



Small Temperature Difference Terminals

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Abstract

Air-source heat pump (ASHP), as an optional product, has already been extensively applied to residences and offices in hot summer and cold winter. However, the existing products confront serious problems under heating conditions in winter. A visible issue is that the air distribution is irrational because the terminals

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in room, in most cases, are installed in an upper space, which makes users feel uncomfortable. Another issue is that the terminals are designed under an inappropriate condition, leading to a low efficiency of the air-source heat pump. In addition, the terminals themselves are not designed with the best structure parameters and hence influence the heat transfer performance a lot.

To solve the problems above, small temperature difference terminals (STDTs), such as small temperature difference fan-coil unit (STDFCU) or floor-heating coil (FHC), have been proposed. In this chapter, a small temperature difference fan-coil unit is designed and manufactured. Besides, its heat transfer performance is tested. An air-source heat pump combined with four selected prototypes is applied to a 100 m² apartment. The system performance and energy consumption in heating mode are measured. According to the results, the room environment can be kept in a comfortable state under the condition that the supply water is kept at 35 °C, which can lead to almost half reduction of energy consumption compared to the condition under which the supply water is kept at 45 °C. Finally, a larger air-source heat pump system is applied to a public building and the performances of STDFCU, and normal fan-coil unit (NFCU) in cooling mode are analyzed from several aspects, including their energy consumption and indoor temperature distribution in summer. Meanwhile, the performances of STDFCU, NFCU, and FHC in heating mode are analyzed including energy consumption and zone comfort in winter. The result shows that ASHP + STDFCU system can save 15–22% energy compared to traditional ASHP + NFCU system and thus has a 13–24% higher COP than ASHP + NFCU system depending on the weather condition in summer. What's more, STDFCU and FHC in heating mode can reduce system energy consumption up to 20%, and the air distribution in the room is more uniform in winter.

Keywords

Air-source heat pump · ASHP · Small temperature difference · Terminals · Fan-coil unit · Residential heating · Energy efficiency · Indoor thermal comfort · Life cycle cost

Introduction to Small Temperature Difference Terminals

Background

Energy crisis has become a problem almost every country confronts nowadays. All the measures promoted to solve the problem can be classified into two types, including the increasing of the production from the supply side and the decreasing of consumption from the demand side. As for the demand side, building energy consumption accounts for quite a considerable part [1]. By the Sankey diagram mapped by Ma et al. building consumes 29% of the total energy consumption in China [2]; furthermore, most of the building energy consumption is for indoor thermal comfort, in other words, air-conditioning. It can be foreseen the percentage will continue to rise due to China's urbanization process. According to the US

energy information administration's international energy outlook, China residential sector will consume up to 20 quadrillion Btu energy in 2040, occupying 24% of the total world use [3]. Thus, investigating the efficiency of space cooling/heating system and possibility to improve it has a great importance [4].

Air-source heat pump (ASHP) system is the most widely used building energy system in China due to its flexibility and feasibility [5]. Looking into all ASHP systems, there are three major kinds, which are listed as follows. Type A is air-to-air heat pump system with refrigerant pipe connecting the outdoor unit and indoor units, featuring variable refrigerant flow (VRF) or variable refrigerant volume (VRV) technology [6]. The so-called multi-split VRF/VRV system can satisfy the same needs for the installation of several individual units yet with less space. This is because this system consists of one outdoor unit and multiple indoor units [7, 8] (VRV is a trademark of a leading VRF manufacturer, and VRF is a generic term used by all of the VRF manufacturers [9]). Masuda et al. [10] developed a control method for a multi-split VRF system with two indoor units. Choi and Kim [11] studied the performance of a multi-split VRF system which has two indoor units with individual EEVs by varying the indoor loads and the compressor speed. Aynur [12, 13] investigated the integration of a multi-split VRF system with a heat pump desiccant unit to test its performance in heating and cooling seasons. Type B is air-to-water heat pump system with water pipe carrying cool or heat to the indoor terminals for air-conditioning. Such system is called fan-coil unit (FCU) system. Investigated issues include novel product development, optimal design methods of FCU systems [14, 15], modeling and measurement of FCU coil performance [16, 17], energy consumption metering of FCU systems [18], indoor air quality, and thermal comfort in FCU-served air-conditioning space [19–21]. Type C is air-to-air heat pump system with air handling unit, fulfilling variable air volume (VAV) with the help of VAV box. The applications of VAV technology are introduced in details in ASHRAE handbook [22]. Hung [23] studied the effect of flow controllers on VAV system performance by simulations and field measurements. Engdahl and Johansson [24] optimized the air supply temperature of a VAV system theoretically by minimizing the system energy use. Ke [25] developed an innovative low-temperature differential variable air volume fan-coil unit (LTD-VAV FCU) to overcome the defects of variable air volume (VAV) and cold air distribution energy-saving technologies.

Many researchers adopt comparison method when they conducted their previous researches. They either choose different air-conditioning (AC) systems for the same building or adopt different operation control strategies. Besides, they also compare energy consumption and thermal comfort in different cases. Zhou [26] simulated the energy consumption of the VRF system, variable air volume (VAV) system, and fan-coil plus fresh air (FPFA) system in Shanghai, China. According to the results of the research, the VRF system consumed 22.2% and 11.7% less energy compared with the VAV system and the FPFA system, respectively. Aynur [27] compared the performance of VAV system and VRF system under the same indoor and outdoor conditions in an existing office building during an entire cooling season by simulation. It has been found that compared with the VAV system, the VRF system stated 27.1–57.9% energy-saving potentials depending on the system configuration and indoor and outdoor conditions.

