

Article

Performance Analysis of Novel Direct-Condensation Heating Panels Integrated with Air Source Heat Pump System on Thermal Economy and System Efficiencies

Suola Shao ^{1,2,*}  and Chengcheng Xu ³

¹ School of Civil Engineering and Architecture, Zhejiang Sci-Tech University, Xiasha Education Area, Qiantang District, Hangzhou 310018, China

² Zhejiang Engineering Research Center of Green and Low Carbon Technologies in Buildings, Hangzhou 310018, China

³ College of Civil Engineering and Architecture, Zhejiang University of Water Resources and Electric Power, Hangzhou 310018, China; chengchengxu_seu@163.com

* Correspondence: solar_shao@163.com; Tel./Fax: +86-057186843374

Abstract: As an efficient heating terminal, direct-condensation terminals are increasingly being applied. This study proposes novel direct-condensation heating panels with different aspect ratios, which have been optimized in structure and reduced in copper tube length. To quantify their thermal superiority, the proposed panels' thermal economic performance and system efficiencies are investigated. Compared with previous heating terminals, the vertical temperature difference provided by the proposed panels is reduced by 1.1 ± 0.2 °C, improving thermal comfort. Meanwhile, the system COP is competitive among existing direct-condensation heating terminals. The average heating capacity per cost of the proposed panels reaches 5.4 W/USD, which is 24.1%~46.3% higher than that of previous panels. In addition, the impact of the aspect ratio shows that the panel with a high ratio has advantages in thermal economic performance and system efficiencies. The annual running cost and the system task efficiency of the panel with a ratio of 2.22 are 74.81 USD and 48.5%, respectively, which are 4.7% lower and 0.9% higher than that with a ratio of 0.45, respectively. The developed direct-condensation heating panels help to optimize the heating technology of air source heat pumps, while the evaluation methodology and aspect ratio results are suitable for optimizing other heating terminals.

Keywords: air source heat pump system; direct-condensation; heating panel; thermal analysis; system efficiencies



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1. Introduction

1.1. Background Information

In recent years, the proportion of building energy consumption amongst the total industrial energy consumption has gradually increased, with over half of it being consumed by heating, ventilation, and air conditioning (HVAC) systems [1,2]. The air source heat pump (ASHP) is a very popular heating technology in HVAC systems, and it has been widely used in the winter heating of buildings all over the world.

However, Lin et al. [3] pointed out that the forced convection caused by the fans of the split air conditioner will lead to discomfort, which reduces work efficiency and psychological motivation. Gendelis et al. [4] found that the indoor predicted percentage of dissatisfied (PPD) provided by the split air conditioner often exceeded 15%. The indoor vertical temperature gradient created by the traditional air conditioner varied from 3.2 °C to 5.9 °C [5], exceeding the 3 °C recommended by the ASHRAE standard [6]. Meanwhile, the indoor air velocity distribution is chaotic with in-built fans [7].

1.2. Direct-Condensation Heating Terminal

Correspondingly, fan-less direct-condensation heating terminals have been proposed. Zhang et al. [8] proposed a novel direct-condensation heating radiator. The radiator surface temperatures were uniform, in which the maximum surface temperature difference was lower than 2.0 °C during experiments. The temperature and velocity field of the novel radiator were better than those of the split air conditioner. For the direct-condensation heating radiator, the maximum indoor PPD was 12.64, and the indoor air speed changed from 0.024 m/s to 0.25 m/s. For the split air conditioner, the maximum PPD was 21.46% and the indoor air speed reached 1.6 m/s [9].

Immediately, many direct-condensation heating devices with different structural improvements were proposed. A refrigerant-driven heat pipe radiator was proposed by Xu et al. [10]. The heat pipe radiator was composed of condenser coils and heat pipes. The heating area was expanded by heat pipes, and the heat transfer intensity was enhanced since the inner temperature of heat pipes was nearly constant. To avoid the heating terminal absorbing indoor heat under defrosting conditions, a direct-condensation radiant floor with built-in phase change material was proposed by Zheng et al. [11]. The results showed that the indoor thermal comfort and the heating capacity of the terminal with phase change material were higher than those without phase change material under defrosting conditions. Similarly, Jiang et al. [12] proposed another direct-condensation radiant floor, and the indoor air temperature fluctuations were slowed down under defrosting conditions. The above research indicates that a heat dissipation area should be required for devices relying on natural convection and radiation heat transfer. In addition, thermal storage material can be selected to store heat for defrosting conditions.

In the authors' previous research [13,14], direct-condensation heating terminals with various structural forms were also examined to be competitive in comparison with traditional air conditioners. However, the length of the copper tube was too long in existing heating terminals, which led to increased flow loss and investment. Therefore, it is necessary to develop a direct-condensation heating terminal with reduced copper pipe length while ensuring heat dissipation performance.

In addition to structural improvements, the thermal performance of the heating terminal can also be improved by the device material [15] and fluid materials [16]. For example, Ordu and Der pointed out that PTFE is the most suitable material for manufacturing flexible pulsating heat pipes [17]. Adnan et al. devoted research to the heat transfer characteristics and influencing physical factors of ternary hybrid fluids [18] since the nanofluid contributes to enhancing the heat transfer intensity to a large extent [19]. In addition to nanofluids, R410A, R134A, and R1234yf are commonly used refrigerants in heat pump systems. In terms of costs and the applicable occasions, R410A with an ozone layer destruction potential of 0 and a global warming coefficient of 2025 is suitable for heating terminals with high flow rates and long heating lengths.

Apart from the thermal performance, the system energy and exergy efficiencies are important evaluation indicators to judge whether a heating terminal is worth researching and promoting. When the outdoor temperature is lower than −5 °C, the system coefficient of performance (COP) of the direct-condensation heating radiator is higher than that of the traditional hot water radiator and air conditioner [8]. Wang et al. [20] examined the system seasonal performance factors of the direct-condensation radiation floor heating system. The results showed that the system seasonal performance factors in Xi'an, Beijing, and Chengdu were 3.9%, 1.9%, and 9% higher than those of conventional ASHP systems, respectively. The COP performance of direct-condensation heating terminals has been compared with that of existing heating terminals, while studies on exergy analysis of the direct-condensation heating terminals are relatively few. Dincer et al. [21] pointed out that exergy efficiency was a key parameter in judging the effectiveness of energy conversion. Larry et al. [22] investigated the exergetic performance of an integrated cascade trigeneration power system, offering valuable insights into energy efficiency and process optimization. Consequently,

the system energy and exergy efficiencies of the direct-condensation heating terminal should be investigated comprehensively.

Therefore, although direct-condensation heating terminals have advantages over traditional heating terminals, there is still room for optimization in terms of structure, thermal performance, and other aspects.

1.3. Objective of This Study

In this paper, a novel direct-condensation heating panel is developed. Compared with previous panels, the copper tube length is cut down while heat dissipation is ensured. To quantify the performance superiority, the thermal economy performance and the system efficiencies of the developed panel are compared with those of previous direct-condensation heating terminals. Meanwhile, the influence of the aspect ratio on panel performance is investigated through experimental data. The proposed direct-condensation heating panel can be applied to ASHP systems to promote energy savings and emission reductions in winter space heating. The length–width ratio investigation is beneficial for the structural optimization of other heating terminals.

2. Experimental Procedure

2.1. Experimental Site Description

Figure 1 shows the experimental site and the heating system. A standard air enthalpy laboratory was selected as the experimental site. The laboratory temperature can be adjusted between $-20\text{ }^{\circ}\text{C}$ and $60\text{ }^{\circ}\text{C}$, while the relative humidity can be adjusted between 20% and 90% with air conditioning systems. Each chamber of the laboratory is $4\text{ m} \times 4\text{ m} \times 3\text{ m}$ ($L \times W \times H$). To avoid the influence of the indoor air conditioning system on the experimental velocity field, a test chamber with the size of $3.5\text{ m} \times 3.5\text{ m} \times 2.5\text{ m}$ ($L \times W \times H$) was built. The rated heat range of the heating terminal allowed for testing in the experimental site is $1.142\text{ kW} \sim 4.981\text{ kW}$.

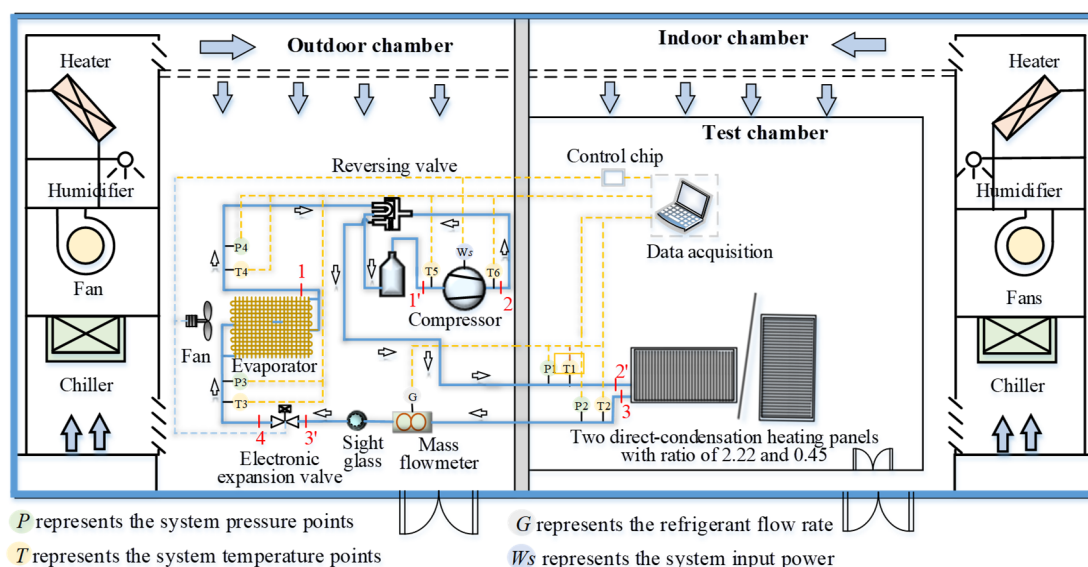


Figure 1. Experimental site for the ASHP system with direct-condensation heating panels using two length–height ratios.

