Performance characteristics of a heat pump for dehumidifying of a cabin in electric vehicles

Jae Hwan Ahn, Ho Seong Lee, Changhyun Baek, Sang Hun Lee, Jongho Jung, Yongchan Kim
Department of Mechanical Engineering, Korea University
Anam-Dong, Sungbuk-Ku, Seoul, 136-713, Republic of Korea
E-mail: yongckim@korea.ac.kr

Abstract
In cold weather, fogging on a windshield in a vehicle disturbs the driver’s view and leads to the threat for the safety of the vehicle and passengers. Air-conditioning units for dehumidifying and engine coolant for heating has been used in combustion engine vehicles. However, in electric vehicles, since the electric heater instead of the engine coolant is used for heating, the dehumidifying and heating process causes higher power consumption. As an efficient dehumidifying and heating, a heat pump system is available. The objective of this study is to investigate the feasibility of the heat pump for dehumidifying and heating and to compare the performance of the heat pump with that of an air-conditioning unit and an electric heater. The heat pump showed higher dehumidifying and heating performance than those of an air-conditioning unit and an electric heater.

Keywords: Dehumidifying, Electric vehicle, Heat pump

1 Introduction
In cold weather, a fog formed in a windshield of a vehicle can disturb the driver’s view, resulting in a serious accident. The fog has to be removed by a dehumidification process. In addition, the cabin temperature has to be increased to satisfy thermal comfort. In electric vehicles, an air-conditioning unit and an electric heater have been used for dehumidifying and heating of the cabin, respectively. The air-conditioning unit cools indoor air and then removes condensed water vapor in the air. After cooling, the air is heated passing through the heater. The air-conditioning unit and the electric heater have a cost advantage because the system doesn’t need to modify the established design and to equip additional devices. However, the electric heater consumes more electric power than required heat load of the cabin due to reheating the cooled air. Although an electric vehicle suffers a limited driving range, the dehumidifying and heating operation aggravate the insufficient driving range. Consequently, developing a heat pump system for dehumidifying and heating in electric vehicles is necessary for energy saving. Several researches on a dehumidifying heat pump system have been conducted in order to improve dehumidifying and heating performance of the heat pump. Tamura et al. [1] suggested a novel CO2 heat pump for the dehumidifying of a vehicle compartment. The CO2 heat pump system used the heat released from the refrigerant in the high-pressure side heat exchanger during the dehumidification process. The performance of the CO2 heat pump system was superior to that of the conventional R-134a system. Hawlader et al. [2]
developed a simulation program to evaluate the performance of a solar-assisted heat pump dryer and water heater considering the influence of different operating variables. Gungor et al. [3,4] investigated the performance of gas engine heat pumps for food drying processes. They evaluated the performance of the dryer using an exergy analysis. Minea [5] investigated the system integration and application potential of drying heat pumps based on a literature survey. The dehumidifying and heating performance of a heat pump (HP) combined with a heater was investigated to replace the conventional air-conditioning (AC) system used in electric vehicles. The performances of the AC and HP systems were measured by varying the indoor air wet bulb temperature. The operating characteristics of the HP system were compared with that of the conventional AC system at various indoor air wet bulb temperatures.

2 Experimental setup and test procedure

2.1 Experimental setup

Figure 1 presents a schematic diagram of the test setup. An AC system and an HP system were designed for the dehumidifying and heating of the cabin in electric vehicles. Basically, the test setup consisted of a twin rotary compressor, an outdoor heat exchanger (ODHX), an indoor condenser (IDCON), an indoor evaporator (IDEVA), an electric air-heater, an electric expansion valve (EEV), and an accumulator. In the AC system, as shown in Fig. 1(a), the discharged refrigerant from the compressor flows into the ODHX that is used as a condenser. After heat rejection in the ODHX, the condensed liquid refrigerant enters the IDEVA through the EEV at the IDEVA inlet. The refrigerant evaporates in the IDEVA and flows into the compressor. The indoor air is cooled and dehumidified through the IDEVA and heated through the electric heater. In the HP system, as shown in Fig. 1(b), the condensate liquid refrigerant in the IDCON flows into the IDEVA through the EEV. The refrigerant evaporates in the IDEVA enters the compressor. The indoor air is cooled and dehumidified through the IDEVA and heated through the electric heater. The designed overall heating capacity of the test setup was 2.5 kW. The overall heating capacity represents the heating capacity provided by the HP and heater, representing the temperature increase from the inlet air temperature of 20 °C to the air temperature at the heater outlet. The hermetic twin rotary compressor had a displacement volume of 24.0 cm³ rev⁻¹. The ODHX was a parallel-flow type, louvered-fin brazed-aluminum heat exchanger of 17 mm in depth, 350 mm in height, and 670 mm in width. The IDCON was a cross-flow type, plate-fin brazed-aluminum heat exchanger of 100 mm in depth, 198 mm in height, and 147 mm in width. The IDEVA was a laminated aluminum heat exchanger of 61 mm in depth, 280 mm in height, and 220 mm in width. The rated capacity of the electric air-heater was 6.0 kW. The EEV, which were used as an expansion device, had orifice diameter of 1.6 mm.

