# PERFORMANCE CHARACTERISTICS OF A SIMULTANEOUS HEATING AND COOLING HEAT PUMP SYSTEM AT EACH OPERATING MODE

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**Abstract:** Cooling load in winter season is significant in many commercial buildings and hotels because of the usage of office equipments and efficient wall insulation. The development of a multi-heat pump that can cover heating and cooling simultaneously for each indoor unit is required. In this study, a 4-room simultaneous heating and cooling heat pump system was designed and its performance including cooling/heating capacity and COP was measured at four operating modes; cooling-only, heating-only, cooling-main, and heating-main modes. The system used R410A as a working fluid and adopted a variable speed compressor and EEV. The experiments were performed by varying outdoor temperature in each operating mode. Problems in the designed heat pump system were analyzed and defined. In addition, the solutions of the problems were suggested to improve system performance and achieve stable operation.

**Key Words:** ***Simultaneous cooling and heating, Heat pump, Heat recovery, System design, Operating mode, Outdoor temperature.***

## 1 INTRODUCTION

Many multi-type heat pumps have been applied to satisfy the needs of cooling and heating of several zones in residential buildings. A multi type heat pump which has one outdoor unit and several indoor units has been used widely to achieve energy conservation and improve human comfort. Generally, the multi type heat-pump operated in a single mode (heating or cooling) at one time. However, a simultaneous heating and cooling heat pump can provide heating and cooling for different zones at the same time. There have been many studies of multi-type heat pump systems. However, the study on simultaneous heating cooling heat pumps was very limited in literature.

In this study, a simultaneous heating and cooling heat pump system was designed and its performance was measured by varying outdoor conditions in four operating modes; cooling-only, heating-only, cooling-main, and heating-main modes. Entire heat recovery mode was not tested because it was not dependent on outdoor temperature. The experiments were performed at outdoor temperatures from 18$^o$C to 43$^o$C for cooling-only and cooling- main modes, and from 2$^o$C to 24$^o$C for heating-only and heating-main modes.

