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**Heating performance analysis of an air source heat pump heating system with
different heating terminals under frosting conditions**

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Abstract

Heating terminal configurations significantly affect the operating performance and indoor thermal response of air source heat pumps (ASHPs). However, most studies have focused on steady heating conditions, with limited consideration of the frosting-defrosting process. Consequently, their effects during frosting-defrosting cycles remain unclear, hindering improvements in ASHP heating performance for different terminals. To address this issue, an ASHP test rig was established in an artificial environment chamber, in which fan coil units (FCUs) and radiant floors (RFs) were connected to the same ASHP unit and operated separately. Then, experiments were performed under preset environmental conditions to compare the frosting-defrosting characteristics, heating performance, and indoor thermal environment of the ASHP system under the FCU and RF heating modes. Results showed that defrosting caused obvious fluctuations in supply and return water temperatures under both the FCU and RF heating modes, with larger fluctuations observed under the FCU heating mode. Compared to those under the RF heating mode, the frosting suppression performance under the FCU heating mode was better, with frosting durations prolonged by 21.7% and 29.8%, respectively. However, better heating performance was obtained under the RF heating mode. The average heating capacity and COP under the RF heating mode were 7.9% and 21.3% higher than those under the FCU heating mode. In addition, the defrosting had less effect on indoor thermal environment under RF heating mode. The indoor temperature and relative humidity remained nearly unchanged under RF heating mode, while fluctuated 3.13–4.02 °C and 6.03%–7.51% under the FCU heating mode.

Keywords: air source heat pump, indoor thermal environment, frosting-defrosting, heating performance, heating terminal

Nomenclatures

c_w	Specific heat capacity of water [kJ/kg·°C]
M_{dfw}	Mass of melted frost [g]
m_w	Water mass flow rate [kg/s]
q	Heating capacity [kW]
q_A	Maximum heating capacity during the stable heating stage after defrosting [kW]
q_B	Heating capacity at defrosting initiation [kW]
RH	Relative humidity [%]
RH_i	Indoor air relative humidity [%]
T	Temperature/Dry-bulb temperature [°C]
T_c	Outdoor coil surface temperature [°C]
T_i	Indoor air temperature [°C]
T_r	Return water temperature [°C]
T_s	Supply water temperature [°C]
t	Operating time [s]
t_f	Frosting duration [s]
v_{dfw}	Average frosting rate [g/min]
W	Input power [kW]
ΔT	Temperature difference between T and T_c [°C]
ΔT_w	Temperature difference between T_s and T_r [°C]
ε_{NL}	Energy loss coefficient caused by frosting-defrosting [%]
ε_q	Maximum attenuation rate of q caused by frosting [%]

Abbreviations

ASHP	Air source heat pump
COP	Coefficient of performance
FCU	Fan coil unit
RF	Radiant floor
$\Delta T-t$	Temperature difference-time defrosting control strategy

Subscripts

A	Initial stable heating state after the preceding defrosting process
ave	Average value
B	Defrosting initiation state

1. Introduction

With the continuous growth of building floor area and residential energy demand, space heating energy consumption accounts for a considerable share of total building energy consumption [1, 2]. Consequently, improving heating efficiency and reducing carbon emissions have become critical challenges for the low-carbon development in buildings [3, 4]. Among various heating technologies, air source heat pumps (ASHPs) use ambient air as a heat source and provide heat at a suitable temperature for space heating with electrical energy input [5, 6]. Owing to their high efficiency, operational flexibility, and low pollutant emissions, ASHPs have been widely used in building heating systems [7, 8]. However, frost formation commonly occurs on the outdoor heat exchangers of ASHPs under cold and humid operating conditions [9, 10]. Frost accumulation increases the air-side thermal resistance, lowers the evaporating temperature, and ultimately degrades the heating capacity (q) and coefficient of performance (COP) of ASHPs [11, 12].

When frost accumulation reaches a critical level, defrosting must be initiated to prevent further deterioration in ASHP operating efficiency [13, 14]. Among existing defrosting methods, reverse-cycle defrosting is commonly adopted in ASHP systems [15]. In this method, the four-way valve is switched to reverse the refrigerant flow direction. The high-temperature refrigerant vapor discharged from the compressor then enters the outdoor heat exchanger and removes the frost layer [16]. However, during reverse-cycle defrosting, the indoor-side heat exchanger switches from a condenser to an evaporator [17]. Consequently, the water-side heat transfer direction is reversed, and the system shifts from delivering heat to the circulating water to absorbing heat from it, thereby temporarily interrupting normal heating operation [18]. This process can reduce both the supply water temperature (T_s) and the heat supplied to the indoor space, and may further decrease the indoor air temperature (T_i) [19]. Therefore, reverse-cycle defrosting not only deteriorates the operating performance of ASHPs but also disturbs the indoor thermal environment, leading to a decline in thermal comfort [20].

Previous studies have attempted to improve the defrosting performance of ASHPs to mitigate heating interruptions and fluctuations in the indoor thermal environment caused by defrosting [21]. These studies can be broadly grouped into two research directions. The first direction aims to optimize the criteria for defrosting initiation and termination, thereby reducing the risk of premature or delayed defrosting [22–24]. Zhao et al. [22] developed a defrosting control approach using machine vision to detect frost growth on the outdoor heat exchanger and improve defrosting decision accuracy. Xu et al. [23] developed an adaptable intelligent frost recognition method based on evaporator image texture features and machine learning to

1 enhance frosting state recognition under complex lighting conditions. Tang et al. [24] proposed a time-
2 current-temperature difference defrosting control method based on outdoor fan operating characteristics,
3 providing a more comprehensive criterion for defrosting initiation. The second direction focuses on
4 optimizing defrosting strategies and maintaining heat supply during defrosting, thereby shortening the
5 defrosting process and lowering energy consumption [25, 26]. Zhang et al. [25] introduced a defrosting
6 method with thermal energy storage, in which heat dissipated by the compressor was recovered to improve
7 defrosting performance and sustain heat supply during defrosting. Kim et al. [26] proposed a defrosting
8 approach that combined dual hot-gas bypass with an accumulator heater to enhance defrosting performance
9 in cold climate conditions. Overall, these studies have improved defrosting efficiency and reduced the
10 adverse effects of defrosting on heating operation. However, most of them have focused on outdoor units
11 of ASHPs, with limited consideration of the influence of heating terminal configurations on ASHP
12 operation.

13 In practical operation, different heating terminals require different T_s values and exhibit distinct heat
14 transfer modes and thermal inertia, which may lead to differences in ASHP heating performance [27]. Fan
15 coil units (FCUs) and radiant floors (RFs) are two commonly used heating terminals in ASHP systems [28].
16 FCUs provide space heating mainly through forced convection and generally require relatively high T_s
17 values [29], whereas RFs provide space heating mainly through thermal radiation and can usually meet
18 indoor heating demand at lower T_s values [30–32]. Several studies have compared ASHP systems coupled
19 with FCUs and RFs. Atienza Márquez et al. [33] used TRNSYS simulations to compare control strategies
20 for an ASHP system equipped with both FCUs and RFs, reporting that the combined FCU + RF
21 configuration improved thermal comfort during the heating season. Hu et al. [28] experimentally compared
22 q , COP, indoor temperature distribution, and thermal comfort of ASHP systems coupled with different
23 heating terminals, and reported better thermal comfort under the RF heating mode. Liu et al. [27] compared
24 FCU, RF, and combined FCU + RF heating modes in a convective and radiant heating system coupled with
25 an ASHP, showing that FCU operation caused larger T_i fluctuations, whereas RF and FCU + RF modes
26 maintained more stable T_i values. However, these studies did not consider the effects of the frosting-
27 defrosting process. During frosting-defrosting operation, the T_s and return water temperature (T_r) of ASHP
28 systems fluctuate considerably. Because FCUs and RFs require different operating water temperatures, the
29 water temperature fluctuations induced by frosting-defrosting may also differ between the two heating
30 modes, thereby causing different effects on ASHP operating performance and the indoor thermal
31 environment. At present, the differences in these effects under different heating terminal configurations

remain unclear.

To address this research gap, an experimental ASHP platform was established in an artificial environment chamber in this work. The single outdoor unit was connected to both FCUs and RFs, and the FCU and RF heating modes were switched by adjusting the corresponding valves. Then, the frosting-defrosting characteristics, heating performance, and indoor thermal environment of the ASHP system under the two heating modes were analyzed. The novelty of this work lies in: (1) revealing the differences in frosting-defrosting characteristics and heating performance of the ASHP system between the FCU and RF heating modes; and (2) clarifying the different effects of FCU and RF heating modes on the indoor thermal environment during frosting-defrosting conditions. The findings of this study can guide the development of defrosting control methods tailored to different heating terminals and help mitigate indoor thermal environment fluctuations associated with different heating terminal configurations.

2. Experimental setup and methods

2.1 Artificial environment chamber

To investigate the heating performance and indoor thermal environment of an ASHP system under different heating terminal configurations, experiments were carried out in an artificial environment chamber located in Zhuhai, Guangdong Province, China. The chamber was configured to reproduce specified outdoor conditions and record thermal environmental parameters. This setup enabled the frosting-defrosting characteristics, heating performance, and indoor thermal environment of the ASHP system to be evaluated under controlled conditions. As illustrated in Fig. 1(a), the artificial environment chamber was divided into an external room and an internal room. Heating, humidification, and refrigeration devices were installed in the external room to simulate the required outdoor environmental conditions. This configuration provided controlled frosting-defrosting conditions around the ASHP outdoor unit.

