

This is an Open Access document downloaded from ORCA, Cardiff University's institutional repository: <https://orca.cardiff.ac.uk/id/eprint/183118/>

This is the author's version of a work that was submitted to / accepted for publication.

Citation for final published version:

Wei, Wenzhe, Zhang, Jianfei, Jin, Xinyue, Zhang, Xinyu, Wang, Wei, Luo, Zhiwen, Sun, Yuying, Li, Zhaoyang and Wang, Shiquan 2026. Experimental investigation of indoor thermal environment dynamics during the frosting-defrosting process of air source heat pumps. *Energy and Buildings* 353, 116849. 10.1016/j.enbuild.2025.116849

Publishers page: <http://dx.doi.org/10.1016/j.enbuild.2025.116849>

Please note:

Changes made as a result of publishing processes such as copy-editing, formatting and page numbers may not be reflected in this version. For the definitive version of this publication, please refer to the published source. You are advised to consult the publisher's version if you wish to cite this paper.

This version is being made available in accordance with publisher policies. See <http://orca.cf.ac.uk/policies.html> for usage policies. Copyright and moral rights for publications made available in ORCA are retained by the copyright holders.



1 Experimental investigation of indoor thermal environment dynamics
2 during the frosting-defrosting process of air source heat pumps

3 Wenzhe Wei ^a, Jianfei Zhang ^a, Xinyue Jin ^a, Xinyu Zhang ^{b,*}, Wei Wang ^{a,c,*}, Zhiwen Luo ^d,
4 Yuying Sun ^a, Zhaoyang Li ^c, Shiquan Wang ^a

5 ^aBeijing Key Laboratory of Green Built Environment and Energy Efficient Technology, Beijing
6 University of Technology, Beijing, China

7 ^bState Key Laboratory of Building Safety and Built Environment, Beijing 100013, China

8 ^cSchool of Mechatronic Engineering, Beijing Polytechnic University, Beijing, China

9 ^dWelsh School of Architecture, Cardiff University, Cardiff, United Kingdom

10 *Corresponding author, E-mail address: mrwangwei@bjut.edu.cn (W. Wang), zxyhit@163.com (X.
11 Zhang)

Abstract

During the defrosting operation of air source heat pumps (ASHPs), heating supply is interrupted. This leads to significant fluctuations in the indoor thermal environment, resulting in thermal discomfort for occupants. However, the dynamic behavior of indoor thermal conditions during the frosting-defrosting cycles remains unclear, which poses a major challenge to improving the heating performance of ASHPs. To solve this problem, two test rigs, one air-water heat pump (AWHP) heating system and one air-air heat pump (AAHP) heating system, were built in an artificial environmental chamber. Space heating experiments were conducted under controlled frosting conditions to investigate the indoor thermal environment during the frosting-defrosting cycles. By doing this work, the indoor thermal environment variation patterns during ASHP frosting-defrosting cycles were revealed for the first time. Results showed that the indoor temperature decreases notably during defrosting for both AWHPs and AAHPs, with greater declines observed for AWHPs. Specifically, the temperature drop ranged from 2.99–4.66 °C for AWHPs and from 1.58–1.79 °C for AAHPs. Furthermore, it was found that for both systems, the lowest indoor temperature does not occur at the end of the defrosting operation, as previously assumed, but rather 4.25–6.00 minutes later for AWHPs and 3.75–4.25 minutes later for AAHPs.

Keywords: air source heat pump, frosting, defrosting-induced indoor temperature fluctuation, heating interruption

Nomenclatures

c_a	specific heat capacity of air (J/(kg·K))
c_w	specific heat capacity of water (J/(kg·K))
m	water mass flow rate (kg/h)
P_D	percentage dissatisfied (%)
q_a	heating capacity for the air- air heat pump (kW)
q_w	heating capacity for the air-water heat pump (kW)
T_a	ambient air temperature (°C)
T_{ar}	return air temperature (°C)
T_{as}	supply air temperature (°C)
T_c	outdoor coil surface temperature (°C)
T_r	return water temperature (°C)
T_s	supply water temperature (°C)
t	time
v	air velocity (kg/m)
ρ	air density (kg/m ³)
ΔT	temperature difference (°C)
ΔT_{ah}	vertical air temperature difference between the head and ankles (°C)
μ	population mean
σ	standard deviation

Abbreviation

AAHP	air-air heat pump
ASHP	air source heat pump
AWHP	air-water heat pump
COP	coefficient of performance
PMV	predicted mean vote

1. Introduction

To reduce building heating energy consumption, the Chinese government has actively promoted the application of energy-saving technologies, such as heat pumps and solar thermal technology, in building heating in recent years [1, 2]. Among various heat pump types, the air source heat pump (ASHP), which extracts heat from ambient air as a low-temperature heating source, has gained significant popularity [3]. ASHPs are widely used due to their advantages, including abundant heat source availability, ease of installation, and low initial investment costs [4-6]. However, when working under low temperature and high humidity environmental conditions, ASHPs always encounter the frosting problem [7-9]. Frosting will not only reduce the air flowrate through the outdoor coil, but also increase the thermal resistance between the ambient air and refrigerant, leading to a significant degradation in its heat transfer capacity [10, 11].

To maintain the high heating performance of ASHPs, periodic defrosting is employed, with reverse-cycle defrosting being the most commonly used method [12, 13]. Reverse-cycle defrosting is achieved by switching the four-way valve, causing high-temperature refrigerant from the compressor to flow into the outdoor coil and melt the frost. During this process, the outdoor coil acts as a condenser while the indoor unit functions as an evaporator. This results in the temporary cessation of indoor heating, or in some cases, heat absorption from indoors, leading to significant indoor temperature drops. Consequently, defrosting can severely disrupt the indoor thermal environment [14], posing challenges to maintaining thermal comfort, particularly in rural buildings with low thermal inertia.

To address these challenges, researchers have focused on two primary strategies: 1) improving defrosting accuracy rate [15-17], and 2) reducing the defrosting duration [18, 19]. Enhancing defrosting accuracy minimizes heating capacity loss during the frosting-defrosting cycle, thereby reducing indoor temperature fluctuations. Wang et al. [20] proposed the optimal defrosting initiation time theory,

1 demonstrating that initiating defrosting at the optimal time minimizes heating losses [21]. Building on this,
2 Li et al. [22] and Wang et al. [23] developed image recognition-based defrosting methods [24]. Tang et al.
3 [25] introduced a control strategy incorporating variations in outdoor fan current, temperature, and coil
4 conditions. These advancements have significantly improved defrosting accuracy [26-28].

5 Reducing the defrosting duration is another effective approach to mitigate indoor temperature
6 fluctuations. Shortening the defrosting period reduces the interruption to space heating [29]. Dong et al.
7 [30] and Zhang et al. [31] used phase-change materials to store heat during frosting, which was later used
8 to accelerate defrosting, reducing its duration by 59.41% and 65.00%, respectively. Wei et al. [32]
9 employed vapor injection technology in quasi-two-stage ASHPs, cutting defrosting time by 20.61% without
10 additional cost.

11 While these studies have advanced defrosting techniques, most experiments were conducted in
12 psychrometric chambers, focusing solely on defrosting performance without considering the impact on
13 indoor thermal environments. Qu et al. [33] assessed indoor thermal comfort during defrosting using the
14 Predicted Mean Vote (PMV) model in a practical space heating project. However, the PMV model was
15 built based on the steady-state thermal environment [34, 35]. It is not suitable for the fluctuating indoor
16 environment during the frosting-defrosting cycles. On the other hand, in the practical space heating process,
17 the outdoor temperature varied all the time. The variation of outdoor temperature affected the heating
18 capacity of ASHP significantly, and further affected the variation of indoor thermal environment. Therefore,
19 the variation rules of indoor thermal environment during frosting-defrosting cycles of ASHP remain unclear,
20 significantly limiting the development of effective defrosting control strategies.

21 To fill the research gap above, this study investigates the indoor thermal environment of ASHPs during
22 dynamic frosting-defrosting cycles using an artificial environmental chamber. A typical rural residential

1 building, representative of those in the Yangtze River Basin, was constructed within the chamber. Two test
2 rigs, an air-water heat pump (AWHP) heating system and an air-air heat pump (AAHP) heating system,
3 were installed to supply space heating. Experiments were conducted under standard frosting conditions
4 ($2/1\text{ }^{\circ}\text{C}$ dry-bulb/wet-bulb temperature) with varying indoor temperature setpoints. The variations in indoor
5 thermal environments for both ASHP types were analyzed. The key contributions of this study are: (1) the
6 first exploration of indoor thermal environment variation patterns during ASHP frosting-defrosting cycles,
7 and (2) the precise control of frosting conditions and indoor environments using a constructed building in
8 an artificial environmental chamber. This work can provide guidance to solve the defrosting-induced indoor
9 temperature fluctuation problem.

10 **2. Experimentations**

11 **2.1 The artificial environment chamber**

12 To investigate the variation patterns of the indoor thermal environment during the dynamic frosting-
13 defrosting cycles, space heating experiments with ASHPs were conducted in an artificial environmental
14 chamber in Zhuhai, as shown in Fig. 1. The chamber consists of two sections: an external room and an
15 internal room. In the external room, there were heaters, humidifiers, and cold coils to create outdoor
16 environmental conditions [36]. Through creating stable environment in it, the effects of outdoor
17 environment on heating performance of ASHP and the building heating load were eliminated, and the
18 experiments on indoor environment variations during frosting-defrosting can be conducted.

19 The internal room, a two-story building, was used to study the indoor thermal environment during
20 frosting-defrosting cycles. It was constructed following building envelope standards for the Hot Summer
21 and Cold Winter climatic region of China. The materials and sizes of the building envelope, including walls,

1 windows and floor, are widely employed in the Yangtze River Basin. Consequently, the thermal inertia of
2 the internal room is the same as that of real buildings. The floor plans of the first and second stories are
3 shown in Figs. 2(a) and 2(b), respectively. An AWHP and an AAHP were installed in this artificial
4 environment chamber, and used to supply heat for internal rooms.

5 There were three rooms on the second story, with areas of 62.08 m², 20.99 m², and 38.92 m², all of
6 which were heated by the AWHP through adding three fan-coils and used for analyzing indoor thermal
7 conditions. On the first story, for AAHPs, apart from the variable refrigerant flow (VRF) heat pump system,
8 one AAHP can supply heat for only one room because it has a single indoor heat exchanger. Meanwhile,
9 for VRF heat pumps, their compressor speed regulations are not allowed by manufacturers, and cannot be
10 used in this study. Consequently, a common AAHP with one indoor heat exchanger was chosen in this
11 work, and used to supply heat for Room 4, with an area of 15.57 m². Thus, Rooms 1–3 on the second story
12 and Room 4 on the first story were selected for investigating the indoor thermal environment during the
13 dynamic frosting-defrosting cycles of AWHP and AAHP systems.