2.1.1. System Description

The experimental heating system comprises an ASHP unit with a customized heat of 3.52 kW and two direct-condensation heating panels with length–height ratios of 0.45 and 2.22. The parameters of the ASHP unit are presented in Table 1.

Table 1. The information of the ASHP unit.

Components of the ASHP Unit	Parameters
Evaporator	Size: 0.72 m × 0.48 m × 0.23 m (L × H × W) Refrigerant: R410A Outdoor Fan: speed range of 0~900 r/s
Electronic expansion valve	Opening degree: 0~100% stepless regulation
Compressor	Displacement: 10.2 cm ³ /rev Frequency: 0~100 Hz
Gas liquid separator	Type: LG-GST102MAA
Reversing valve	Type: SHF-50-79(P)
Sight glass	Type: DanFoss SGH6 014-1660
Flowmeter	Type: DMF-1 mass flowmeter, Range: 0~100 kg/h.

The outdoor air parameters were regulated by the air conditioning system in the lab. The condensation pressure of the experimental ASHP system could be controlled by adjusting the outdoor fan speed, the compressor regulation frequency, and the opening degree of the EEV in the ASHP unit. Correspondingly, the heat source temperature for the direct-condensation heating panels with different length–height ratios can be controlled to remain consistent.

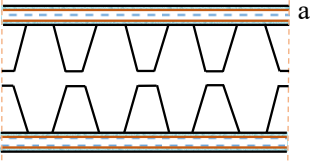
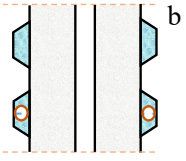
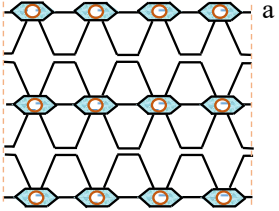
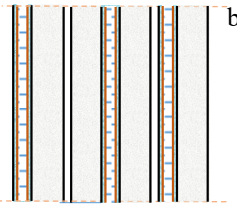
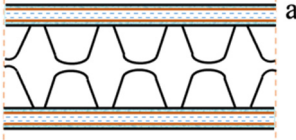
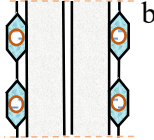
2.1.2. Detailed Heating Panel Description

As shown in Figure 2, the heating panel has two steel shells with a plurality of channels. Copper tubes are arranged in the channel, while thermal storage material water is filled in the gaps of the channels. Behind each steel shell are vertically welded composite fins. Due to the height variations of people in different activity states, the longest panel dimension is consistent with the regular room bed length of 2 m, while the shortest panel dimension is in line with the human sitting height of 0.9 m. Accordingly, the minimum length–height ratio is 0.45, and the heating panel is 0.9 m × 0.1 m × 2 m (L × W × H) with copper tubes arranged horizontally. The maximum length–height ratio is 2.22, and the panel is 2 m × 0.1 m × 0.9 m (L × W × H) with copper tubes arranged vertically. The surface area of the steel shells of two panels with different length–height ratios is the same. Meanwhile, the panels with different aspect ratios have the same heat transfer area for the same total length of vertical welded fins. Compared with previous heating terminals, the structural improvements of the developed panels are listed in Table 2. Compared with previous heating terminals [13,14], the copper tubes are laid out in intervals, reducing the copper tube length without affecting the heat exchange area. Meanwhile, the back steel plate of the panel is designed to be flat so that the fins are welded with line contact rather than point contact [14].

2.2. Testing Scheme

The testing scheme is developed using the testing method for radiators (GB/T 13754-2017 [23]) and the testing standard for air conditioners (GB/T 7725-2022 [24]).

Table 2. The comparison between the developed panels and previous direct-condensation heating terminals.

Type		Channel Version ($A \times \chi$)	Copper Tube Layout Style	Fins Layout Style	Universal Fixed Dimensions
Developed panel	Information	Isosceles trapezoidal ($128 \text{ mm}^2 \times 52 \text{ mm}$)	Interval piping	Wire welding	1. Diameter of copper tube: $\varphi 6.35 \times 0.5 \text{ mm}$ 2. Composite fin size: $3 \text{ cm} \times 1 \text{ cm}$ (Length $\times$ Span)
	Ratio of 0.45				
	Profile diagram				
	Ratio of 2.22				
Previously proposed panel	Information	Hexagonal ($128 \text{ mm}^2 \times 48.8 \text{ mm}$)	Full channel piping	Wire welding	1. Diameter of copper tube: $\varphi 6.35 \times 0.5 \text{ mm}$ 2. Composite fin size: $3 \text{ cm} \times 1 \text{ cm}$ (Length $\times$ Span)
	Type A [13]				
	Profile diagram				
	Information	Hexagonal ($128 \text{ mm}^2 \times 48.8 \text{ mm}$)	Full channel piping	Spot welding	
Previously proposed panel	Type B [14]				1. Diameter of copper tube: $\varphi 6.35 \times 0.5 \text{ mm}$ 2. Composite fin size: $3 \text{ cm} \times 1 \text{ cm}$ (Length $\times$ Span)
	Profile diagram				

Notes: ^a represents the transverse section diagram, ^b represents the longitudinal section diagram.

The measurement points of the system pressures, the system temperatures, the system refrigerant flow rates, and the system input power are arranged in Figure 1. The location of the indoor air temperature measurement points and panel temperature measurement points are shown in Figure 3. The anemometers and humidity auto meter are located in the center of the laboratory. The indoor air reference temperature point is located at a height of 0.75 m in the center of the laboratory. Seven measuring points are located at the center of each enclosure structure to test the surface temperature while 29 measuring points are arranged in the indoor laboratory to explore the indoor air temperature distribution. In Figure 3b, 20 measuring points are set to test the surface temperatures of the panel. Meanwhile, each heating panel has one temperature measuring point placed in the water layer and three measuring points arranged at the fin end. To ensure the accuracy of the measurement, all measuring points are wrapped in aluminum foil to improve the thermal conductivity between the points and the heating panel, making the thermocouple reading closer to the actual temperature. Meanwhile, when the deviation between the temperature measured 6 times within 30 min and the mean value is less than $\pm 0.2\text{ }^{\circ}\text{C}$, the pressure deviation is less than $\pm 0.2\text{ kPa}$, and the refrigerant flow deviation is less than $\pm 1\%$, the experimental measurement values are considered stable and thus recorded.