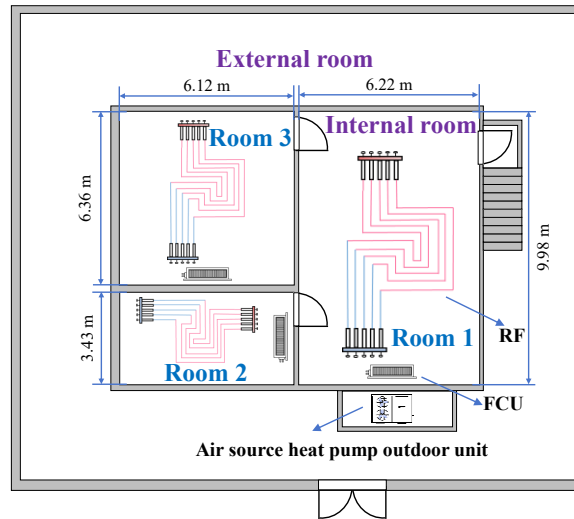
The internal room served as the indoor-side test space for the ASHP heating experiments. It contained a two-story residential test building constructed in accordance with the design standards for China's Hot-Summer and Cold-Winter Climate Zone. The internal room was employed to analyze the influence of different heating terminals on the indoor thermal environment under frosting-defrosting conditions. The ASHP unit was arranged in the external room and connected to both FCUs and RFs to provide heating for the internal room, as illustrated in Fig. 1(b). In this study, FCUs and RFs were selected because they represent two typical heating terminals with distinct heat transfer mechanisms and thermal inertia characteristics, while radiator terminals were not included because they were not configured in the present

1 experimental platform. This experimental setup therefore enabled a comparative assessment of the FCU
 2 and RF heating modes.

3 The internal room contained three test rooms, designated as Room 1, Room 2, and Room 3. Their floor
 4 areas were 62.08 m², 20.99 m², and 38.92 m², respectively. Under each test condition, the three rooms were
 5 heated simultaneously using the same type of heating terminal, either FCUs or RFs. This arrangement
 6 enabled the ASHP heating performance and indoor thermal environment to be compared under different
 7 heating terminal configurations.



(a) Three-dimensional view of the artificial environment chamber



(b) Floor plan of Rooms 1–3

Fig. 1. Configuration of the artificial environment chamber and arrangement of Rooms 1–3.

9 2.2 Experimental ASHP system

10 FCUs and RFs are two commonly used heating terminals in ASHP systems. RFs primarily transfer
 11 heat through thermal radiation, whereas FCUs rely primarily on forced convection. During heating
 12 operation, RFs generally require lower T_s values, which is beneficial for improving system performance. In
 13 addition, their large thermal inertia may help smooth T_i variations. By contrast, FCUs usually require higher
 14 T_s values, which may reduce the heating efficiency of the ASHP system. Moreover, their lower thermal
 15 inertia may lead to more pronounced T_i fluctuations. As a result, FCUs and RFs may exhibit substantial

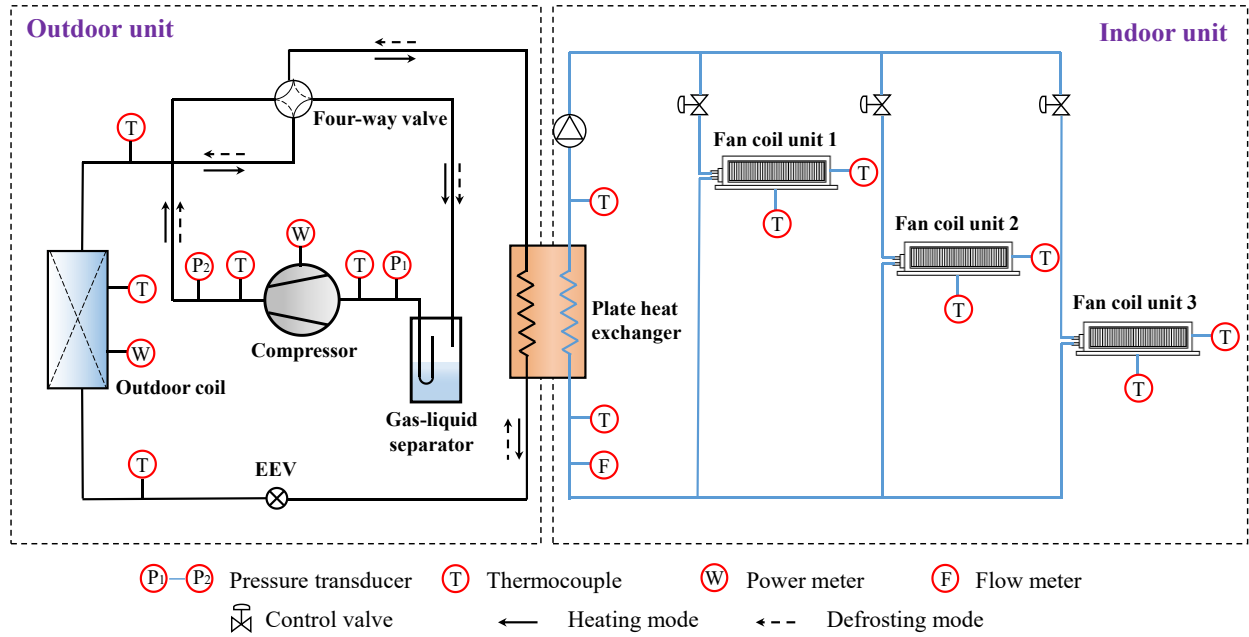
1 differences in operating performance and in their effects on the indoor thermal environment during frosting-
2 defrosting cycles.

3 The schematic diagrams of the experimental ASHP system are shown in Fig. 2(a) and (b). To ensure
4 consistent experimental conditions, no buffer tank was installed in the ASHP system. This eliminated the
5 potential influence of its heat storage capacity and allowed the effects of different heating terminal
6 configurations on heating performance and indoor thermal environment to be isolated more clearly.

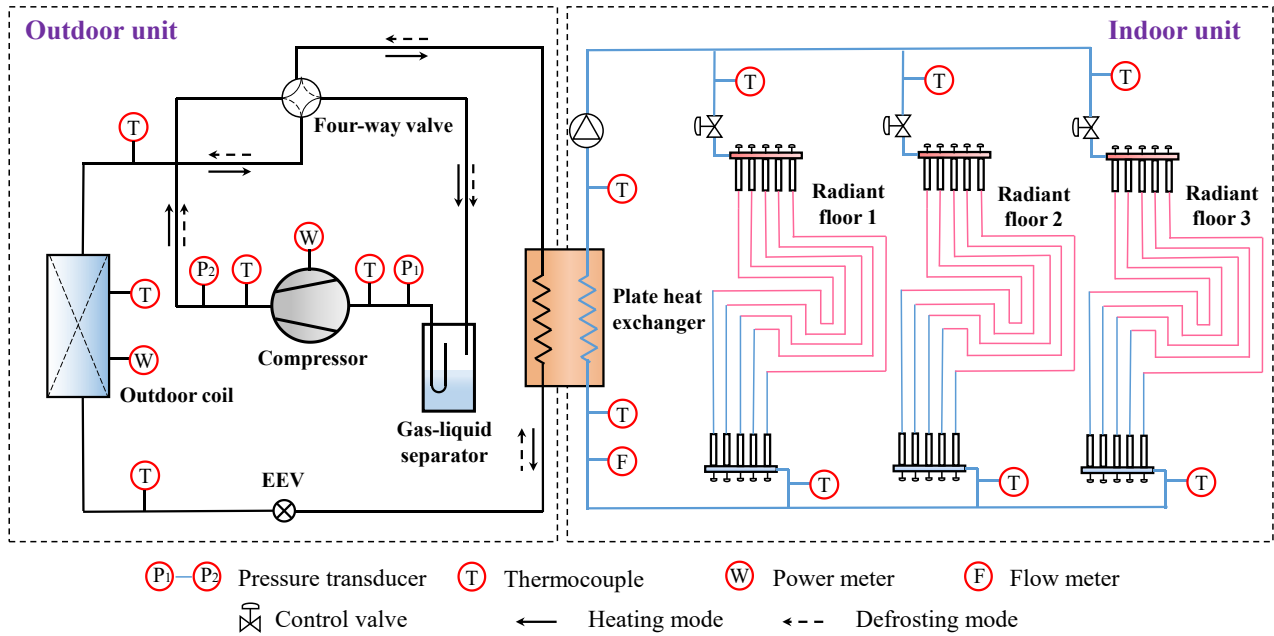
7 The experimental ASHP system consisted of one outdoor unit, three FCUs, and three RFs. The FCUs
8 and RFs were installed in Rooms 1–3, with one FCU and one RF in each room to provide space heating.
9 The two heating terminal configurations were connected to the same outdoor unit, and the operating mode
10 was switched by adjusting the corresponding valves. Therefore, the ASHP system could be operated in
11 either the FCU heating mode or the RF heating mode.

12 The experimental ASHP unit had a rated q value of 18 kW and a rated COP of 3.1. The refrigerant was
13 R410A, and the detailed specifications are listed in Table 1. During the frosting period under each terminal
14 configuration, the operating terminals were independently controlled according to the measured T_i values
15 in Rooms 1–3. Therefore, the comparative analysis of ASHP system performance and indoor thermal
16 environment under different heating terminal configurations was not affected by the number of operating
17 terminals.

18 The experimental ASHP unit adopted the reverse-cycle defrosting method, and the temperature
19 difference-time (ΔT - t) control strategy was used to initiate defrosting [34]. Defrosting was initiated when
20 ΔT exceeded the preset threshold of 7.0 °C and t exceeded 40 min. Once the ASHP system entered
21 defrosting mode, the four-way valve reversed the refrigerant flow direction. The high-temperature
22 refrigerant vapor from the compressor then entered the outdoor heat exchanger and removed the
23 accumulated frost. When the outdoor heat exchanger surface temperature (T_c) reached 20 °C, the ASHP
24 terminated defrosting and resumed heating operation. The same defrosting control strategy was applied
25 under two heating terminal configurations, ensuring a consistent defrosting criterion for comparing ASHP
26 frosting-defrosting characteristics under the FCU and RF heating modes. During defrosting, the circulating
27 water pump remained in operation under both heating modes. The FCUs continued operating with their
28 fans running, while water circulation through the RF loops was also maintained. Therefore, the heating
29 terminals were not shut down during defrosting.



(a) ASHP with FCU heating mode



(b) ASHP with RF heating mode

Fig. 2. Schematics of the ASHP system under different heating terminal configurations.