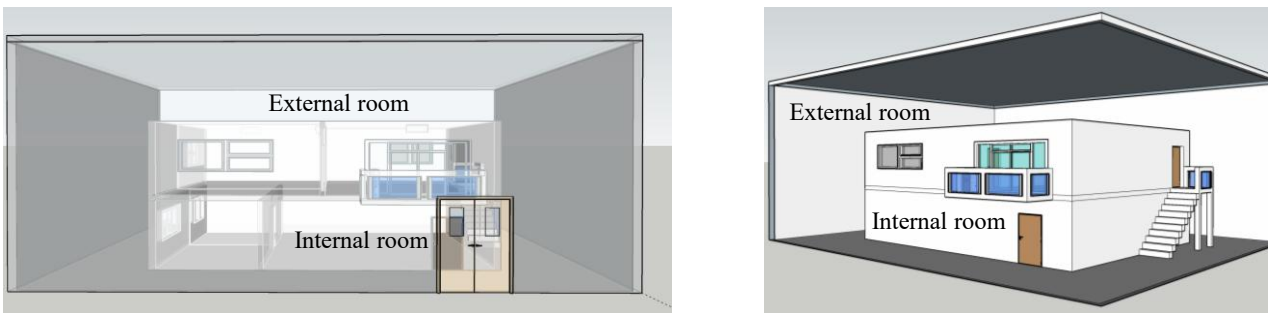


Fig. 1 Three-dimensional diagram of artificial environmental chamber

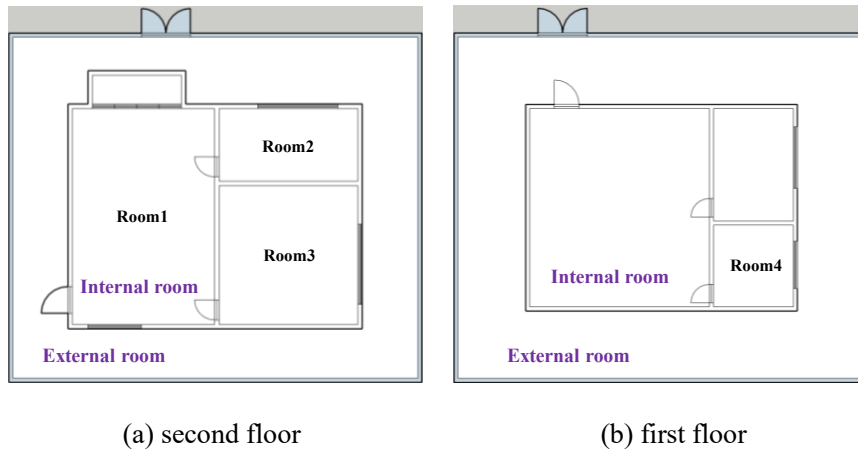


Fig. 2 Floor plan of the artificial environment chamber

2.2 The experimental air source heat pumps

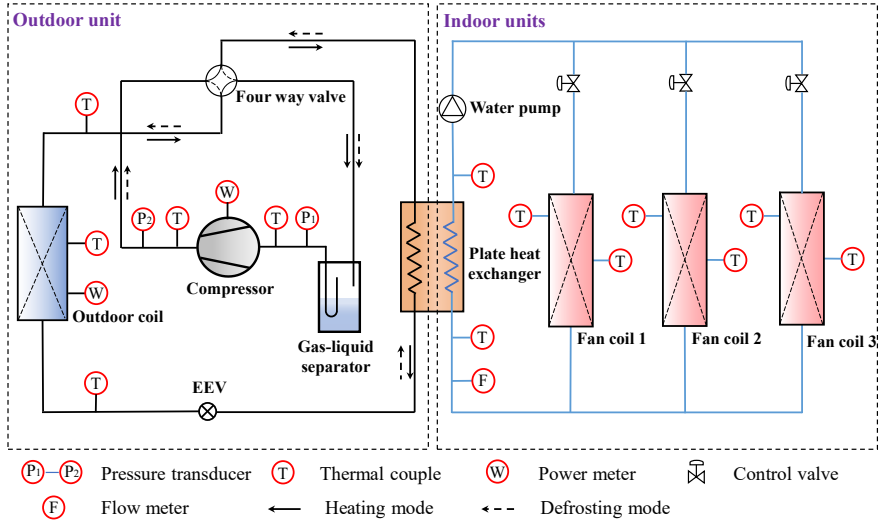
At present, ASHPs are mainly classified into two types: AWHP and AAHP. During the defrosting process, significant differences exist in their operation, which leads to distinct impacts on the indoor thermal environment. In AWHP systems, the circulation water temperature is high at the start of defrosting, storing a substantial amount of heat. Meanwhile, there is no communication line between the outdoor unit and indoor units for AWHPs. During defrosting, the indoor fan-coils remain operational and continue supplying heat to the indoor environment. In contrast, the indoor unit of the AAHP is turned off during defrosting, halting heat supply to indoors. In some cases, the AAHP may even absorb heat from indoors via natural-convection heat transfer. As a result, the impact of the frosting-defrosting cycles on the indoor thermal environment varies significantly between AWHPs and AAHPs.

To comprehensively investigate these differences and reveal the variation patterns of the indoor thermal environment during the dynamic frosting-defrosting cycles, an AWHP and an AAHP were selected for experiments. Their schematic diagrams are shown in Figs. 3(a) and 3(b), respectively.

The AWHP heating system consists of one outdoor unit and three indoor fan-coils, which are installed in three rooms on the second floor of the artificial environmental chamber to provide heating. The AWHP

has a rated heating capacity of 18 kW and a coefficient of performance (COP) of 3.1. It uses R410A as the refrigerant. To make the water capacity in AWHP heating system similar to those in practical projects, a small water tank was adopted and connected to the circulation water system, and the total water capacity was 65 L.

The AAHP heating system comprises one outdoor unit and one indoor unit, with the indoor unit installed in Room 4 on the first floor of the artificial environmental chamber. This system provides heating with a rated heating capacity of 3.5 kW and a COP of 3.3. As the defrosting performances of ASHPs with different refrigerants are nearly the same [37, 38], the refrigerant difference between the AWHP and AAHP will not affect the results of this work. During the frosting process, the three fan-coils of AWHP were regulated based on the indoor temperature independently, which was the same as the control method of fan-coil in AAHP. Meanwhile, during the defrosting process, the work states of the three indoor fan-coils were same, and the indoor thermal environment would not be affected by the number of fan-coils. Therefore, the number of indoor fan-coils would not lead to the control asymmetry problem. Detailed specifications of the outdoor units for both the AWHP and AAHP are provided in Table 1.



(a) air-water heat pump

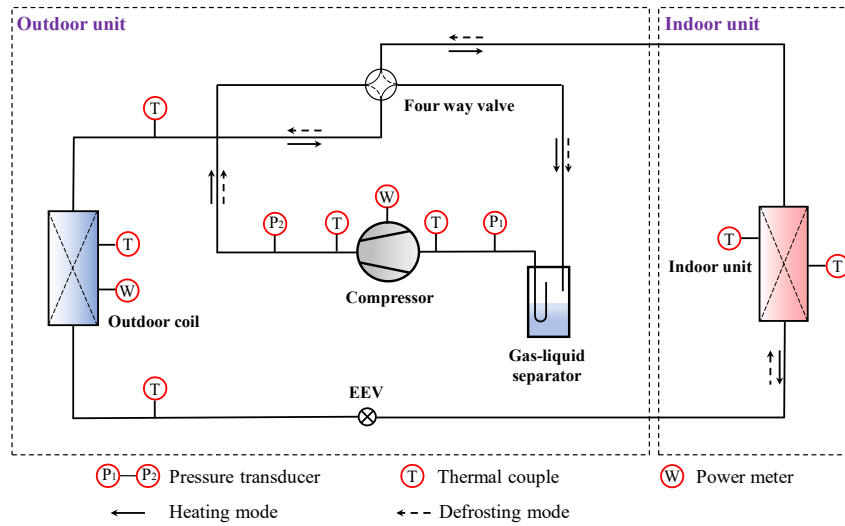


Fig. 3 Schematic diagrams of the experimental air source heat pumps

Table.1 Specifications of the outdoor units.

Parameters	Values/Details	
	air-water heat pump	air-air heat pump
Refrigerant	R410A	R32
Rated heating capacity (kW)	18	3.5
Rated COP	3.1	3.3
Rated condition	7/6 °C	7/6 °C
Compressor type	scroll	rotor
Compressor speed (r/s)	40-100	20-100
Type of outdoor coil	finned tube	finned tube
Tube type	inner grooved tube	inner grooved tube
Tube material	copper	copper
Fin material	aluminum	aluminum

For both the AWHP and AAHP, the reverse-cycle defrosting is adopted. When the defrosting is required, the four-way valve switches, and the high temperature refrigerant generated by compressor will flow into the outdoor heat exchanger to melt the frost layer. The most widely used temperature difference-time ($\Delta T-t$) defrosting control method is adopted by both ASHPs. The ΔT denotes the temperature difference between ambient air and the outdoor heat exchanger surface. During the frosting process, ΔT increases gradually as frost layer grows. Therefore, it can be used as an indicator to reflect frosting level. For the two experimental ASHPs, when ΔT exceeds 7 °C and the operation time exceeds 40 min, the

1 compressor speed decreases rapidly to 0 and prepare to defrosting through switching the four-way valve.

2 During the defrosting process, when the tube surface temperature of the lowest refrigerant circuit of outdoor

3 heat exchanger (T_c) reaches 20 °C, the defrosting operation of both ASHPs terminates. Then, the four-way

4 valve switches again, and the ASHP resumes to heating mode. To reduce the pressure difference between

5 the two sides of four-way valve during its switching, the AWHP is turned off for 2 min before each

6 switching. Meanwhile, due to its lower pressure difference between the two sides of four-way valve, AAHP

7 only shuts down for 1 min before each switching. The schematic diagram of the compressor speed variation

8 during the defrosting process is shown in Fig. 4.

9 During the defrosting process, the indoor fan-coils of AWHP are turned on and continue to supply

10 heating for indoors, while the indoor unit of AAHP is shut down to avoid absorbing heat from indoors. As

11 the temperature of indoor fan-coil of AAHP is very low after defrosting, its fan is not turned on until its

12 surface temperature reaches 35 °C to avoid blowing cold wind.

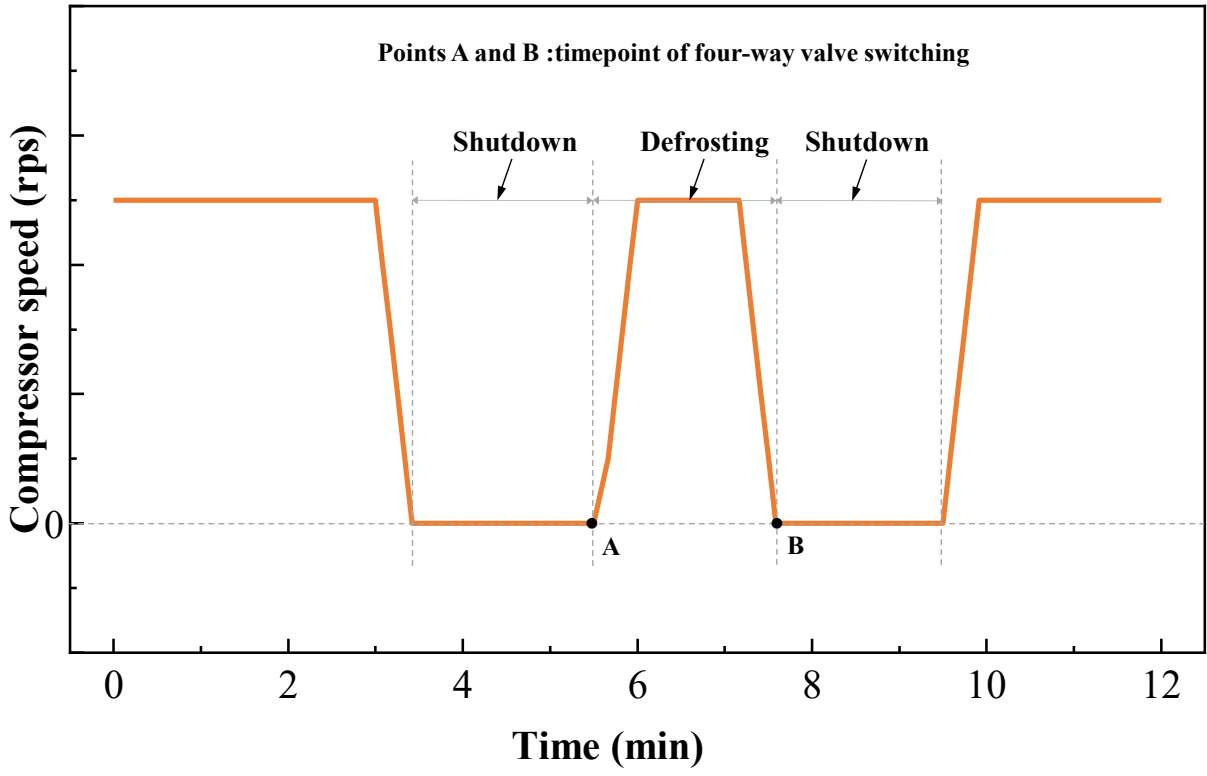


Fig. 4 Schematic diagram of compressor speed during the defrosting process

2.3 Data acquisition system

To obtain the experimental data, plenty of measuring instruments were installed to monitor the operating parameters of the two ASHPs, and the indoor and outdoor environmental parameters. Specific measurement parameters include:

1) Outdoor environment side

One temperature & humidity sensor was installed near the AWHP to monitor ambient temperature and relative humidity.

2) Outdoor unit side

One smart power meter was installed to measure the power input. Two pressure transducers were installed to monitor the suction pressure and discharge pressure. Five thermocouples were installed to measure the refrigerant temperature at different locations as displayed in Fig. 3.

3) Circulating water side of AWHP

Two thermocouples were installed on supply and return water pipes to monitor water temperature. One electromagnetic flow meter was installed on the supply water pipe to measure the water flowrate.

4) Indoor environment side

To systematically investigate the variations of indoor thermal environment during the dynamic frosting-defrosting cycles, plenty of thermocouples and relative humidity sensors were installed in indoors. Their installation methods are as follows:

(1) The plane distributions of thermocouples in rooms supplied heating by AWHP and AAHP are shown in Figs. 5 (a) and 6 (a). At each measurement point, a vertical line was set up, and five thermocouples were installed at the heights of 0.3 m, 0.8 m, 1.3 m, 1.8 m, and 2.3 m. There are 270 (9×6×5), 150 (6×5×5), and 90 (6×3×5) thermocouples installed in Room 1, Room 2, and Room 3, respectively. Meanwhile, there

are 210 (7×6×5) thermocouples installed in Room 4. In addition, three thermocouples are installed at the inlet and outlet of the fan coil unit of the AAHP respectively to measure the inlet and outlet air temperature, as shown in Fig. 3.

(2) The plane distributions of relative humidity sensors in rooms supplied heating by AHP and AAHP are shown in Figs. 5 (b) and 6 (b). According to the *Standard of Test Methods for Thermal Environment of Buildings* (JGJ/T 347-2014), there were respectively five, two, and three relative humidity sensors installed at the midpoints, trisection, or quadrisection points of diagonal for Room 1, Room 2, and Room 3. Meanwhile, there is one relative humidity sensor installed at the midpoint of diagonal for Room 4. At present, since the ASHPs don't have the function of adjusting indoor humidity, the humidity variations during frosting-defrosting cycles cannot be solved without additional humidification device. Consequently, the humidity variation was not main concern of this work, and all the relative humidity sensors were installed at the height of 0.6 m without considering its vertical variation.

