scroll compressor and high-stage scroll compressor), two heat exchangers (condenser and evaporator), an inter-stage heat exchanger, a flash tank, and two expansion devices (inter-stage EEV and low-stage EEV). The EEV was used as the expansion device to control mass flow rate actively in the heat pump system.

Figure 3 shows the P-h diagram of the single-stage vapor compression cycle in the cooling mode operation. The single-stage cycle consisted of a compressor, condenser, receiver, expansion device, evaporator, and accumulator, which was the same as the conventional heat pump system. In the cooling tests, the refrigerant was charged at the water temperature of 25℃ corresponding to the river temperature in summer season. The refrigerant was added into the heat pump in 200 g increments until the maximum COP was obtained. The EEV opening was adjusted to control superheat at each operating condition.

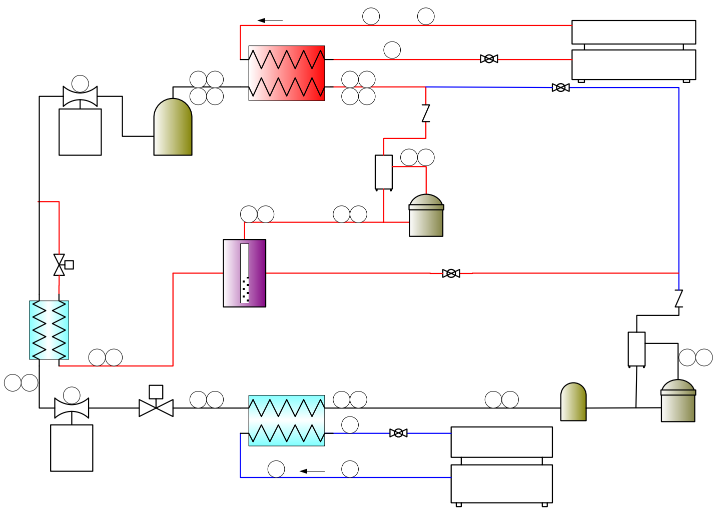


Figure 2 : Schematic of the heat pump system.

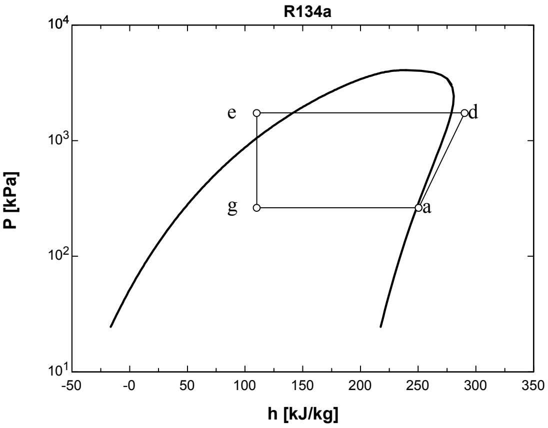



Figure 3 : P-h diagram for the cooling.

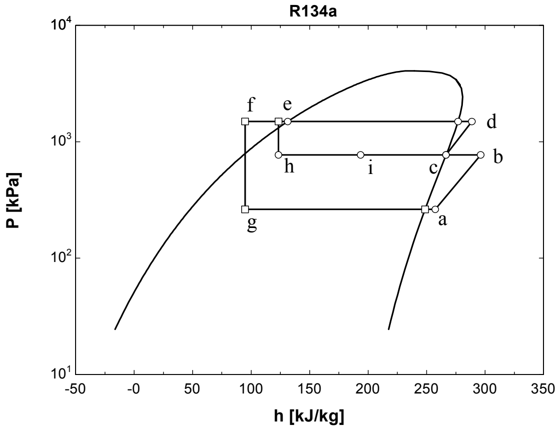



Figure 4 : P-h diagram for the heating.

Table 1 : Specifications of experimental setup

| Items | | Model | Specifications |
|---|---|---|---|
| Compressor | High-stage | Copeland ZR47K3E-TF5-501 | 1RT, 3Φ -220V-60Hz |
| | Low-stage | Copeland ZR18K4E-TF5-961 | 3RT, 3Φ -220V-60Hz |
| EEV | Inter-stage | Saginomiya DKV18 | 5 – 9 kW |
| | Low-stage | Fujikoki BD24SH | 10 – 15 kW |
| Heat exchanger | Condenser | Alfalaval plate type | 16 kW |
| | Evaporator | Alfalaval plate type | 12 kW |
| | Inter-stage | Alfalaval plate type | 2 kW |