Table 1 Specifications of the ASHP outdoor unit.

Items	Values
Rated condition (°C)	7/6 (dry-bulb/wet-bulb temperature)
Rated heating capacity (kW)	18
Rated COP	3.1
Type of outdoor coil	Finned-tube
Compressor type	Scroll
Compressor speed (rps)	40–100
Refrigerant	R410A

2.3 Data acquisition system

To ensure the reliability of the experimental data, a data acquisition system was installed in the artificial environment chamber to record the ASHP operating parameters, outdoor environmental conditions, and indoor environmental parameters. All monitored parameters were synchronously recorded at a sampling interval of 5 s throughout the experiments, enabling the rapid transient variations during the frosting-defrosting process to be captured. The monitoring arrangements are described as follows:

(1) ASHP operating parameters

The compressor suction and discharge lines were instrumented with pressure transducers and thermocouples to record the corresponding pressures and temperatures. A smart power meter recorded the instantaneous input power (W). An additional thermocouple was used to record T_c . The frost growth and removal process on the outdoor heat exchanger was visually recorded using a video camera during frosting-defrosting cycles. The defrost water was collected and weighed using an electronic scale to quantify the mass of melted frost in each cycle (M_{dfw}). In addition, refrigerant temperatures in the outdoor unit were measured at two representative positions using thermocouples. Two thermocouples were arranged on the circulating water pipes to measure T_s and T_r . The total water flow rate (m_w) was measured by an electromagnetic flow meter mounted on the main return water pipe. The measurement points are presented in Fig. 2.

(2) Outdoor environment

The outdoor air temperature and relative humidity were recorded by a temperature & humidity sensor positioned 0.5 m from the air inlet surface of the outdoor heat exchanger of the outdoor unit, outside the influence of the outlet airflow.

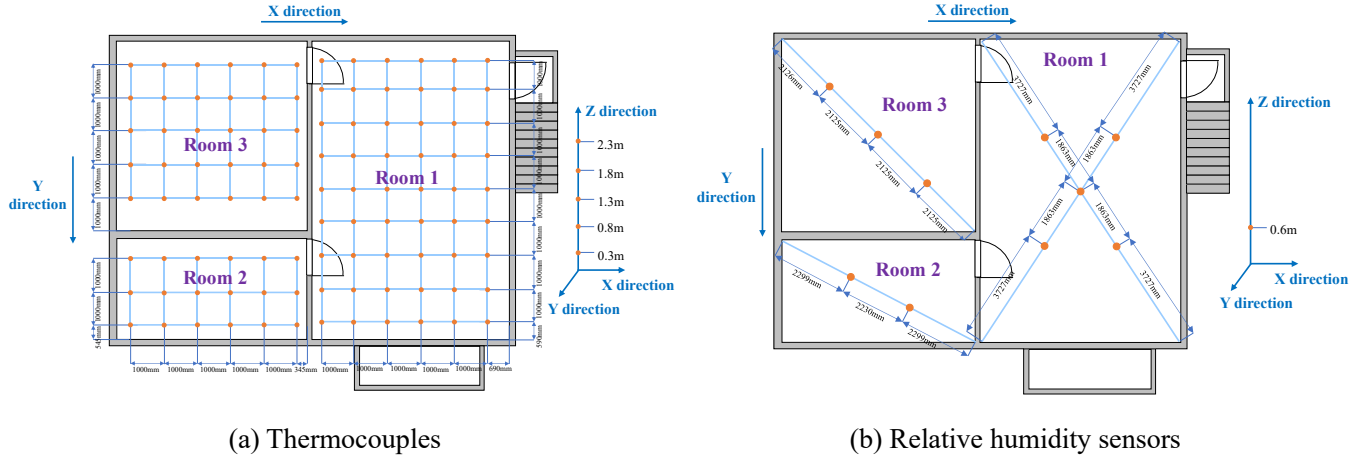
(3) Indoor environment

Indoor thermal conditions in Rooms 1–3 were monitored using multiple thermocouples and temperature & humidity sensors. The sensor arrangement was as follows:

1) Fig. 3(a) shows the T_i measurement points arranged in the horizontal plane of Rooms 1–3. At each measurement location, thermocouples were arranged vertically along a plumb line. Thermocouples were installed at heights of 0.3 m, 0.8 m, 1.3 m, 1.8 m, and 2.3 m along each vertical line. A total of 270 ($9 \times 6 \times 5$), 90 ($6 \times 3 \times 5$), and 150 ($6 \times 5 \times 5$) thermocouples were installed in Rooms 1–3, respectively, to measure the spatial indoor temperature distribution. As illustrated in Fig. 2(a), the air temperatures at the inlet and outlet of each FCU were recorded using two thermocouples. As shown in Fig. 2(b), the supply and return water temperatures of each RF floor heating loop in Rooms 1–3 were recorded using

1 thermocouples placed at the corresponding manifolds.

2 2) Fig. 3(b) shows the indoor relative humidity (RH_i) measurement points arranged in the horizontal
 3 plane of Rooms 1–3. Following the method specified in JGJ/T 347-2014 [35], 5, 2, and 3 relative humidity
 4 sensors were arranged in Rooms 1–3, respectively, at the midpoint, trisecting points, or quartering points
 5 along the room diagonals. All relative humidity sensors were positioned at a consistent height of 0.6 m
 6 above the floor.



7 **Fig. 3.** Layout of indoor temperature and relative humidity measurement points in Rooms 1–3.

8 To ensure data reliability, all measuring instruments were calibrated periodically, and selected
 9 instruments were randomly checked for verification. Table 2 summarizes the main technical parameters of
 10 the measuring instruments.

11 **Table 2** Details of the measuring instruments.

Name	Measuring range	Accuracy	Number
Thermocouple	-20–100 °C	±0.1 °C	531
Temperature & humidity sensor	Temperature: 0–50 °C	±0.15 °C	11
	Relative Humidity: 0–100%	±3%	
Electromagnetic flow meter	0.1–30 m³/h	0.5% FS	1
Power meter	0–6.6 kW	0.25% FS	2
Pressure transducer	0–3.0 MPa	0.3% FS	2
Video camera	10 megapixels	/	1
Electronic scale	0–10 kg	±1 g	1

12 2.4 Experimental conditions and procedures

13 (1) Experimental conditions

14 With reference to GB/T 25127-2020 [36] and ANSI/AHRI Standard 550/590-2023 [37], the ASHP
 15 frosting performance test was conducted under a representative outdoor condition of 2/1 °C dry-bulb/wet-
 16 bulb temperature. This condition promotes rapid frost accumulation on the outdoor heat exchanger and is
 17 suitable for characterizing typical frosting behavior. The external room was maintained at 2/1 °C throughout

the experiments to ensure consistent and comparable test conditions.

During the heating season in China's Hot-Summer and Cold-Winter Climate Zone, T_i values in buildings heated by ASHP systems are usually maintained within 22–24 °C [19]. Considering the differences in floor area and corresponding heating loads among the three rooms, different target temperatures within this range were adopted. Specifically, a lower target temperature was assigned to the room with a larger floor area to reduce the differences in heating load among the three rooms. Accordingly, the target indoor temperatures for Rooms 1–3 were set at 22 °C, 24 °C, and 23 °C, respectively, under both FCU and RF heating modes. To account for possible fluctuations in T_i during actual operation, an allowable deviation of ± 2 °C was adopted for each room. Thus, the temperature control ranges for Rooms 1–3 were 22 ± 2 °C, 24 ± 2 °C, and 23 ± 2 °C, respectively. Table 3 provides the detailed indoor and outdoor environmental settings.

(2) Experimental procedures

Preliminary tests were performed before the formal experiments to adjust the ASHP operating parameters so that q could match the building heating load. After the outdoor environment reached the preset 2/1 °C condition, the ASHP was started, and q was regulated by adjusting the compressor speed and outdoor fan speed. When the T_i values in Rooms 1–3 stabilized within their target ranges, the ASHP was considered to have provided sufficient heating output for the building load. To ensure consistent initial conditions across all formal tests, a complete defrosting operation was performed before each test to eliminate the influence of accumulated frost.

The preliminary tests indicated that the same compressor and outdoor fan speeds, namely 70 rps and 1750 rpm, respectively, enabled the ASHP to approximately match the combined heating demand of Rooms 1–3 under both FCU and RF heating modes. The circulating water flow rates were maintained at 1.5 m³/h and 2.0 m³/h under the FCU and RF heating modes, respectively. Therefore, all subsequent experiments used these operating parameters, which are provided in Table 3. All formal experiments were conducted only after the ASHP reached stable operating conditions. By the end of the preliminary tests, T_i , outdoor temperature, and W had stabilized within the predefined thresholds.

Indoor environmental parameters were continuously monitored throughout the experiments to track transient indoor thermal variations during the frosting-defrosting process. To avoid additional fluctuations in q caused by parameter adjustments, the compressor and outdoor fan were manually controlled to operate at constant speeds during all tests. Once the ASHP operating conditions satisfied the predefined defrosting initiation conditions specified in Section 2.2, the ASHP automatically entered defrosting mode. To improve

data robustness and reduce experimental uncertainty, the tests under each operating condition were performed over two consecutive frosting-defrosting cycles.

