Experimental study on the performance of a simultaneous heating and cooling multi-heat pump with the variation of operation mode

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\begin{abstract}
The cooling load in the winter season becomes significant in commercial buildings and hotels because of the wide usage of office equipment and improved wall insulation. In this study, a simultaneous heating and cooling multi-heat pump having four indoor units and an outdoor unit was designed and tested in five operation modes: cooling-only, heating-only, cooling-main, heating-main, and entire heat recovery. The performance of the system with R410a was optimized by adjusting the system's control parameters. In the cooling-main mode, the rate of the bypass flow to the heating-operated indoor unit was optimized by controlling the EEV opening of the outdoor unit. In the heating-main mode, the mass flow rate to the cooling-operated indoor unit was optimized by adjusting the EEV opening in the outdoor unit. In the entire heat recovery mode, the compressor speed was controlled to improve the system COP with appropriate heating and cooling capacities.
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1. Introduction

A multi-heat pump, which has one outdoor unit and several indoor units, has been used to cover the cooling load of a building in the summer season and the heating load in the winter season. The operation mode of the multi-heat pump can be changed from the cooling to the heating mode by switching the direction of the refrigerant flow using a 4-way valve. A conventional multi-heat pump can provide either only-cooling or only-heating at any time. However, the cooling load in the winter season becomes significant in commercial buildings and hotels because of the wide usage of office equipment having high heat flux and because of the improved wall insulation. Therefore, the function of the multi-heat pump has to be extended to cover both heating and cooling loads simultaneously in different zones of a building. Currently, research continues to develop a highly efficient simultaneous heating and cooling multi-heat pump that can provide the required heating and cooling to different zones of a building at the same time. The major focus in recent research has been on the development of cutting edge technologies to realize optimum refrigerant flow control and high system efficiency and reliability.

Previous studies on multi-heat pumps focused on the improvement of the system performance in heating-only or cooling-only operation. The technologies used in the development of multi-heat pumps mainly focused on compressor capacity control, refrigerant flow rate control, and alternative refrigerants. Compressor speed control has been widely adopted in multi-heat pumps to modulate the compressor capacity. In the early stage of compressor development, alternating current (AC) inverter technology was widely used for compressor speed control (Nagatomo, 1998; Aprea et al., 2006). However, recently, the use of direct current (DC) inverter technology has become increasingly more common because of its energy saving efficiency and high control accuracy. The control of the refrigerant flow rate was remarkably improved by using the electronic expansion valve (EEV) (Qifang et al., 2007; Park et al., 2007), which can control the refrigerant flow rate precisely according to the variations of the operating conditions. The compressor inverter and EEV are essential components in the development of multi-heat pumps (Choi and Kim, 2003; Park et al., 2001).

Even though many studies on multi-heat pumps have been conducted, studies on simultaneous heating and cooling multi-heat pumps are very limited in open literature. Especially, the system optimization and flow distribution control of simultaneous heating and cooling multi-heat pumps according to the variations of the operation mode need to be studied more comprehensively to develop a highly efficient and reliable system.

In this study, a simultaneous heating and cooling multi-heat pump was designed and its performance was measured for five operation modes, namely the cooling-only, heating-only, cooling-main, heating-main, and entire heat recovery modes. The performance of the simultaneous heating and cooling multi-heat pump with R410a (Park et al., 2003; Kim and Bullard, 2001) was analyzed and then, improved by optimizing its control parameters such as the compressor rotation speed and EEV opening.

2. Experimental setup and test procedure

2.1. Experimental setup

Fig. 1 shows the schematic diagram of experimental setup. A simultaneous heating and cooling multi-heat pump using R410a was designed to have a cooling capacity of 8.0 kW in the cooling-only mode. The pump system consisted of an outdoor unit (ODU), a mode change unit (MCU), and four indoor units (IDU). The outdoor unit consisted of a bluish direct current (BLDC) type rotary compressor, an oil separator, a liquid–gas separator, an accumulator, a finned-tube heat exchanger and an EEV. Each indoor unit consisted of a finned-tube heat exchanger and an EEV. The heat exchangers for the indoor and outdoor units used micro-fin tubes and slit-fins. Each indoor heat exchanger had an evaporation capacity of 2.15 kW at the evaporating temperature of 7.2 °C and an air flow rate of 6.0 m³/min. The outdoor heat exchanger had a condensation capacity of 11.34 kW at the condensing temperature of 54.4 °C and an air flow rate of 37.5 m³/min.