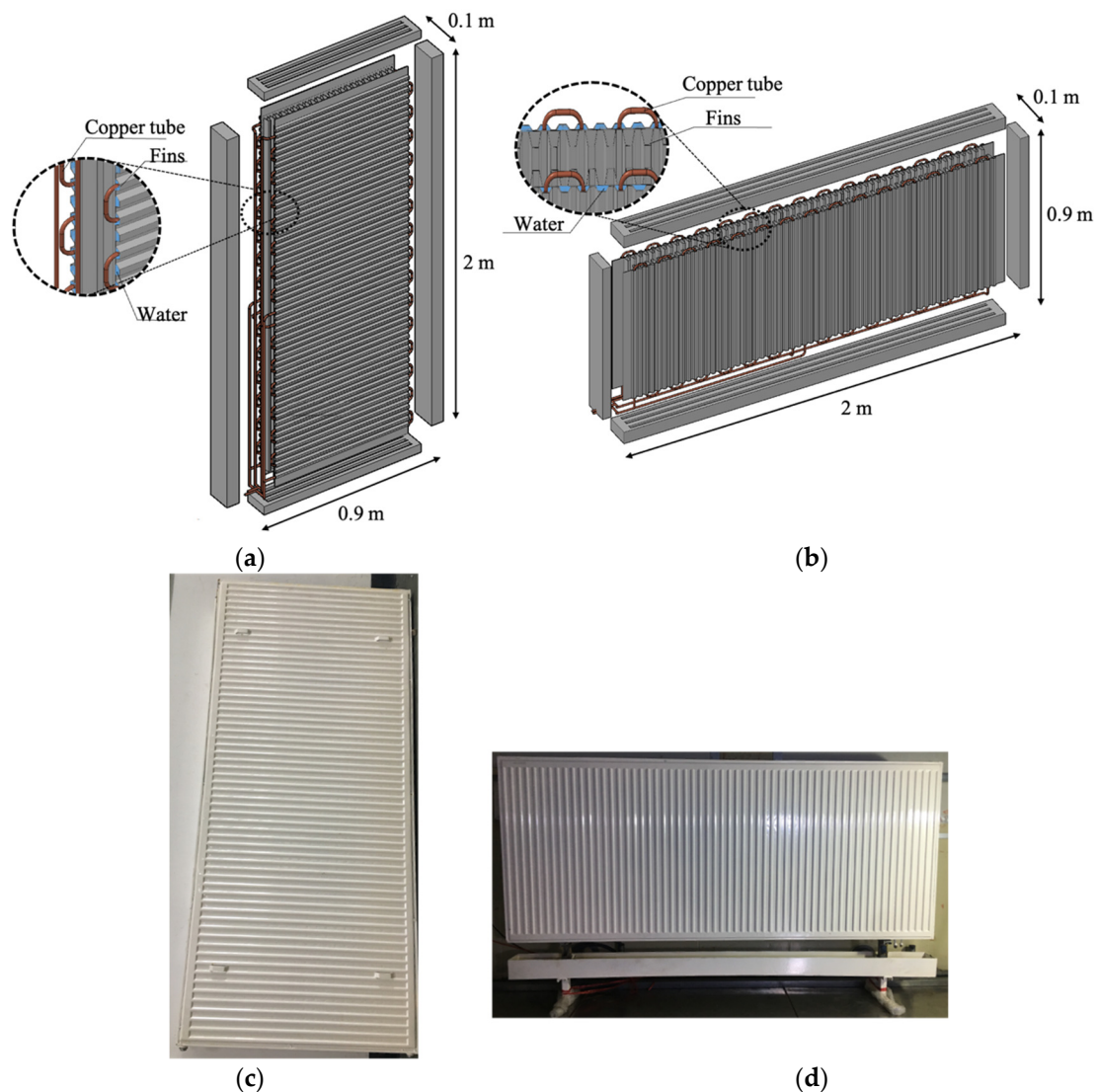


Figure 2. The developed direct-condensation heating panels. (a) Sketch map of the panel with a length–height ratio of 0.45. (b) Sketch map of the panel with a length–height ratio of 2.22. (c) Physical map of the panel with a ratio of 0.45. (d) Physical map of the panel with a ratio of 2.22.

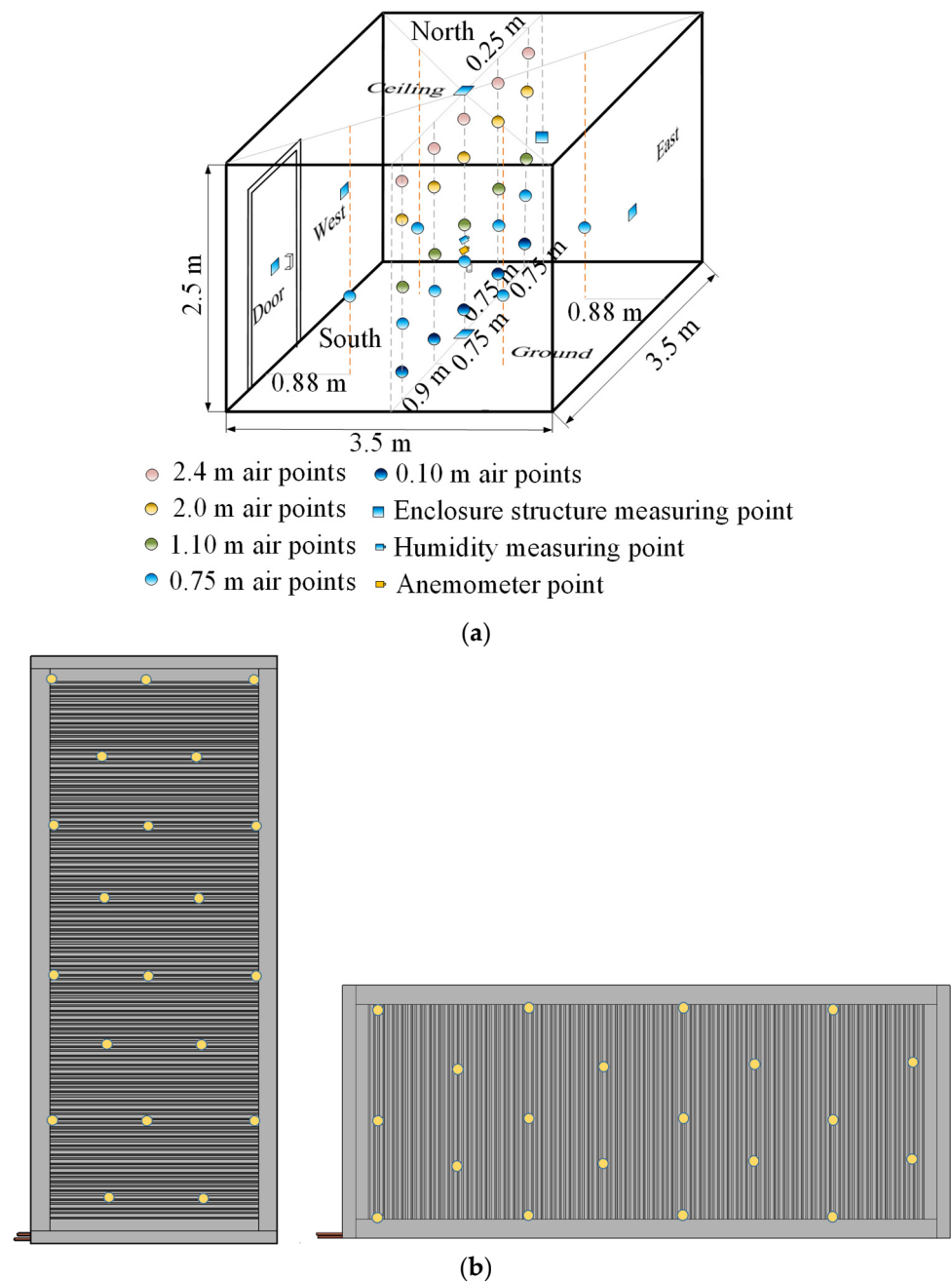


Figure 3. The arrangements of experimental measurements. (a) Measuring points arrangement in the test chamber. (b) Measuring points arrangements of the two panels.

2.3. Uncertainty Analysis

Measurement uncertainties originate from gross, systematic, and random errors. Gross error is caused by human negligence in data reading and recording, which can be eliminated in this paper. The system error originates from the limited precision of the instruments and the improper use of the instrument. In this paper, improper use is avoided by improving the test methods and standardizing the use of the instrument. Although random error cannot be eliminated, it is reduced by multiple measurements. In the experiments, the measurements were performed six times. The uncertainties of direct measurements originating from random error and the limited precision of the instruments are calculated as follows [25]:

$$\gamma_d = \gamma_{ra} + \gamma_{ins} \quad (1)$$

$$\gamma_{ra} = \frac{\sqrt{\sum_{i=1}^n (x_i - \bar{x})^2}}{\sqrt{n(n-1)}} \quad (2)$$

where γ_d is the uncertainties of direct measurements, %. Γ_{ra} is the random error, %. Γ_{ins} is the system error caused by instrument precision, %. $\bar{x}$ is the mean value.

The uncertainties of indirect measurements are given by the square root of the uncertainties of direct measurements [26], which is defined as follows:

$$\gamma_{ind} = \sqrt{\left(\frac{\gamma_{d_1}}{d_1}\right)^2 + \left(\frac{\gamma_{d_2}}{d_2}\right)^2 + \cdots + \left(\frac{\gamma_{d_m}}{d_m}\right)^2} \quad (3)$$

where γ_{ind} is the uncertainties of indirect measurements, %. $D_1 \sim d_m$ are the direct measurements associated with the indirect measurements.

The uncertainties of the measurements are listed in Table 3.

Table 3. Ranges of experimental measurements.

Parameters	Instrument Precision	Experimental Value Range	Error Range
T	$\pm 0.5^\circ\text{C}$	$12^\circ\text{C} \sim 80^\circ\text{C}$	$\pm 0.63\% \sim 4.16\%$
P	0.1% FS	4.0 bar $\sim$ 29.0 bar	$\pm 0.10\%$
RH	$\pm 1\%$	20% $\sim$ 60%	$\pm 1.67\% \sim 5\%$
G	$\pm 0.002\text{ kg/h}$	49 kg/h $\sim$ 83 kg/h	$\pm 0.0025\% \sim 0.004\%$
W_s	$\pm 0.01\text{ kW}$	0.5 kW $\sim$ 1.4 kW	$\pm 0.71\% \sim 2.00\%$
Q	-	2.3 kW $\sim$ 3.5 kW	$\pm 0.64\% \sim 4.16\%$
COP	-	2.0 $\sim$ 4.8	$\pm 0.96\% \sim 4.62\%$

3. Evaluation Methodology

3.1. Thermal Comfort Evaluation

The indoor thermal indexes (PMV-PPD), the local dissatisfaction rate caused by vertical air temperature difference (PD_2), and floor surface temperature (PD_3) are taken to evaluate the thermal comfort of the novel heating panels with two length–height ratios.