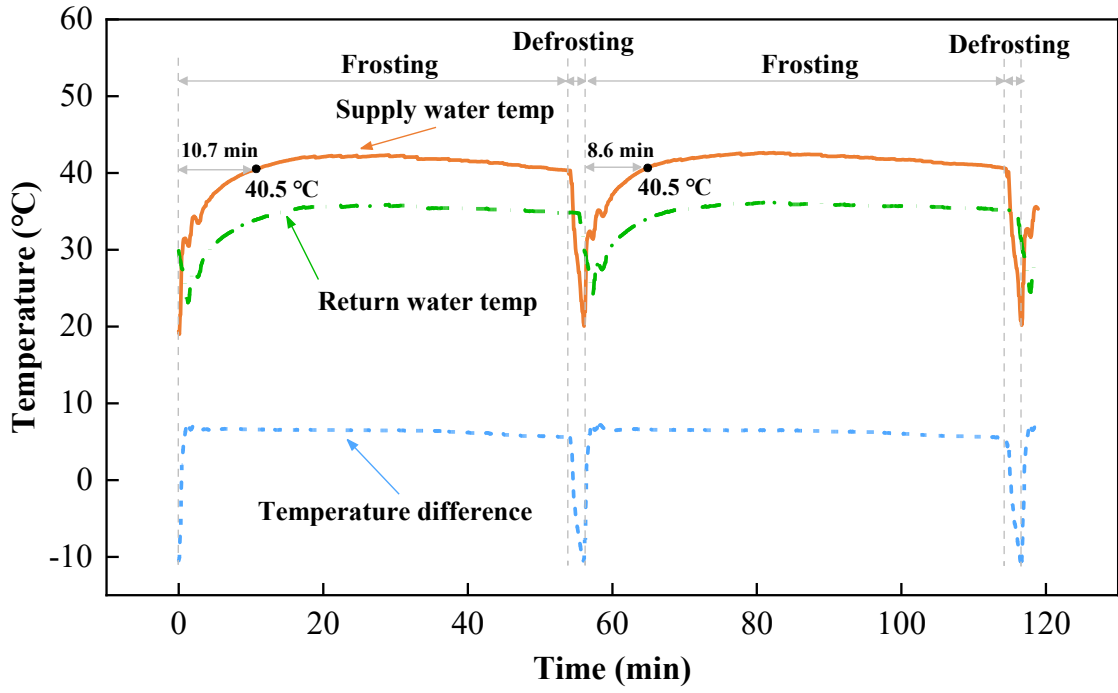
Table 3 Experimental conditions under different heating terminal configurations.

Heating terminal configuration	Compressor speed (rps)	Outdoor fan speed (rpm)	Circulating water flow rate (m ³ /h)	Outdoor environmental condition (°C)	Indoor temperature setpoints (°C)
ASHP with FCU heating mode	70	1750	1.5	2 (dry-bulb temperature)/ 1 (wet-bulb temperature)	Room 1: 22 Room 2: 24 Room 3: 23
ASHP with RF heating mode	70	1750	2.0	2 (dry-bulb temperature)/ 1 (wet-bulb temperature)	Room 1: 22 Room 2: 24 Room 3: 23

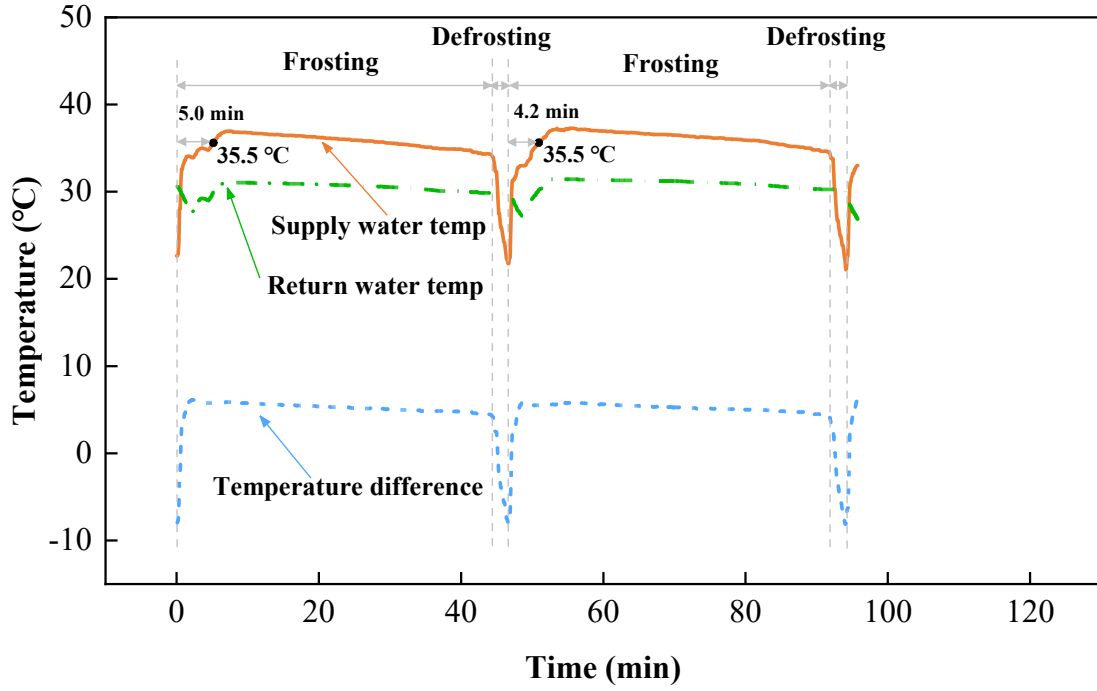
3. Experimental results and discussion

3.1 Comparison of frosting-defrosting characteristics of the ASHP

Fig. 4(a) displays the changes in T_s and T_r , and the temperature difference between them (ΔT_w) under the FCU heating mode during two consecutive frosting-defrosting cycles. The test period covered two complete frosting-defrosting cycles, beginning when the ASHP resumed heating after the pre-test defrosting operation and ending when normal heating operation was restored after the second defrosting process. Since a complete defrosting operation had been performed before the formal test, the accumulated frost on the outdoor heat exchanger had been removed. Accordingly, T_s and T_r were relatively low at the beginning of the test, with initial values of 19.36 °C and 29.86 °C, respectively. After the ASHP resumed heating operation, T_s and T_r increased gradually and then approached steady levels of about 40 °C and 35 °C, respectively, during the stable heating periods of the two frosting-defrosting cycles.



(a) ASHP under the FCU heating mode



(b) ASHP under the RF heating mode

Fig. 4. Variations in water temperatures of the ASHP system under the FCU and RF heating modes during frosting-defrosting cycles.

As shown in Fig. 5, T_c decreased continuously during the frosting process under the FCU heating mode, dropping from $-2.18\text{ }^{\circ}\text{C}$ to $-5.10\text{ }^{\circ}\text{C}$ in the first cycle and from $-2.14\text{ }^{\circ}\text{C}$ to $-5.20\text{ }^{\circ}\text{C}$ in the second cycle. The frosting durations were 53.8 min and 57.9 min in the two cycles, respectively. As frosting progressed, the

1 ASHP satisfied the defrosting initiation criteria and subsequently entered defrosting mode. During
 2 defrosting, the four-way valve switched the indoor heat exchanger from a condenser to an evaporator. As
 3 the refrigerant extracted heat from the circulating water, T_s and T_r fluctuated markedly. The maximum T_s
 4 fluctuation amplitudes were 20.32 °C and 20.37 °C in the two frosting-defrosting cycles, respectively, and
 5 the corresponding T_r fluctuation amplitudes were 10.56 °C and 10.34 °C. Once the defrosting termination
 6 criteria were satisfied, refrigerant flow was reversed again, and the ASHP resumed heating operation,
 7 marking the beginning of the heating recovery stage.

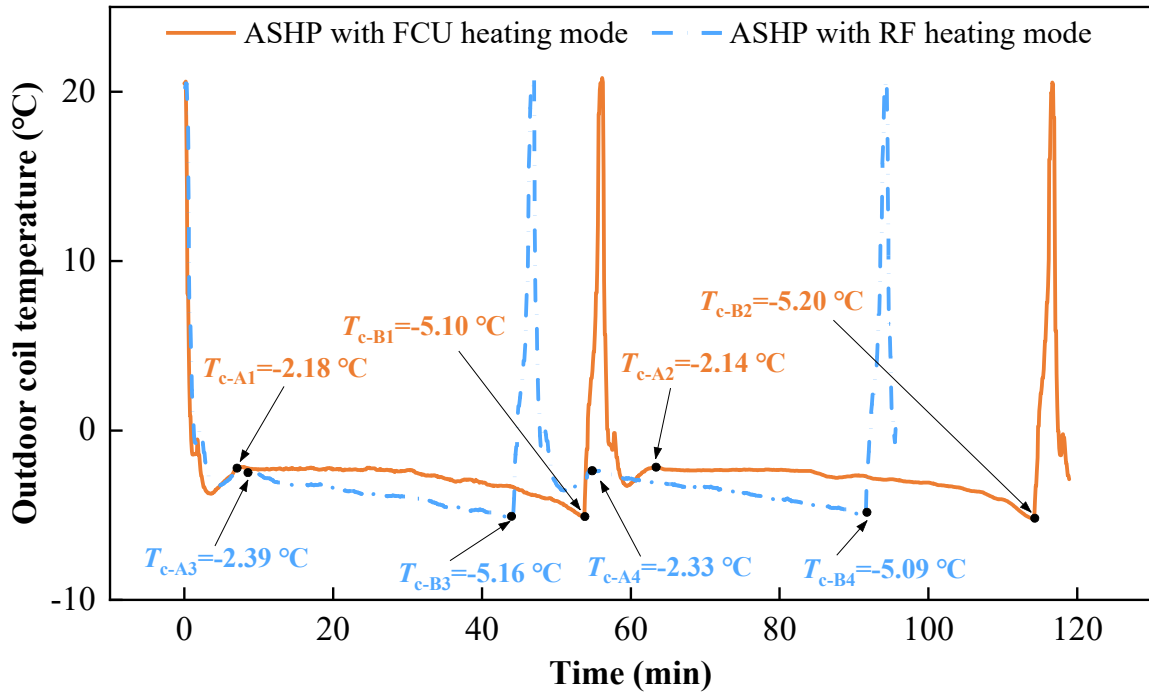


Fig. 5. Variations in the outdoor coil temperature of the ASHP system under the FCU and RF heating modes.

As presented in Fig. 5, under the FCU heating mode, the defrosting duration in each frosting-defrosting cycle was approximately 3 min. Although the defrosting process, which corresponded to an interruption of heat supply from the ASHP, was relatively short, T_r decreased to about 24.0 °C during both frosting-defrosting cycles because the circulating water continued to flow through the operating FCUs (Fig. 4(a)) while supplying heat to the refrigerant circuit during this period. After defrosting ended, the system switched back to heating operation and underwent a relatively long thermal recovery period. The heating operation of the ASHP was considered to have approximately recovered to its normal state after T_s reached 40.5 °C again. The corresponding thermal recovery times were 10.7 min and 8.6 min in the two frosting-defrosting cycles, respectively. Therefore, under the FCU heating mode, the abnormal heating duration caused by the defrosting process consisted of the defrosting period and the subsequent thermal recovery period, with a total duration of approximately 11.6–13.7 min.