For individual heating and cooling operations of the indoor units, an MCU was installed between the outdoor unit and the indoor units. It consisted of header pipes, branch pipes and solenoid valves. On/off operations of the solenoid valves were determined according to the operation mode of the indoor
units. Three connection pipes of discharge-gas, suction-gas and liquid pipes were installed between the MCU and the outdoor unit, whereas the conventional multi-heat pump had only two connection pipes of gas and liquid pipes between the indoor units and the outdoor unit. An EEV, consisting of a stepping motor and a needle valve, was adopted for the optimum control of the refrigerant flow rate, and its orifice diameters were 1.4 mm for the indoor unit and 1.8 mm for the outdoor unit, with the control resolution of 500 steps. The accumulator was installed at the compressor suction to prevent wet-compression and the oil-separator was installed at the compressor discharge to allow proper oil return to the compressor.

As shown in Table 1, the simultaneous heating and cooling multi-heat pump has five operation modes: heating-only, cooling-only, heating-main, cooling-main, and entire heat recovery. In the heating-only or the cooling-only mode, all the indoor units operate in the same mode. In the heating-main mode, the number of heating-operated indoor units is more than that of cooling-operated indoor units, and vice versa in the cooling-main mode. In these operation modes, the total or partial heat absorbed by the cooling-operated indoor units is reused for heating; this is referred to as heat recovery operation. In the entire heat recovery mode, the number of cooling-operated indoor units is the same as that of heating-operated indoor units and there is no refrigerant flow to the outdoor heat exchanger. Therefore, all the heat absorbed from the cooling-operated indoor units is reused for heating in the entire heat recovery mode.

In the cooling-only mode, all indoor units operate in the cooling mode. All solenoid valves for cooling operation, which are represented as “C”, are opened and all solenoid valves for heating operation, which are marked as “H”, are closed. Vapor refrigerant discharged from the compressor flows into the outdoor heat exchanger through the high pressure gas pipe. The condensed liquid refrigerant from the outdoor unit expands through the indoor heat exchangers and then evaporates in the indoor heat exchangers. The indoor heat exchangers work as evaporators and the outdoor heat exchanger operates as a condenser in this mode. As the operation mode changes from cooling-only to cooling-main, the solenoid valves “C” in the heating-operated indoor units are closed. Simultaneously, both the solenoid valve in the bypass line of the outdoor unit and the solenoid valves “H” in the heating-operated indoor units are opened and then the bypassed refrigerant enters the heating-operated indoor unit. The heating-operated indoor unit works as a condenser and the other indoor units operate as evaporators. The liquid-phase refrigerant condensed in the heating-operated indoor unit flows to the MCU through the fully-opened EEV, thus merging with the liquid-phase refrigerant condensed in the outdoor heat exchanger and then, the merged refrigerant flows to the cooling-operated indoor units through the EEVs.

In the heating-only mode, all indoor units operate in the heating mode. All solenoid valves “H” are opened and all solenoid valves “C” are closed. Vapor refrigerant discharged from the compressor enters the indoor units, 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 operation mode changes from heating-only to heating-main, the solenoid valve “H” in the cooling-operated indoor unit is closed. The refrigerant passing through the heating-operated indoor units divides into two directions. Some of the refrigerant goes through the cooling-operated indoor unit by opening the solenoid valve “C” and the remainder flows into the outdoor unit. The outdoor heat exchanger and the cooling-operated indoor units work as evaporators, and the other heating-operated indoor units work as condensers.

In the entire heat recovery mode, half of the indoor units are in the cooling mode and the other half are in the heating mode. The vapor refrigerant discharged from the compressor flows into the heating-operated indoor units, which work as condensers. The condensed refrigerant from these units enters the other cooling-operated indoor units, 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 the entire-heat recovery mode.