## 2 EXPERIMENTAL SETUP AND TEST METHODS

### 2.1 Experimental Setup

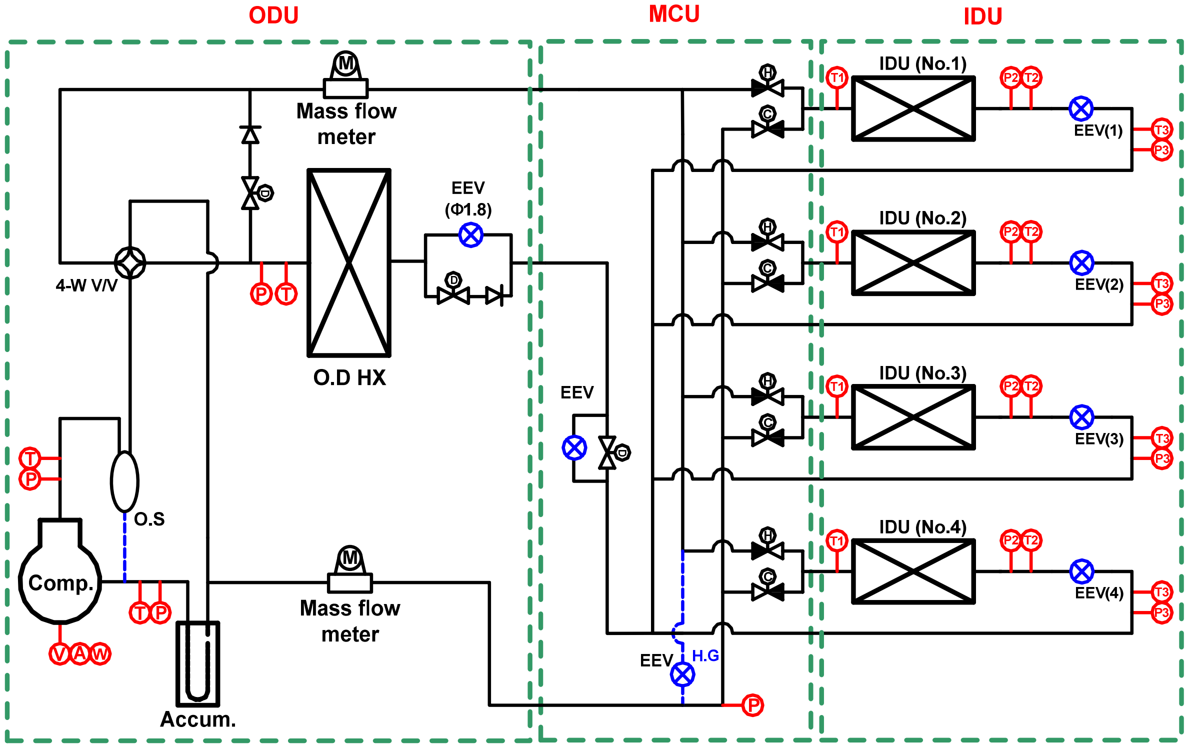


**Figure 1: Schematic diagram of the experimental setup.**

As shown in Figure 1, the simultaneous heating and cooling heat pump system consists of a BLDC type rotary compressor, an outdoor unit(ODU), four indoor units(IDU), an oil separator, a liquid-gas separator, an accumulator in the ODU, and MCU(mode change unit). The designed capacity was 8 kW with R410A as a working fluid. The projected system COP was 3.2. Each indoor unit included a fin and tube type heat exchanger with the designed capacity of 2 kW and an electronic expansion valve (EEV). The designed simultaneous heating and cooling heat pump can operate at five operating modes; cooling-only, heating-only, cooling-main, heating-main, and entire heat recovery modes. The heating-only and cooling-only modes are the same as conventional operating modes in a heat pump.

In cooling-only mode, all indoor units are in cooling operation. Vapor refrigerant discharged from the compressor flows into the outdoor heat exchanger through high pressure gas pipe. The condensed liquid refrigerant from the outdoor unit expands through indoor EEV and then evaporates in the indoor heat exchanger. The indoor heat exchangers work as evaporators, and the outdoor heat exchanger works as a condenser in this mode. As the operating mode changes from cooling-only to cooling-main mode, the solenoid valve "C" in the indoor unit under heating operation is closed. Simultaneously, both the solenoid valve in the bypass line of the outdoor unit and the solenoid valve "H" in the indoor unit under heating operation are opened and then the bypassed refrigerant enters into the indoor unit under heating operation. The indoor unit under heating operation works as a condenser and the other indoor units work as evaporators.

In heating-only mode, all indoor units are in heating operation. Vapor refrigerant discharged from the compressor enters the indoor heat exchangers which work as condensers. The condensed refrigerant from the indoor units expands in the indoor and outdoor EEVs. The expanded refrigerant enters the outdoor unit which works as an evaporator. As the operating mode changes from heating-only to heating-main mode, the solenoid valve "H" in the indoor unit under cooling operation is closed. The refrigerant passing through the indoor units under heating operation divides into two directions. Some of the refrigerant goes through the indoor unit under cooling operation by opening the solenoid valve "C". The other flows into the outdoor unit. The outdoor heat exchanger and the indoor units under cooling operation work as evaporators, and the other indoor units under heating operation work as condensers.

In entire-heat recovery mode, the half of indoor units is in cooling mode and the other half is in heating mode operation. Vapor refrigerant discharged from the compressor flows into the indoor units under heating operation which works as condensers. The condensed refrigerant from these units enters the other indoor units under cooling operation which work as evaporators. Since the absorbed energy from the evaporators balances the rejected energy from the condensers, the outdoor heat exchanger does not operate in entire-heat recovery mode.