The thermal indexes PMV-PPD, PD_2 , and PD_3 are calculated as follows [27]:

$$\text{PMV} = [0.303 \times \exp(-0.036M) + 0.0275] \times S \quad (4)$$

$$S = M - W - C_c - R - E \quad (5)$$

$$C_c = f_{cl} \times h_c \times (t_{cl} - t_{ai}) \quad (6)$$

$$R = 3.96 \times 10^{-8} \times f_{cl} \times [(t_{cl} + 273.15)^4 - (t_r + 273.15)^4] \quad (7)$$

$$E = 0.0014 \times M \times (34 - t_{ai}) + 0.0173 \times M \times (5.867 - P_{ai}) + 0.42 \times (M - W - 58.2) + E_{dif} \quad (8)$$

$$E_{dif} = 3.05[5.377 - 0.007(M - W) - P_{ai}] \quad (9)$$

$$f_{cl} = 1 + 0.3 \times I_{cl} \quad (10)$$

$$t_{cl} = 35.7 - 0.028(M - W) - I_{cl}(R + C) \quad (11)$$

$$h_c = 2.38 \times (t_{cl} - t_{ai})^{0.25} \quad (12)$$

$$\text{PPD} = 100 - 95 \times \exp(-0.03353 \times \text{PMV}^4 - 0.2179 \times \text{PMV}^2) \quad (13)$$

$$\text{PD}_2 = 100 / 1 + \exp(5.76 - 0.856 \Delta t_{ai,v}) \quad (14)$$

$$\text{PD}_3 = 100 - 94 \times \exp(-1.378 + 0.118 \times t_g - 0.0025 \times t_g) \quad (15)$$

where M is the human energy metabolic rate, $\text{W} \cdot \text{m}^{-2}$; W is the mechanical work, $\text{W} \cdot \text{m}^{-2}$; C_c is the convection heat transfer between the human body and the indoor air, $\text{W} \cdot \text{m}^{-2}$; R is the radiation heat transfer between the human body and the indoor chamber, $\text{W} \cdot \text{m}^{-2}$; E

is the evaporation heat of the human body, $W \cdot m^{-2}$; E_{dif} is the skin diffusion evaporation loss, $W \cdot m^{-2}$; P_{ai} is the water vapor pressure of indoor air, calculated by ref. [28], Pa; f_{cl} is the clothing area coefficient; I_{cl} is the clothing insulation, $0.14 m^2 \cdot K \cdot W^{-1}$ in winter; h_c is the indoor heat transfer coefficient in natural convection, calculated by ref. [29], $W \cdot m^{-2} \cdot K^{-1}$; and $\bar{t}_r$, t_{cl} , t_{ai} , $\Delta t_{ai,v}$, and t_g are the mean radiant temperature, the clothing surface temperature, the indoor air temperature, the vertical air temperature difference, and the ground surface temperature, respectively, $^{\circ}C$.

3.2. Thermal Economic Evaluation

The heating capacities of the direct-condensation heating panels (Q), the refrigerant flow loss (ΔP), the system initial capital cost (C_{ICC}), the heating capacity under per cost (C_{pr}), and the system cost-effectiveness ratio (C_{CER}) are used to examine the thermal economy performance of the proposed heating panels.

The ΔP and heating capacities of the heating terminals are calculated as follows [30]:

$$\Delta P = P_{in} - P_{out} \quad (16)$$

$$Q = G \times (h_{in} - h_{out}) \quad (17)$$

$$Q_{rad} = A \times \sigma \times \varepsilon \times (T_s^4 - T_{AUST}^4) \quad (18)$$

$$t_{AUST}^* = \frac{\sum_{i=1}^7 A_i t_i}{\sum_{i=1}^7 A_i} \quad (19)$$

where Q and Q_{rad} are the total heating capacity and the radiation heating capacity, respectively, W ; G is the refrigerant flow rate, $kg \cdot s^{-1}$; h_{in} and h_{out} are the refrigerant inlet and outlet enthalpy, respectively, $kJ \cdot kg^{-1}$; σ is the Stephen Boltzmann constant $5.67 \times 10^{-8} W \cdot m^{-2} \cdot K^{-4}$; ε is the radiant surface emissivity; A and A_i are the apparent surface area of the heating panel and the area of the enclosure structures, respectively, m^2 ; and T_s and t_i are the average surface temperature of the panel and the temperature of the enclosure structure, respectively, $^{\circ}C$.

The system C_{ICC} and C_{pr} are calculated as follows:

$$C_{ICC} = C_{SYS} + C_{ADD} \quad (20)$$

$$C_{ADD} = 15\% C_{ICC} \quad (21)$$

$$C_{pr} = \frac{Q}{C_{ICC}} \quad (22)$$

where C_{SYS} is the cost of the system components, USD. C_{ADD} is the additional costs of valves, installation, and so on, USD.

The system C_{CER} corresponds to the energy consumption (C_{ws}) of the total heating capacity of the heating terminal, which can be defined as follows:

$$C_{CER} = \frac{C_{ws}}{Q} \quad (23)$$

3.3. System Efficiency Evaluation

The system coefficient of performance (COP) is an evaluation indicator of system energy efficiency, while the exergy destruction of the system components, the system efficiency (ϕ), and the task efficiency (η) are used to investigate the exergy efficiencies of the ASHP system with direct-condensation heating panels using different length–width ratios.

3.3.1. Energy Efficiency

The COP is given as follows:

$$\text{COP} = \frac{Q_{\text{DCRP}}}{W_s} \quad (24)$$

where W_s is the input power of the system, W.

3.3.2. Exergy Efficiency

The T - s diagram of the ASHP system with direct-condensation heating panels in the steady-state test is shown in Figure 4. The cycle points in the T - s diagram are shown in Figure 1. Based on the T - s diagram, the system exergy destruction can be calculated with the following assumptions:

- The system keeps a steady operation without chemical reactions;
- The process from 3' to 4 is isenthalpic (i.e., $h_{3'} = h_4$);
- The outdoor environment is considered as the dead state [31];
- The power consumption of the outdoor fan is ignored in comparison with that of the compressor;
- The pressure drop in the evaporator and condenser is ignored;
- Air is treated as an ideal gas.

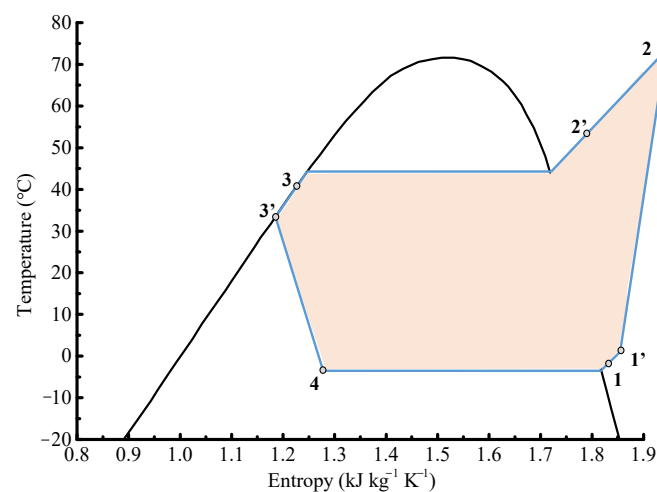


Figure 4. The T - s diagram of the ASHP system with direct-condensation heating panels.

The exergy transfer rate of heat and work (Ex_{heat} and Ex_{work}) are defined as follows [32]:

$$Ex_{\text{heat}} = \sum \left(1 - \frac{T_0}{T_{\text{ai}}}\right) Q \quad (25)$$

$$Ex_{\text{work}} = W_s \quad (26)$$

where T_0 is the ambient temperature, °C.

The exergy destruction for each component of the system is defined as follows:

$$Ex_{\text{des,eva}} = G[(h_4 - h_1) - T_0(s_4 - s_1)] \quad (27)$$

$$Ex_{\text{des,com}} = W_s + G[(h_{1'} - h_2) - T_0(s_{1'} - s_2)] \quad (28)$$

$$Ex_{\text{des,con}} = G[(h_{2'} - h_3) - T_0(s_{2'} - s_3)] - Q\left(1 - \frac{T_0}{T_{\text{ai}}}\right) \quad (29)$$

$$Ex_{\text{des,EEV}} = G[(h_{3'} - h_4) - T_0(s_{3'} - s_4)] \quad (30)$$

$$Ex_{\text{des,pl}} = G[(h_1 - h_{1'}) - T_0(s_1 - s_{1'}) + (h_2 - h_{2'}) - T_0(s_2 - s_{2'}) + (h_3 - h_{3'}) - T_0(s_3 - s_{3'})] \quad (31)$$

The system exergy efficiency φ_{sys} employed here is associated with the coefficient of performance in the reversed Carnot cycle (COP_{rev}), which can be expressed as follows [33]:

$$\varphi_{\text{sys}} = \frac{\text{Recovered Exergy}}{\text{Supplied Exergy}} = \frac{\sum (1 - \frac{T_0}{T_{\text{ai}}})Q}{W_s} = \frac{\text{COP}}{\text{COP}_{\text{rev}}} \quad (32)$$

$$\text{COP}_{\text{rev}} = \frac{T_{\text{ai}}}{T_{\text{ai}} - T_0} \quad (33)$$

The task efficiency (η) is the ratio between COP and the ideal Carnot heat pump performance ($\text{COP}_{\text{rev-euq}}$), which is expressed as follows:

$$\eta = \frac{\text{COP}}{\text{COP}_{\text{rev-euq}}} \quad (34)$$

$$\text{COP}_{\text{rev-euq}} = \frac{T_{\text{con}}}{T_{\text{con}} - T_{\text{eva}}} \quad (35)$$

where T_{eva} and T_{con} are the evaporation and condensation temperatures, respectively, °C.

4. Results and Discussion

4.1. Thermal Comfort of the Proposed Heating Panels

4.1.1. Comparison with Previous Heating Terminals

The indoor temperature distributions of the proposed panels are shown and compared in Figure 5. As t_{ai} increases from 16 °C to 22 °C, the indoor vertical temperature difference provided by the proposed heating panels ranges from 2.0 °C to 3.8 °C. Although the panel with a high aspect ratio has larger indoor temperature differences due to heat exchange with more cold air, the temperature distributions provided by the proposed heating panels all meet the indoor thermal comfort requirements of BS EN ISO 7730-2005 [29]. In the working area (0.1 m~2 m), the vertical temperature differences provided by the developed heating panels are all below 3 °C, and the indoor horizontal temperature differences are less than 2.4 °C, as shown in Figure 5b.