2.2. Measuring equipment and test procedure

The performance of the simultaneous heating and cooling multi-heat pump was measured in five operation modes. The operating status of each indoor unit according to the operation mode is shown in Table 1. Test conditions for the indoor units under heating and cooling operations were chosen based on ISO/DIS 15042 (2005). Table 2 shows the test conditions for each operation mode.

The performance of the simultaneous heating and cooling multi-heat pump was measured by an air-enthalpy type calorimeter. The outdoor unit was installed in a psychrometric chamber, and each indoor unit was installed in separate air handling units, which can control the temperature and humidity of air entering the unit. The measurement devices for the air flow rate, temperature and humidity were equipped in each indoor unit to measure the performance of each indoor unit. A resistance temperature sensor (Pt 100 Ω) was used to measure the dry and wet bulb temperatures of air to an accuracy of ±0.15 °C. The air flow rate of the indoor unit was measured by using a nozzle with a diameter of 76.2 mm. The pressure difference across the nozzle was measured by using an electronic differential pressure transducer with an accuracy of ±0.25%. The power consumption of the multi-heat pump was measured by using an electronic power meter with an accuracy of ±0.2%.

Air flow rate and air-side cooling and heating capacities were calculated according to ISO/DIS 15042 (2005). The COP of the system was calculated by Eq. (1). The estimated uncertainties of the capacity and COP of the system were to be approximately 3.5% by single-sample analysis according to ASHRAE Guideline 2 (1986).

\[
\text{COP} = \frac{q_{w} + q_{a}}{W}
\]

Refrigerant temperatures, pressures and flow rates were measured to analyze the operating characteristics of the simultaneous heating and cooling multi-heat pump at various operating conditions. The refrigerant temperature was measured by a T-type thermocouple with an accuracy of ±0.3 °C, and a thermocouple sensor was inserted inside the pipe using a T-union. The refrigerant pressure was measured by using a gauge pressure transducer with an accuracy of ±0.13%. The refrigerant flow rate was measured by using a Coriolis mass flow meter with an accuracy of ±0.1%.

<table>
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3. Results and discussion

3.1. Performance characteristics in cooling-only and cooling-main operations

The operating characteristics of the system in the cooling-only mode were measured at the test conditions specified in Table 2. Based on the preliminary tests of the system according to the variations of the refrigerant charge and EEV opening, the maximum system COP in the cooling-only mode was observed at the refrigerant charge of 3900 g and the EEV opening of 110 steps in the indoor units. Therefore, the tests in the cooling-only mode were conducted at the refrigerant charge of 3900 g and the compressor speed of 3500 rpm, while the EEV openings of the indoor and outdoor units were maintained at 110 and 500 steps (full opening), respectively. Experiments in the cooling-main mode were also conducted at the same operating conditions as the cooling-only mode, except that the EEV opening of the heating-operated indoor unit was modified to 500 steps.

Fig. 2 compares the cycle characteristics of the cooling-only mode with those of the cooling-main mode on the pressure–enthalpy diagram. When the operation mode changed from the cooling-only to cooling-main, the evaporating pressure decreased due to the decrease in the number of cooling-operated indoor units. The condensing pressure also decreased because the heating-operated indoor unit worked as a condenser in the cooling-main mode. The compression ratio in the cooling-main mode was increased by 11.4% from that in the cooling-only mode because the decreasing rate of the evaporating pressure was larger than that of the condensing pressure. The refrigerant mass flow rate in the cooling-main mode was decreased by 20.6% from that in the cooling-only mode due to the decreases of the compressor volumetric efficiency and EEV inlet pressure of
the cooling-operated indoor units. The power consumption in the cooling-main mode was decreased by 11.1% from that in the cooling-only mode due to the decrease of refrigerant flow rate.

The total capacity, average cooling capacity, and COP of the system in the cooling-only mode were 8248 W, 2062 W, and 3.12, respectively. The total capacity includes heating and cooling capacities of all indoor units. The average cooling capacity denotes an averaged value for each cooling-operated indoor unit. The total capacity in the cooling-main mode was 8120 W, which was 1.6% lower than that in the cooling-only mode. The average cooling capacity in the cooling-main mode was 2276 W, which was 10.4% higher than that in the cooling-only mode. However, the heating capacity in the cooling-main mode was 1292 W, which was much lower than the designed capacity of 2.0 kW, because the flow rate of the bypass refrigerant to the heating-operated indoor unit was very low, only 15.4% of the total flow rate. The COP in the cooling-main mode was 3.46, which was 10.9% higher than that in the cooling-only mode due to the heat recovery operation in the cooling-main mode.