## 2.2 Experimental Procedure

The outdoor unit was installed in a psychrometric chamber and each indoor unit was installed in separate air handling units which can control temperature and humidity entering the unit. To measure the system performance, we installed pressure transducers and thermocouples at the inlet and outlet of each component. Optimum refrigerant charge was determined based on cooling-only mode operation, which was used throughout the experiments. The optimum refrigerant charge amount was 3900 g. The compressor frequency was fixed at 50Hz during the tests. EEV opening was adjusted from 0 to 500 steps.

Heating and cooling capacities, power consumption, COP, refrigerant temperatures and pressures at major locations of the system were monitored by using a data acquisition system. The data presented in this paper are arithmetic average values stored for 30 min under steady state conditions. Airside cooling and heating capacities were determined by utilizing the air enthalpy method (ASHRE Standard 116,1993) based on both air flow rate and enthalpy difference across the heat exchanger. The air flow rate was determined by measuring the pressure difference between the inlet and outlet of the indoor unit. The COP can be expressed by

$$COP = \frac{Q_{\text{total cooling}} + Q_{\text{total heating}}}{W} \quad (1)$$

Table 1 shows operating status of each indoor unit with respect to operating mode. In this study, cooling-only, cooling-main, heating-only, and heating-main modes were tested.

**Table 1; Operating status of indoor units with operating mode**

| | IDU (No.1) | IDU (No.2) | IDU (No.3) | IDU (No.4) |
|---|---|---|---|---|
| Cooling-only operation | Cooling | Cooling | Cooling | Cooling |
| Cooling-main operation | Cooling | Cooling | Cooling | Heating |
| Heating-main operation | Cooling | Heating | Heating | Heating |
| Heating-only operation | Heating | Heating | Heating | Heating |

**Table 2; Test conditions**

| Outdoor temperature | | Indoor temperature | |
|---|---|---|---|
| Cooling-only operation<br>Cooling-main operation | Heating-only operation<br>Heating-main operation | Cooling mode | Heating mode |
| DB 18°C/ WB 12°C | DB 2°C/ WB 1°C | DB 27°C<br>WB 19°C | DB 20°C<br>WB 15°C |
| DB 24°C/ WB 18°C | DB 7°C/ WB 6°C | | |
| DB 35°C/ WB 24°C | DB 18°C/ WB 12°C | | |
| DB 43°C/ WB 26°C | DB 24°C/ WB 18°C | | |

As shown in Table 2, outdoor temperature was varied from 18°C to 43°C and from 2°C to 24°C in the cooling-only/main and heating-only/main modes, respectively, while indoor temperature was fixed at 27°C/19°C (cooling), 20°C/15°C (heating).

## 3 RESULTS AND DISCUSSION

### 3.1 Performance characteristics in cooling-only and cooling-main modes

Figure 2 shows the variations of total capacity and COP with outdoor temperature in cooling-only mode. Both COP and total capacity decreased with the increase of outdoor temperature, while power consumption increased with the increase of outdoor temperature. These trends are also observed in a conventional heat pump. With the increase of outdoor temperature from 18°C to 43°C, COP and total capacity decreased by 56% and 27%, respectively.

Figure 3 shows the variations of total capacity, average heating capacity, average cooling capacity and COP with outdoor temperature in cooling-main mode. As outdoor temperature increased, total capacity gradually increased by 45% in cooling-main mode, because increasing rate of the average heating capacity was higher than that of the cooling capacity. The increase of the heating capacity was due to the increase of bypassed mass flow rate as shown in Figure 4. Power consumption decreased with the increase of outdoor temperature. In spite of the increase of power consumption, COP increased until outdoor temperature reached about 35°C. The effect of mass flow rate on COP was more dominant than that of power consumption increment.

Total mass flow rate increased with the increase of outdoor temperature. The total mass flow rate in cooling-only mode was higher than that in cooling-main mode because compressor volumetric efficiency and compression ratio in cooling-only mode were higher than those in cooling-main mode. With the increase of outdoor temperature from 18°C to 43°C, the total mass flow rate and bypass mass flow rate increased by 78% and 41%, respectively.