Fig. 4(b) displays the changes in T_s , T_r , and ΔT_w under the RF heating mode during two consecutive frosting-defrosting cycles. The test period was selected using the same criterion as that in Fig. 4(a), beginning when the ASHP resumed heating operation after the preliminary defrosting process and ending when the system recovered to normal heating operation after the second defrosting process. Similar to the FCU heating mode, both T_s and T_r were relatively low at the beginning of the test due to the preceding defrosting process, with initial values of 22.64 °C and 30.61 °C, respectively.

During the stable heating periods, T_s and T_r were maintained at approximately 35 °C and 30 °C, respectively, in the two frosting-defrosting cycles. Based on the variations in T_c under the RF heating mode presented in Fig. 5, T_c decreased continuously during the frosting process, dropping from -2.39 °C to -5.16 °C in the first cycle and from -2.33 °C to -5.09 °C in the second cycle. The frosting durations were 44.2 min and 44.6 min in the two cycles, respectively. As frosting progressed, defrosting was initiated, and the resulting heating interruption lasted approximately 3 min in each cycle, close to that in the FCU heating mode. During defrosting, T_s and T_r also exhibited fluctuations. The maximum T_s fluctuation amplitudes were 12.54 °C and 13.22 °C in the two frosting-defrosting cycles, respectively, and the corresponding T_r fluctuation amplitudes were 2.57 °C and 3.29 °C, respectively. When T_s recovered to 35.5 °C, the heating operation of the ASHP was considered to have approximately returned to its normal state. The corresponding thermal recovery times were 5.0 min and 4.2 min in the two frosting-defrosting cycles, respectively. Therefore, under the RF heating mode, the abnormal heating duration caused by defrosting, including the defrosting period and the subsequent thermal recovery time, was approximately 7.2–8.0 min.

Comparison of Fig. 4(a) and (b) shows that the ASHP exhibited similar overall water temperature variation trends under the FCU and RF heating modes. Specifically, during the heating recovery stage, T_s and T_r gradually increased, whereas both decreased sharply when the system entered the defrosting process. However, clear differences were observed between the two heating terminal configurations in the following aspects.

(1) The T_s decrease amplitudes and abnormal heating durations under the FCU heating mode were both greater. The maximum T_s decreases caused by defrosting in the two experimental cycles were 20.32 °C and 20.37 °C under the FCU heating mode, compared with 12.54 °C and 13.22 °C under the RF heating mode. The T_s decrease amplitudes under the FCU heating mode were 62.0% and 54.1% greater than those under the RF heating mode. The abnormal heating durations caused by defrosting were 11.6–13.7 min under the FCU heating mode, compared with only 7.2–8.0 min under the RF heating mode. Therefore, the abnormal heating duration under the FCU heating mode was 61.1%–71.3% longer than that under the RF

1 heating mode.

2 (2) Compared with the RF heating mode, the FCU heating mode exhibited better frosting suppression
3 performance. The frosting durations were 53.8 min and 57.9 min under the FCU heating mode, compared
4 with 44.2 min and 44.6 min under the RF heating mode in the two cycles, respectively. Thus, the frosting
5 durations under the FCU heating mode were prolonged by 21.7% and 29.8%, respectively, compared with
6 those under the RF heating mode.

7 These results indicate that different heating terminals resulted in substantial differences in the frosting-
8 defrosting cycle duration, water temperature fluctuations, and thermal recovery time of the ASHP system.

9 Fig. 6 demonstrates the frost distribution on the outdoor coil surface under the FCU and RF heating
10 modes during different frosting-defrosting cycles. At 0 min, no obvious frost was observed on the outdoor
11 coil surface. As the frosting process continued, frost gradually accumulated on the outdoor coil surface and
12 expanded from local regions to the entire heat exchanger surface.

13 Under the same heating terminal configuration, the frost morphology and frost development process
14 on the outdoor coil surface were generally consistent among different frosting-defrosting cycles, indicating
15 relatively small cycle-to-cycle differences in frosting characteristics. However, clear differences in the frost
16 accumulation rate were observed between the two heating terminal configurations. Specifically, at 10 min,
17 visible frost had already appeared on the outdoor coil surface under the RF heating mode, whereas frost
18 formation under the FCU heating mode remained insignificant. As the frosting time increased, this
19 difference became more pronounced. At 40 min, the outdoor coil surfaces in both frosting-defrosting cycles
20 under the RF heating mode were almost completely covered by frost. In contrast, under the FCU heating
21 mode, almost complete frost coverage was not observed on the outdoor coil surface until 50 min.

22 Overall, compared with the FCU heating mode, frost accumulated more rapidly on the outdoor coil
23 surface under the RF heating mode, indicating that the RF heating terminal configuration accelerated the
24 frosting process of the ASHP outdoor heat exchanger.

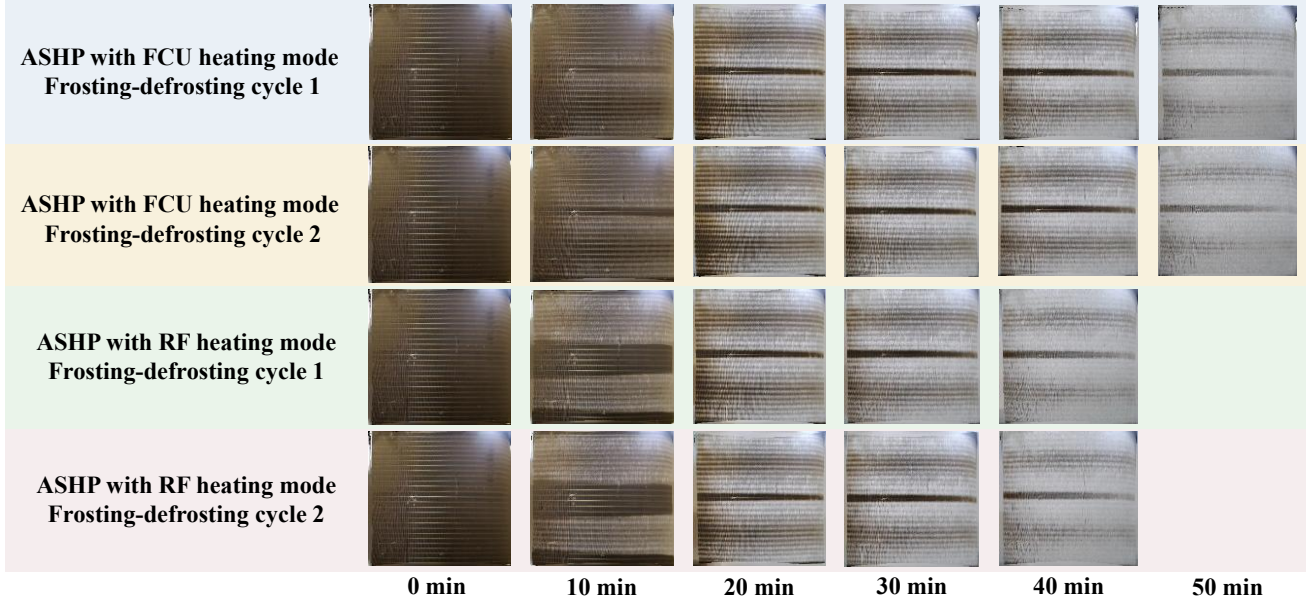


Fig. 6. Photographs of frost development on the outdoor coil surface under different heating terminal configurations during frosting-defrosting cycles.

The M_{dfw} values under the FCU and RF heating modes during different frosting-defrosting cycles were obtained by collecting and weighing the defrost water, as described in Section 2.3. The corresponding average frosting rates (v_{dfw}) were then obtained from Eq. (1) [38], and the calculated values are presented in Fig. 7.

$$v_{dfw} = \frac{M_{dfw}}{t_f} \quad (1)$$

As indicated in Fig. 7, the M_{dfw} values under the FCU heating mode were 3238 g and 3472 g in the two frosting-defrosting cycles, respectively, whereas the corresponding values under the RF heating mode were 2895 g and 2917 g, respectively. The v_{dfw} values under the FCU heating mode were 60.19 g/min and 59.97 g/min, respectively, while those under the RF heating mode were 65.50 g/min and 65.40 g/min. Compared with the FCU heating mode, v_{dfw} under the RF heating mode increased by approximately 8.9% on average over the two cycles. Faster frost accumulation under the RF heating mode caused T_c to decrease more rapidly, as presented in Fig. 5. As a result, ΔT reached the defrosting initiation threshold earlier. Therefore, although v_{dfw} under the RF heating mode was higher, its frosting duration was shorter, resulting in a slightly smaller M_{dfw} than that under the FCU heating mode. This result further confirms that the RF heating mode accelerated frosting on the ASHP outdoor heat exchanger relative to the FCU heating mode, which is consistent with the frost development recorded in Fig. 6.