In the cooling-main mode, the bypass refrigerant flow rate to the heating-operated indoor unit has to be controlled optimally to improve the heating capacity and system efficiency. The bypass flow rate was increased by decreasing the EEV opening of the outdoor unit. The bypass flow ratio \( R_{bp} \) was defined as the ratio of the bypass flow rate to the total flow rate, \( R_{bp} = m_{bp}/m_{t} \). As the bypass flow ratio increased from 15.4% to 38.0%, the flow rate to the heating-operated indoor unit was increased by 84.3%. However, at the bypass flow ratio of 38.0%, the total flow rate was decreased by 25.3% from that at the bypass flow ratio of 15.4%, due to the uncondensed refrigerant at the exit of the heating-operated indoor unit. Fig. 3 shows the variations of the average cooling and heating capacities according to the bypass flow ratio of the refrigerant into the heating-operated indoor unit. With the increase of the bypass flow ratio, the heating capacity increased until the bypass flow ratio reached 31.0%. However, the average cooling capacity decreased as the bypass flow ratio increased beyond 31.0% due to the rapid decrease of the total flow rate. At the bypass flow ratio of 27.0%, the average cooling and heating capacities were 2284 W and 2070 W, respectively, which satisfied the designed capacity of 2.0 kW.

Fig. 4 shows the variations of power consumption, total capacity, and COP according to the bypass flow ratio in the cooling-main mode. The power consumption decreased as the bypass flow ratio increased beyond 31% due to the decrease of the total flow rate. The total capacity and COP yielded maximum values at the bypass flow ratio of 27%. The maximum total capacity in the cooling-main mode was 8922 W, which was 8.2% higher than that in the cooling-only mode. The power consumption at the bypass flow ratio of 27% was 2343 W, which was 11.3% lower than that in the cooling-only mode. With the optimization of the flow rate of the refrigerant into the heating-operated indoor unit, the maximum COP in the cooling-main mode was 3.81, which was an increase of 22.1% from that in the cooling-only mode.
3.2. Performance characteristics in heating-only and heating-main operations

Based on the preliminary tests of the system according to the variation of the EEV opening at the refrigerant charge of 3900 g and the test conditions specified in Table 2, the maximum system COP in the heating-only mode was observed at the EEV opening of 320 steps in the outdoor unit. Therefore, the tests in the heating-only mode were conducted at the refrigerant charge of 3900 g and the compressor speed of 3500 rpm, while the EEV openings of the indoor and outdoor units were maintained at 500 (full opening) and 320 steps, respectively. Experiments in the heating-main mode were also conducted at the same operating conditions as the heating-only mode, except that the EEV opening of the cooling-operated indoor unit was modified to 110 steps.

Fig. 5 compares the cycle characteristics of the heating-only mode with that of the heating-main mode on the pressure–enthalpy diagram. In the heating-main mode, the evaporating pressure increased due to the cooling operation of an indoor unit. Simultaneously, the condensing pressure also increased due to the decrease in the number of heating-operated indoor units. The total flow rate in the heating-main mode was increased by 11.2% from that in the heating-only mode due to the increase of the compressor suction pressure. The average flow rate to the heating-operated indoor unit in the heating-main mode was increased by 48.4% from that in the heating-only mode, while the flow rate to the cooling-operated indoor unit was decreased by 43.6% from the average flow rate to the heating-operated indoor unit. In the heating-main mode, the imbalance in refrigerant distribution resulted in lower cooling capacity.

The total capacity, average heating capacity, and COP in the heating-only mode were 7948 W, 1987 W, and 3.29, respectively. The average heating capacity denotes an averaged value for each heating-operated indoor unit. The total capacity in the heating-main mode was 9748 W, which was 22.6% higher than that in the heating-only mode. The average heating capacity in the heating-main mode was 2662 W, which was 34.0% higher than that in the heating-only mode. However, the cooling capacity in the heating-main mode was 1762 W, which was 33.8% lower than the average heating capacity in the heating-main mode. Therefore, the cooling capacity needs to be increased to satisfy the design value of 2.0 kW. The COP in the heating-main mode was 3.63.