Figure 4 shows the P-h diagram of the two-stage vapor compression cycle in the heating mode. The refrigerant from the suction line at state *a* was compressed by the low-stage compressor to the flash tank, which was located at an inter-stage pressure. The superheated refrigerant discharged from the low-stage compressor was mixed with the two-phase refrigerant that was bypassed from the condenser outlet and was expanded in the inter-stage EEV to the inter-stage heat exchanger in the flash tank. The inlet temperature of the high-stage compressor decreased due to the mixing process in the flash tank. The inter-stage heat exchange process allowed the separation of the inter-stage heat exchanger from the flash tank. This resulted in making a compact flash tank with the reduction of heat exchange in the flash tank. In the heating tests, the refrigerant was charged at the water temperature of 12℃ corresponding to the river temperature in winter season. Table 1 shows the specifications of major components.

## 3. RESULTS AND DISCUSSION

### 3.1 Cooling Mode

The refrigerant charge amount strongly affects the performance of the system. Figure 5 shows the variations of cooling capacity and compressor power as a function of refrigerant charge amount. The optimum charge amount in the cooling mode was 6.7 kg. For undercharged conditions, the compressor efficiency decreased with the increase of superheat due to the rise of the temperature at the compressor inlet. Besides, the cooling capacity decreased with the reduction of refrigerant flow rate. For overcharged conditions, the evaporating and condensing pressures increased with the increase of refrigerant charge amount. The superheat was kept constant with the decrease of EEV opening. Since the difference between condensing and evaporating pressure increased, the cooling capacity decreased with the reduction of the compressor efficiency.

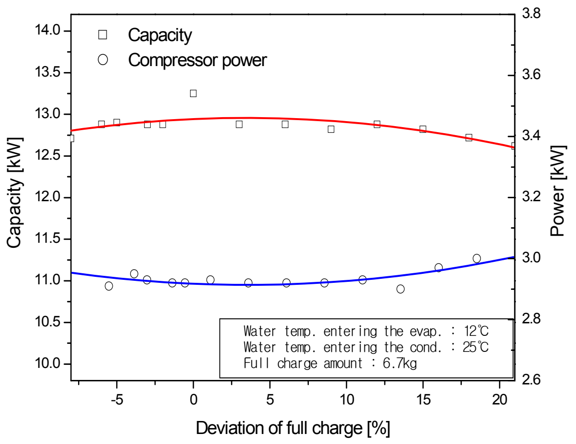



Figure 5 : Cooling capacity with charge amount.

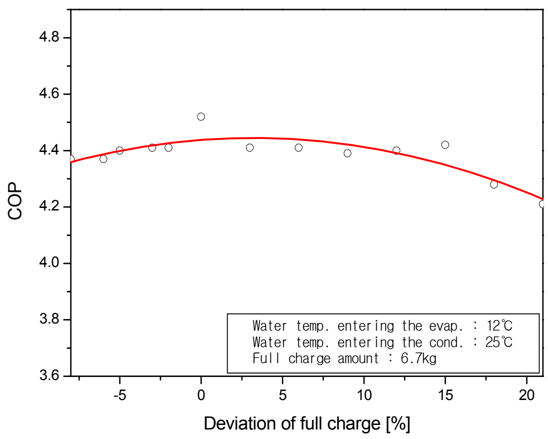



Figure 6 : COP with charge amount.

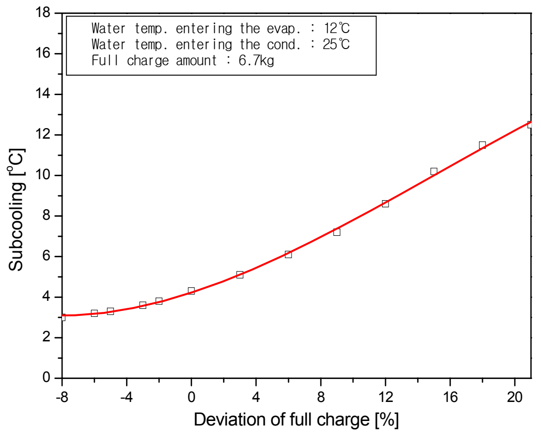



Figure 7 : Variation of subcooling with charge amount.

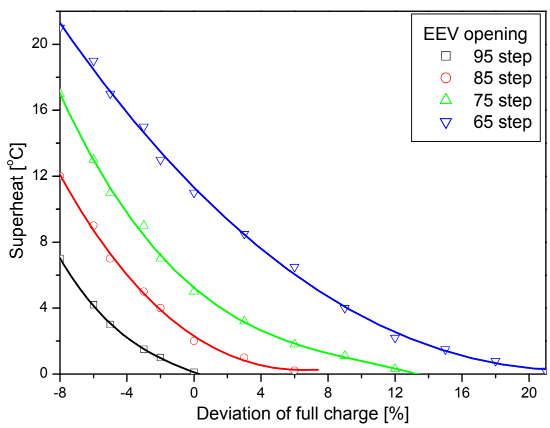



Figure 8 : Variation of superheat with charge amount.