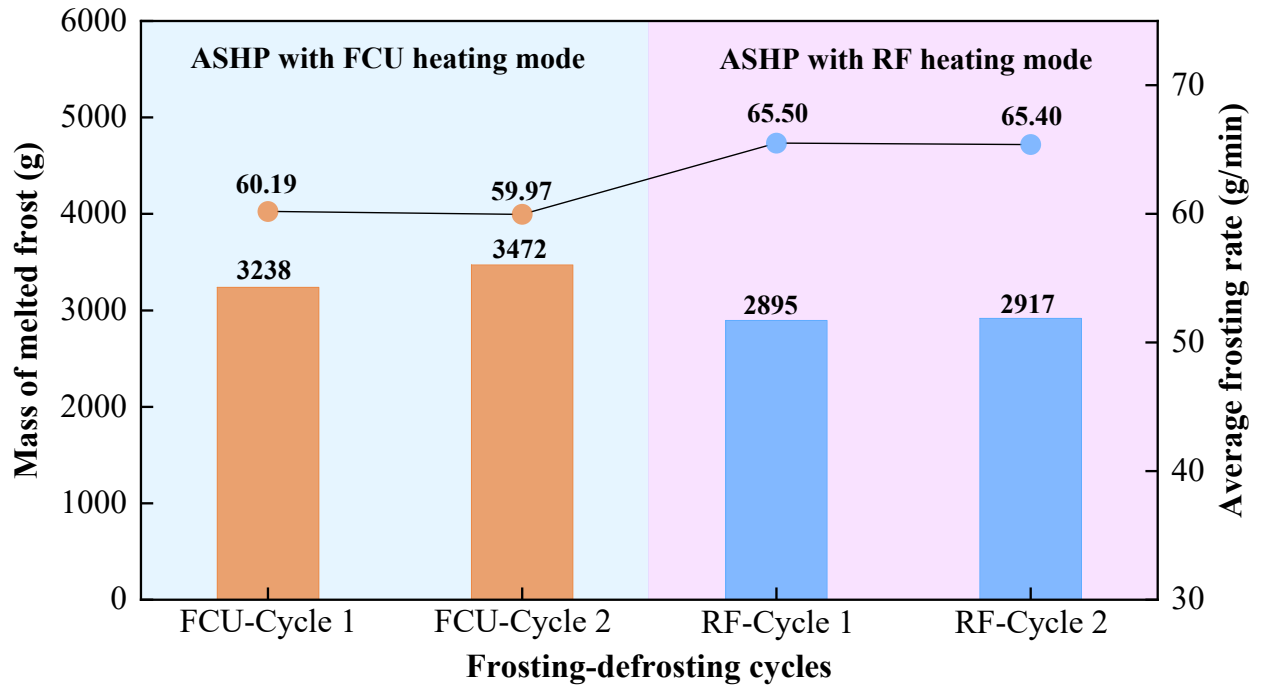


Fig. 7. Mass of melted frost and average frosting rate under the FCU and RF heating modes during frosting-defrosting cycles.

3.2 Heating performance analysis of the ASHP

To clarify the influence of different heating terminals on ASHP heating performance under frosting-defrosting conditions, the dynamic variations in q were analyzed. Based on T_s and T_r in Fig. 4 and m_w measured according to Section 2.3, q under different heating terminal configurations was determined from Eq. (2) [39, 40]. The calculated results are presented in Fig. 8.

$$q = c_w m_w \Delta T_w \quad (2)$$

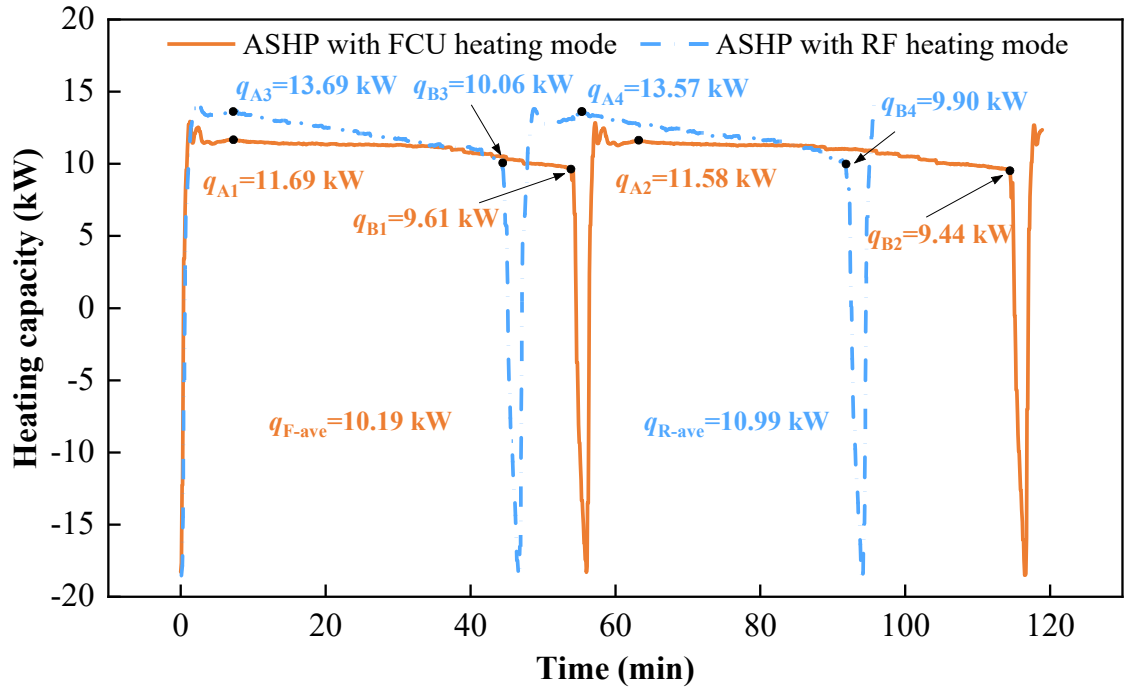


Fig. 8. Variations in heating capacity of the ASHP system under the FCU and RF heating modes during frosting-defrosting cycles.

After defrosting ended, q increased rapidly and then gradually approached a stable level. The maximum q during the stable heating stage after defrosting was defined as q_A , corresponding to point A in each frosting-defrosting cycle. Under the FCU heating mode, the q_A values were 11.69 kW and 11.58 kW in the two frosting-defrosting cycles, respectively, compared with 13.69 kW and 13.57 kW under the RF heating mode. As frosting progressed, q decreased gradually. The q at the onset of defrosting was defined as q_B , corresponding to point B. Under the FCU heating mode, the q_B values were 9.61 kW and 9.44 kW in the two cycles, respectively. The corresponding maximum attenuation rates of heating capacity caused by frosting (ε_q) were 17.8% and 18.5%, respectively, as calculated using Eq. (3) [41]. Under the RF heating mode, q decreased to 10.06 kW and 9.90 kW, respectively, with corresponding ε_q values of 26.5% and 27.0%, respectively.

$$\varepsilon_q = \frac{q_A - q_B}{q_A} \quad (3)$$

Notably, negative q values were observed during defrosting. This occurred because the plate heat exchanger operated as an evaporator during defrosting, and the ASHP therefore extracted heat from the circulating water instead of supplying heat to it. Under both heating modes, the minimum q values during defrosting were approximately -18 kW, corresponding to a peak heat absorption capacity of about 18 kW. Considering the two complete frosting-defrosting cycles, including the thermal recovery, frosting, and

defrosting periods, the average q was 10.19 kW under the FCU heating mode and 10.99 kW under the RF heating mode, indicating an increase of approximately 7.9%.

To quantitatively assess the influence of the frosting-defrosting process on ASHP heating performance, the energy loss coefficient caused by frosting-defrosting (ε_{NL}) was introduced and defined according to Refs. [42, 43] to characterize the degradation in heating performance under different heating terminal configurations, as given in Eq. (4). In Eq. (4), the integration interval from 0 to t corresponds to one complete frosting-defrosting cycle, including the thermal recovery period after the previous defrosting process, the subsequent frosting period, and the following defrosting period. Here, q_A denotes the maximum heating capacity during the stable heating stage after the previous defrosting process, corresponding to point A in Fig. 8. Therefore, ε_{NL} represents the total heating energy loss caused by the complete frosting-defrosting process. The results are shown in Fig. 9. The ε_{NL} values under the FCU heating mode were 12.5% and 12.4% in the two frosting-defrosting cycles, respectively, whereas those under the RF heating mode were 19.5% and 19.2%, respectively. Compared with the FCU heating mode, the average ε_{NL} under the RF heating mode increased by 55.4%.

$$\varepsilon_{NL} = \frac{\int_0^t (q_A - q) dt}{\int_0^t q_A dt} \times 100\% \quad (4)$$

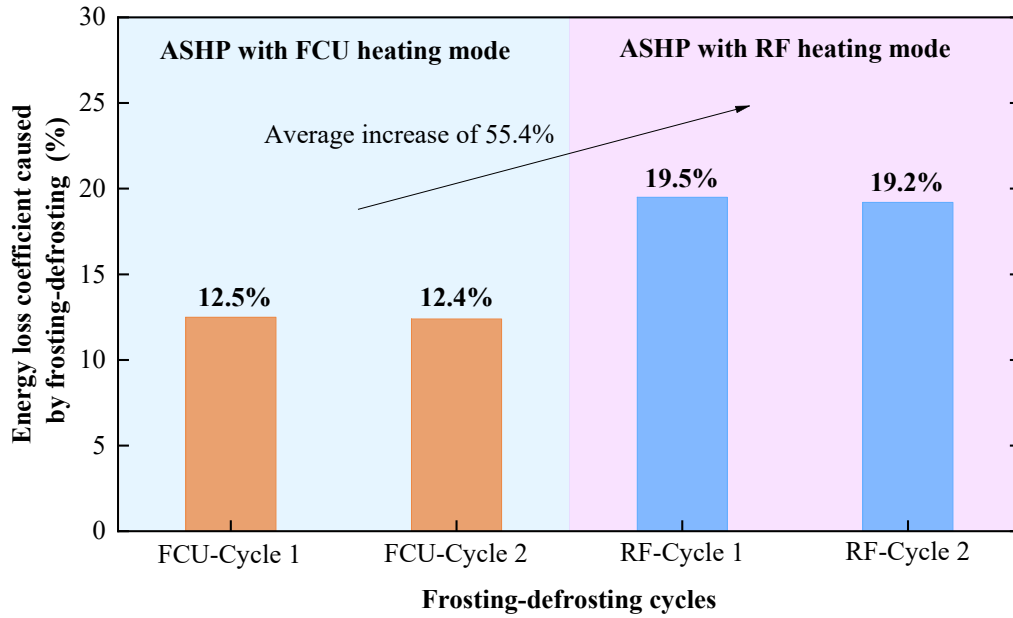


Fig. 9. Energy loss coefficient caused by frosting-defrosting under the FCU and RF heating modes.