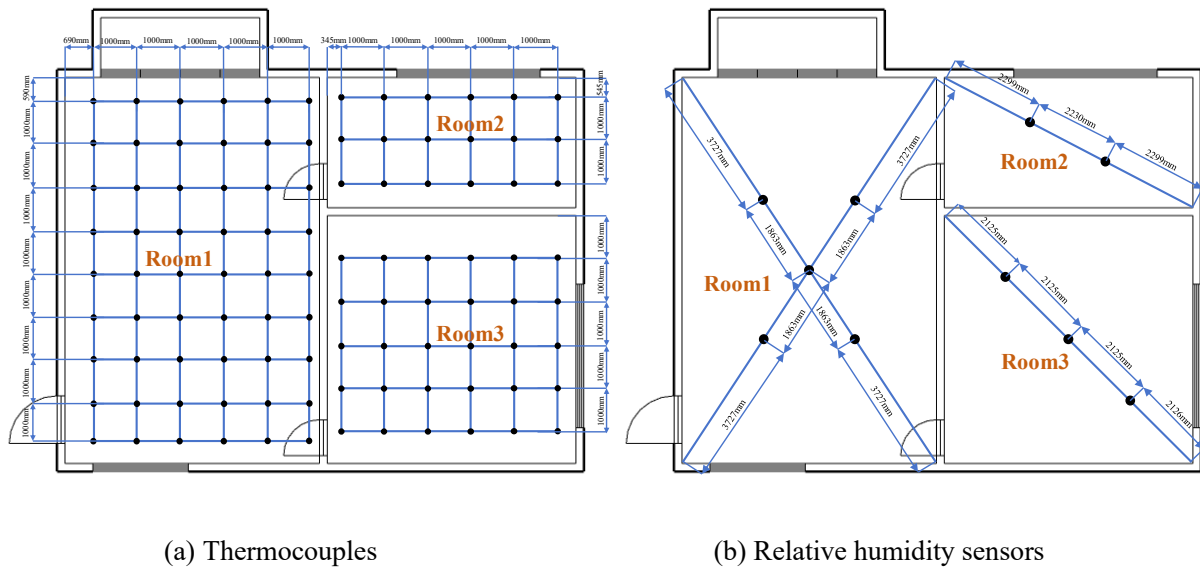


Fig. 5 Floor plan of thermocouples and relative humidity sensors layout in the rooms used AHP

In addition to the aforementioned measuring instruments, a black bulb temperature and an hot-wire anemometer were installed in each room to monitor the indoor black bulb temperature and air velocity. The

measuring instruments mentioned above were calibrated every half year, and some of them were randomly selected for calibration to ensure their accuracy.

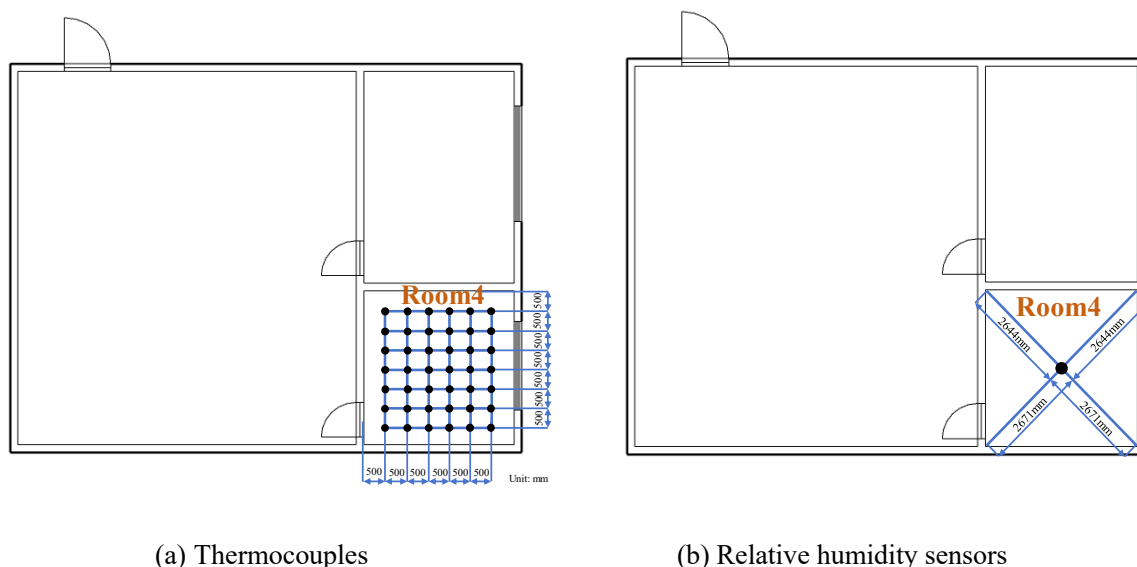


Fig. 6 Floor plan of thermocouples and relative humidity sensors layout in the rooms used AAHP

Table. 2 Detailed specifications of measuring instruments

Name	Measuring range	Error	Number
Thermocouple	-20–100 °C	± 0.1 °C	738
Temperature & humidity sensor	Temp: 0–50 °C RH: 0–100 %	± 0.15 °C ± 3 %	33
Electromagnetic flow meter	0.1–3.0 m ³ /h	0.5% FS	1
Power meter	0–6.6 kW	0.25% FS	4
Black bulb temperature	0–80 °C	± 0.1 °C	13
Hot-wire anemometer	0–30 m/s	± 0.5 %	31
Pressure transducer	0–3.0 MPa	0.3% FS	4

2.4 Experimental conditions and procedures

In the United States, European Union, China, etc., the frosting performance tests of ASHPs are typically conducted under the standardized condition of 2/1 °C (dry-bulb/wet-bulb temperature). Under this condition, the frosting of ASHP initiates naturally with a fast speed, and can represent typical frosting characteristics. To ensure the universality and comparability of experimental results, the outdoor environmental parameters in this study (i.e., the conditions in the external room of the artificial

1 environmental chamber) were also set at 2/1 °C.

2 In the Hot Summer and Cold Winter climatic region, the indoor temperature for ASHP heating systems
3 during space heating is typically maintained between 22 °C and 24 °C. Accordingly, in this study, the
4 indoor temperature for Rooms 1–3 (heated by the AWHP) were set at 22 °C, 23 °C, and 24 °C, respectively.
5 For Room 4, which was heated by the AAHP, the indoor temperature was set at 23 °C. To account for
6 typical indoor temperature fluctuations, a temperature return difference of ± 2 °C was applied for all rooms.
7 Thus, the actual indoor temperature range for Rooms 1–3 was 22 ± 2 °C, 23 ± 2 °C, and 24 ± 2 °C, while for
8 Room 4, it was 23 ± 2 °C. The detailed environmental settings for both outdoor and indoor conditions are
9 provided in Table 3.

10 The experimental process is illustrated in Fig. 7. First, the outdoor environmental conditions were
11 stabilized at 2/1 °C. Preliminary experiments were then conducted to regulate the ASHP heating capacity.
12 After each ASHP was turned on, its heating capacity was adjusted by modifying the compressor speed and
13 outdoor fan frequency. Once the indoor temperature of each room stabilized at the target setpoint, it was
14 assumed that the heating capacity of the ASHP matched the building heating load. To prevent frosting from
15 influencing subsequent experiments, both ASHPs underwent one complete defrosting operation before the
16 preliminary experiments concluded.

17 During the preliminary experiments, it was determined that when the AWHP operated at a compressor
18 speed of 70 rps and an outdoor fan frequency of 1750 rpm, its heating capacity approximately matched the
19 building heating load. Similarly, for the AAHP, a compressor speed of 50 rps and an outdoor fan frequency
20 of 450 rpm were found to meet the required heating load. These operating parameters were adopted for
21 subsequent experiments and are detailed in Table 3. After the determinations of compressor speed and
22 outdoor fan frequency, the preliminary experiments took about 2 hours every day. Before, the end of the

preliminary experiments, the indoor/outdoor temperatures and power input of ASHPs were all stabilized within their thresholds.

Finally, experiments on the indoor thermal environment during the dynamic frosting-defrosting cycles were conducted. To eliminate the influence of heating capacity fluctuations caused by compressor speed and outdoor fan frequency adjustments, both parameters were fixed at constant values manually throughout the testing process. Defrosting was initiated automatically once the operation parameters of the ASHPs met the defrosting initiation criteria described in Section 2.2. To ensure the reliability of results and mitigate random errors, each experiment was repeated four times for both the AWHP and AAHP.

Table. 3 Experimental conditions for AWHP and AAHP heating systems

Types	Compressor rotate speed (rps)	Outdoor fan speed (rpm)	Outdoor environmental condition (°C)	Indoor setting temperature (°C)
Air-water heat pump	70	1750	2/1	Room 1: 22
				Room 2: 23
				Room 3: 24
Air-air heat pump	50	450		23

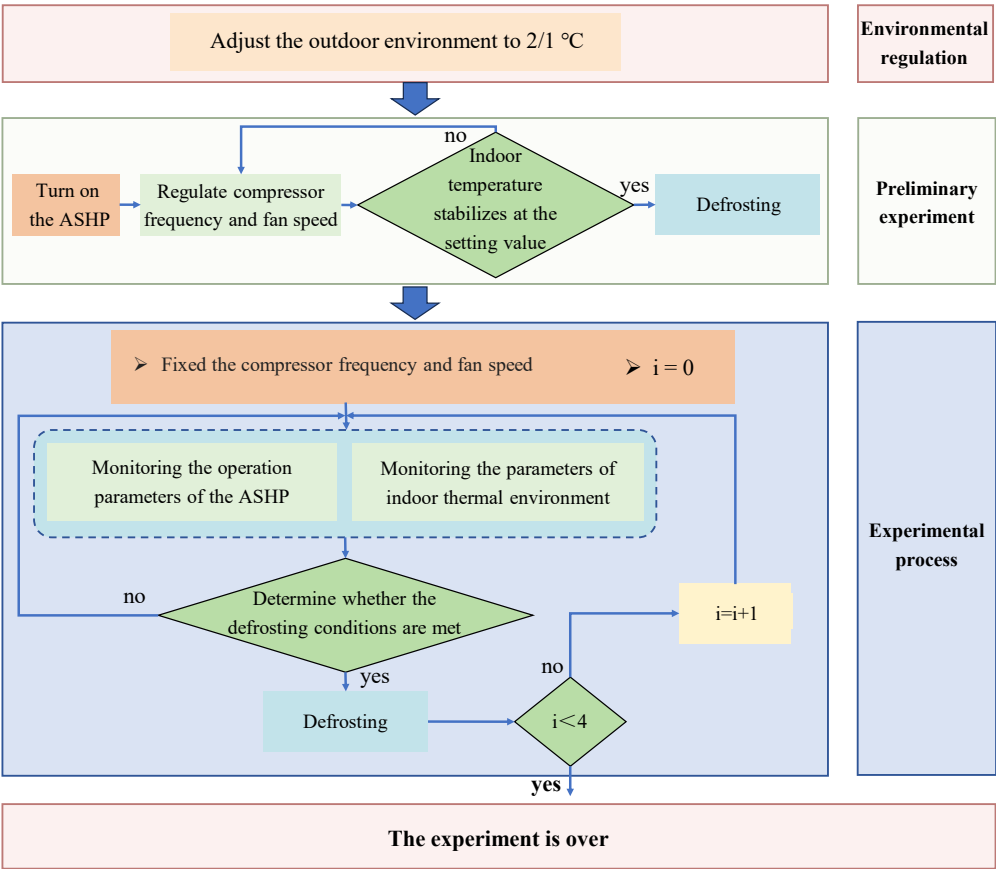


Fig. 7 Experimental process

3 Experimental results

3.1 Heating performance analysis of ASHP

Fig. 8 illustrates the variations of supply and return water temperature during the four groups of repeated frosting-defrosting experiments of the AWHP. The experiment started when the air-water heat pump restarted to supply heating after the defrosting operation in the preliminary experiment, and stopped when the supply water temperature reached 38.0 °C after the defrosting operation in the fourth group of experiment. As expressed in Section 2.4, before the experiments, a defrosting operation was executed at the end of preliminary experiment to eliminate the effect from frost accumulated on outdoor coil. This defrosting operation resulted in the low supply and return water temperature at the start time of experiments, which are 17.1 °C and 14.8 °C, respectively. Then, as the AWHP restarted to supply heating, both the supply and return water temperature exhibited progressive increases. In all the four groups of experiments, the supply and return water temperature ultimately stabilized at approximately 40 °C and 34 °C. At the time of 47.75, 98.75, 151.67, and 202.83 min, the operation parameters of AWHP met the initiation criteria of defrosting expressed in Section 2.2. The AWHP was shut down automatically for 2 min to decrease the pressure difference between the two sides of four-way valve. During the 2-min shutdown, the supply water temperature decreased quickly, and became equal to the return water temperature. Subsequently, the four-way valve switched, and the defrosting operation started. During the defrosting process, the indoor heat exchanger (plate heat exchanger in Fig. 3) became the evaporator, and the refrigerant in AWHP absorbed heat from the circulating water. Consequently, the supply water temperature became lower than the return water temperature. During the four groups of experiments, when the defrosting went on 1.33 min, 1.17 min, 1.41 min, and 1.59 min respectively, the heat absorption of refrigerant from the circulating water reached

1 the maximum. The corresponding temperature difference between supply and return water temperature was
 2 -12.2 °C, -12.8 °C, -12.7 °C, and -12.7 °C in turn. When the surface temperature of outdoor coil reached
 3 20 °C, the defrosting terminated, and the AWHP was shut down automatically for 2 min again. Then, the
 4 four-way valve switched again, and the AWHP restarted to supply heating.