Yao [28] established specific energy models of the primary equipment in HVAC system and made 1-year-round energy simulations of VAV systems and FCU systems of a small office building. This program evaluated the energy-saving performance of the VAV system in six cities of China. The results showed that VAV system consumes 5–10% less energy compared with FCU system in different cities. Pan [29] performed simulations and calculations of the buildings' energy consumption and energy cost via two different outdoor air control strategies of a typical VAV system as well as a fan-coil unit (FCU) system in Shanghai in normal days. The results of the research illustrate that FCU system consumes 3% more energy than VAV system.

Several recently published research papers note that using small temperature difference terminals, such as low-temperature radiators and low-temperature lift heat exchangers [30, 31], which both are kinds of compact and effective heat exchangers, can change the ASHP outlet water temperature a lot, leading to a raise in system coefficient of performance (COP) correspondingly. In order to solve the residential heating problems in winter, Wang [32, 33] proposed a novel concept of small temperature difference fan-coil unit that is combined with air-source heat pump system and undertook some demonstration project in residence and small office. According to the results of the operation, it has been shown that using small temperature difference fan-coil units in air-source heat pump system can increase the COP in the system a lot, leading to a great reduction of energy consumption. Besides, indoor thermal comfort [34–37], as a fundamental goal of the HVAC system, has been obtaining more and more attention. Different indoor terminals usually cause different air distributions, influencing the different indoor thermal comfort. The installation location of indoor terminals will also influence the temperature and humidity distribution, especially in winter. This chapter will discuss the energy-saving potential and indoor thermal comfort when small temperature difference terminals are adopted.

Concept of Small Temperature Difference Terminal

Small Temperature Difference Fan-Coil Unit

Nowadays, air-source heat pumps are widely used in residential, commercial, and industrial spaces, and the terminals that transfer heat to or from the conditioned space are usually fan-coil units. A normal fan-coil unit is shown in Fig. 1. The fan recirculates air from the space through the coil, containing either hot or chilled water. The unit should be equipped with an insulated drain pan if it will be used for cooling.

When the heating and cooling media are supplied to a common coil, it is called a two-pipe system (one supply and one return) or a heating/cooling changeover system, and it can only either heat or cool, depending on the temperature of supply water. A four-pipe distribution system has dedicated supply and return piping for each coil and generally has a better system performance. It can provide all-season availability of heating and cooling at each unit. However, the four-pipe system has a higher initial cost and a more complex structure. When a fan-coil system is applied to residential and some commercial buildings such as small offices, a two-pipe system is more popular due to its easier installation, more compact structure,

Fig. 1 Normal fan-coil unit

and lower initial cost. What's more, the residential and light commercial buildings often have a stable heating or cooling load, and different zones in those buildings have little difference in load. That is to say, there is no need for simultaneous heating and cooling. As a result, the two-pipe fan-coil system is widely used in those buildings in the world. This chapter takes such system as its focus. Although two-pipe system has abovementioned advantages, it has to be designed carefully when it's used for heating and cooling in the whole year. Otherwise, it will waste the energy.

The normal fan-coil unit is designed under the condition of 7 °C supply water in cooling mode, according to Chinese national standards [38, 39]. Choosing 7 °C as a design condition comes from the following reason. The fan-coil unit needs to undertake the task of dehumidification as well as cooling, and the method of dehumidification is always condensation, which means the supply water temperature has to be low enough so that the moisture can be removed. It makes sense when the fan-coil unit is only used for cooling in hot and high-humidity zone. However, if the unit is used for not only cooling in summer but also heating in winter, things will be different. If a building has a high heating load in winter as much as cooling load in summer, the supply water temperature in winter has to be high to ensure a comfort zone because the fan-coil unit is designed under a condition of very low temperature in cooling mode and has a relatively poor heat transfer performance due to the principle of prior dehumidification. Moreover, with the development of energy-efficient building and people's increasing demand for indoor air quality, the infiltration and natural ventilation can't satisfy the fresh air requirement anymore. As a result, adopting fresh air system will become a trend in residential buildings. It's a good choice to combine it with fan-coil unit system.

If dehumidification is undertaken by fresh air system, the fan-coil system will only be used to deal with the sensible heat during the whole year. Therefore, the fan-coil unit can be designed under a better condition without the limit of dehumidification. The air-source heat pump system will be benefited a lot, if the supply water

temperature can be higher in summer and lower in winter. These units designed under non-normal conditions are called small temperature difference fan-coil units.

It's well known that the fin-tube heat exchanger is the most important component in fan-coil unit. The heat transfer process is quantified through the following equation:

$$Q = U \times A \times LMTD \quad (1)$$

Where

Q = amount of heat transferred, W

U = heat transfer coefficient, W/m²·K

A = effective surface area for heat transfer, m²

$LMTD$ = log-mean temperature difference across the coil surface, °C

Increasing any one of these variables (heat transfer coefficient, surface area, or log-mean temperature difference) leads to more heat transfer. However, there are some differences among the three parameters. The surface area is completely a construction parameter, which is decided immediately when the fan-coil unit is designed. Heat transfer coefficient is influenced by many factors, some of which are construction parameters such as fin type, tube type, as well as some other parameters, while the other of which are operation conditions such as the velocity of air and water. Log-mean temperature differences are totally controlled by the operation condition, supply water temperature, return water temperature, inlet air temperature, and outlet air temperature. In the context of a cooling or heating coil, $LMTD$ describes the differences between the temperatures of the air passing across the coil fins and the water flowing through the coil tubes:

$$LMTD = \frac{TD_2 - TD_1}{\ln(TD_2/TD_1)} \quad (2)$$

Where

TD_1 = leaving air and entering water temperature difference at the coil, °C

TD_2 = entering air and leaving water temperature difference at the coil, °C

If the heat transfer performance is improved, the supply water temperature can be higher and hence to satisfy the same cooling demands in cooling mode. The heat transfer characteristics of normal and small temperature difference fan-coil unit are shown in Fig. 2. The heat transfer process is simplified. Therefore, only fin-tube heat exchangers are shown in the Fig. 2a. The inlet water temperature can be lower when the environmental temperatures are the same because the STDFCU has a larger heat transfer area and also a higher heat transfer coefficient. Temperature difference in cooling mode between water and air is shown in Fig. 2b. It can be seen that the inlet water temperature of NFCU is 7 °C, while the outlet air temperature is 13 °C, which