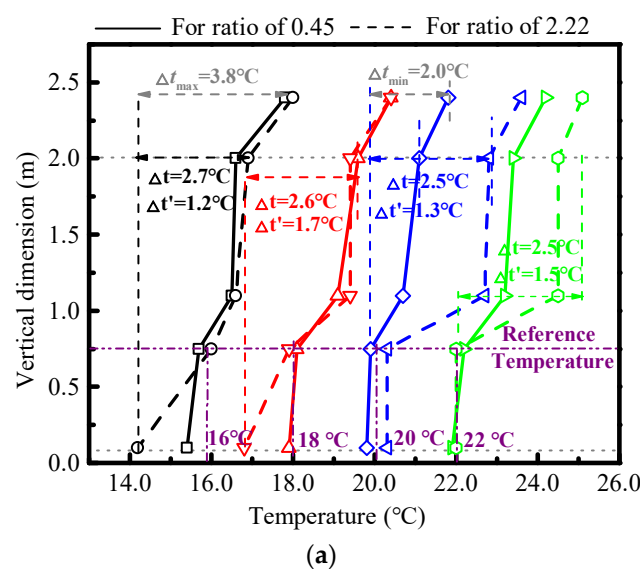


Figure 5. Cont.

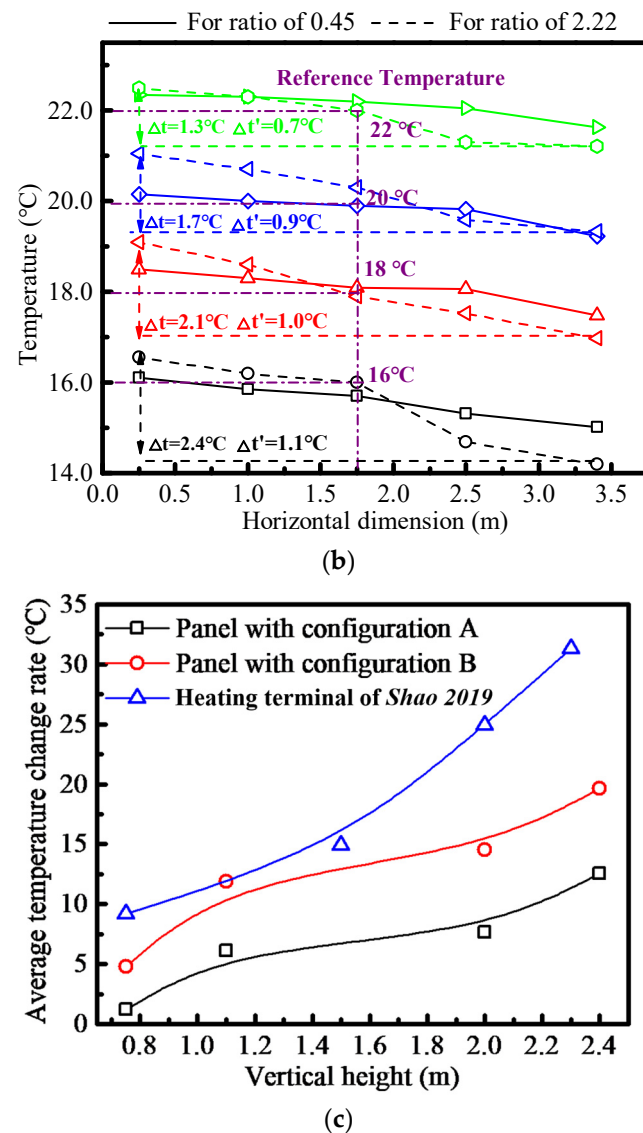


Figure 5. Indoor temperature analysis. (a) Vertical temperatures of the proposed panels. (b) Horizontal temperatures of the proposed panels. (c) Comparisons of temperature change rate.

Meanwhile, the indoor temperature distributions provided by the proposed heating panels are more uniform than that of a previous heating terminal. As shown in Figure 5c, compared with the indoor air temperature at a height of 0.1 m, the indoor temperature growth rate increased from 9.18% to 31.3% as the indoor height increased from 0.75 m to 2.3 m in ref. [14]. However, the temperature growth rate of the proposed heating panel with the ratio of 0.45 changes from 1.2% to 12.5% and changes from 4.8% to 19.6% with the ratio of 2.22 as the height increases from 0.75 m to 2.4 m. The indoor temperature growth rate of the proposed heating panels is 15.2% lower than that provided by the previous heating terminal.

4.1.2. Comparison of the Proposed Heating Panels with Two Aspect Ratios

The effects of the aspect ratios of the proposed heating panels on indoor thermal comfort are analyzed in detail in Figure 6 and Table 3. The comparisons are conducted under the same heating conditions, in which the inlet t_{con} is $41.2 \pm 0.1^\circ\text{C}$, t_{ai} is 18°C , and the ambient temperature (t_0) is 7°C . In Figure 6, the indoor PMV ranges from -1.08 to 0.58 , with an average value of -0.27 for the heating panel with a ratio of 0.45. For the heating panel with a ratio of 2.22, the indoor PMV changes from -0.48 to 1.10 , with an average

value of 0.46. The indoor *PMV* indicates that the direct-condensation heating panel with a higher ratio is more suitable for people who need a warmer environment.

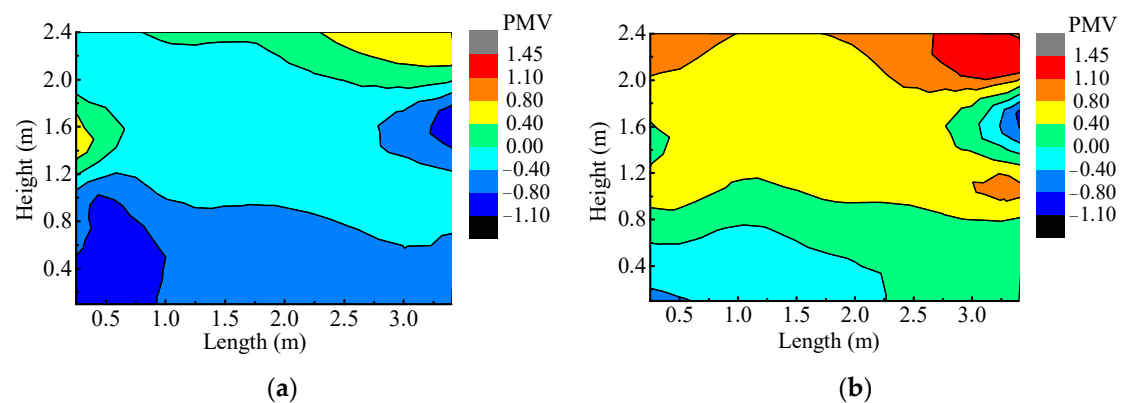


Figure 6. Comparisons of indoor *PMV* variations under typical heating conditions. (a) The panel with the aspect ratio of 0.45. (b) The panel with the aspect ratio of 2.22.

The indoor *PPD*, the *PD*₂, and the *PD*₃ provided by the heating panels with two aspect ratios are presented in Table 4. The *PD*₃ and the ranges of the indoor *PPD* are similar for the two heating panels. Additionally, 35% of the indoor *PPD* under the ratio of 2.22 meets the requirements of thermal category A, which is 10% higher than that under the ratio of 0.45. However, 15% of the indoor *PPD* under the ratio of 2.22 is out of three thermal categories and only 5% of the indoor *PPD* under the ratio of 0.45 is out of three thermal categories. Meanwhile, the *PD*₂ indicates that the thermal performance of the panel with a ratio of 0.45 is superior to that with a ratio of 2.22.

Table 4. Summaries of indoor comfort indicators in the typical heating case.

Index	Position	Ratio	Range (%)	Proportions of Thermal Categories		
				A	B	C
PPD	H: 0.1 m~2 m	0.45	5.0~29.5	25%	35%	35%
		2.22	5.0~28.3	35%	20%	30%
<i>PD</i> ₂	H: 0.1 m~2 m	0.45	1.3~5.0	80%	20%	0
		2.22	2.8~20.6	20%	40%	20%
<i>PD</i> ₃	-	0.45	9.9	100%	-	-
		2.22	9.6	100%	-	-

4.2. Heat Transfer Performance and Economy of the Proposed Heating Panels

4.2.1. Comparison with Previous Heating Terminals

The heating capacities of the proposed heating panels and previous heating terminals are compared at *t*₀ of −7 °C and *t*_{ai} of 18 °C. As shown in Figure 7, the *Q* values of the direct-condensation heating panels are much higher than that of the previous heating terminal in ref. [13]. The heating capacity difference between the novel panels and the heating terminal in ref. [14] is not significant under the same heating conditions. However, the heating capacity under per cost (*C*_{pr}) of the proposed direct-condensation heating panels is competitive among existing heating terminals. As listed in Table 5, the *C*_{ICC} of the ASHP system with the proposed novel heating panels is 470.0 USD, while the *C*_{ICC} of the system with the previous heating terminals in ref. [13] and ref. [14] are 592.7 USD and 435.9 USD, respectively. Correspondingly, the average *C*_{pr} of the novel panels with two ratios is 5.4 W/USD at the inlet *t*_{con} of 40.3 ± 0.3 °C, which is 24.1% higher than that of the terminal in ref. [13] and 46.3% higher than that of the terminal in ref. [14] under the same heating conditions. The comparisons of the *Q* and *C*_{pr} indicate that the thermal economy performance of the proposed novel panels is superior to that of previous heating terminals.