Figure 6 shows the COP of the system as a function of refrigerant charge amount. For undercharged conditions, the COP decreased with the reduction of refrigerant flow rate and compressor efficiency due to the increase of superheat. For overcharged conditions, the COP decreased with the rise of compressor power consumption resulted from the increase of refrigerant mass flow rate. For the capillary tube system, the COP decreased by 16.1% and 4.8% at -20% and +20% of the full charge, respectively, as the charge amount deviated from the full charge (Farzard and O'Neal, 1994). However, the COP was reduced within 2% at -8% and +21% of the full charge because the EEV actively controlled mass flow rate and the receiver compensated optimum refrigerant amount.

Figure 7 shows subcooling as a function of refrigerant charge amount. For undercharged conditions, the subcooling gradually increased with the increase of refrigerant charge amount due to an accumulation of refrigerant in the condenser, while, for overcharged conditions, it increased rapidly.

Figure 8 represents the superheat at the low-stage compressor inlet as a function of refrigerant charge. The superheat decreased as the refrigerant charge increased due to an increase of mass flow rate through the evaporator at the same EEV opening. For the optimal compressor efficiency, the superheat should be kept constant by adjusting EEV opening. In the cooling mode, maintaining of the superheat within 5~10℃ can optimize the performance of the heat pump and enhance the reliability of the system.

### 3.2 Heating Mode

Figures 9 and 10 show the variations of heating capacity and COP, respectively, as a function of refrigerant charge amount. The optimum charge amount in the heating mode was 10.1 kg. The COP of the heating mode decreased within 3% at -15% and +15% of full charge.

Figure 11 shows the variations of the COP with inter-stage EEV opening. The water temperatures entering the condenser and evaporator were kept constant. The inter-stage pressure increased with the increase of inter-stage EEV opening. The mass flow rate of the high-stage cycle increased with the increase of condensing pressure, while that of the low-stage cycle decreased with the reduction of evaporating pressure. Therefore, the heating capacity increased with the increase of condensing pressure and mass flow rate of the high-stage cycle. However, the power consumption of the high-stage compressor increased with the rise of mass flow rate and pressure difference between the condenser and evaporator. The power consumption of the low-stage compressor increased with the increase of superheat due to the reduction of evaporating pressure and mass flow rate. Therefore, the optimal COP of the two-stage compression cycle represented as a function of the inter-stage pressure.

Figure 12 shows the variation of the COP with low-stage EEV opening. As the low-stage EEV opening increased, the evaporating pressure increased and the superheat in the low-stage compressor decreased. Inter-stage and condensing pressures increased with the increase of mass flow rate in the low-stage cycle. Therefore, the heating capacity and the COP increased with the rise of low-stage EEV opening. However, the reduction of superheat in the low-stage compressor limited the increase of EEV opening.

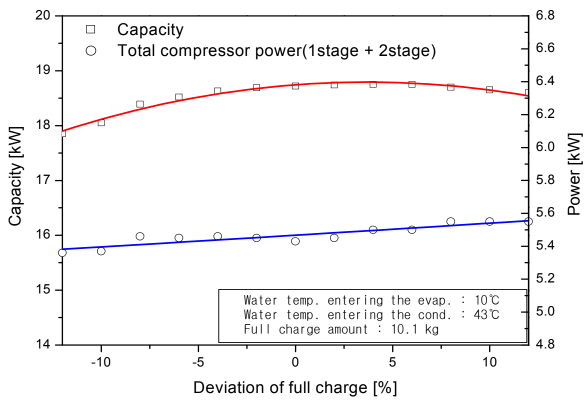



Figure 9 : Heating capacity with charge amount.