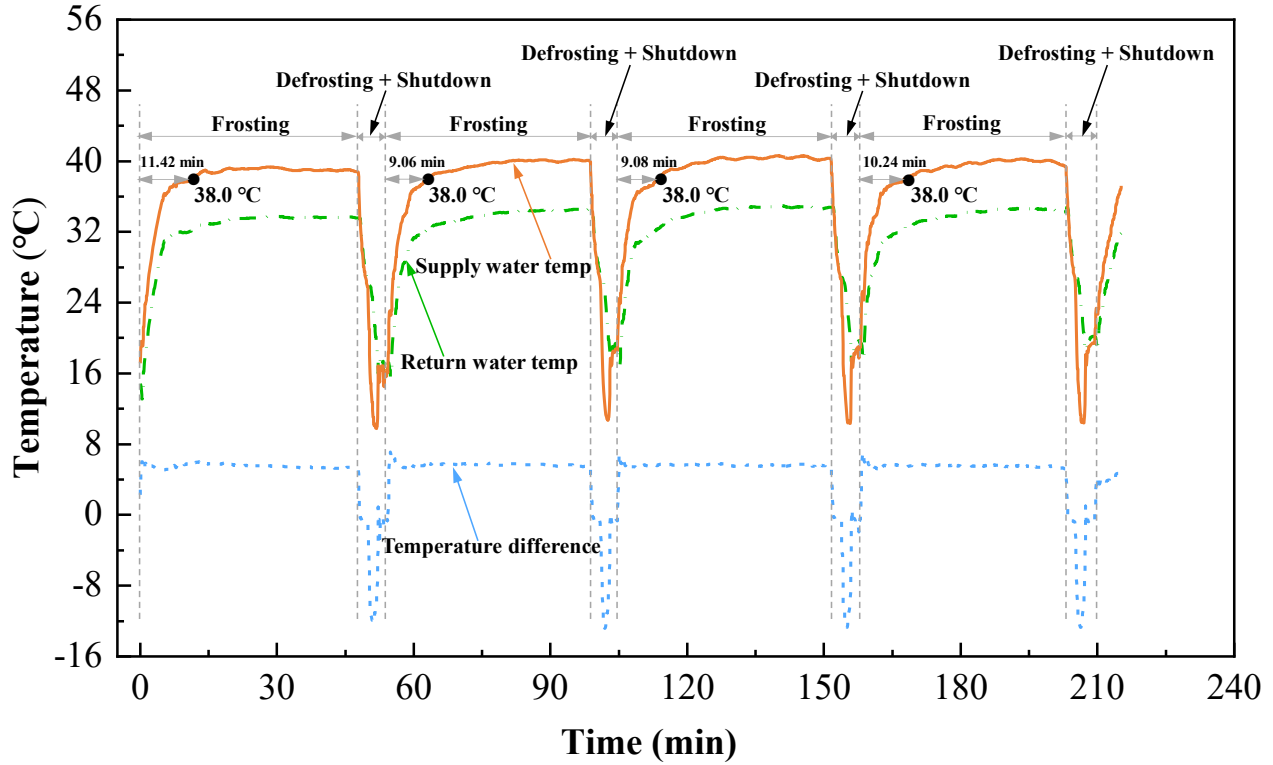


Fig. 8 Water temperature variations of the AWHP during experimental process

7 In the four groups of experiments, the defrosting durations, i.e. the time duration between two four-
 8 way valve switching, were 2.25 min, 2.08 min, 2.25 min, and 2.34 min respectively. Considering the two
 9 2-min shutdown before each four-way valve switching, the total interruption intervals of space heating
 10 caused by defrosting were 6.25 min, 6.08 min, 6.25 min and 6.34 min, respectively. Although the total
 11 interruption intervals are not long, the fan-coils and refrigerant absorbed heat from the circulating water
 12 simultaneously. This made return water temperature drop to approximately 16.0 °C in all the four groups
 13 of experiments. After the second four-way valve switching, the AWHP restarted to supply heating, and
 14 experienced a heating-resume period to make the supply water temperature rise to 38.0 °C, at which point

1 the heating performance of AHP was considered to have returned to normal. In the four groups of
2 experiments, the heating-resume period was 8.75-10.00 min. Therefore, the total abnormal space heating
3 durations caused by defrosting, including defrosting, two shutdowns, and heating-resume period, were
4 14.83-16.34 min.

5 Fig. 9 illustrates the supply and return air temperature variations during the four groups of frosting-
6 defrosting experiments of the AAHP. The total experimental duration was 228 min. The experiment was
7 also started when the AAHP restarted to supply heating after the defrosting operation in the preliminary
8 experiment, and stopped when the supply air temperature reached 36.0 °C after the defrosting in the fourth
9 group of experiment. During the four groups of experiments, when the space heating of AAHP reached
10 stability, the supply and return air temperature maintained at around 36.0 °C and 25.0 °C, respectively. The
11 defrosting operation initiated at the time of 44.00 min, 93.58 min, 151.33 min, and 214.17 min, and the
12 defrosting durations were 2.25 min, 2.42 min, 2.75 min, and 2.91 min in turn. As expressed in Section 2.2,
13 the AAHP experienced a 1-min shutdown before the two four-way valve switching. Consequently, for
14 AAHP, the total interruption intervals of space heating caused by defrosting were 4.25, 4.42, 4.75, and 4.91
15 min. During the defrosting process, the indoor fan-coil was turned off. This made the air temperature near
16 its inlet just decrease slightly. However, owing to the natural-convection heat transfer, the air temperature
17 at the outlet of indoor fan-coils decreased significantly to around 20.0 °C. When AAHP had restarted to
18 supply heating, the supply air temperature increased rapidly. In the four groups of experiments, the heating-
19 resume durations, during which time the supply air temperature returned to 34.0 °C, were 6.33 min, 5.67
20 min, 5.67 min, and 5.09 min in turn. Therefore, the total abnormal space heating durations caused by
21 defrosting, including defrosting, two shutdowns, and heating-resume period, were 10.00-10.58 min.

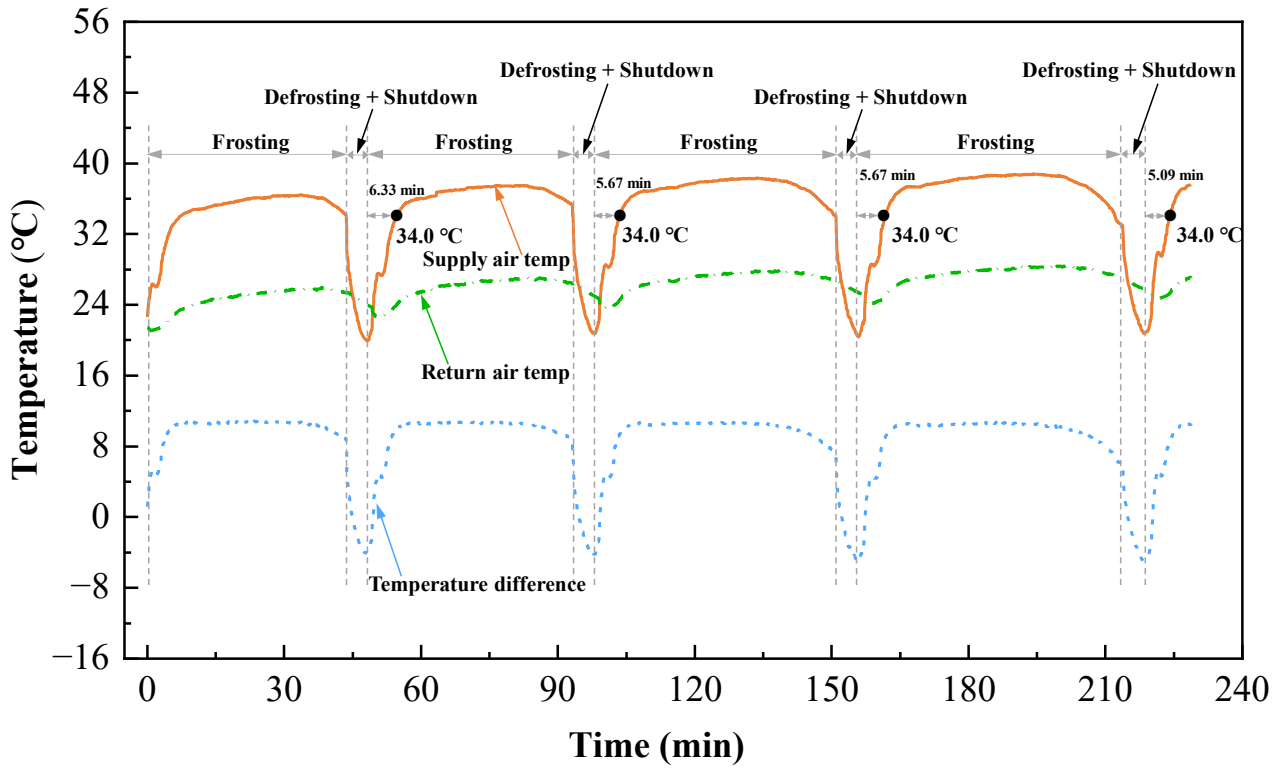


Fig. 9 Air temperature variations of the AAHP during experimental process

The comparison of Figs. 8 and 9 reveals that the variation trends of supply and return water temperature in AWHP heating system agreed well with the variation trends of supply and return air temperature in AAHP heating system. Specifically, the temperature increased gradually during the frosting process, while dropped rapidly during the defrosting process. However, the recovery speed of temperatures varies a lot between the AWHP heating system and AAHP heating system.

Based on the supply/return water and supply/return air temperature shown in Figs. 8 and 9, the heating capacity during the experimental process was calculated for both the AWHP and AAHP using Eqs. (1) and (2), respectively. The results are presented in Fig. 10. After each defrosting operation, the heating capacity increased rapidly. When the heating capacity became stable, the heating capacity of AWHP reached 9.82–10.01 kW. Subsequently, as the frost accumulation progressed, the heating capacity gradually decreased. At the start time of the defrosting operation, the heating capacity declined to 9.56 kW, 9.67 kW, 9.61 kW, and 9.37 kW, with the decay rates of 4.50%, 2.52%, 3.81%, and 4.58%, respectively. Upon the defrosting

starting, the heating capacity decreased sharply. During the defrosting operation, the plate heat exchanger of AWHP (as shown in Fig. 3) worked as the evaporator, and the AWHP started to absorb heat from the circulation water. That is, when its working mode transitioned from frosting to defrosting, the AWHP changed from releasing heat into the water to absorbing heat from the water, and the heating capacity became from positive to negative. As the circulation water was relatively high as shown in Fig. 8, AWHP can absorb lots of heat from the water. At the time of 51.08 min, 101.92 min, 155.08 min, and 206.42 min, the heat absorbed from circulating water reached the peak values. The peak values were 21.08 kW, 22.22 kW, 21.94 kW, and 22.03 kW respectively. The total heating capacity of AWHP under frosting and non-frosting conditions were also calculated, as displayed in Table 4. The decline rate of heating capacity caused by frosting-defrosting was 20.08%-23.32%.

$$q_w = c_w m(T_s - T_r) / 3600 \quad (1)$$

$$q_a = c_a \rho v(T_{as} - T_{ar}) / 3600 \quad (2)$$

For the AAHP, the heating capacity also demonstrated rapid recovery after the defrosting operation. After the heating capacity became stable, the heating capacity in the four groups of experiments reached around 2.58 kW, 2.55 kW, 2.53 kW, and 2.51 kW respectively. With the growth of frost layer on the outdoor heat exchanger surface, the heating capacity progressively declined. At the start time of the defrosting operation, the heating capacity decreased to 2.26 kW, 2.21 kW, 2.18 kW, and 2.14 kW, respectively. The decay rates in the four groups of experiments were 12.40%, 13.33%, 13.83%, and 14.74%, respectively. As displayed in Fig. 9, the return air temperature was higher than the supply air temperature obviously. However, since the indoor fan-coil of AAHP was turned off during defrosting operation as expressed in Section 2.4, it could not absorb heat from indoors, and the heating capacity remained zero. The total heating capacity of AAHP under frosting and non-frosting conditions were also calculated, and the results were

also shown in Table 4. The decline rate of heating capacity caused by frosting-defrosting was 15.87%-16.45%.