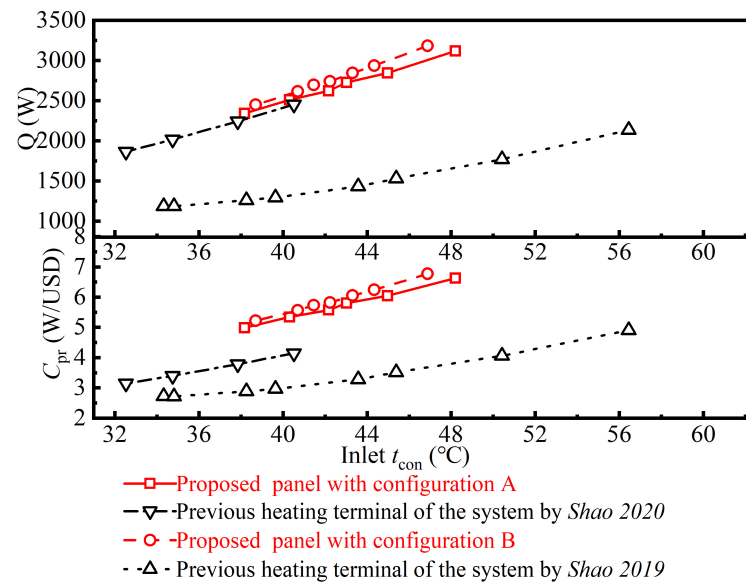


Figure 7. Comparisons of total heating capacities between the proposed direct-condensation heating panels and previous heating terminals [13,14].

Table 5. The C_{ICC} of the ASHP system with proposed novel panels and previous heating terminals.

Type	C_{sys}					C_{ADD}	C_{ICC}
	Components	Heat Pump Unit	Copper Tube	Steel Panel ^a	Fin ^b		
	Unit Price	288.5 USD/Unit	1.44 USD/m	3.60 USD/m ²	2.16 USD/m		
System with novel heating panels	Parameters	1 unit	50 m	$4 \times 2 \times 0.9 \text{ m}^2$	$2 \times 60 \times 0.07 \times 0.72 \text{ m}^2$	15% C_{ICC}	$C_{sys} + C_{ADD}$
	Total price			399.5 USD		70.5 USD	470.0 USD
System with the previous heating terminal in ref. [13]	Parameters	1 unit	117 m	$6 \times 1.6 \times 0.9 \text{ m}^2$	$144 \times 0.07 \times 0.72 \text{ m}^2$	15% C_{ICC}	$C_{sys} + C_{ADD}$
	Total price			503.8 USD		88.9 USD	592.7 USD
System with the previous heating terminal in ref. [14]	Parameters	1 unit	40 m	$4 \times 0.9 \times 1.6 \text{ m}^2$	$50 \times 0.07 \times 1.4 \text{ m}^2$	15% C_{ICC}	$C_{sys} + C_{ADD}$
	Total price			370.5 USD		65.4 USD	435.9 USD

Notes: ^a Calculation formula for the steel panel represents the number of the steel plates multiplied by panel length then multiplied by panel height. ^b Calculation formula of the fin represents the fin number multiplied by the fin length then multiplied by the fin height.

4.2.2. Comparison of the Proposed Heating Panels with Two Aspect Ratios

The effects of the length–width ratio on the thermal performance of the proposed heating panels under different inlet condensation temperatures are investigated in Figures 8 and 9. Although the total Q of the panel with a higher aspect ratio is superior to that with a lower aspect ratio in Figure 7, the radiation heating capacity (Q_{rad}) of the panel with a ratio of 0.45 is slightly higher than that with a ratio of 2.22 under the same heating conditions in Figure 8. The proportion of Q_{rad} to total Q ranges from 14.8% to 18.0%, with an average value of 16.7% for the panel with a ratio of 2.22. For the panel with a ratio of 0.45, the proportion of Q_{rad} ranges from 16.8% to 20.7%, with an average value of 18.6%. The weakening of the total Q of the panel with a lower aspect ratio is due to the rise in hot air flow, which leads to a decrease in the average heat transfer temperature difference between the panel and the indoor air. The higher proportion of Q_{rad} indicates that denser or higher fins are required for the panel with a lower aspect ratio to compensate for the reduced heat dissipation led by the aspect ratio difference.

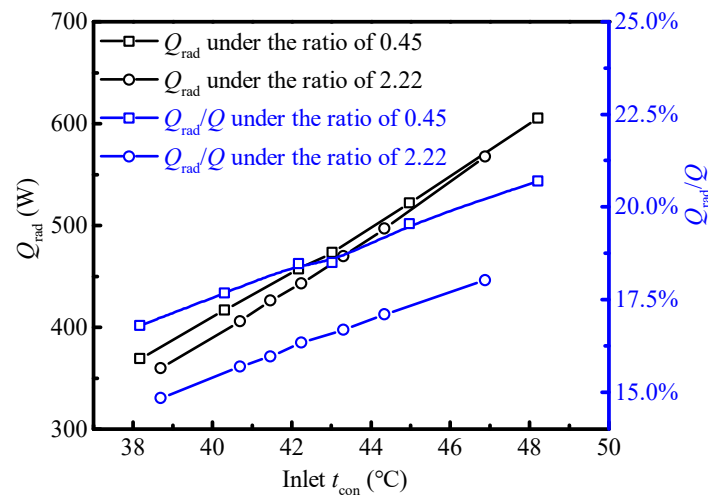


Figure 8. Effects of the aspect ratio on the radiation heating capacities of the proposed heating panels under different inlet t_{con} .

The flow losses of the direct-condensation heating panel with a lower aspect ratio are less than those with a higher aspect ratio. As shown in Figure 9a, the ΔP of the heating panel with a ratio of 2.22 increases from 128 kPa to 149 kPa, while the ΔP of the heating panel with a ratio of 0.45 changes from 109 kPa to 130 kPa. The average deviation of ΔP of the two panels is 17 kPa. As exhibited in Figure 9b, the average Δt caused by ΔP is 2.27 °C for the panel with a ratio of 2.22, and the average temperature gradient rate ($\Delta t/\text{inlet } t_{con}$) is 5.34%. For the panel with a ratio of 0.45, the average Δt caused by ΔP is 1.96 °C, while the average temperature gradient rate is 4.61%. The larger pressure loss not only results in more ineffective heat energy but also makes more liquid refrigerant flash into the gaseous refrigerant, leading to an increased proportion of gaseous refrigerant after EEV throttling. To guarantee the heat demand, the system input power must be increased, which is unfavorable for the long-term operation of the system.

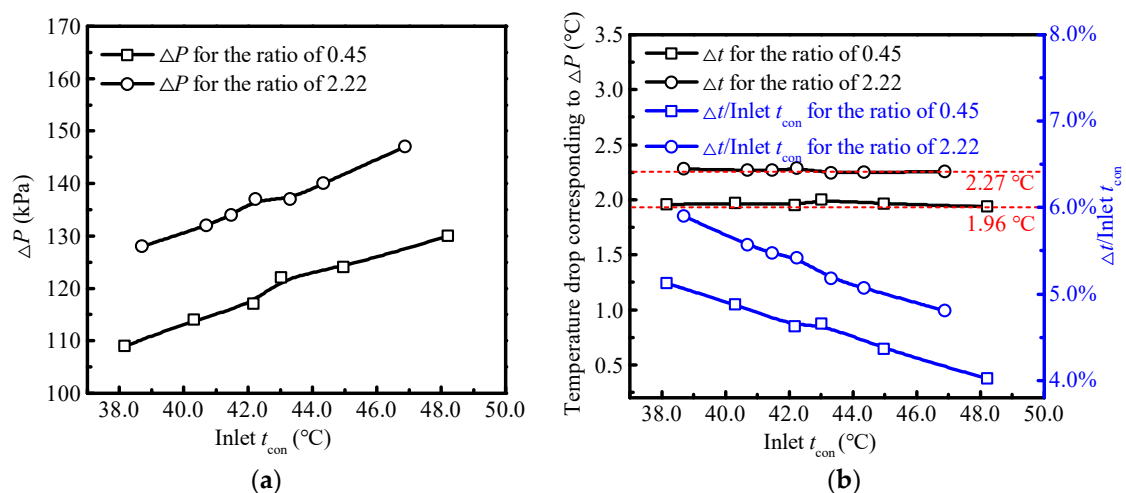


Figure 9. Effects of the aspect ratio on the flow characteristics of the proposed heating panels under different inlet t_{con} . (a) Variations of ΔP . (b) Variations of Δt corresponding to ΔP .

The system C_{CER} is used to examine the economic performance differences of the proposed heating panels with two aspect ratios. A total of 16 cases are chosen, in which t_{ai} is maintained at 18 °C while t_0 changes from −15 °C to 10 °C. As shown in Figure 10, the C_{CER} of the heating panel with a ratio of 2.22 ranges from 0.018 USD/kWh to 0.034 USD/kWh, with an average value of 0.0269 USD/kWh. The C_{CER} of the heating panel with a ra-

tio of 0.45 ranges from 0.019 USD/kWh to 0.036 USD/kWh, with an average value of 0.0282 USD/kWh.