A resistance temperature sensor with an accuracy of ±0.2 °C was used to measure the refrigerant temperature, and a digital pressure transducer with an accuracy of ±0.5% was utilized to measure the refrigerant pressure. A mass flow meter with an accuracy of ±0.2% was used to measure the mass flow rate of the refrigerant. A digital power meter with an accuracy of ±0.5% was applied to measure the power used for the compressor and the electric heater. A digital pressure transducer with an accuracy of ±0.3% was applied to measure the pressure difference across the nozzle. The nozzle
method [6] was utilized to measure the air flow rate.

2.2 Test procedure
The refrigerant charge amounts in the AC and HP systems were determined based on preliminary tests, yielding the maximum COP at the standard cooling and heating test conditions [7] and at a rated compressor frequency of 60 Hz. The indoor air dry bulb temperature was fixed at 20 °C. In the AC and HP systems, the operating characteristics were measured by varying the indoor air wet bulb temperature from 13 °C to 17 °C. The superheat at the outlet of the IDEVA was controlled at 5 °C by modulating the EEV opening. The heater at the IDEVA outlet was controlled to satisfy the designed overall heating capacity of 2.5 kW.

The moisture extraction rate (MER) was evaluated using Equation (1) in terms of the air flow rate and the humidity ratio difference across the IDEVA. As given in Equation (2), the overall heating capacity was evaluated using the air flow rate and the air temperature difference between the IDEVA inlet and the heater outlet. The COP was evaluated using Equation (3).

\[ MER = \frac{Q_{\text{air}}}{V} (HR_{\text{air,in,eva}} - HR_{\text{air,out,eva}}) \] (1)

\[ q_{h,o} = \frac{Q_{\text{air}}}{V} c_p (T_{\text{air,out,heater}} - T_{\text{air,in,eva}}) \] (2)

\[ COP = \frac{q_{h,o}}{W_j} \] (3)

3 Results and discussion
The dehumidifying and heating performances of the conventional AC system and the HP system were analyzed at various indoor air wet bulb temperatures.

Figure 2 shows the suction pressure, discharge pressure, and refrigerant mass flow rate according to the indoor air wet bulb temperature in the AC system. As the indoor air wet bulb temperature increased, the suction and discharge pressures in the AC system increased because of an increased evaporation heat transfer rate with an increased latent heat transfer rate in the IDEVA. The refrigerant mass flow rate increased with increasing indoor air wet bulb temperature due to the increase in the refrigerant density at the compressor inlet.

Figure 3 shows the power consumption in the heater, total power consumption, and COP with the indoor air wet bulb temperature in the AC system. Both the total power consumption and the power consumption in the heater decreased with increasing indoor air wet bulb temperature. As the indoor air wet bulb temperature increased, the COP in the AC system increased due to the decreased total power consumption.

Figure 4 shows the suction pressure, discharge pressure, and refrigerant mass flow rate according to the indoor air wet bulb temperature in the HP system. As the indoor air wet bulb temperature...
increased, the suction and discharge pressures in the HP system increased because of an increased evaporation heat transfer rate with an increased latent heat transfer rate in the IDEVA. However, the HP system showed a higher increasing incline in the discharge pressure according to the indoor air wet bulb temperature than the AC system due to the increase in the air temperature at the IDCON inlet in the HP system. Therefore, the compression ratio of the HP system was 36% higher than that of the AC system at the indoor air wet bulb temperature of 15 °C. In the HP system, the refrigerant mass flow rate increased with increasing indoor air wet bulb temperature due to the increase in the refrigerant density at the compressor inlet. In addition, the refrigerant mass flow rate in the HP system was higher than that in the AC system at a given indoor air wet bulb temperature due to the higher suction pressure in the HP system.

Figure 5 shows the power consumption in the heater, total power consumption, and COP with indoor wet bulb temperature in the HP system. Both the total power consumption and the power consumption in the heater decreased with increasing indoor air wet bulb temperature. In addition, the HP system showed lower total power consumption than the AC system at a given indoor air wet bulb temperature due to the additional heating capacity in the condenser of the HP system. At the indoor air wet bulb temperature of 15 °C, the total power consumption in the HP system decreased by 65% compared to that in the AC system. As the indoor air wet bulb temperature increased, the COP in the HP system increased due to the decreased total power consumption. In addition, the HP system showed higher COP than the AC system at a given indoor air wet bulb temperature due to the decreased power consumption in the heater. At the indoor air wet bulb temperature of 15 °C, the COP in the HP system were 180% higher than that in the AC system.

Figure 6 shows the performance ratio of the HP system at the indoor air wet bulb temperatures of 15 °C. The performance ratio was defined by the ratio of each index of the HP system to the AC system. The MER of the HP system was 24% lower than that of the AC system. The total power consumption of the HP system was 59% lower than that of the AC system. Therefore, the COP of the HP system was 145% higher than that of the AC system. HP system because of the higher optimized CSR in the DHP system. The HP system showed superior performance to the AC system in the dehumidifying and heating operation.

4 Conclusions

In electric vehicles, air-conditioning units and electric heaters have been used for the dehumidifying and heating of the cabin. An HP system is proposed as an effective dehumidifying and heating system to replace the conventional AC system. The dehumidifying and heating performances of the AC and HP systems were measured according to the indoor air wet bulb temperature. The HP system showed lower total power consumption and higher COP than the AC system. The COP of the HP system was 145% higher than those of the AC system.

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