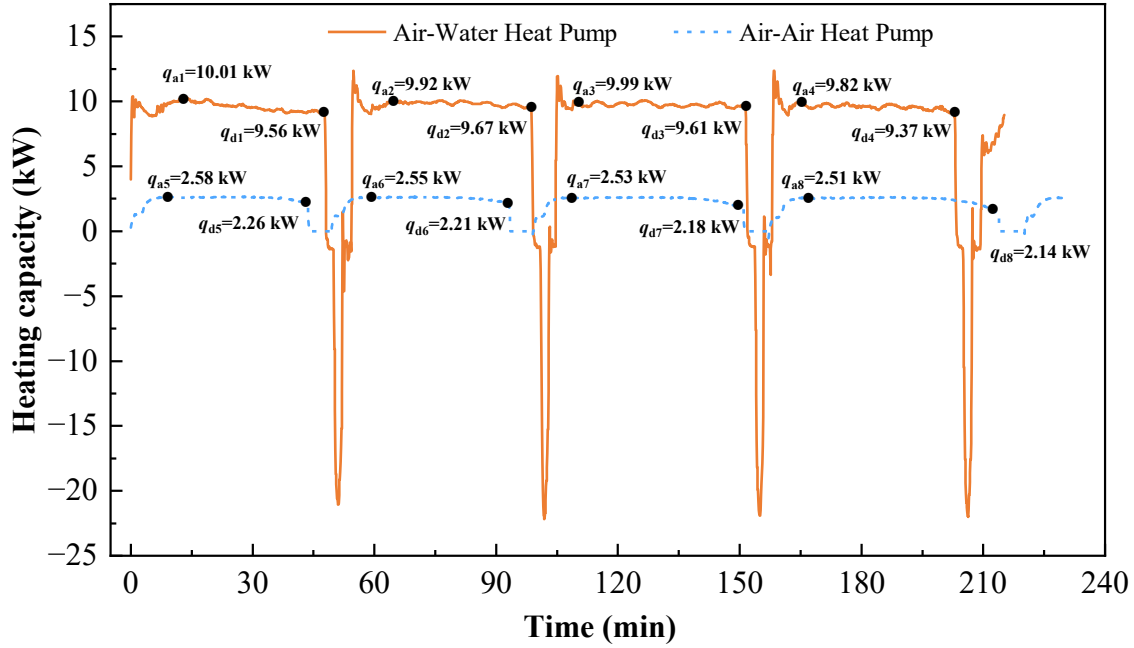


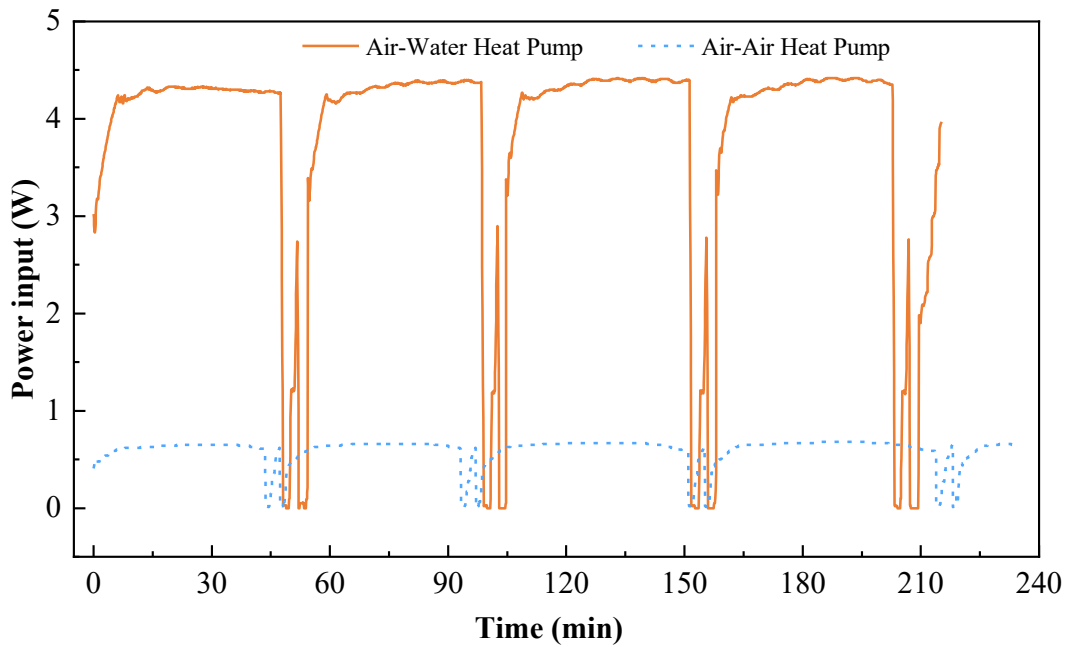
Fig. 10 Variation of heating capacity during frosting-defrosting cycles in AWHP and AAHP

Table 4 Decline rate of the total heating capacity of AWHP and AAHP during four frosting-defrosting cycles

Item	Total heating capacity of AWHP (kJ)			Total heating capacity of AAHP (kJ)		
	Without frosting	With frosting	Decline rate	Without frosting	With frosting	Decline rate
1st cycle	32632.6	25023.45	23.32%	7636.28	6399.45	16.20%
2nd cycle	29908.8	23902.45	20.08%	7573.5	6366.35	15.94%
3rd cycle	31918.05	25085.75	21.41%	8866.64	7408.4	16.45%
4th cycle	30294.72	23539.55	22.30%	8710.704	7328.55	15.87%

Fig. 11 illustrates the power input variations of the AWHP and AAHP during the frosting-defrosting cycles. During the frosting process, the power input remained relatively stable at approximately 4.3 kW for the AWHP and approximately 0.65 kW for the AAHP. Once the ASHPs reached the defrosting initiation criteria, owing to the shutdown of the compressor and outdoor fan, the power input of AWHP and AAHP both immediately dropped to 0. After the 2-min shutdown for AWHP and 1-min shutdown for AAHP, the four-way valve switched, and the defrosting started. During the defrosting process, the power input increased rapidly to approximately 2.7 kW for AWHP and approximately 0.62 kW for AAHP. After

1 defrosting, the power input declined sharply again to 0. After the other 2-min shutdown for AWHP and 1-
 2 min shutdown for AAHP, the four-way valve switched again, and the power input increased gradually to
 3 supply heating.



4 **Time (min)**
 5 Fig.11 Power input variations of ASHPs during experimental process

6 **3.2 Variations of indoor thermal environment**

7 Figs. 12 and 13 present the variations in indoor air temperature and relative humidity in the four rooms
 8 during the experiments. As described in Section 2.3, 270, 150, 90, and 210 thermocouples were respectively
 9 installed in the four rooms to monitor the indoor temperature. Since the temperature varied across different
 10 locations, the average values of all thermocouple readings in each room are displayed in Figs. 12 and 13.
 11 Throughout the experiment, although the temperature in all four rooms overall exhibited a slight upward
 12 trend, they remained stable near the setpoint values listed in Table 3. For both the AWHP and AAHP, a
 13 significant drop in indoor temperature was observed during the defrosting process. During the four groups
 14 of experiments, the decline magnitudes in each room are summarized in Table 5. The maximum indoor
 15 temperature drop reached 4.66 °C in rooms heated by the AWHP (Rooms 1-3), while it was only 1.79 °C

1 for the room heated by the AAHP (Room 4). Furthermore, the average temperature decline in each room
2 during four groups of frosting-defrosting was calculated. The average temperature drop in the three rooms
3 heated by the AWHP was 3.61 °C, 3.28 °C and 4.27 °C respectively, while only 1.71 °C in the room heated
4 by the AAHP. In Table 5, the standard deviations of indoor temperature drop in four rooms were also
5 displayed. The values in rooms heated by the AWHP (Rooms 1-3) were significantly greater than that in
6 the room heated by AAHP (Room 4), which meant the differences among the indoor temperature drops at
7 various locations in rooms heated by the AWHP were larger. Considering the average temperature drops
8 and standard deviations in the four rooms, it can be concluded that the temperature drop in the rooms heated
9 by the AWHP is significantly larger than that in the room heated by the AAHP.

10 Additionally, the temperature recovery rate in the room heated by AWHP was notably slower than
11 that heated by AAHP. Taking Room 1 as an example, in the first three groups of experiments, the time for
12 the indoor temperature to recover to the value before defrosting operation was 31.17 min, 31.50 min, and
13 29.58 min, respectively. Meanwhile, for Room 4, the time was 19.33 min, 24.92 min, and 25.54 min,
14 respectively.

15 To exhibit the temperature differences at various locations during frosting-defrosting cycles, the
16 standard deviations of indoor temperature at different time in the first frosting-defrosting cycle were
17 calculated as expressed in Table 6. At the defrosting termination time, the standard deviations were the
18 lowest in all the four rooms. They were 0.66 °C, 0.74 °C, 0.62 °C, and 1.59 °C, respectively. The value in
19 Room 4 was significantly greater than those in Rooms 1-3, which indicated that the temperature difference
20 in rooms heated by AAHP was higher at the defrosting termination time. Then, the standard deviations
21 increased gradually in the frosting process, and became stable at the 20th min. In all the four rooms, the
22 values at the period from 20th min to the time of defrosting beginning were approximately 2.0 °C. This

1 meant that the temperature differences at various locations in rooms heated by AWHP and AAHP are
2 similar in this period.

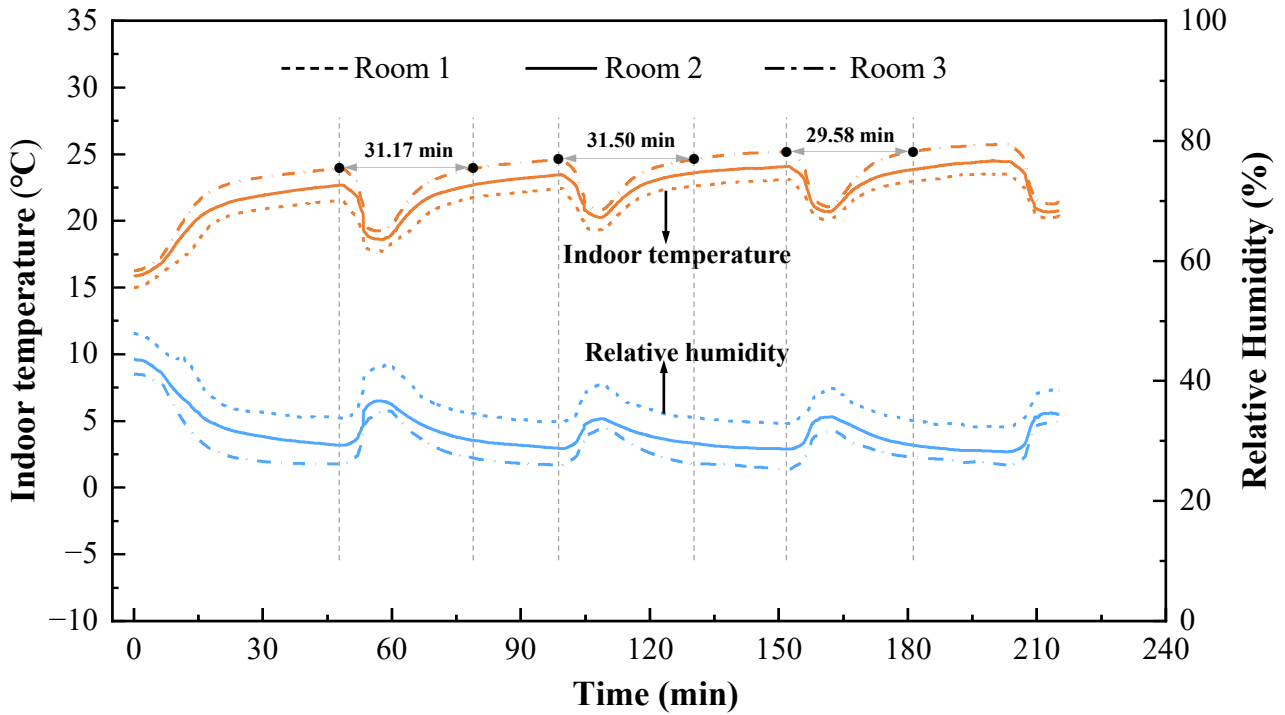


Fig.12 Variations of indoor temperature during experimental process of AWHP

Table 5 Temperature drop magnitude and standard deviation after each frosting-defrosting cycles

Item	1 st defrosting $\mu \pm \sigma$ (°C)	2 nd defrosting $\mu \pm \sigma$ (°C)	3 rd defrosting $\mu \pm \sigma$ (°C)	4 th defrosting $\mu \pm \sigma$ (°C)
Room 1	4.06±1.565	3.18±1.701	3.41±1.769	3.79±1.512
Room 2	3.73±1.297	3.12±1.209	2.99±1.26	3.28±1.538
Room 3	4.66±1.969	3.78±2.058	4.16±2.207	4.49±1.996
Room 4	1.58±0.662	1.7±0.634	1.77±0.536	1.79±0.621

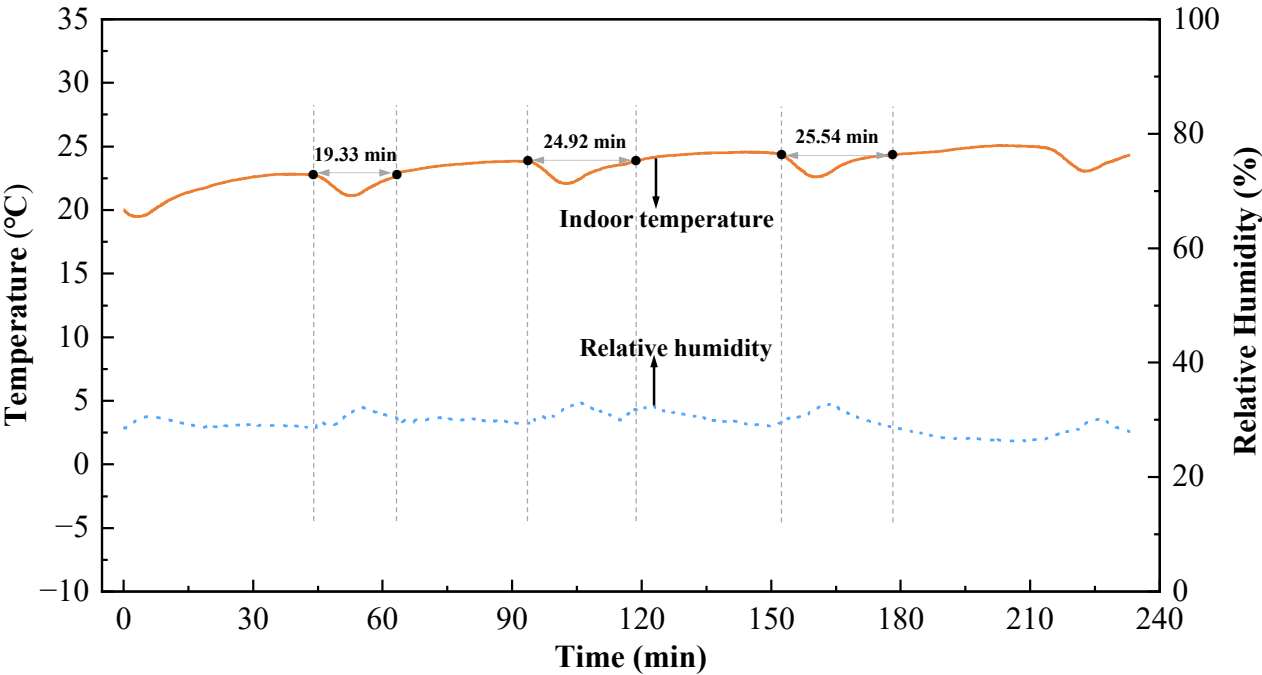
* μ is the average temperature drop value, and σ is the standard deviation.