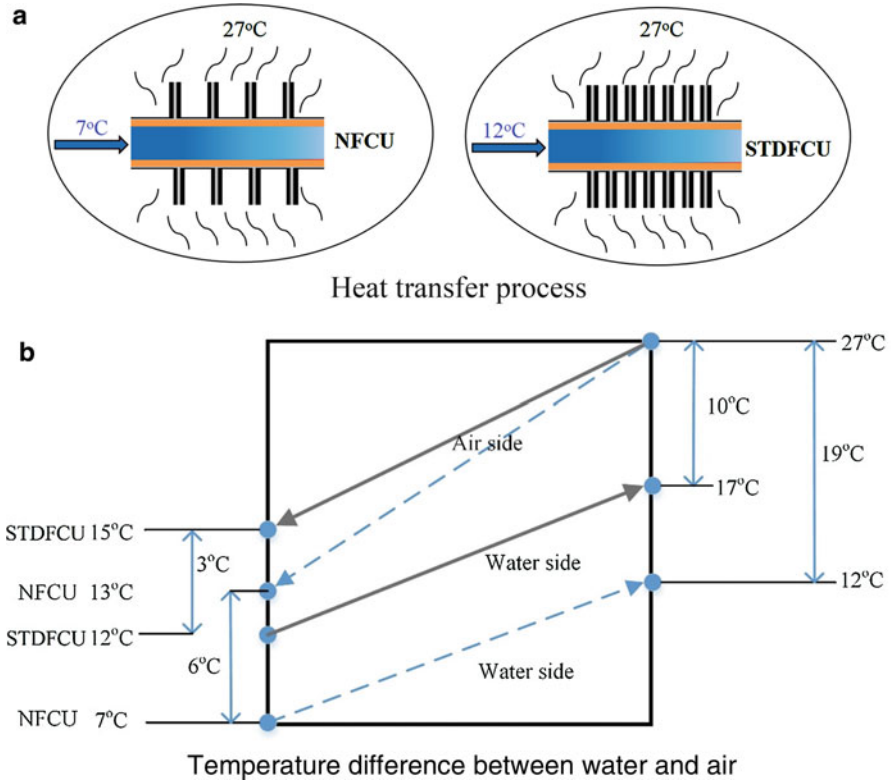


Fig. 2 Heat transfer characteristics of two fan-coil units in cooling mode. (a) Heat transfer process. (b) Temperature difference between water and air

is low enough to ensure dehumidification. According to Chinese national standard [38], the temperature difference of inlet and outlet water temperatures should be 5 °C, considering the balance of initial cost and the life cycle cost of the whole system. As a result, the outlet water temperature is 12 °C, while the inlet air temperature is 27 °C, which equals to the room air temperature. The log-mean temperature difference across the coil surface can be calculated according to Eq. 2. The result is 11.3 °C through the calculation. As for the STDFCU, since it doesn't have to handle the latent load and has a relatively higher heat transfer performance, the inlet water temperature can be set higher, and it's 12 °C in this case. The outlet air temperature is 15 °C. By the same token, the outlet water temperature of STDFCU is 5 °C higher than the inlet, which is 17 °C. At last, the *LMTD* can be calculated. The final result is 5.8 °C, which is about half of the normal one. According to the heat transfer process (Eq. 1), it seems that the product of *U* and *A* of STDFCU is twice of NFCU. However, actually, the amount of heat transferred is indifferent in the two fan-coil system. The STDFCU will be used combining with fresh air system, which will handle the latent heat, taking up 30% of the total heat normally. As a result, the

heat transfer performance of STDFCU has to be enhanced by 35% to ensure the room comfort. The method will be mentioned later. In the end, one thing needs to be explained in cooling mode, the air flow rate of STDFCU should be higher than that of NFCU because the supply air temperature is slightly lower according to Fig. 2b. It will increase the power consumption of fan in principle. However, a novel fan and motor system is adopted in STDFCU as a compensation that will be introduced later. The same goes to the heat transfer characteristics of two fan-coil units in heating mode. Supply water temperature can be decreased because of the heat transfer enhancement of STDFCU. The heat transfer process and temperature difference between water and air in heating mode are shown in Fig. 3a, b. The *LMTD* can be calculated according to Eq. 2. The results are 12 °C and 6.2 °C for NFCU and STDFCU, respectively.