In the heating-main mode, the cooling capacity can be increased by increasing the refrigerant flow rate to the cooling-operated indoor unit. Two controlling methods were considered in this study to increase the flow rate to the cooling-operated indoor unit: (a) decreasing the EEV opening in the outdoor unit and (b) increasing the EEV opening in the cooling-operated indoor unit. Experiments with the application of the first method were performed by decreasing the EEV opening in the outdoor unit from 320 steps (64% of full opening), while the EEV opening in the cooling-operated indoor unit was maintained at 110 steps (22% of full opening). The flow rate to the cooling-operated indoor unit increased with the decrease of the EEV opening in the outdoor unit, even though the total flow rate decreased. The power consumption increased with the decrease of the EEV opening in the outdoor unit because of the increase of the compression ratio. Experiments with the application of the second method were also conducted by increasing the EEV opening in the cooling-operated indoor unit from 110 steps, while the EEV opening in the outdoor unit was maintained at 320 steps. As the EEV opening in the cooling-operated indoor unit was increased, the flow rate to the cooling-operated indoor unit increased and the total flow rate also slightly increased. The power consumption of the system showed no difference with the variation of the EEV opening.

Fig. 6 shows the variations of average cooling and heating capacities according to the EEV openings ($\Phi_{EEV}$ for ODU and IDUc) and compressor speed in the heating-main mode. The compressor speed ratio (CSR) was defined as the ratio of the actual compressor speed to the reference value of 3500 rpm. Both the cooling and heating capacities increased with the increase of the compressor speed due to the increase of the mass flow rate at all EEV openings. As the EEV opening in the outdoor unit ($\Phi_{EEV}$ of ODU) decreased from 64% to 20% at the compressor speed of 3500 rpm (CSR = 100%), the cooling capacity increased from 1762 W to 2889 W, an increase of 64.0% due to the decrease of the superheat region in the...
The average heating capacity increased from 2662 W to 2869 W, an increase of 7.8% due to the increase of the condensing pressure. However, the condensing pressure increased significantly with the decrease of the EEV opening in the outdoor unit. As the EEV opening in the cooling-operated indoor unit ($\Phi_{\text{EEV}}$ of IDU$_{C}$) increased from 22% to 80% at the compressor speed of 3500 rpm (CSR = 100%), the cooling capacity increased from 1762 W to 2723 W, an increase of 54.6%, while the average heating capacity showed no difference with constant condensing pressure. The variation of cooling capacity at high EEV openings above 60% was relatively small because the superheat region did not exist at these EEV openings.

Fig. 7 shows the variation of the system COP according to the EEV openings ($\Phi_{\text{EEV}}$ for ODU and IDU$_{C}$) and compressor speed in the heating-main mode. As expected, the system COP increased with the decrease of compressor speed due to the increase of the compressor efficiency, even though both the cooling and heating capacities decreased due to the reduction of mass flow rate. Therefore, the compressor speed must be optimized to improve the system COP with appropriate cooling and heating capacities. When the EEV opening in the outdoor unit ($\Phi_{\text{EEV}}$ of ODU) was varied, the optimum EEV opening in the outdoor unit was 30% and the optimum compressor speed ratio was 70% with consideration of the system COP and required capacities. At these optimum conditions, the system COP was 5.06, which was 53.8% higher than that in the heating-only mode. In addition, the average heating and cooling capacities were 2026 W and 2042 W, respectively, satisfying the designed capacities. When the EEV opening in the cooling-operated indoor unit ($\Phi_{\text{EEV}}$ of IDU$_{C}$) was varied, the optimum EEV opening in the cooling-operated indoor unit was 40% and the optimum compressor speed ratio was 80% with consideration of the system COP and required capacities. At these optimum conditions, the COP was 4.37, which was 32.8% higher than that in the heating-only mode. In addition, the average heating and cooling capacities were 2068 W and 2102 W, respectively, satisfying the designed capacities. To improve the system COP in the heating-main mode, the method of decreasing the EEV opening in the outdoor unit by modulation of the compressor speed is highly recommended. However, the condensing pressure should be monitored precisely so as not to increase over a limit.