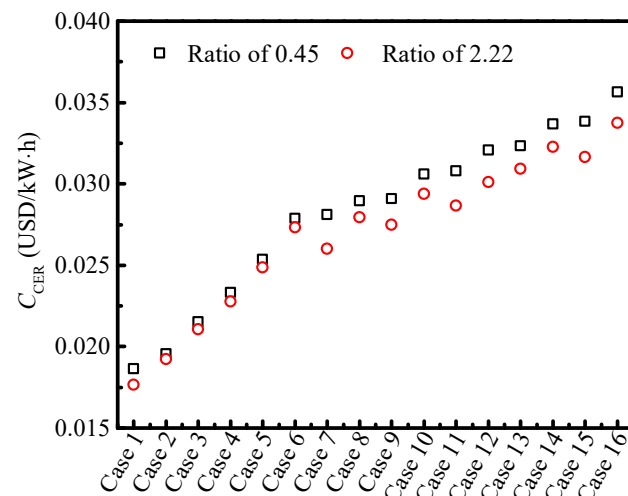


Figure 10. System C_{CER} comparisons of direct-condensation heating panels with two aspect ratios.

As shown in Figure 11, the total heating load of a 25 m² residential room is 2776.9 kWh during the heating period [34]. Based on the average C_{CER} of the ASHP system with the heating panels using two aspect ratios, the annual running cost for the heating panel with a ratio of 0.45 is 78.34 USD, while it is 74.81 USD for the panel with a ratio of 2.22. The deviation of the annual running cost between the two aspect ratios is 4.7%.

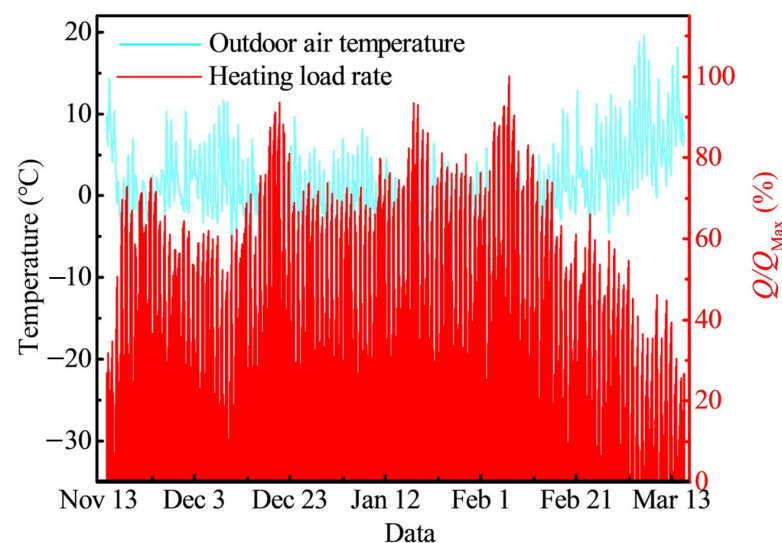


Figure 11. Hourly load rates of a residential room with an area of 25 m² [34].

As mentioned above, the indoor thermal comfort and thermal economy performances of the proposed novel direct-condensation heating panels are superior to those of previous direct-condensation heating terminals. In addition, the proposed heating panel with a high aspect ratio is more competitive than that with a low aspect ratio for thermal economy performance, whereas the panel with a low aspect ratio is more prominent in reducing flow losses.

4.3. System Efficiency Investigations of the Proposed Heating Panels

4.3.1. System COP Comparisons with Previous Heating Terminals

The system COP of the heating panels is examined in Figure 12. As shown in Figure 12, the COP deviations of the proposed heating panels are higher than those of previous heating terminals. Under t_0 of 7 °C, the system COP of the proposed panel with a ratio of 0.45 decreases from 4.05 to 2.48 as the inlet t_{con} increases from 42.5 °C to 55.0 °C. The COP of the panel with a ratio of 2.22 changes from 4.72 to 2.90 as the inlet t_{con} increases from 38.3 °C to 51.6 °C. Under the inlet t_{con} of 44.8 ± 0.5 °C and t_0 of 7 °C, the average COP deviation of the proposed heating panel is 3.72, which is 0.04 higher than that in ref. [13] and 0.73 higher than that in ref. [14].

Under t_0 of −7 °C, the system COP of the panel with a ratio of 0.45 changes from 2.72 to 2.02 as the inlet t_{con} increases from 38.2 °C to 48.2 °C, while it changes from 2.88 to 2.12 under the ratio of 2.22 as the inlet t_{con} increases from 38.7 °C to 46.9 °C. Under the inlet t_{con} of 40.1 ± 0.5 °C and t_0 of −7 °C, the average COP deviation of the proposed heating panel is 2.64, which is 0.21 higher than that in ref. [13] and 0.36 higher than that in ref. [14]. The COP variations demonstrate that the proposed novel heating panels are more competitive than previous heating terminals in system energy efficiency. Meanwhile, the COP deviation also indicates that the proposed heating panel with a higher ratio is better than that of the panel with a lower ratio.

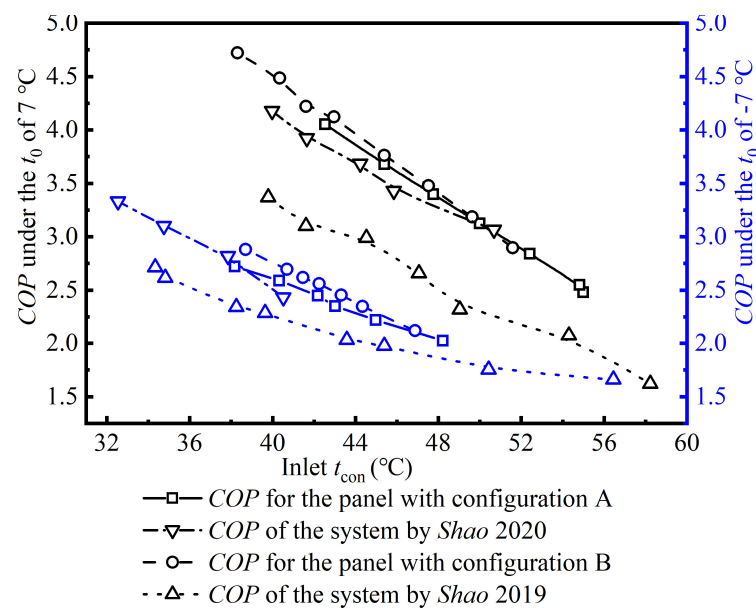


Figure 12. System COP comparisons between the novel direct-condensation heating panels and previous heating terminals [13,14].

4.3.2. System Exergy Efficiencies of the Proposed Heating Panels with Two Aspect Ratios

The exergy destructions of each component in the ASHP system with two heating panels are compared in Tables 6 and 7. The comparison is conducted under the same heating conditions with t_{ai} of 20 °C, t_0 of 7 °C, and the inlet t_{con} of 45.4 °C. The dead state is set to t_0 at 7 °C and P_0 at 101.3 kPa. As listed in Table 5, the maximum exergy destruction (Ex_{des}) occurs in the compressor, followed by the condenser in the two heating systems. For the panel heating system with a ratio of 0.45, the total exergy destruction ($Ex_{des,sys}$) is 654.5 W, and the exergy destruction of the condenser ($Ex_{des,con}$) accounts for 34.3% of the $Ex_{des,sys}$. For the panel heating system with a ratio of 2.22, the $Ex_{des,sys}$ is 641.1 W and $Ex_{des,con}$ accounts for 35.1% of the $Ex_{des,sys}$ under the same heating conditions. The system exergy efficiency φ_{sys} of the panel with a ratio of 0.45 is 16.3%, which is a bit lower than that with a higher ratio.

Table 6. Property data of the ASHP system with the direct-condensation heating panels using two aspect ratios.

Panel Type	Number	Fluid	Position	State	Flow Rate m (kg/s)	Temperature (°C)	Pressure (kPa)	Specific Enthalpy (kJ/kg)	Specific Entropy (kJ/kg K)	Exergy Rate (kW)
Ratio of 0.45	0	R410A	-	Dead state	-	7	101.3	455.16	2.12	
	1		Evaporator outlet	Vapor	0.01773	5.30	922.3	423.44	1.80	1.18
	1'		Compressor inlet	Vapor	0.01773	9.20	922.3	427.88	1.82	1.18
	2		Compressor outlet	Vapor	0.01773	58.70	2751.3	446.56	1.78	1.68
	2'		Panel inlet	Vapor	0.01773	52.00	2751.3	436.02	1.75	1.66
	3		Panel outlet	Liquid	0.01773	43.10	2630.3	272.15	1.24	1.30
	3'		EEV inlet	Liquid	0.01773	41.40	2630.3	268.78	1.23	1.30
	4		EEV outlet/ Evaporator inlet	Mixture	0.01773	5.72	957.3	268.78	1.25	1.20
Ratio of 2.22	1	R410A	Evaporator outlet	Vapor	0.01764	4.70	917.3	422.89	1.80	1.17
	1'		Compressor inlet	Vapor	0.01764	8.60	917.3	427.35	1.82	1.17
	2		Compressor outlet	Vapor	0.01764	58.90	2751.3	446.86	1.79	1.68
	2'		Panel inlet	Vapor	0.01764	52.30	2751.3	436.52	1.75	1.65
	3		Panel outlet	Liquid	0.01764	42.10	2606.3	270.19	1.23	1.29
	3'		EEV inlet	Liquid	0.01764	40.30	2606.3	266.67	1.22	1.28
			EEV outlet/ Evaporator inlet	Mixture	0.01764	5.99	965.3	266.67	1.24	1.20

Table 7. Exergy destructions under the same heating conditions.