It's necessary to decrease the water temperature demand of fan-coil units as low as possible in an air-source heat pump system. Taking heating mode, for example, a lower water temperature can lead to a lower condensing temperature and a higher system COP. The heat pump cycle is illustrated in Fig. 4. For a normal fan-coil unit,

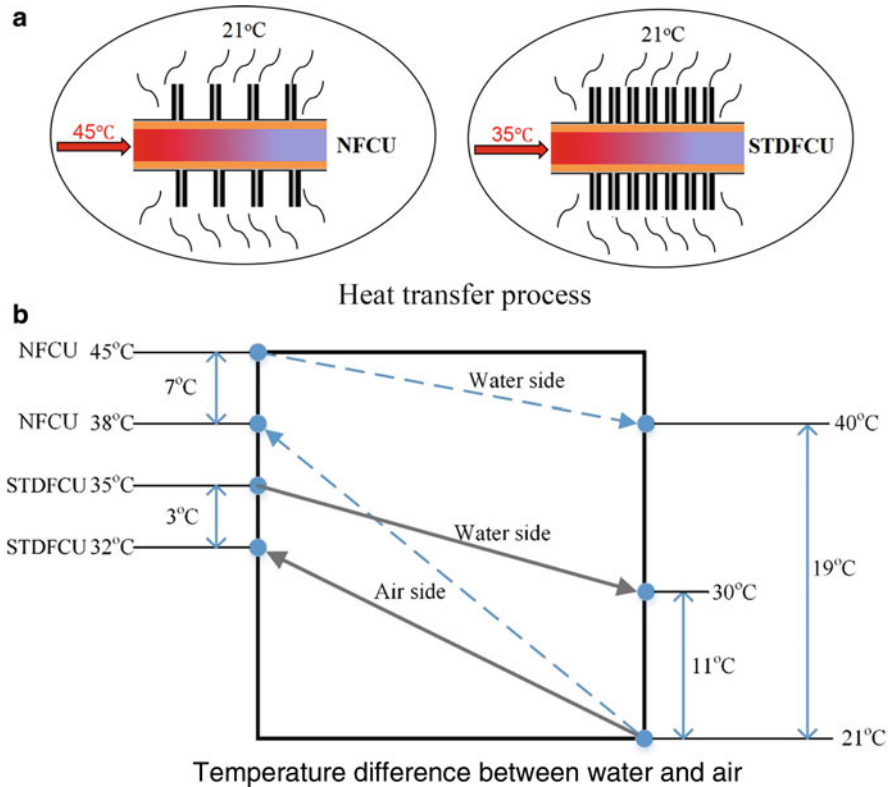
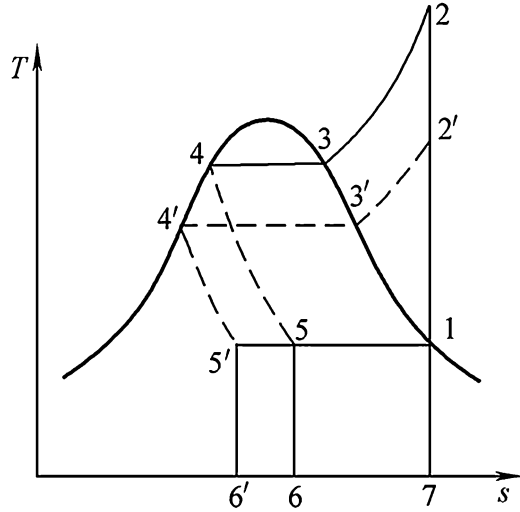


Fig. 3 Heat transfer characteristics of fan-coil units in heating mode. (a) Heat transfer process. (b) Temperature difference between water and air

Fig. 4 Heat pump cycle



the water needs to be in a high temperature and hence satisfies the zone heating demand. The system condensing temperature is high. The system with a normal fan-coil unit operates through the cycle 1-2-3-4-5-1, while the system with a STDFCU operates through the cycle 1-2'-3'-4'-5'-1. According to the heat pump cycle, the system COP can be valued by the heating coefficient.

$$\epsilon'_c = \frac{\bar{T}_c}{\bar{T}_c - \bar{T}_e} \tag{3}$$

For the system with NFCUs,

$$\epsilon'_{cn} = \frac{\bar{T}_{2-3-4}}{\bar{T}_{2-3-4} - \bar{T}_{1-5}} \tag{4}$$

For the system with STDFCUs,

$$\epsilon'_{cl} = \frac{\bar{T}_{2'-3'-4'}}{\bar{T}_{2'-3'-4'} - \bar{T}_{1-5'}} \tag{5}$$

The average temperature from state 2 to 4 is higher than that from 2' to 4'. The average temperature from state 1 to 5 equals to that from 1 to 5'. According to Eqs. 4 and 5, the heating coefficient of system with STDFCUs is higher than that with normal fan-coil units, signifying that STDFCUs can help to reduce system energy consumption with the same zone heating demands.

Figure 5 illustrates the influence of water temperature on theoretical COP of air-source heat pump system with different refrigerants. The COP is calculated at the

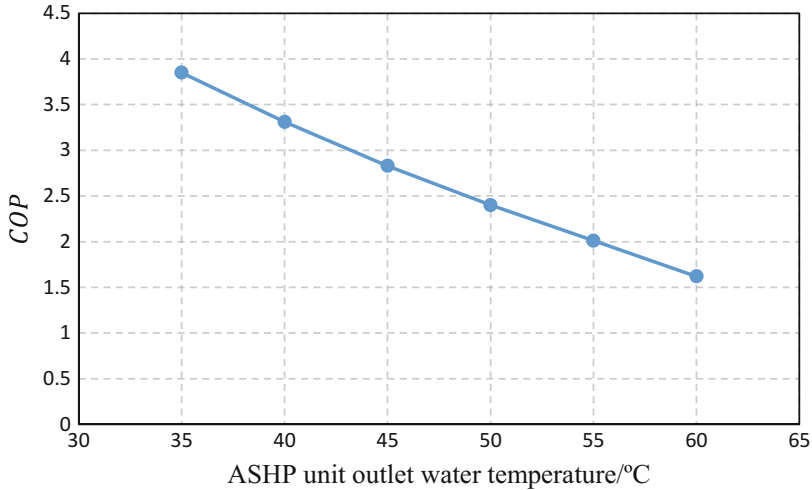


Fig. 5 Influence of ASHP outlet water temperature on ASHP Unit (heating mode)

outdoor temperature of 7 °C. The evaporating and condensing temperatures are determined through empirical method [40]. It can be seen that the system COP decreases a lot with the increase of the ASHP unit outlet water temperature. For example, if the water temperature is 60 °C, the system COP is only about 1.6. However, if the water temperature decreases to 35 °C, the system COP will increase nearly to 4, which is a visual progress and can save much energy.