Figure 5 shows the variations of suction and discharge pressures for both cooling-only and cooling-main modes as a function of outdoor temperature. In both cases, the discharge and suction pressures increased with the rise of outdoor temperature. As the operation mode changed from cooling-only to cooling-main mode, evaporating pressure decreased due to the reduction in the number of the indoor units under cooing operation. As the outdoor

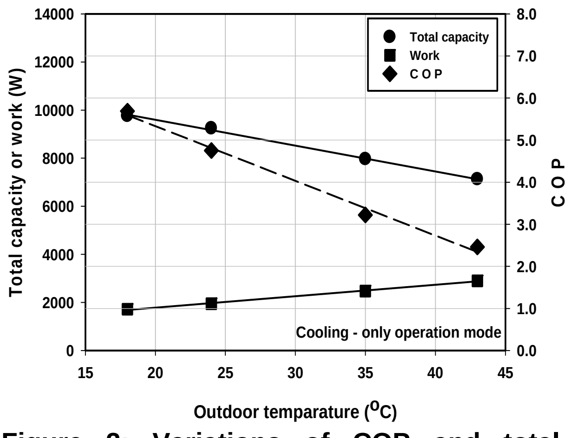


**Figure 2: Variations of COP and total capacity with outdoor temperature in cooling-only mode.**

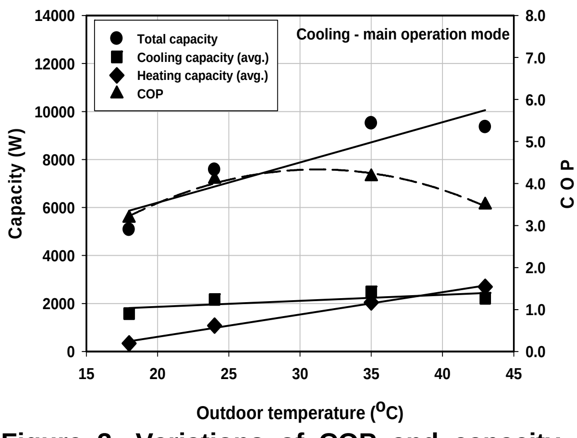


**Figure 3: Variations of COP and capacity with outdoor temperature in cooling-main mode.**

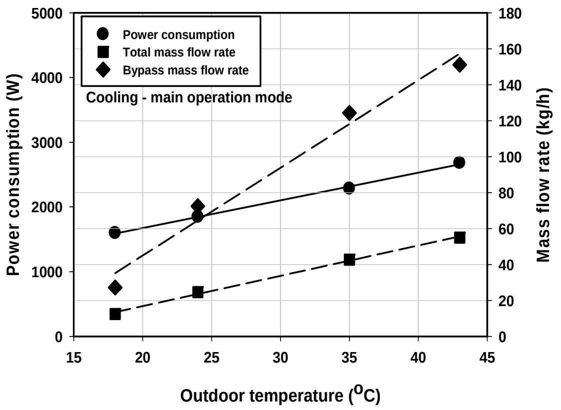


**Figure 4: Variations of power consumption and mass flow rate with outdoor temperature in cooling-main mode.**

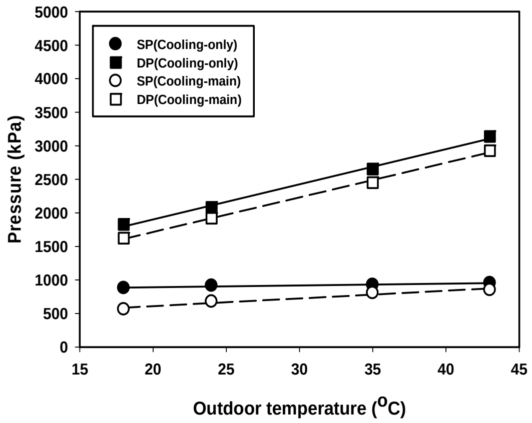


**Figure 5: Variations of suction and discharge pressures in cooling-only/ cooling-main modes.**

temperature increased, the heat transfer rate decreased in the outdoor unit, which was used as an evaporator, due to the decrease of the temperature difference between air and refrigerant. In addition, condensing pressure increased due to the increase of outdoor temperature.