Fig. 10 illustrates the changes in W under different heating terminal configurations during frosting-defrosting cycles. During the stable heating periods of the two cycles, W values under the FCU heating mode were maintained at approximately 4.42 kW and 4.47 kW, respectively, whereas the corresponding

values under the RF heating mode were 4.15 kW and 4.20 kW, respectively. As frosting progressed, W decreased slightly during the later stage of each heating operation. When the defrosting initiation criteria were satisfied, the ASHP switched to defrosting mode, accompanied by a rapid decrease in W . After defrosting ended, the four-way valve reversed again, and the system returned to heating operation, during which W gradually recovered to a stable level. Over the two complete frosting-defrosting cycles, the average W was 4.24 kW under the FCU heating mode and 3.78 kW under the RF heating mode. Overall, although W exhibited similar variation trends under the two heating modes, both the stable W values and the average W values over the complete frosting-defrosting cycles were slightly lower under the RF heating mode than under the FCU heating mode.

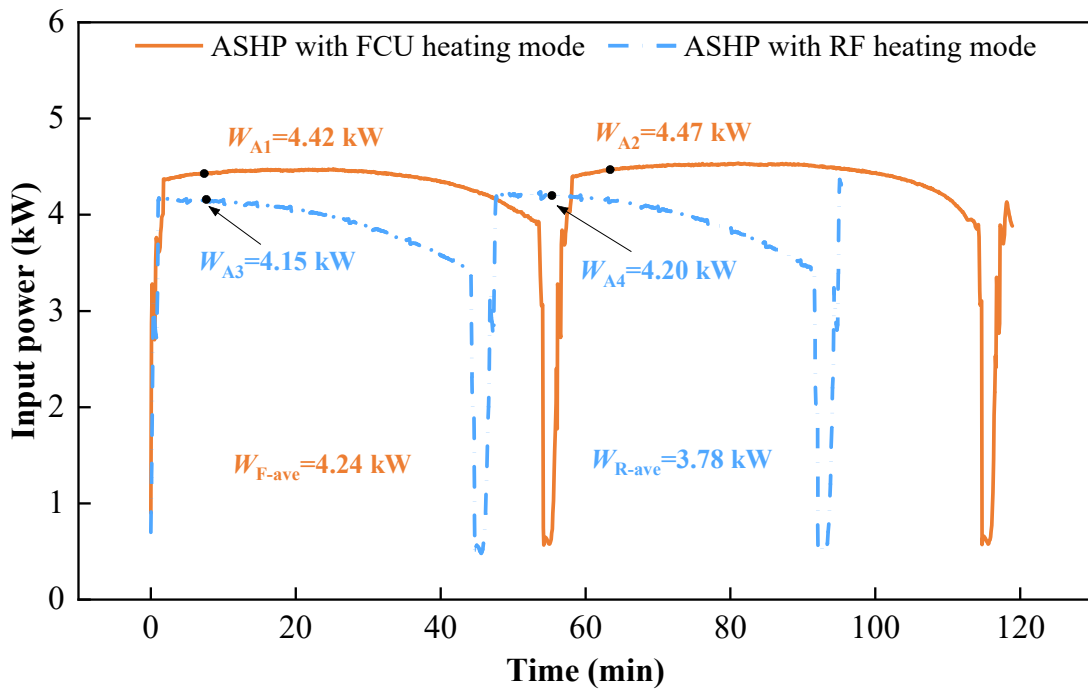


Fig. 10. Variations in input power of the ASHP system under the FCU and RF heating modes during frosting-defrosting cycles.

The variation in COP under the FCU and RF heating modes during different frosting-defrosting cycles was calculated using Eq. (5) [44, 45], and the corresponding COP variation is shown in Fig. 11. During the frosting-defrosting cycle, the average COP was calculated as the ratio of the total heating energy to the total electrical energy input, as shown in Eq. (6) [44, 45]. Based on Figs. 8 and 10, the RF heating mode exhibited higher q and lower W than the FCU heating mode during both the stable heating periods and the complete frosting-defrosting cycles. Therefore, the COP under the RF heating mode was higher than that under the FCU heating mode during both the stable heating periods and the complete frosting-defrosting cycles. Specifically, during the stable heating periods, the COP values under the FCU heating mode were 2.64 and

2.59 in the two frosting-defrosting cycles, respectively, whereas the corresponding values under the RF heating mode were higher, reaching 3.29 and 3.24, respectively. In addition, over the two complete frosting-defrosting cycles, the average COP under the RF heating mode was 2.91, which was approximately 21.3% higher than the value of 2.40 obtained under the FCU heating mode. This indicates that the RF heating mode achieved better overall heating performance under frosting-defrosting operating conditions.

$$COP = \frac{q}{W} \quad (5)$$

$$COP_{ave} = \frac{\int_0^t q dt}{\int_0^t W dt} \quad (6)$$

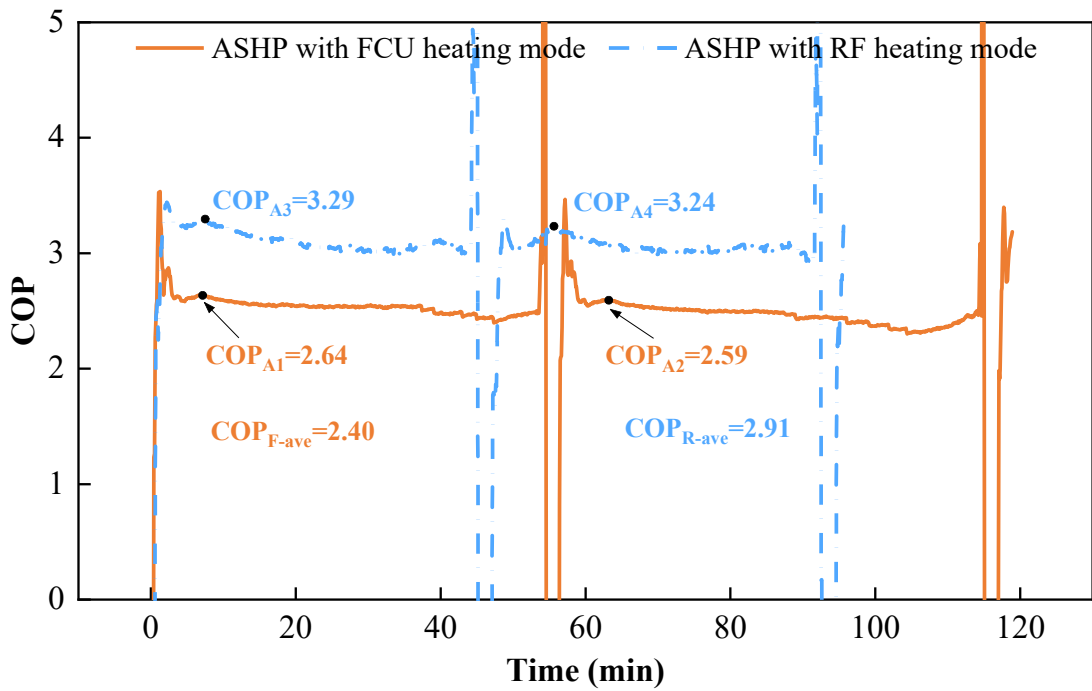


Fig. 11. Variation in COP of the ASHP system under the FCU and RF heating modes during frosting-defrosting cycles.

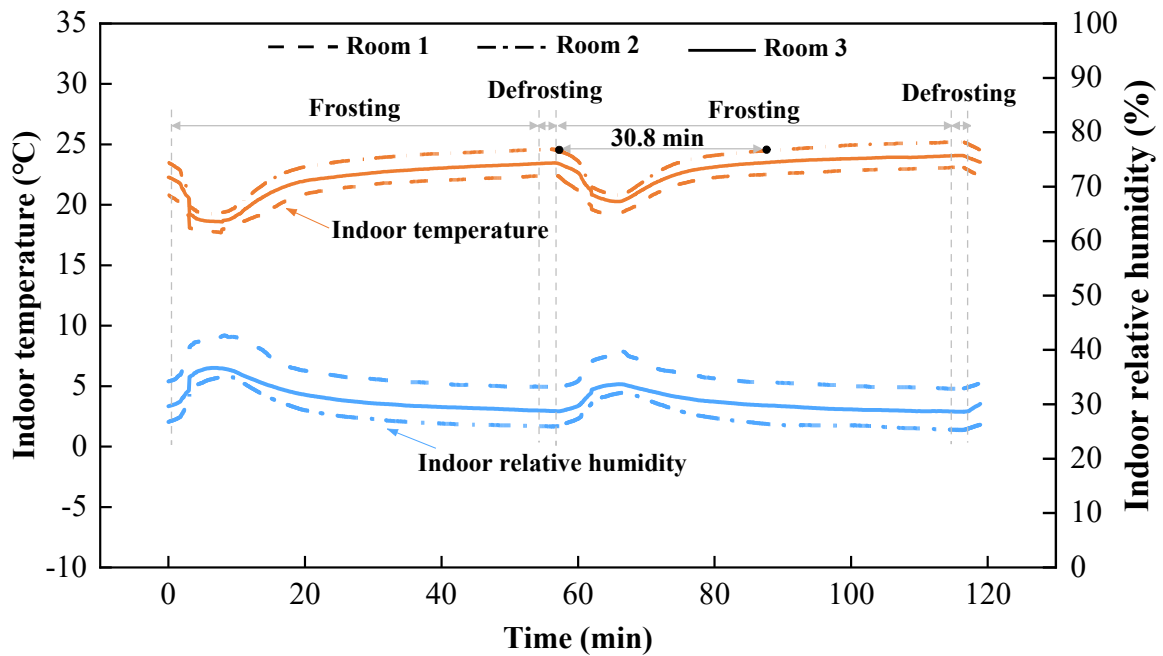
Overall, compared with the FCU heating mode, the RF heating mode operated with lower T_s and T_r values. The RF heating mode showed weaker frosting suppression performance, resulting in an 8.9% increase in v_{dfw} and a higher ε_{NL} . Nevertheless, over the two complete frosting-defrosting cycles, the average q under the RF heating mode was still 7.9% higher than that under the FCU heating mode. In contrast, the FCU heating mode required higher T_s and T_r values, which increased W by 12.2%. Consequently, the average COP under the FCU heating mode was lower than that under the RF heating mode during frosting-defrosting operation.