In a word, the concept of small temperature difference fan-coil unit compromises between initial cost and operation cost. With the development of air-source heat pump system, it becomes normal that air-source heat pump is used for both heating and cooling in summer and in winter, which means the system operates longer than before. It is of great importance to consider the performance of the system both in summer and in winter. The proposal of small temperature difference fan-coil unit implies that more heat transfer enhancement technologies are applied to the terminal, and more heat transfer areas are adopted. That is to say, to ensure a more energy-efficient terminal and air-source heat pump system by increasing the initial cost will become a trend in the world. Figure 6 shows the relative contribution of operating costs to a customer's total life cycle A/C costs in both the USA and India. The life cycle A/C costs include non-refrigerant equipment cost, refrigerant cost, installation cost, and operating cost. For a country in the hot-humid climate such as India, the A/C system's energy consumption can take up to 80% of a customer's lifecycle costs. It's the same with hot summer and cold winter climate zones: air-source heat pump system operates in the whole year, and the energy consumption is too much. It makes sense to optimize the operation condition and increase the system energy efficiency by increasing the initial cost to achieve a balance of the life cycle A/C cost.

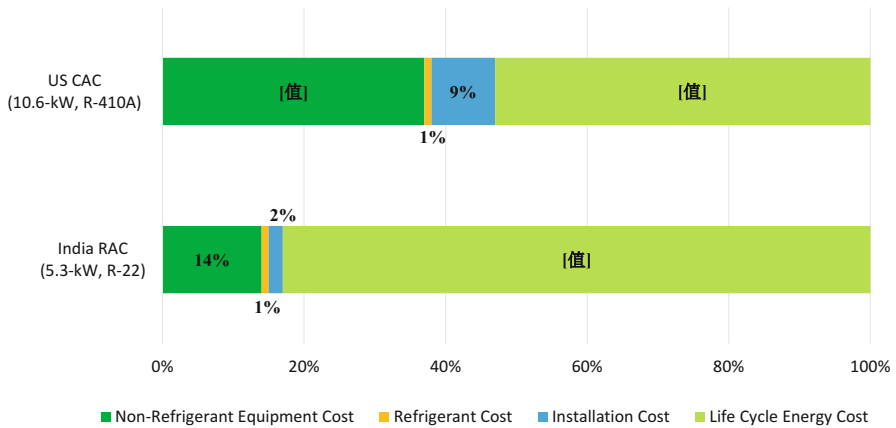


Fig. 6 Relative contribution of equipment, refrigerant, installation, and operating costs to customer life cycle A/C costs [41]

Floor-Heating Coil

Another kind of small temperature difference terminal is floor-heating coil that can also be combined with air-source heat pump system. This system has its advantages in heating. The typical structure of floor-heating system is shown in Fig. 7. It consists of five layers. Floor is the first layer that transfers heat to zone directly. Leveling blanket is the second layer that can ensure the floor is being flat. The third layer is a packing layer that packs the heating coil and serves as a conductor. The fourth layer is an insulating layer, preventing the heat from being transferred to downside space, while the last layer is a floor slab. Hot water flows through the heating coil and transfers heat to floor through conduction. The whole floor heats up the zone through radiation and natural convection.

The floor has a large area and a high heat transfer performance. The supply water of this terminal can be decreased, and the energy efficiency of air-source heat pump system can be increased. Different from small temperature difference fan-coil unit, the floor-heating coil transfers heat mainly through radiation. It has following characteristics. Firstly, the overall thermal comfort of the indoor space is good. The air distribution is more uniformed compared with the ventilation heating system in the height direction, which is conducive to improve the thermal comfort. Meanwhile, the temperature fluctuation in the room is smaller due to the higher thermal inertia of floor-heating system. Secondly, the walls and ceiling are heated through radiation first. People get heat not only from air but also from these surfaces. As a result, the indoor air temperature can be decreased, and it is conducive to energy conservation. Thirdly, floor-heating coil occupies little indoor space. The traditional heating equipment will occupy a certain amount of indoor area which will influence the decoration and normal life as the use of floor-heating coil doesn't occupy any of the living space and can improve the efficiency of space utilization. It's a good choice to adopt floor-heating coil as a heating device in winter for the sake of its thermal comfort and energy saving. However, this heating method also has some

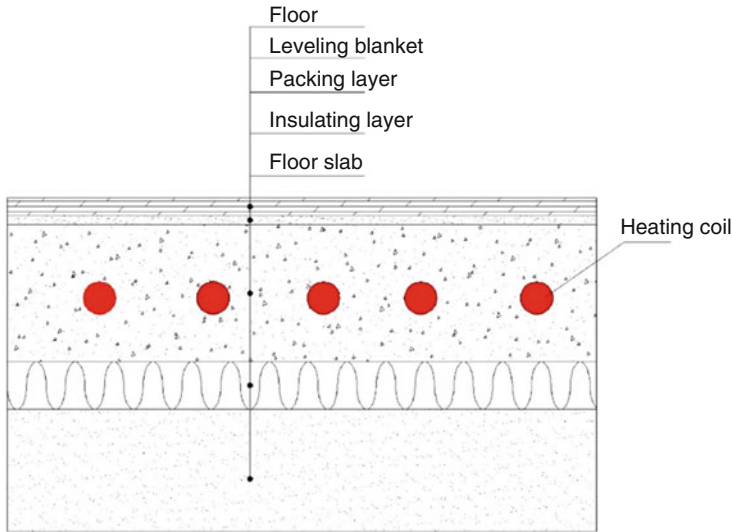


Fig. 7 Typical structure of floor-heating system

restrictions. The floor-heating system is hard to be installed and not suitable for the retrofit of an existing building. What's more, for those places which need both heating and cooling, such as hot summer and cold winter climate zones, an extra cooling device is needed because the floor we discussed above cannot be used for cooling considering the thermal comfort and problem of condensation. As a result, the initial cost will be greatly increased, which appears to be not so necessary because of the proposal of small temperature difference fan-coil unit.