Panel Type	Component	Ex_{in} [W]	Ex_{out} [W]	Ex_{dest} [W]
Ratio of 0.45	Evaporator	1204.5	1175.1	29.4
	Compressor	1965.2	1683.3	281.9
	Condenser (heating panel)	1655.8	1431.3	224.5
	Expansion valve	1295.7	1204.5	91.3
	Pipeline	1683.3	1655.8	27.5
	System	7804.5	7150.0	654.5
Ratio of 2.22	Evaporator	1199.5	1166.6	32.9
	Compressor	1946.5	1675.4	271.1
	Condenser (heating panel)	1648.5	1423.4	225.1
	Expansion valve	1284.7	1199.5	85.1
	Pipeline	1675.4	1648.5	26.9
	System	7754.6	7113.5	641.1

The influences of inlet t_{con} and t_{ai} on the system exergy efficiency (φ_{sys}) and system task efficiency (η) are estimated in Figure 13. As shown in Figure 13a, the heating conditions are conducted under t_{ai} of 20 °C and t_0 of 7 °C, in which the system COP decreases with the enhanced inlet t_{con} , while the COP_{rev} is unchanged. Thus, the system exergy efficiency decreases with the enhanced inlet t_{con} . As the inlet t_{con} increases from 38.3 °C to 55.0 °C, the system φ_{sys} of the heating panel with a ratio of 2.22 is a bit higher than that with a ratio of 0.45. The average deviation of the φ_{sys} between the two heating terminals is 0.45%. Meanwhile, as the inlet t_{con} increases from 42.5 °C to 55.0 °C, the system η decreases from 47.0% to 37.1% for the heating panel with a ratio of 0.45. As the inlet t_{con} increases from 38.3 °C to 51.6 °C, the system η decreases from 48.4% to 41.2% for the heating panel with a ratio of 2.22. The average deviation of the system η is 0.8%. In Figure 13b, the system exergy efficiency increases with increased t_{ai} . The phenomenon is attributed to the rising indoor air temperature while the inlet t_{con} and t_{out} are maintained at 41.2 ± 0.1 °C and 7 °C, respectively. As t_{ai} increases from 16 °C to 22 °C, the φ of the heating panel with a ratio of 2.22 changes from 12.8% to 21.8%, which is higher than that with a ratio of 0.45. In addition, the average system η for the panel with a ratio of 0.45 is 47.6%, while it is 48.5% for the panel with a ratio of 2.22.

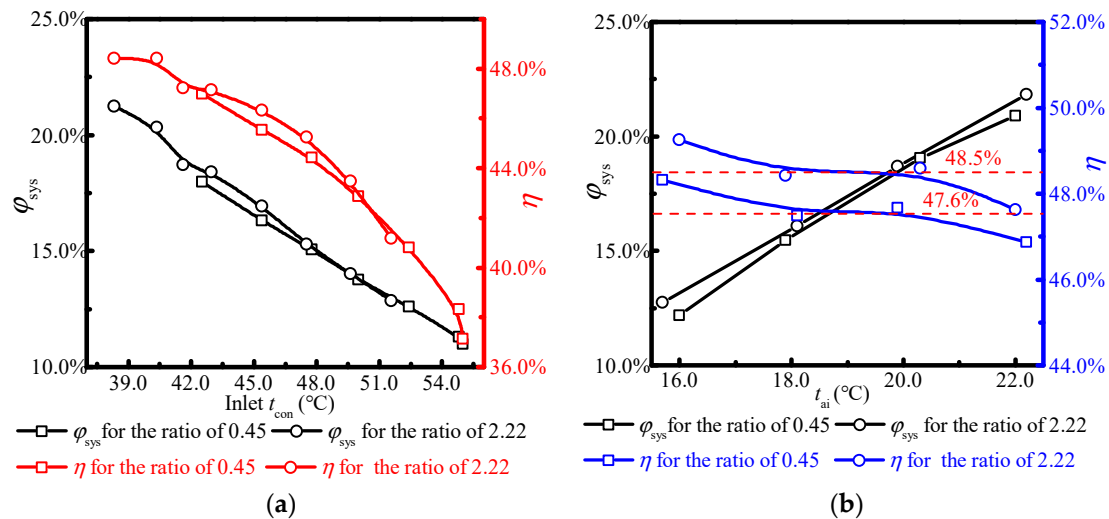


Figure 13. Exergy efficiency variations for the direct-condensation heating panels with two aspect ratios. (a) Under different inlet t_{con} . (b) Under different t_{ai} .

As mentioned above, the system energy efficiency of the proposed novel direct-condensation heating panels are competitive among existing direct-condensation heating terminals. The system COP and exergy efficiency variations demonstrate that the system efficiency of the panel with a higher ratio is better than that of the panel with a lower ratio.

5. Conclusions

A direct-condensation heating panel with lower copper tube length is proposed in this study. The performance of the proposed heating panel is tested in a modified standard air enthalpy laboratory, and the thermal performance and system energy efficiency of the novel heating panel are compared with those of previous direct-condensation heating terminals. Meanwhile, the effects of the length–width ratio on the performance of the heating panel are examined from the aspects of thermal performance and system efficiencies. The main conclusions are drawn as follows:

1. The proposed novel panel is competitive in indoor thermal comfort, thermal economy performance, and system efficiencies. The indoor temperature growth rate provided by the proposed heating panels is 15.2% lower than that of a previous heating terminal. The average heating capacity per cost is 5.4 W/USD, which is 24.1%~46.3% higher than that of previous panels. The system COP of the proposed heating panel is 0.04 to 0.73 higher than that of other direct-condensation heating terminals under the same heating conditions.
2. The proposed heating panel with an aspect ratio of 0.45 provides a more comfortable indoor thermal environment than that with an aspect ratio of 2.22, while the panel with a high ratio creates a warmer local environment.
3. Although the panel with a low aspect ratio is more prominent in reducing flow losses, the thermal economic performance and system efficiency of the proposed heating panel with an aspect ratio of 2.22 are more competitive than that with an aspect ratio of 0.45. The annual running cost for the panel with a ratio of 0.45 is 4.7% higher than that with a ratio of 2.22. Meanwhile, the average system η for the panel with a ratio of 2.22 is 48.5%, which is 0.9% higher than that with a ratio of 0.45.

The novel direct-condensation heating panel proposed in this study, coupled with the ASHP system, has advantages in thermal economy and system efficiency, which helps to reduce energy consumption and carbon emissions of winter space heating. The investigation of the length–width ratio promotes the structural optimization of direct-condensation heating terminals. The evaluation methodology can be applied to performance assessments of other heating terminals. It is worth noting that the findings such as thermal and flow

performances and system efficiencies will be affected by heat fluid differences and system differences. In future research, more experiments will be conducted in real residential houses to carry out annual operation analysis and carbon emission calculations.

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Nomenclature

A	Area (m^2)
C	Cost (USD)
C_{ADD}	System additional costs (USD)
C_c	Convection heat transfer between human body and indoor air ($\text{W}\cdot\text{m}^{-2}$)
C_{CER}	System cost-effectiveness ratio ($\text{USD}\cdot\text{kWh}^{-1}$)
C_{ICC}	Initial capital cost (USD)
C_{SYS}	System investment cost (USD)
C_{Ws}	System energy prices corresponding to energy consumption (USD)
E	Evaporation heat of the human body ($\text{W}\cdot\text{m}^{-2}$)
E_{dif}	Skin diffusion evaporation loss ($\text{W}\cdot\text{m}^{-2}$)
Ex	Rate of exergy (kW)
f	Coefficient
G	Refrigerant flow rate ($\text{kg}\cdot\text{s}^{-1}$)
h	Specific enthalpy ($\text{kJ}\cdot\text{kg}^{-1}$)
I	Proportion of evaluation indicators
M	Metabolic rate ($\text{W}\cdot\text{m}^{-2}$)
P	Pressure (Bar)
P_{ai}	Water vapor pressure (Pa)
Q	Heating capacity (W)
q	Heat flux (W)
R	Radiation heat transfer between human body and indoor chamber ($\text{W}\cdot\text{m}^{-2}$)
s	Specific entropy ($\text{kJ}\cdot\text{kg}^{-1}\cdot^\circ\text{C}^{-1}$)
T	Temperature ($^\circ\text{C}$)
W_s	system input power (W)
<i>Greek symbols</i>	
Δ	Variation
α	Heat transfer coefficient ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$)
γ	Uncertainties (%)
ε	Radiant surface emissivity
η	Task efficiency (%)
σ	Stephen Boltzmann constant, $5.67 \times 10^{-8} \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-4}$
φ	Exergy efficiency (%)
χ	Wetted perimeter (mm)
<i>Subscript</i>	
0	Dead state
AUST	Comprehensive temperature of the building envelope
ai	air
cl	Clothing

con	Condenser
com	Compressor
d	Direct measurements
des	Destruction
eva	Evaporator
EEV	Electronic expansion valve
in	Inlet
ind	Indirect measurements
ins	Instruments
nc	Natural convection
oc	Occupying construction area
out	Outlet
pip	pipeline
pr	Production
ra	Random
rad	Radiation
rev	Reversed Carnot cycle
s	Surface
<i>Acronyms</i>	
ASHP	Air source heat pump
HVAC	Heating, ventilation, and air conditioning
COP	System coefficient of performance
PD ₂	Percentage dissatisfied caused by vertical temperature difference
PD ₃	Percentage dissatisfied caused by floor temperature
PMV	Predicted mean vote
PPD	Predicted percentage of dissatisfied
RH	Relative humidity

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