In Figs. 12 and 13, it can be observed that the variation trend of relative humidity in all the four rooms was opposite to that of indoor temperature. For instance, when the indoor temperature in Room 4 reached the highest value, the relative humidity dropped to the lowest value. Throughout the experimental process, for all the four rooms, noticeable fluctuations in indoor relative humidity could be observed. Specifically, the relative humidity in the rooms heated by AWHP increased by 13.73%–25.01%, while 9.05%–10.66% in the room heated by AAHP. This indicates that the frosting-defrosting of AWHP has a greater impact on

1 indoor relative humidity. As the heating resumed after the defrosting operation, the relative humidity
 2 gradually decreased again, and reached its minimum value just before the next defrosting operation.

3 Table 6 Standard deviations of indoor temperature at different time in the first frosting-defrosting cycle

Standard deviation (σ)	defrosting termination	10 th min	20 th min	30 th min	40 th min	defrosting beginning
Room 1	0.66	1.82	1.97	1.99	1.98	1.99
Room 2	0.74	1.81	1.84	2.06	2.0	2.08
Room 3	0.62	2.02	2.07	2.25	2.35	2.42
Room 4	1.59	1.78	2.01	2.14	2.17	2.16



4
5 Fig.13 Variations of indoor relative humidity during experimental process of AAHP

6 4 Discussion

7 4.1 Time point of the lowest indoor temperature

8 Before conducting this work, it was a common view that the indoor temperature would increase
 9 gradually once the defrosting terminated, and the lowest indoor temperature appeared at the end moment
 10 of defrosting. However, combining Figs. 9, 10, 12 and 13, it can be found that the indoor temperature
 11 continues to decrease after defrosting, and its lowest value appears at a later moment. To reveal the time
 12 point of the lowest indoor temperature, the supply water temperature, return water temperature, and the

1 indoor temperature during the first frosting-defrosting cycle of AWHP are replotted in Fig. 14. Meanwhile,
2 the supply air temperature, return air temperature, and indoor temperature during the first frosting-
3 defrosting cycle of AAHP are replotted in Fig. 15.

4 In Fig. 14, it can be seen that the AWHP was shut down at the time of 47.75 min. After an approximate
5 2-min shutdown, it started to defrost. During the shutdown period, both the supply water temperature and
6 return water temperature decreased sharply from 38.80 °C and 33.58 °C. The lower supply water
7 temperature will decrease the heat dissipated by indoor fan-coils. Thus, the indoor temperature started to
8 decrease since the time of 49 min. At the moment of 50.33 min (0.58 min since defrosting operation
9 initiation), the supply water temperature decreased to 18.1 °C, and became lower than the indoor air
10 temperature of all the three rooms. This indicates that the indoor fan-coils do not dissipate heat to three
11 rooms, but absorb heat from them. Owing to the coupled effect of heat absorption of fan-coil and heat
12 dissipation to outdoor environment, the decline speed of indoor temperature became faster. This was the
13 main reason for the greater indoor temperature decrease of air-water heat pump, as displayed in Fig. 12. At
14 the time of 52 min, the defrosting terminated. After defrosting, although AWHP was shut down and stopped
15 absorbing heat from circulating water, the supply water temperature was as low as approximately 17.0 °C.
16 The lower supply water temperature enhanced the heat absorption capacity of indoor fan-coils.
17 Consequently, the decline speed of indoor temperature continued to increase until the AWHP restarted to
18 supply heating at the time of 54.00 min.

19 After the time of 54.00 min, the AWHP restarted to supply heating. Although its heating capacity
20 increased quickly after defrosting as displayed in Fig. 10, the thermal mass of the circulation water was
21 very large, which resulted in the slow increase in circulation water temperature. The heat release of the
22 indoor fan-coils was proportional to the temperature difference between the circulating water and the air.

Owing to the low temperature of circulation water caused by its thermal mass, the heat release of indoor fan-coils was low, and further led to a continuous decline in indoor temperature until the circulation water temperature rose to a certain value. At the moments of 57.08 min, 56.25 min, and 56.92 min, the indoor temperature of the three rooms reached the lowest values of 18.62 °C, 17.78 °C, and 19.25 °C. This means the lowest indoor temperature appeared 4.25-5.08 min after defrosting termination. The appearing time of the lowest indoor temperature after defrosting termination during the other 3 frosting-defrosting cycles was also counted, as expressed in Table 7. From all the 4 frosting-defrosting cycles, it can be obtained that the lowest indoor temperature appears at 4.25-6.00 min after the defrosting termination of AWHP. Furthermore, the relationship between the delay time of the lowest indoor temperature and its setting value was analyzed, and there was no obvious pattern.

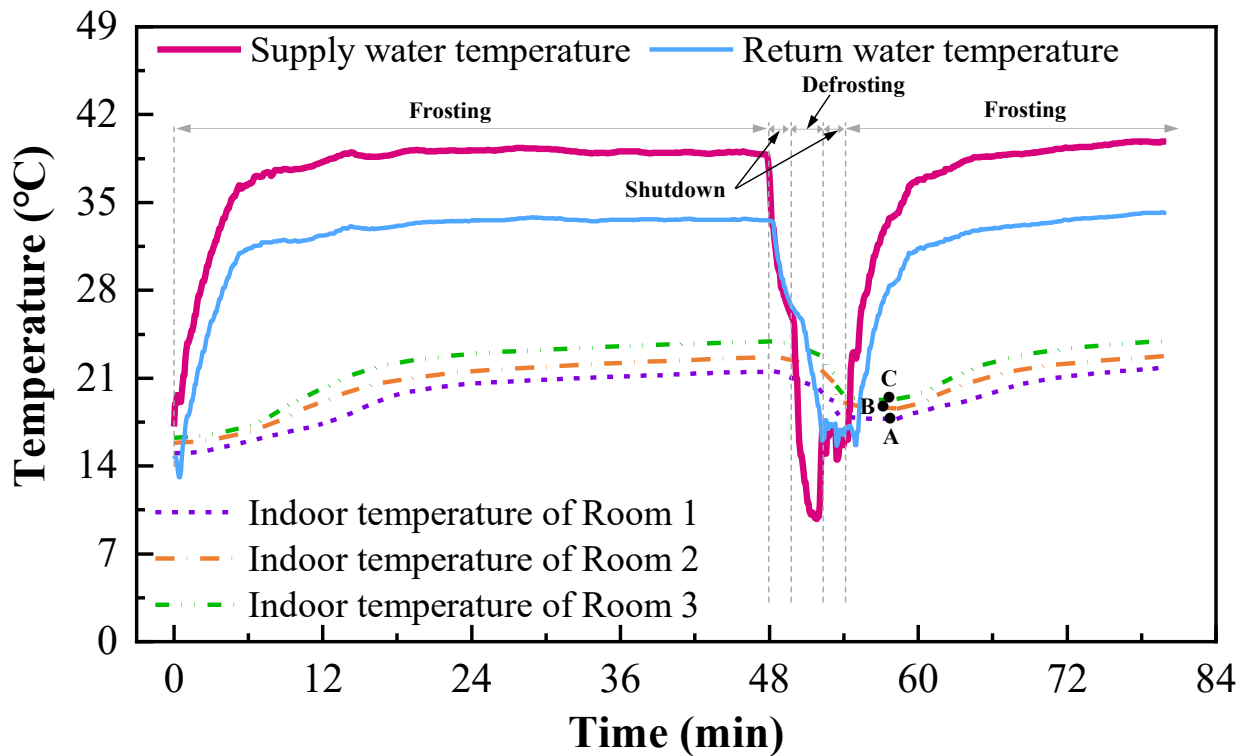


Fig. 14 Variations of water temperature and indoor temperature during the first frosting-defrosting cycle

For AAHP, although it was transitioned to heating mode after defrosting, the fan of indoor coil was still shut down owing to its low temperature, as expressed in Section 2.2. Consequently, the indoor

1 temperature continued to decrease after defrosting, as illustrated in Fig. 15. Once the indoor fan-coil
 2 restarted to run, its heat release became greater than building heating capacity soon, and the indoor
 3 temperature started to increase. The lowest indoor temperature appeared at the time of 4.17 min after
 4 defrosting termination. In Table 7, the lowest indoor temperature during other three frosting-defrosting
 5 cycles was also given. The lowest indoor temperature during the four frosting-defrosting cycles appeared
 6 at the moment of 4.17 min, 4.25 min, 3.84 min, and 3.75 min after defrosting termination. Therefore, for
 7 AAHP, the lowest indoor temperature appeared 3.75-4.25 min after defrosting termination for AAHP.

8 Through the discussion above, it could be obtained that for both the AWHP and AAHP, the lowest
 9 indoor temperature did not appear at the termination time as previously thought[32, 33], but at 4.25-6.00
 10 min after defrosting operation for AWHP, while at the 3.75-4.25 min after defrosting termination for AAHP.

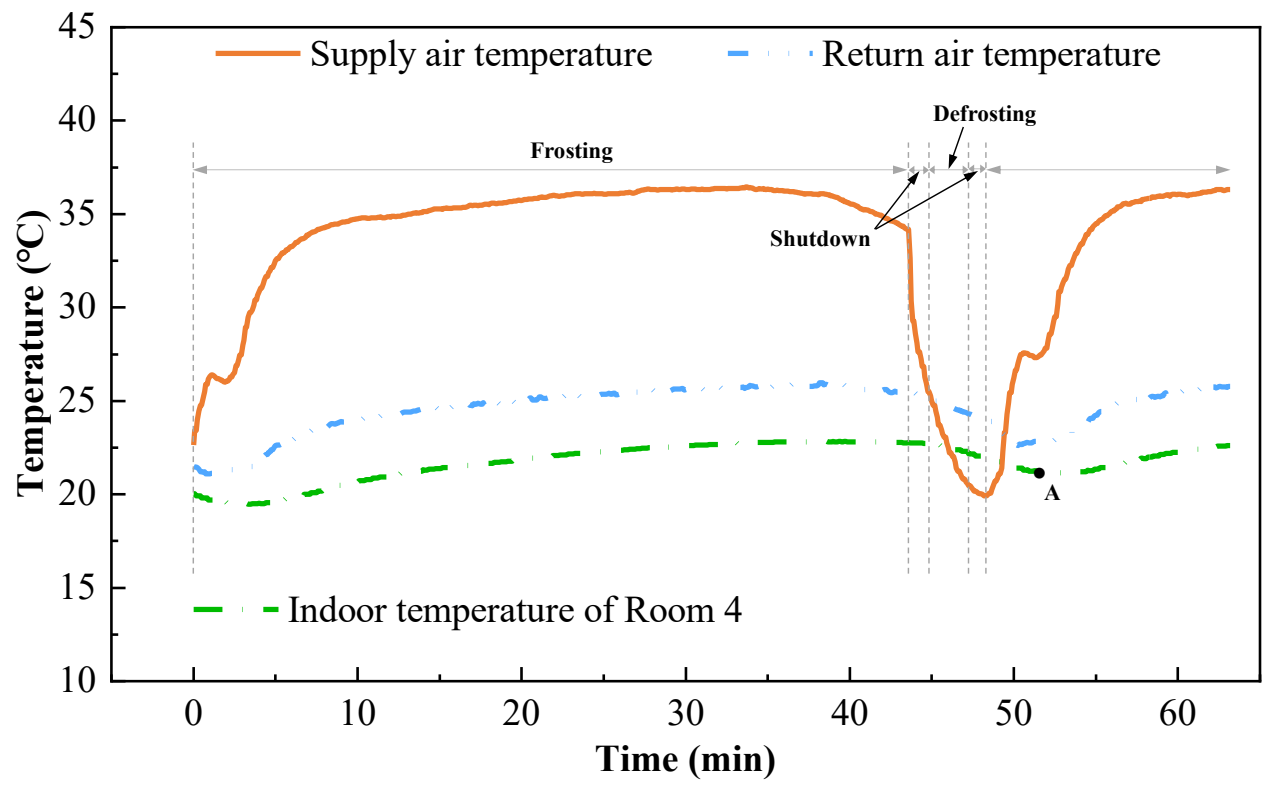


Fig.15 Variations of air temperature and indoor temperature during the first frosting-defrosting cycle

Table 7 Time lag between defrost termination and the lowest room temperature

Item	1 st defrosting (min)	2 nd defrosting (min)	3 rd defrosting (min)	4 th defrosting (min)
Room 1	5.08	6.00	5.75	5.58
Room 2	4.25	4.50	4.83	5.08
Room 3	4.92	5.59	5.75	5.58
Room 4	4.17	4.25	3.84	3.75

4.2 Effect of defrosting on indoor temperature at different heights

As shown in Fig. 12, the average indoor temperature exhibits significant fluctuations during the frosting-defrosting cycles. However, it cannot reveal the vertical temperature variations. Excessive vertical temperature difference affects human thermal comfort. Therefore, it is necessary to investigate the temperature variation at different heights during the frosting-defrosting cycles. To obtain this goal, in this section, the effect of defrosting on indoor temperature at different heights is analyzed firstly. Then, the percentage dissatisfied (P_D) caused by vertical temperature at sitting and standing postures is predicted.