### 3.3 Performance characteristics in entire heat recovery operation

The performance of the system in the entire heat recovery mode was measured at the test conditions specified in Table 2. In the entire heat recovery mode, the EEV opening in the cooling-operated indoor units was set to 110 steps and that in the heating-operated indoor units was set to 500 steps (full opening) at the refrigerant charge of 3900 g. The compressor speed was varied from 1050 to 2800 rpm to obtain appropriate cooling and heating capacities corresponding to the design values. In addition, when the compressor speed was increased to 3500 rpm, the system became unstable due to the high condensing pressure up to 4500 kPa. Therefore, the compressor speed must be controlled optimally with respect to the operation mode to achieve high system reliability and performance.

Fig. 8 shows the variations of average capacity and power consumption according to compressor speed ratio (CSR) in the entire heat recovery mode. The average cooling capacity at the compressor speed ratio of 50% was 2367 W, which was 14.8% higher than that in the cooling-only mode. The average heating capacity at the compressor speed ratio of 50% was 2233 W, which was 12.4% higher than that in the heating-only mode. The power consumption at the compressor speed ratio of 50% was 1197 W, which was 54.7% and 50.5% lower than those in the cooling-only and heating-only modes, respectively. In the entire heat recovery mode, the average capacity of each indoor unit satisfied the designed capacity of 2.0 kW at the compressor speed ratio of 50%, resulting in lower power consumption.

Fig. 9 shows the variation of COP according to the compressor speed ratio in the entire heat recovery mode. The system COP in the entire heat recovery mode decreased with the increase of the compressor speed ratio because the increasing rate of the total capacity with the increase of the compressor speed was higher than that of the power.
consumption. In the present experiments, the optimum compressor speed ratio in the entire heat recovery mode was 50% with consideration of the system COP and required capacities. The system COP at the compressor speed ratio of 50% was 7.69, which was 146.5% and 133.7% higher than those in the cooling-only and heating-only modes, respectively. This COP improvement was due to lower power consumption from the decrease of compressor speed. In addition, the variation of the COP in the entire heat recovery mode was more sensitive to the compressor speed than that in the cooling-only and heating-only modes. Therefore, in the entire heat recovery mode, compressor speed should be controlled more precisely to operate the system effectively at optimum operating conditions.

4. Conclusions

A simultaneous heating and cooling multi-heat pump was designed to provide heating and cooling to different zones of a building at the same time and its performance was measured in five operation modes: cooling-only, heating-only, cooling-main, heating-main, and entire heat recovery. The performance of the simultaneous heating and cooling multi-heat pump with R410a was improved by optimizing control parameters such as the compressor speed and EEV opening. In the cooling-main mode, the flow rate of the bypass refrigerant to the heating-operated indoor unit was optimized by controlling the EEV opening of the outdoor unit, and the maximum performance of the system was observed at the bypass flow ratio of 27%. The maximum total capacity and COP in the cooling-main mode were 8.922 W and 3.81, respectively, which were 8.2% and 22.1% higher than those in the cooling-only mode due to heat recovery. For the heating-main mode, two methods for controlling the mass flow rate in the cooling-operated indoor unit were proposed and tested to determine their effect on the increase of cooling capacity. The first method of decreasing the EEV opening in the outdoor unit gave 13.6% higher system COP than the second method of increasing the EEV opening in the cooling-operated indoor unit. In the first method, the optimum EEV opening in the outdoor unit and compressor speed ratio were selected as 30% and 70%, respectively. The optimized system COP was 5.06, which was 53.8% higher than that in the heating-only mode, with the average heating and cooling capacities of 2026 W and 2042 W, respectively, satisfying the design capacity of 2.0 kW. In the entire heat recovery mode, the optimum compressor speed ratio was selected as 50% with consideration of the system COP and required capacities. The optimized system COP at the compressor speed ratio of 50% was 7.69, which was 146.5% higher than that in the cooling-only mode, with the average cooling and heating capacities of 2367 W and 2233 W, respectively, satisfying the design capacities. This COP improvement was due to lower power consumption that resulted from the decrease of compressor speed.

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