At the outdoor temperature of 18$^{o}$C, suction pressure of the compressor in cooling-only mode was approximately 560 kPa. It was higher than that in cooling-main mode at outdoor temperatures from 18$^{o}$C to 43$^{o}$C. In cooling-only mode, the suction pressures at outdoor temperatures of 18$^{o}$C to 43$^{o}$C were 33% and 7.7% higher than those in cooling-main mode, respectively. Therefore, as outdoor temperature increased, cooling-main mode operation showed higher condensing pressure than cooling-only mode operation.

### 3.2 Performance characteristics in heating-only and heating-main modes

Figure 6 shows the variations of total capacity and COP according to outdoor temperature in heating-only mode. Both total capacity and COP decreased by 16% and 38%, respectively, with the reduction of outdoor temperature from 24$^{o}$C to 2$^{o}$C. In heating-only mode, the increase of total capacity was due to the rise of the temperature difference between outdoor air and evaporating refrigerant.

Figure 7 shows the variations of total capacity, average cooling capacity, average heating capacity and COP according to outdoor temperature in heating-main mode. Total capacity was gradually elevated with the increase of outdoor temperature and COP showed small variation.

In heating-main mode, heating capacity increased and cooling capacity decreased compared with those in heating-only mode. Therefore, the increase of condensing pressure in heating-main mode was higher than that in heating-only mode. With the increase of outdoor temperature, heating capacity also increased, while cooling capacity decreased because bypass flow rate did not increase in the indoor unit under cooling operation. As shown in Figure 8, although total capacity increased continuously, bypass flow rate was remained nearly constant at 40 kg/h.

Figure 9 shows the variations of suction and discharging pressures with outdoor temperature in heating-only and heating-main modes. As the operating mode changed from heating-only to heating-main mode, condensing pressure decreased due to the reduction in the number of the indoor units under heating operation. Both suction and discharge pressures increased continuously in heating-only and heating-main modes. The increase of condensing pressure