As expressed in Section 3.2, the fluctuation range of indoor temperature in rooms heated by AWHP is larger. Therefore, in this section, the vertical temperature distributions in rooms heated by AWHP are chosen and analyzed. Fig. 16 illustrates the air temperature at different positions in Room 3 at the starting point of defrosting operation in the first group of experiment. It could be observed clearly that the temperature gradient in the vertical direction was obvious, while the temperature difference among different positions at the same height was relatively small. Specifically, the average temperature at the height of 2.3 m was 22.63 °C, and it was 18.56 °C at the height of 0.3 m.

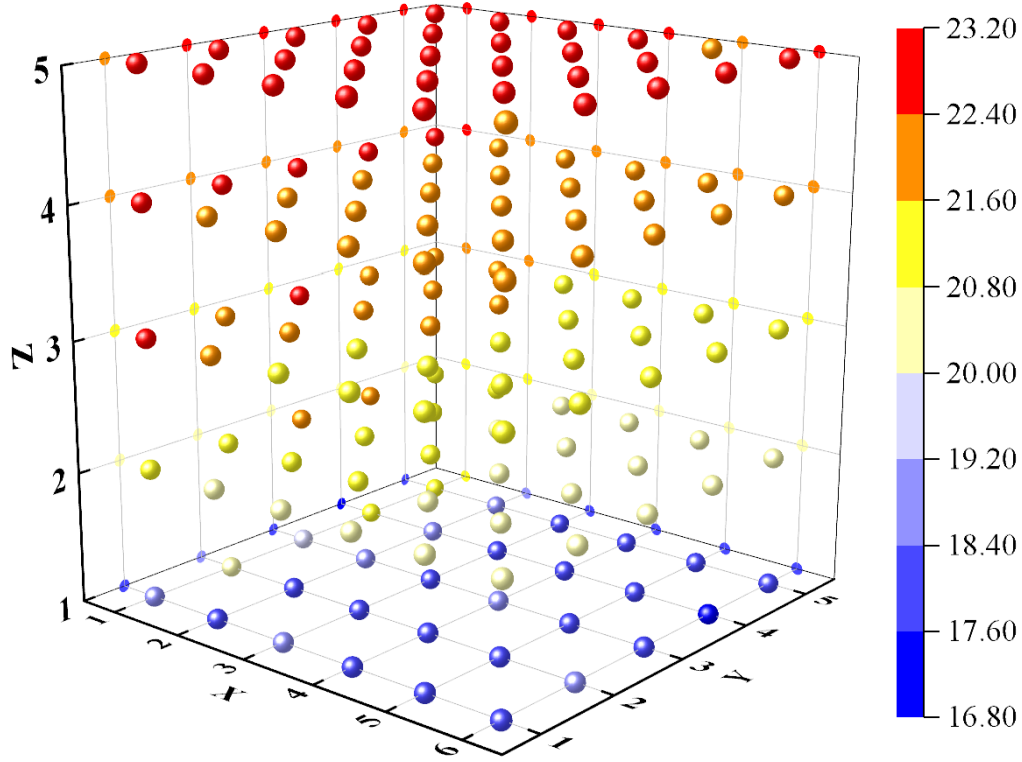


Fig.16 Air temperature distribution at different positions in Room 3.

To further reveal the variation of vertical temperature, Fig. 17 presents the average air temperature at the five heights in Room 3 throughout the entire experimental process. The average temperature gradually rises with the increase of height, and the temperature difference among different heights undergoes significant changes during the frosting-defrosting cycles. After each defrosting operation, the average air temperature at all the five heights increases gradually. As the increase velocity at higher height is greater, the temperature difference among different heights becomes larger gradually. In all the four groups of experiments, the temperature differences reach the peak values at the initiation time of defrosting operation. During the defrosting process, the average air temperature at all the five heights decreases rapidly, and the decrease velocity at higher height is much greater. This phenomenon reduces the vertical temperature difference rapidly. Taking the second group of experiment as an example, at the defrosting initiation time of 98.75 min, the vertical temperature difference between the heights of 2.3 m and 0.3 m reached the maximum value of 4.33 °C. Then, this temperature difference decreased rapidly. As expressed in Section

4.1, during a short period after defrosting operation, the heating capacity of AWHP remains lower than the building heating load. This makes the vertical temperature difference continue to decrease. At the time of 2.92 min after defrosting termination, the vertical temperature difference reached the minimum value of 1.96 °C.

In Fig. 17, it can also be observed that at a higher height, the minimum average air temperature after defrosting appeared earlier. Taking the second defrosting operation as an example, the minimum values at the height of 0.3 m, 0.8 m, 1.3 m, 1.8 m and 2.3 m appeared at the 109.25 min, 109.00 min, 108.92 min, 107.42 min and 107.08 min, respectively, which were 4.42 min, 4.17 min, 3.85 min, 2.59 min and 2.25 min after the defrosting termination. This is mainly because the outlets of the fan-coils are located at a higher position, and the hot supply air affects the upper air earlier.

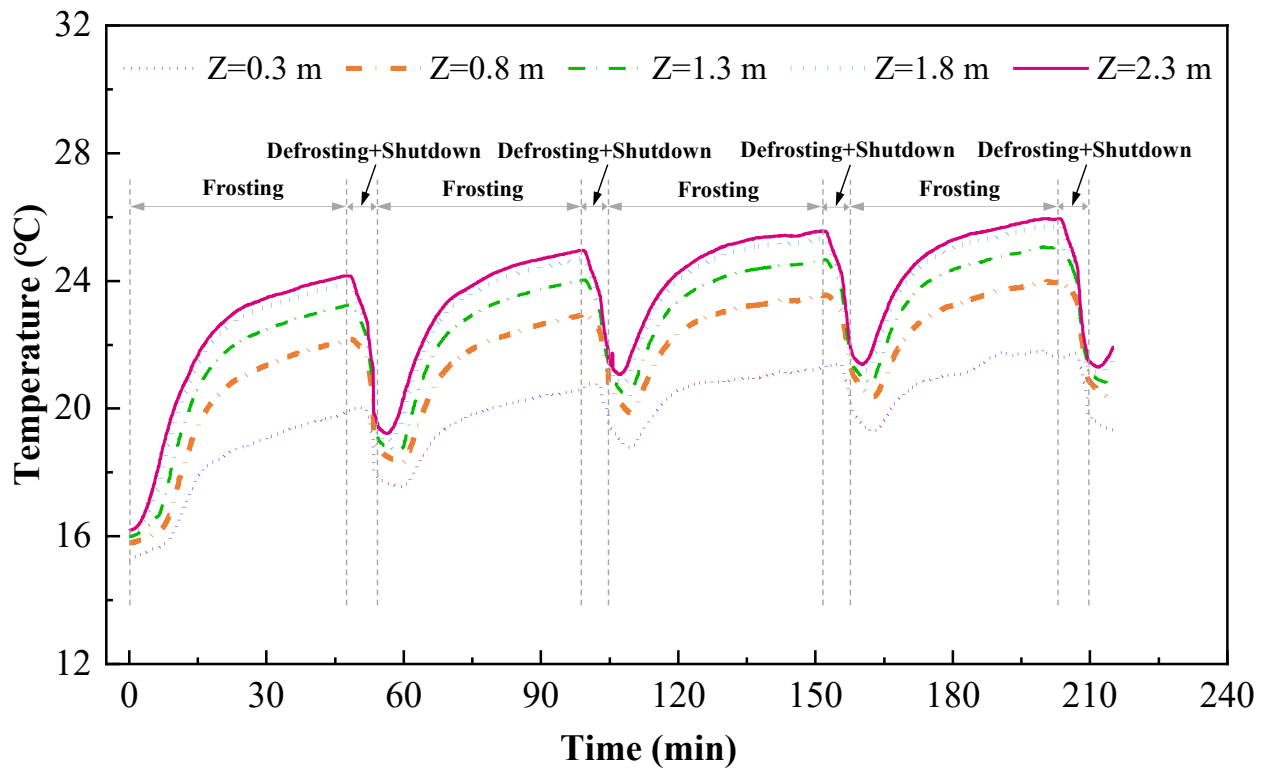


Fig.17 Average air temperature variations at vertical heights in Room 3 throughout the entire experimental process

As displayed in Figs. 12-13, the most obvious characteristics of indoor thermal environment during the frosting-defrosting is the short-term sharp fluctuations. This makes the PMV-PDD model unable to be

1 used to evaluate the indoor thermal comfort in this work. However, based on the air temperature variations
 2 at different vertical heights in Fig. 17, the percentage dissatisfied (P_D) caused by vertical temperature
 3 difference in Room 3 at standing and sitting postures could be calculated using Eq. (3) respectively, and
 4 the results are shown in Fig. 18. The vertical temperature difference at the standing posture is calculated
 5 based on the average air temperatures at the heights of 1.8 m and 0.3 m, while based on the average air
 6 temperature at the heights of 1.3 m and 0.3 m for the sitting posture. Owing to the greater vertical
 7 temperature difference at standing posture, its P_D value is significantly greater than that at the sitting posture.
 8 In all the four groups of experiments, the P_D caused by vertical temperature difference remained at a high
 9 level for a period before defrosting operation. The maximum values at the standing and sitting postures are
 10 9.96 and 5.98, respectively. As demonstrated in Fig. 17, the vertical temperature difference decreases
 11 rapidly once the defrosting operation initiates, and this makes the P_D value decrease rapidly. Taking the
 12 second group of experiment as an example, at the initiation time of defrosting operation, the P_D values at
 13 the standing and sitting postures were 9.28 and 5.41 respectively. By the time of 2.84 min after defrosting
 14 termination, the P_D values at standing and sitting postures both dropped to their lowest points, which were
 15 1.36 and 1.23 respectively.

$$P_D = \frac{100}{1 + \exp(5.76 - 0.856\Delta T_{ah})} \quad (3)$$

17 Where, P_D is the percentage dissatisfied (%); ΔT_{ah} is the vertical air temperature difference between the
 18 head and ankles (°C).

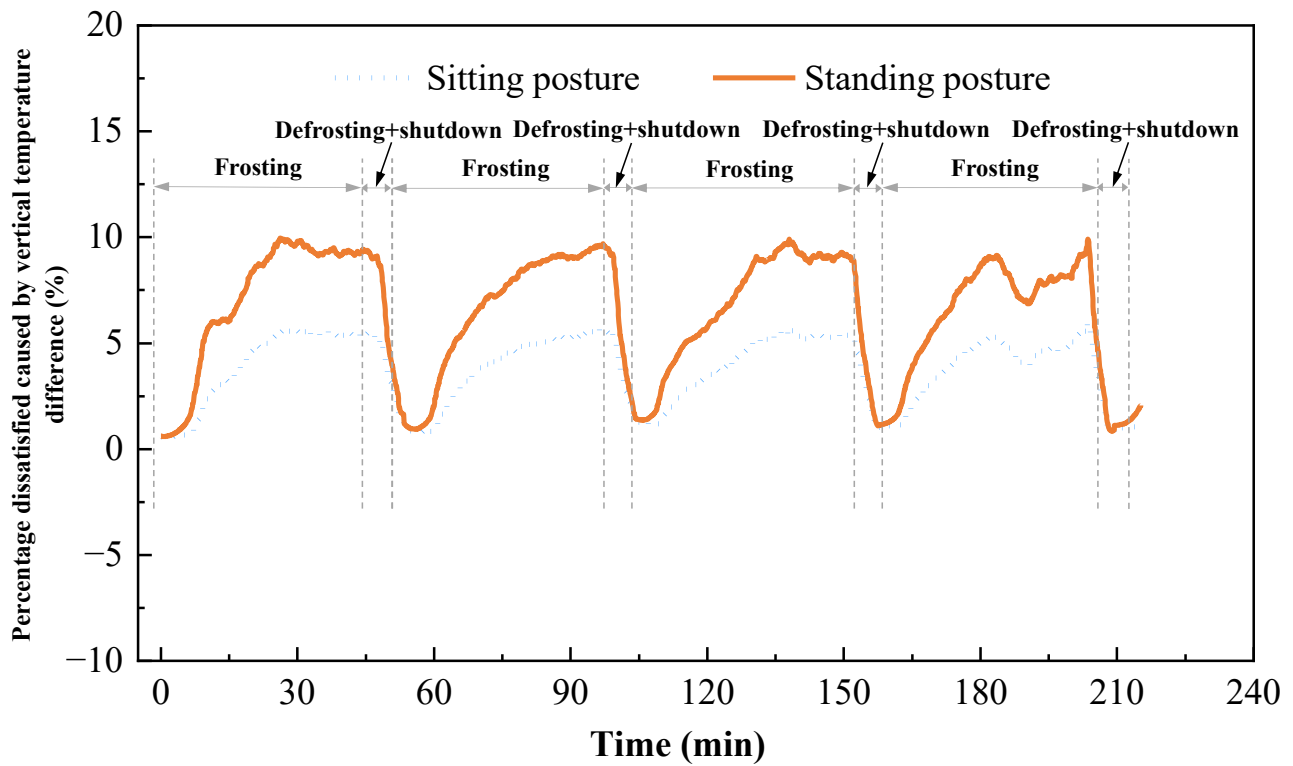


Fig.18 Variations of percentage dissatisfied caused by vertical temperature difference in Room 3

5 Conclusions

To reveal the of indoor thermal environment variation rules during the dynamic frosting-defrosting cycles of air source heat pump (ASHP), two test rigs, an air-water heat pump (AWHP) heating system and an air-air heat pump (AAHP) heating system, were built in an artificial environmental chamber. Through creating the frosting environmental condition, the practical space heating experiments of ASHP during frosting-defrosting cycles were conducted. Then, based on the experimental results, the indoor thermal environment variation rules of the two ASHPs and their differences were analyzed. The main findings are as follows.