Design and Performance Study of STDFCU

Design of STDFCU

The newly designed STDFCU consists a fin-tube heat exchanger, a cross-flow fan, a brushless DC (direct current) motor, and a condensate plate. The fin-tube heat exchanger serves as the most important component in the whole unit. It's mentioned above that increasing U-factor and the heat transfer area of coil can enhance the performance of heat exchanger. U-factor is also called heat transfer coefficient. It is determined by three factors. Firstly, it is determined by the air-side heat transfer resistance between the passing air stream and the fin surfaces. Secondly, it is decided by the water-side heat transfer resistance between the inside surfaces of the copper tubes and the circulating fluid. Thirdly, it is decided by thermal conductance of aluminum fins and copper tubes of the coil. Among the three factors, air-side heat transfer resistance is the largest one. It affects the heat transfer coefficient most. By optimizing the fin geometry, the air-side heat transfer resistance can be decreased.

Besides, the overall heat transfer coefficient can be improved. Coil fins are usually stamped into waveforms resembling corrugated cardboard or louvered forms, as what is shown in Fig. 8. These waveforms create turbulence in the passing air stream, which lessens the resistance to heat transfer. What's more, the pressure performance is also an important factor that will affect the power consumption of the unit. It's often opposite to the heat transfer performance in relation to structure parameters. As a result, it is of great importance to decide a parameter and hence to have the best heat transfer performance and keep the pressure drop as low as possible. A model of fin-tube heat exchanger is shown in Fig. 9.

Brushless DC motor is adopted in STDFCU. Brushless direct current technology or BLDC motors are also known as electronically commutated (EC) motors, which consist of a motor combined with an inverter. Having no rotor-induced current, as what it is in case of a traditional fixed-speed AC motor-driven fan-coil unit, leads to a significant reduction in power consumption. Compared with AC motor, the energy efficiency of BLDC is improved, and the power consumption can be reduced by 70%. Different from the normal fan-coil unit, a cross-flow fan is adopted to make the unit compact and the air flow more uniform. The model of fan and motor is shown in Fig. 10.

Design of Fin-Tube Heat Exchanger

The structure of STDFCU is determined. The value is shown in Table 1. The parameters d_o and d_i refer to outer and inner diameter of copper tube, respectively. The parameters δ_f and s_f refer to fin thickness and fin pitch, respectively. The fin has a type of corrugation and is made of aluminum. The parameters s_1 and s_2 refer to tube pitch and row spacing, respectively. The parameter n refers to row number along the direction of air flow. The tubes are arranged as what they are shown in Fig. 11. The parameters L_1 and L_2 refer to the width and length of fin, respectively. They are determined according to tube arrangement.

After the parameters of fin-tube heat exchanger have been determined, the overall size of STDFCU can be calculated. First of all, it is necessary to analyze the design goal. Normal fan-coil units are designed to satisfy the cooling capacity in summer.