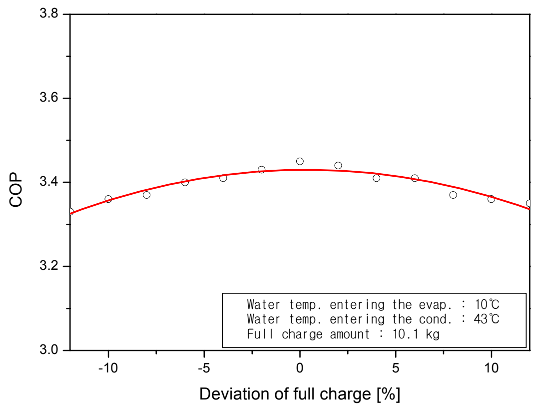



Figure 10 : COP with charge amount

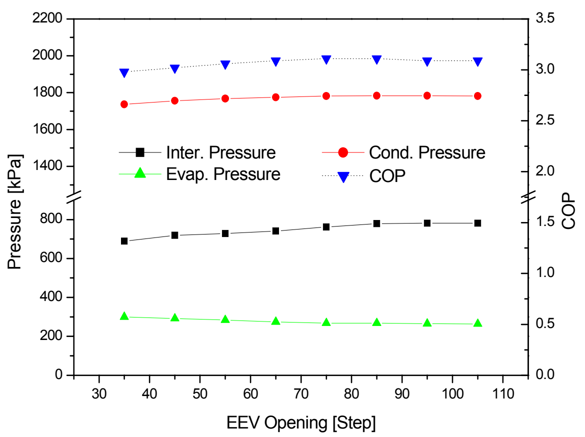



Figure 11 : COP with inter-stage EEV opening.

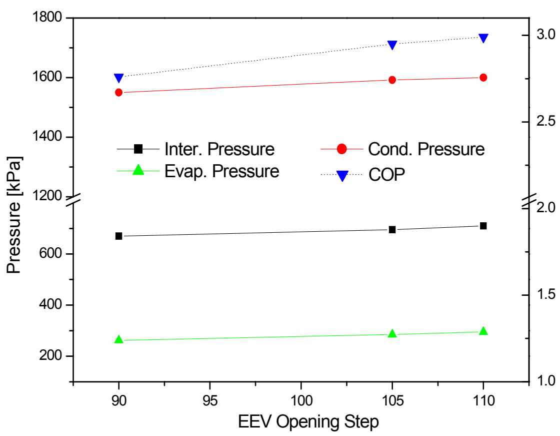



Figure 12 : COP with low-stage EEV opening.

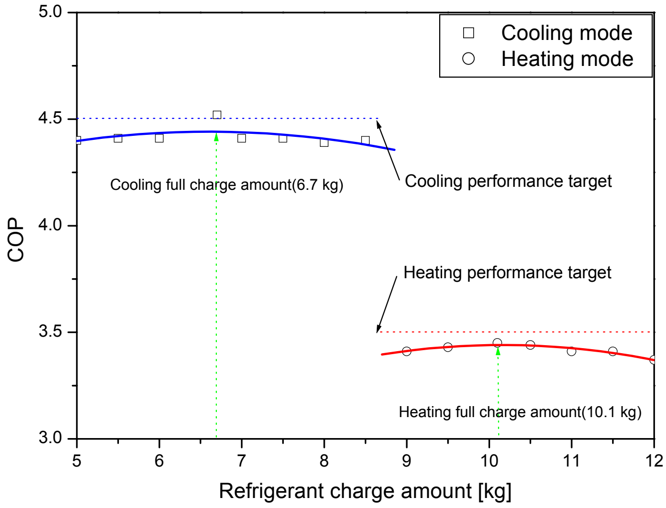



Figure 13 : COP with charge amount.

Figure 13 represents the target COP in the cooling mode and heating modes. The maximum COP of this system was about 4.52 in the cooling mode. The performance of the present system was 23% higher than that (COP 3.5) of the conventional refrigeration system using atmospheric air as the heat source. The maximum COP was about 3.45 in the heating mode. The difference of full charge between the heating and the cooling mode was 3.4 kg. When the heating operation was converted into the cooling, the overcharged refrigerant should be removed to operate at optimal cooling cycle.

## 6. CONCLUSIONS

To characterize the effects of refrigerant charge on the performance of the two-stage heat pump, the experiments were carried out by varying refrigerant charge and EEV opening in the cooling mode and heating modes. In the cooling mode, the maintaining of the superheat within 5~10℃ could optimize the performance of the heat pump and enhance the reliability of the system. The COP was 4.52 at the full charge amount of 6.7 kg. In the heating mode, the heating capacity of the two-stage compression system increased with the increase of low-stage EEV opening. However, the increase of the low-stage EEV opening was limited by the superheat in the low-stage compressor, which should be within 5~10℃. The optimal inter-stage pressure was obtained when the temperature difference between the condenser and the flash tank was controlled within 28~30℃. The maximum COP was 3.45 at the optimum charge of 10.1 kg. The two-stage heat pump could be operated optimally in the cooling and heating modes by controlling refrigerant amount in the inter-stage heat exchanger and flash tank.

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