3.3 Variations in the indoor thermal environment

Fig. 12 demonstrates the dynamic changes in T_i and RH_i in Rooms 1–3 under the FCU heating mode

1 during the experimental process. As described in Section 2.3, the T_i and RH_i values presented in the figure
 2 were obtained by averaging all corresponding measurement points in Rooms 1–3. These values were used
 3 to characterize the overall indoor thermal and moisture conditions. The results indicate that although T_i
 4 values in the three rooms generally exhibited a slight upward trend, they fluctuated within a narrow range
 5 around the target values specified in Table 3.

6 During defrosting, the refrigerant extracted heat from the circulating water in the plate heat exchanger.
 7 As presented in Fig. 4(a), the maximum decreases in T_s during defrosting were 20.32 °C and 20.37 °C in
 8 the two frosting-defrosting cycles, respectively, and T_s fell markedly below T_r . Consequently, the heat
 9 supply to the indoor rooms was interrupted, and T_i subsequently decreased. It is worth noting that, owing
 10 to the thermal storage capacity of the circulating water and indoor heating terminals, T_i did not decrease
 11 immediately after defrosting began but exhibited a delayed response. In the two frosting-defrosting cycles,
 12 the maximum T_i drops caused by defrosting were 3.09 °C and 3.17 °C in Room 1, 4.25 °C and 3.79 °C in
 13 Room 2, and 3.68 °C and 3.19 °C in Room 3, respectively. Accordingly, the average maximum T_i drops in
 14 Rooms 1–3 were 3.13 °C, 4.02 °C, and 3.44 °C, respectively. This finding suggests that, under the FCU
 15 heating mode, the defrosting process caused noticeable disturbances to the indoor temperature stability of
 16 Rooms 1–3.



17
 18 **Fig. 12.** Variations in indoor temperature and relative humidity in Rooms 1–3 under the FCU heating mode during the
 19 experimental period.

20 After defrosting ended, the ASHP resumed heating operation, and T_i gradually increased. However,
 21 although the actual defrosting duration was relatively short, the thermal disturbance in the circulating water

caused by defrosting lasted 11.6–13.7 min, as shown in Fig. 4(a). Consequently, a longer recovery period was required for T_i to return to its pre-defrosting level. Taking the second frosting-defrosting cycle in Room 2 as an example, the T_i recovery process lasted 30.8 min, which was far longer than the actual defrosting duration. This indicates that, under the FCU heating mode, the impact of defrosting was not limited to the defrosting stage itself but extended to the subsequent water temperature recovery period.

RH_i in Rooms 1–3 varied inversely with T_i . During defrosting, T_i decreased, leading to an obvious increase in RH_i in all three rooms. In the two frosting-defrosting cycles, the average maximum increases in RH_i in Rooms 1–3 were 7.51%, 7.21%, and 6.03%, respectively. After defrosting ended, as the ASHP resumed heating operation, T_i gradually increased, while RH_i decreased and reached a relatively low level before the next defrosting process. These results indicate that, under the FCU heating mode, the defrosting process caused obvious fluctuations in T_i and RH_i , thereby reducing the stability of the indoor thermal environment.

Fig. 13 displays the dynamic changes in T_i and RH_i in Rooms 1–3 under the RF heating mode during the experimental process. Similar to Fig. 12, the T_i and RH_i values presented in the figure were obtained by averaging all corresponding measurement points in Rooms 1–3, as described in Section 2.3. The results indicate that T_i values in the three rooms exhibited a slight upward trend but remained close to the target values specified in Table 3.

During defrosting, the refrigerant also extracted heat from the circulating water in the plate heat exchanger. However, owing to the strong thermal storage capacity of the radiant floor, the heating interruption had a limited effect on T_i . As a result, both T_i and RH_i remained nearly unchanged during defrosting. This indicates that, under the RF heating mode, the defrosting process did not have a noticeable effect on the indoor environment.

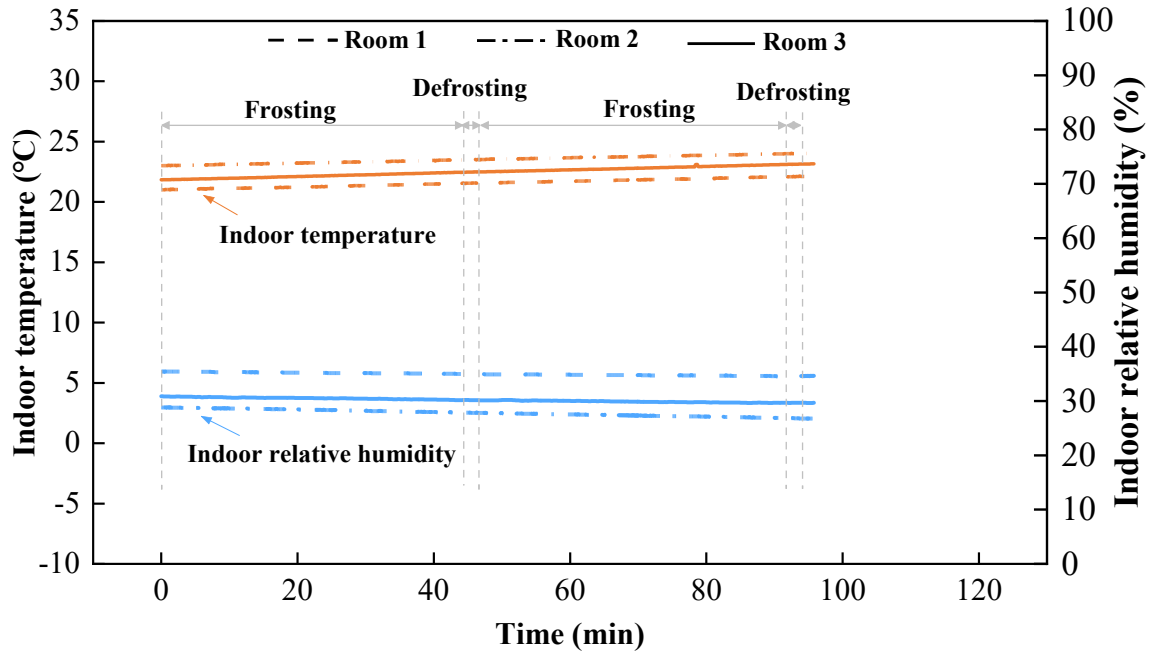


Fig. 13. Variations in indoor temperature and relative humidity in Rooms 1–3 under the RF heating mode during the experimental period.

As presented in Figs. 12 and 13, the indoor thermal environment exhibited different responses under the FCU and RF heating modes during frosting-defrosting operation. Under the FCU heating mode, the limited heat storage capacity of the FCUs led to noticeable disturbances in indoor thermal environment stability during defrosting. In contrast, under the RF heating mode, the strong thermal storage capacity of the RFs effectively buffered the heating interruption caused by defrosting, thereby maintaining a more stable indoor thermal environment.

4. Conclusions

To reveal the effects of different heating terminal configurations on the heating performance and indoor thermal environment of the ASHP under frosting-defrosting conditions, an experimental ASHP platform was established in an artificial environment chamber, and comparative experiments were performed. According to the experimental findings, the frosting-defrosting characteristics, heating performance, and indoor thermal environment responses under the two heating terminal configurations were systematically analyzed. The main findings are as follows:

(1) Defrosting caused obvious decreases in T_s under both FCU and RF heating modes, with larger fluctuations observed under the FCU heating mode. The maximum decreases in T_s caused by defrosting were 20.32 °C and 20.37 °C in the two experimental cycles under the FCU heating mode, compared with 12.54 °C and 13.22 °C under the RF heating mode. Compared with the results under the RF heating mode,

the T_s fluctuation amplitudes were 62.0% and 54.1% greater.

(2) Compared to that under the RF heating mode, the frosting suppression performance of ASHP under the FCU heating mode was better. Under the 2/1 °C outdoor condition, the frosting durations were 44.2 min and 44.6 min under the RF heating mode, compared with 53.8 min and 57.9 min under the FCU heating mode, respectively. Compared with the RF heating mode, the FCU heating mode prolonged the frosting durations by 21.7% and 29.8% respectively, indicating that the FCU heating mode exhibited better frosting suppression performance.

(3) Under frosting-defrosting conditions, the ASHP achieved higher average q and COP values under the RF heating mode than under the FCU heating mode. During the two complete frosting-defrosting cycles, the average q was 10.19 kW under the FCU heating mode and 10.99 kW under the RF heating mode, representing an increase of 7.9%. Meanwhile, the average COP under the RF heating mode was 2.91, which was approximately 21.3% higher than the value of 2.40 obtained under the FCU heating mode.

(4) The indoor thermal environment under the FCU and RF heating modes showed obviously different responses during the defrosting process. Owing to the lower heat storage capacity of the FCUs, the average maximum T_i drops under the FCU heating mode in Rooms 1–3 ranged from 3.13 °C to 4.02 °C, and the average maximum increases in RH_i ranged from 6.03% to 7.51%. In contrast, both T_i and RH_i remained nearly unchanged under the RF heating mode.

Under frosting-defrosting conditions, this study compared the frosting-defrosting characteristics, heating performance, and indoor thermal environment of an ASHP system under the FCU and RF heating modes. The findings can guide terminal-specific defrosting control methods and help mitigate indoor thermal environment fluctuations associated with different heating terminals. However, this study has several limitations: (1) the experiments were conducted under a representative outdoor condition of 2/1 °C, with fixed compressor and outdoor fan speeds; (2) only two consecutive frosting-defrosting cycles were considered. Future work should examine a wider range of environmental conditions, variable-speed ASHP operation, longer-term frosting-defrosting tests, and additional heating terminal configurations.

Acknowledgements

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