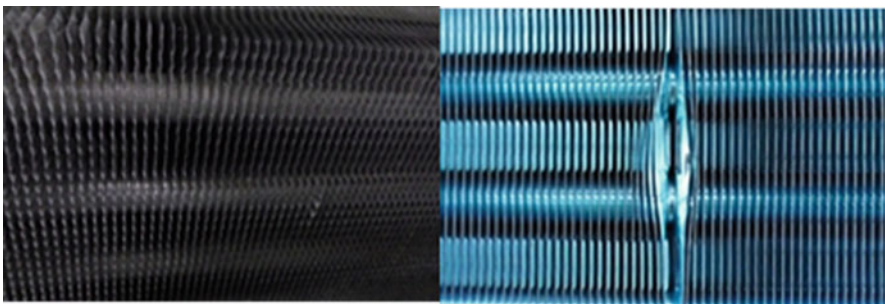


Fig. 8 Typical geometries for coil fins

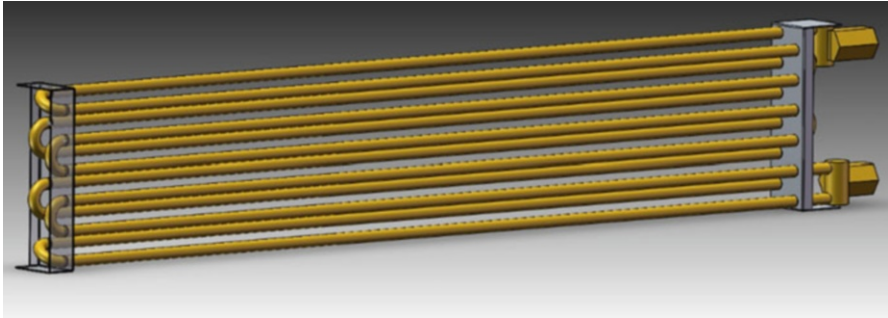


Fig. 9 Fin-tube heat exchanger

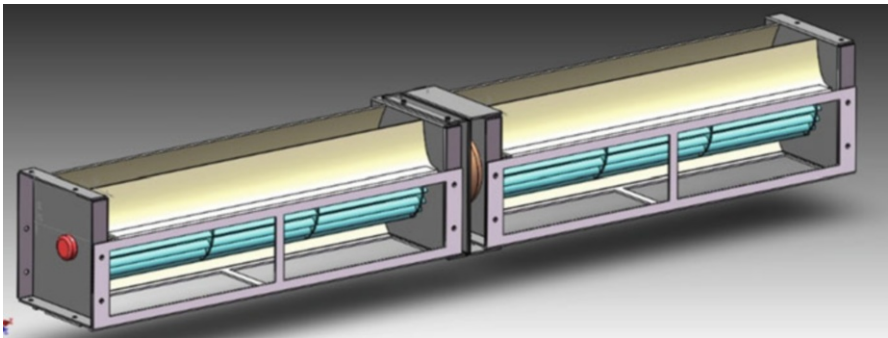


Fig. 10 Brushless DC motor and cross-flow fan

Table 1 Value of structure parameters

d_o (mm)	d_i (mm)	δ_f (mm)	s_f (mm)	s_2 (mm)	s_f (mm)	L_1 (mm)	L_2 (mm)
9.52	8.82	0.115	25	21.6	1.8	45	150

However, STDFCU is designed to meet heating demand in winter first. To satisfy the heating demand for residence with an area of 20 m² in Central and South China, the design goal of heating capacity is set to 2000 W.

Then, it comes to the determination of design parameters. To combine with air-source heat pump, it is a must to decrease the supply water temperature, which is designed to be 45 °C as what was mentioned above. The inlet air temperature is designed to be 18 °C, while air flow rate is designed to be 450 m³/h. The water flow rate is designed to be 0.3 m³/h. The design parameters are shown in Table 2.

Up to now, all the parameters have been determined. Therefore, the overall size of fin-tube heat exchanger can be determined also. The heat transfer area of fin-tube heat exchanger is 6 m², and the effective length of one tube is 0.74 m.

Fig. 11 Arrangement of tubes

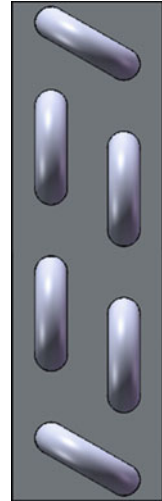


Table 2 Design parameters of fin-tube heat exchanger

Parameter	Value
Heating capacity	2000 W
Supply water temperature	45 °C
Inlet air temperature	18 °C
Air flow rate	450 m ³ /h
Water flow rate	0.3 m ³ /h

Design of Fan and Motor

The fan of normal fan-coil unit is made of metal and is driven by AC motor. But it’s known that brushless DC motor has an advantage of low power consumption. Besides, plastic fan with metal shell makes less noise, which has been partly absorbed by the two composite materials. What’s more, plastic fan is much lighter and cheaper compared with metal fan, which is very helpful to reduce the weight and cost of fan-coil unit. And a lighter fan leads to lower power consumption when it’s being started. Eventually, the plastic fan and brushless DC motor are adopted. The photo of cross-flow fan is shown in Fig. 12.

So far, the overall size of fan-coil unit can be determined, just as what is shown in Table 3. It shows that this prototype is extremely compact and it’s quite suitable to be installed in a family. The photo of small temperature difference fan-coil unit is shown in Fig. 13.

Performance Study of STDFCU

The performance of STDFCU has been tested based on Chinese national standard [38]. The rated cooling and heating capacity have been tested. Besides, the air and

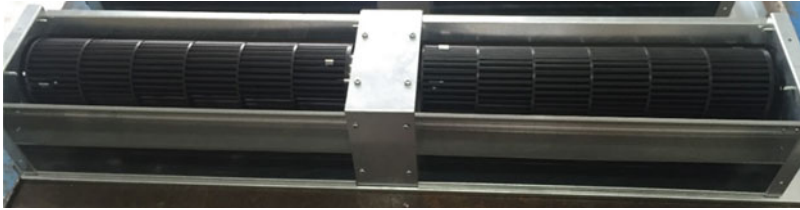


Fig. 12 Photo of cross-flow fan

Table 3 Overall size of STDFCU

Parameter	Value/mm
Length	860
Width	128
Height	348



Fig. 13 Photo of STDFCU

water flow rate under different conditions and the heating capacity under low-temperature heating conditions have also been tested.

The Rated Cooling and Heating Capacity

Although the prototype is designed under low-temperature heating conditions, its rated heating and cooling capacity need to be tested under the standard condition and hence to make a comparison with the normal fan-coil unit. The parameters of rated condition [38] are shown in Table 4. The heating and cooling capacities of STDFCU are tested in the enthalpy lab, while experimental test process is shown in Fig. 14.

The results are shown in Table 5. According to the statistics, the STDFCU has a cooling capacity of 1950 W, a heating capacity of 3070 W, and an input power of 18.73 W under the rated conditions. And one phenomenon needs to be noticed is that the air flow rate under cooling condition is lower than what it is under heating condition. It's

Table 4 Parameters of rated condition

Items	Cooling	Heating
Inlet air dry-bulb temperature/°C	27	21
Inlet air wet-bulb temperature /°C	19.5	–
Supply water temperature/°C	7	60
Supply and return water temperature difference/°C	5	–
Water flow rate/(kg/h)	Based on water temperature difference	Same as cooling condition
Fan speed	High gear	
Outlet static pressure/Pa	0	0



Fig. 14 The experimental test process of STDFCU

because that the cooling condition is a wet condition. Therefore, condensate water from air adheres to the surface of fins, which increases the air-side resistance.

The Low-Temperature Heating Test

The STDFCU is designed to combine with air-source heat pump in winter. Therefore, the temperature of supply water should be lower and hence to assure that the system will operate well and the energy consumption will be decreased. The heating capacity has been tested when the design supply water is 45 °C. The result is shown in Table 6. It shows that the heating capacity is 2090 W and reaches the design goal.