(1) During the defrosting process, the indoor temperature using both AWHP and AAHP decreases obviously, and the decline amplitude in rooms using AWHP is greater. During the four groups of frosting-defrosting experiments, the temperature decline in rooms using AWHP is 2.63-5.48 °C, with an average of

1 3.72 °C. Meanwhile, the temperature decline in the room using AAHP is 1.58-1.79 °C, with an average of
2 1.71 °C.

3 (2) During frosting-defrosting cycles, the lowest indoor temperature does not appear at the termination
4 time of defrosting operation as previously thought, but a period later. For both the AWHP and AAHP, the
5 indoor temperature will continue to decrease after the termination time of defrosting operation. The lowest
6 temperature in rooms using AWHP appears at the 4.25-6.00 min after defrosting termination, while at the
7 3.75-4.25 min of defrosting termination for AAHP.

8 (3) For AWHP, during the late stage of its defrosting operation, the indoor fan-coils will absorb heat
9 from indoor air, rather than supply heating. At the 0.58th min of defrosting operation, the supply water
10 temperature decreased to 18.10 °C, and became lower than indoor air temperature. The circulating water
11 started to absorb heat from indoor air. Owing to the coupled effect of heat absorption of fan-coil and heat
12 dissipation to outdoor environment, the indoor temperature decline speed was greater.

13 (4) The defrosting operation has greater effect on the air temperature in the upper part of rooms, and
14 makes the vertical temperature difference decrease significantly. During the four sets of experiments, the
15 average air temperature at the height of 2.3 m decreases by 4.3-5.1 °C, while decreases 2.0-2.8 °C at the
16 height of 0.3 m. The vertical temperature difference decreases obviously. This makes the dissatisfaction
17 rate of vertical temperature difference decrease from 9.28% in standing posture and 5.41% in sitting posture
18 to 1.39% and 1.26%.

19 In summary, the decline amplitude in indoor temperature, the appearing time of the lowest indoor
20 temperature, and the reason of the indoor temperature decline during the frosting-defrosting of AWHP and
21 AAHP were revealed in this work. These results can guide the manufactures to improve the control
22 strategies of ASHPs, such as the operating state of indoor fans and the setting value of indoor temperature

at the beginning of defrosting operation, to decrease the indoor temperature fluctuation during the frosting-defrosting cycles.

During conducting the experiments in this work, the indoor thermal comfort was also investigated through questionnaire survey. Owing to the limitation of the article length, only the indoor thermal environment part is introduced in this paper, and the indoor thermal comfort analysis part cannot be expressed. In the next paper, the indoor thermal comfort part will be introduced in detail.

Acknowledgements

This work was financially supported by the National Natural Science Foundation of China (No. 52408098) and the National Support Program for Young Talent in Built Environment and Energy Engineering (2025-2027).

References

- [1] X.L. Yu, Q. Lv, L. Zhang, X.L. Wang, X.Z. Meng, Z.H. Zong, Study of thermal performance of air-source heat-pump heating for suburban residential buildings in Beijing, *Science and Technology for the Built Environment*, 26 (2020) 975-986.
- [2] Y. Xu, Y. Huang, N. Jiang, M. Song, X. Xie, X. Xu, Experimental and theoretical study on an air-source heat pump water heater for northern China in cold winter: Effects of environment temperature and switch of operating modes, *Energy and Buildings*, 191 (2019) 164-173.
- [3] G. Ma, T. Lu, F. Liu, J. Niu, S. Xu, Experimental study on hot liquid subcooling defrosting of an air source heat pump with multi-connected outdoor units, *Energy and Buildings*, 291 (2023) 113104.
- [4] Q. Yu, Z.L. Yang, J.D. Song, F.H. Xing, J.H. Zhou, H.J. Qi, Modeling and operational characteristics analysis of a dual-source synergetic heat pump based on air energy and biomass waste heat, *Energy*, 309 (2024) 133087.
- [5] A. Rogeau, J. Vieubled, L. Ruche, R. Girard, A generic methodology for mapping the performance of various heat pumps configurations considering part-load behavior, *Energy and Buildings*, 318 (2024) 114490.
- [6] W. Wang, Y.M. Cui, Y.Y. Sun, S.M. Deng, X. Wu, S.M. Liang, A new performance index for constant speed air-source heat pumps based on the nominal output heating capacity and a related modeling study, *Energy and Buildings*, 184 (2019) 205-215.
- [7] X. Bai, Z. Ge, S. Deng, M. Qu, X. Sang, L. Zhang, An experimental study on the defrosting performances of an ASHP having a novel dual-fan outdoor coil under different levels of frosting evenness along airflow direction, *Energy and Buildings*, 318 (2024) 114464.
- [8] P. Wei, Z. Wang, H. Feng, S. Wang, J. Fan, Frost suppression performance of an air source heat pump using sensible heat from indoor air to preheat outdoor air, *Building Services Engineering Research & Technology*, 44 (2023) 537-555.

- [9] S. Liang, W. Wang, Y. Sun, Y. Lin, Q. Luo, S. Deng, Field investigations on frosting suppression for variable-capacity ASHPs through optimizing their operations and configurations, *Energy and Buildings*, 224 (2020) 110266.
- [10] W.Z. Wei, Z.L. Feng, Y.Y. Sun, W. Wang, X.Y. Jin, J.F. Zhang, C.M. Dai, Experimental investigation on the space heating performance of air source heat pump with vapor injection under frosting condition, *Journal of Building Engineering*, 90 (2024) 111696.
- [11] Y. Lin, W. Wang, J.F. Luo, Y.Y. Sun, S.M. Deng, A novel variable speed ASHP frosting suppression operation method based on the frosting suppression performance map, *Applied Thermal Engineering*, 234 (2023) 121254.
- [12] J.C. Ma, D. Kim, J.E. Braun, W.T. Horton, Development and validation of a dynamic modeling framework for air-source heat pumps under cycling of frosting and reverse-cycle defrosting, *Energy*, 272 (2023) 127030.
- [13] M. Qu, M. Lu, Z. Li, X. Song, J. Chen, D. Ziviani, J.E. Braun, Thermal energy storage based (TES-based) reverse cycle defrosting control strategy optimization for a cascade air source heat pump, *Energy and Buildings*, 219 (2020) 110014.
- [14] J.K. Dong, L. Zhang, S.M. Deng, B. Yang, S. Huang, An experimental study on a novel radiant-convective heating system based on air source heat pump, *Energy and Buildings*, 158 (2018) 812-821.
- [15] Y.J. Xu, H.R. Zhang, K. Wu, H.Q. Jin, M.J. Song, X. Shen, Comparative study on adaptable intelligent frost recognition method for air-source heat pump and cold chain based on image texture features under complex lighting conditions, *International Journal of Refrigeration*, 175 (2025) 273-287.
- [16] H. Zhao, Z.H. Liu, Y.F. Sang, J.Z. Chang, X.J. Zheng, J. Jurasz, W.D. Zheng, A visual defrosting control method for air source heat pump system based on machine vision, *Energy*, 302 (2024) 131778.
- [17] Z.K. Li, Y.Y. Sun, W. Wang, W.Z. Wei, An improved equivalent temperature drop method for evaluating the operating performances of ASHP units under frosting conditions considering their configuration and operation, *Energy and Buildings*, 331 (2025) 115370.
- [18] W.Z. Wei, Z.L. Feng, L. Ni, W. Wang, X.Y. Jin, R. Tang, S.N. Zhao, Frosting suppression performance enhancement of air source heat pump through improving relative capacity, *Applied Thermal Engineering*, 236 (2024) 121698.
- [19] Y. Tan, W. Wei, W. Wang, Y. Sun, S. Wang, Z. Li, Z. Li, R. Tang, C. Zhang, S. Wei, Experimental and theoretical research on evaluation criterion for defrosting initiation time accuracy of air source heat pump, *Applied Thermal Engineering*, 278 (2025) 127235.
- [20] W. Wang, S.Q. Zhang, Z.Y. Li, Y.Y. Sun, S.M. Deng, X. Wu, Determination of the optimal defrosting initiating time point for an ASHP unit based on the minimum loss coefficient in the nominal output heating energy, *Energy*, 191 (2020) 116505.
- [21] J. Klingebiel, C. Höges, S. Göbel, P. Bannmüller, T. Boelsen, V. Venzik, C. Vering, D. Müller, Optimal defrost initiation for air-source heat pumps: Evaluating the improvement potential of common defrosting controllers, *Energy*, 324 (2025) 135871.
- [22] Z.Y. Li, W. Wang, Y.Y. Sun, S.Q. Wang, S.M. Deng, Y. Lin, Applying image recognition to frost built-up detection in air source heat pumps, *Energy*, 233 (2021) 121004.
- [23] S.Q. Wang, W.Z. Wei, J.S. Xie, W. Wang, Y.Y. Sun, Z.Y. Li, Y. Lin, C.Y. Huang, S.M. Deng, Space heating performance analysis of air source heat pump integrated with image gray recognition-based defrosting controller, *Applied Thermal Engineering*, 236 (2024) 121536.
- [24] Y.J. Xu, Y. Xie, X.P. Wang, X. Shen, M.J. Song, W. Hang, Study on a novel defrost control method based on the surface texture of evaporator image with gray-level cooccurrence matrix, new

characterization parameter combination and machine learning, *Energy and Buildings*, 292 (2023) 113173.

[25] R. Tang, W.Z. Wei, P. Sun, J.H. Zhao, W. Wang, Y.Y. Sun, S.M. Deng, A novel time-current-temperature difference (T-I-T) defrosting control method for air source heat pump based on outdoor fan operation characteristics in space heating, *Applied Thermal Engineering*, 259 (2025) 124850.

[26] S.Q. Wang, W.Z. Wei, W. Wang, Y.Y. Sun, Z.Y. Li, C.M. Dai, C.Y. Huang, R. Tang, S.M. Deng, Frosting-defrosting performance variations of air source heat pumps with the type of refrigerants: R410A and R32 as an example, *Applied Thermal Engineering*, 262 (2025) 125189.

[27] L. Chen, W.P. Wang, X.Y. Yang, H.C. Liu, R.H. Ou, Dynamic modeling and defrost optimization for air source heat pumps: A deep learning and autoregression approach, *Energy and Buildings*, 322 (2024) 114689.

[28] G. Milev, A. Al-Habaibeh, S. Fanshawe, F.L. Siena, Investigating the effect of the defrost cycles of air-source heat pumps on their electricity demand in residential buildings, *Energy and Buildings*, 300 (2023) 113656.

[29] S. Wang, W. Wei, J. Luo, W. Wang, Y. Sun, Z. Li, C. Huang, R. Tang, S. Deng, A prediction model of the optimal defrosting initiating time for the air source heat pumps with different configurations and operations, *Energy and Buildings*, 326 (2025) 115079.

[30] J.K. Dong, Y.Q. Jiang, Y. Yao, X.D. Zhang, Operating performance of novel reverse-cycle defrosting method based on thermal energy storage for air source heat pump, *Journal of Central South University of Technology*, 18 (2011) 2163-2169.

[31] Z. Long, D. Jiankai, J. Yiqiang, Y. Yang, A novel defrosting method using heat energy dissipated by the compressor of an air source heat pump, *Applied Energy*, 133 (2014) 101-111.

[32] W.Z. Wei, L. Ni, Q. Dong, W. Wang, J.Y. Ye, L.F. Xu, Y.H. Yang, Y. Yao, Experimental investigation on improving defrosting performance of air source heat pump through vapor injection, *Energy and Buildings*, 256 (2022) 111696.

[33] M.L. Qu, L.A. Xia, S.M. Deng, Y.Q. Jiang, Improved indoor thermal comfort during defrost with a novel reverse-cycle defrosting method for air source heat pumps, *Building and Environment*, 45 (2010) 2354-2361.

[34] S.-h. Cao, P.-p. Ming, X. Zhao, Fuzzy comprehensive evaluation of human thermal comfort in simulating natural wind environment, *Building and Environment*, 188 (2021) 107447.

[35] Z. Fang, X. Feng, Z. Lin, Investigation of PMV Model for Evaluation of the Outdoor Thermal Comfort, in: 10TH INTERNATIONAL SYMPOSIUM ON HEATING, VENTILATION AND AIR CONDITIONING, ISHVAC2017, 2017, 2457-2462.

[36] W. Wang, H.R. Di, R. Tang, W.Z. Wei, Y.Y. Sun, C.M. Dai, Effect of supply water temperature on frosting performance of air source heat pump and indoor thermal environment in space heating, *Building and Environment*, 267 (2025) 112258.

[37] Z. Li, W. Wei, W. Wang, Y. Sun, S. Wang, R. Tang, Y. Lin, C. Huang, S. Deng, Determination of the defrosting duration ratio for defrosting performance evaluation of air source heat pump, *Journal of Building Engineering*, 95 (2024) 110305.

[38] Y. Zhang, H. Tang, H. Li, S. Wang, Integration and interaction of next-generation AI-focused data centers with smart grids and district energy systems: The state-of-the-art, opportunities and challenges, *Renewable and Sustainable Energy Reviews*, 224 (2025) 116097.