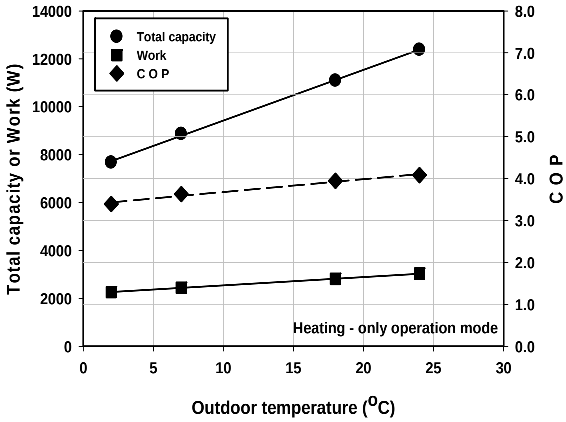


**Figure 6: Variations of COP and total capacity with outdoor temperature in heating-only mode.**

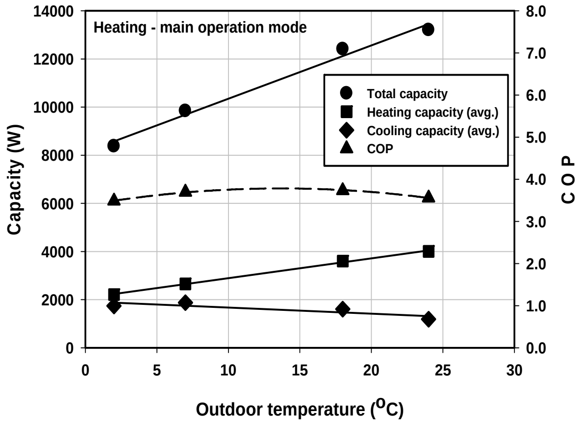


**Figure 7: Variations of total capacity and COP with outdoor temperature in heating-main mode.**

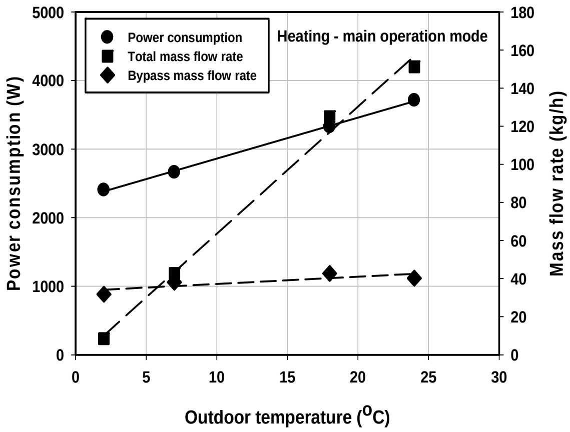


**Figure 8: Variations of mass flow rate and bypass mass flow rate in heating-only/main modes.**

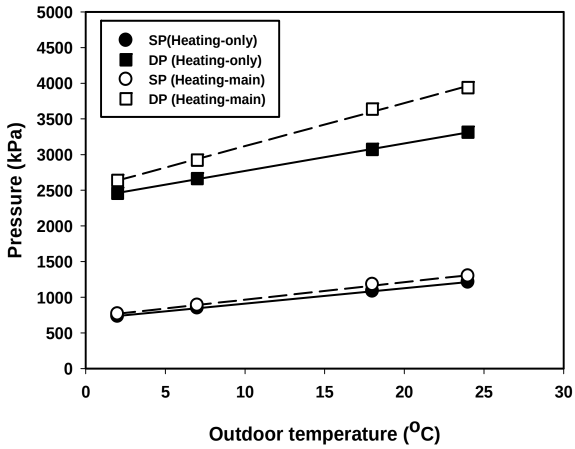


**Figure 9: Variations of suction and discharge pressures in heating-only/main modes.**

was higher than that of evaporating pressure. The increase of discharge pressure in heating-main mode was higher than that in heating-only mode because the number of heating-operation indoor units in heating-main mode was smaller than that in heating-only mode.

## 4 CONCLUSIONS

The performance of the simultaneous heating and cooling heat pump system was measured by varying outdoor temperature in four operating modes. In cooling-only and heating-only modes, outdoor temperature showed more strong effects on COP than those in cooling-main and heating-main modes. The increase of condensing pressure with outdoor temperature in cooling-main mode was higher than that in heating-main mode. In cooling-main mode, bypass flow rate increased by 78% with the rise of outdoor temperature. However, the bypass flow rate with respect to outdoor temperature remained nearly constant in heating-main mode. The measured data can be used as a database to develop a mode-change control algorithm in a simultaneous heating and cooling heat pump system. The present data suggested that the system performance of the simultaneous heating and cooling heat pump system can be enhanced by applying an adequate mode-change control parameter.

## 5 ACKNOWLEDGEMENTS

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## 6 REFERENCES

Park, T.J., Lee, M.K., Jeong, J.H., Chang, K.S., 2003. "Comparison of performance variation between R-22 and R410A refrigeration systems." Korean Journal of Air-Conditioning and Refrigeration Engineering, Vol.15, No.3, pp. 166-176.

M. Hosoz, M.Direk, 2006. "Performance evaluation of an integrated automotive air conditioning and heat pump system." Energy Conversion and Management, Vol.47, pp. 545-559.

ASHRAE Guideline 2, 1986. Engineering analysis of experimental data, ASHRAE, Atlanta(GA), USA.

ASHRAE Standard 116, 1993, Methods of testing for seasonal efficiency of unitary air-conditioner and heat pump, ASHRAE, Atlanta(GA), USA.