Performance Study of the ASHP System Combined with STDFCU

Prototype Description

The prototype designed and manufactured above is adopted to combine with an air-source heat pump. Besides, a shell with inlet and outlet deflector is installed to make the prototype a real product. The final appearance of small temperature difference

Table 5 Results of rated cooling and heating capacity experiment

Items	Cooling	Heating
Inlet air dry-bulb temperature/ $^{\circ}\text{C}$	26.97	20.97
Inlet air wet-bulb temperature/ $^{\circ}\text{C}$	20.01	–
Outlet air dry-bulb temperature/ $^{\circ}\text{C}$	17.54	42.01
Outlet air wet-bulb temperature/ $^{\circ}\text{C}$	15.52	–
Inlet water temperature/ $^{\circ}\text{C}$	7.19	60.04
Outlet water temperature/ $^{\circ}\text{C}$	12.69	51.11
Water-side pressure loss/kPa	35.96	35.96
Water flow rate/(m^3/h)	0.312	0.312
Air flow rate/(m^3/h)	415.6	440
Input power/W	18.73	18.73
Cooling/heating capacity/kW	1.95	3.07

Table 6 Results of low-temperature heating experiment

Items	Heating
Inlet air dry-bulb temperature/ $^{\circ}\text{C}$	17.99
Outlet air dry-bulb temperature/ $^{\circ}\text{C}$	32.84
Inlet water temperature/ $^{\circ}\text{C}$	44.89
Outlet water temperature/ $^{\circ}\text{C}$	38.81
Water-side pressure loss/kPa	29.227
Water flow rate/(m^3/h)	0.306
Air flow rate/(m^3/h)	440
Heating capacity/kW	2.09

fan-coil unit is shown in Fig. 15. The overall size of the final prototype is presented in Table 7.

ASHP System for Residence

In order to figure out the demands of adopting STDFCU in air-source heat pump system, the terminals manufactured above are used for residence in this part. In this research, the system performances under different conditions were studied. Besides, this research also studied the influence of water temperature on system energy consumption.

System Description

An air-source heat pump system is installed in a 100 m^2 apartment in Shanghai Jiao Tong University combined with small temperature difference fan-coil unit developed above. According to the tested heating rate above, four STDFCUs have been adopted in the system to satisfy the heating and cooling demands of the apartment.



Fig. 15 Photo of the final prototype

Table 7 Overall size of the final prototype

Parameter	Value/mm
Length	1160
Width	180
Height	480

The building plan, along with the location of four terminations, is shown in Fig. 16. This apartment includes two bedrooms, a living room, a dining room, and a kitchen. The indoor temperatures are controlled by indoor temperature controllers. Moreover, five temperature and humidity loggers are allocated in different places of the apartment to collect data.

The schematic of the air-source heat pump system is shown in Fig. 17. The system includes three major parts, including an air-source heat pump unit with a rated heating capacity of 9.5 kW, a 200 L water tank, and four small temperature difference fan-coil units. The water tank plays a critical role in the air-source heat pump system. On one hand, it serves as a heat exchanger to combine the heat pump unit and the indoor terminals. On the other hand, it works as a buffer to stabilize the water temperature when the outdoor unit is in a floating state. The photo of outdoor unit is shown in Fig. 18.

The inlet and outlet water temperatures are measured through T-type thermocouples. An ultrasonic flow meter is used to measure the water flow rate. A digital power meter is used to measure the overall power consumption, including the power consumption of the compressor and the water pump.

With operation parameters got from the test system, the system performance under different operation conditions can be then evaluated. The heating capacity of the ASHP Q_{ASHP} is calculated according to Eq. 6. The COPs of heat pump units and the whole system (including ASHP, water pump, and indoor units) are calculated according to Eqs. 7 and 8.

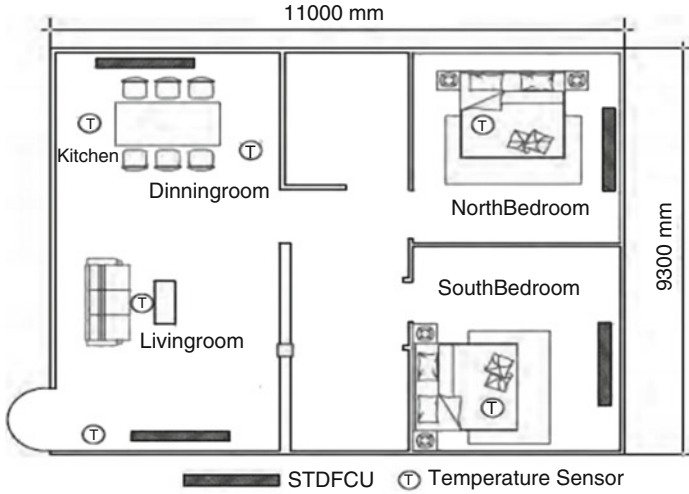


Fig. 16 Building plan

$$Q_{ASHP} = c m_{ASHP} (T_{in, ASHP} - T_{out, ASHP}) \quad (6)$$

$$COP_{ASHP} = \frac{Q_{ASHP}}{W_{ASHP}} \quad (7)$$

$$COP_{system} = \frac{Q_{ASHP}}{W_{ASHP} + W_{pump} + W_{indoor}} \quad (8)$$

Where

c [kJ/kg · °C] = specific heat capacity of water

m_{ASHP} [kg/s] = ASHP mass flow rate

$T_{in, ASHP}$ [°C] = ASHP supply water temperature

$T_{out, ASHP}$ [°C] = ASHP return water temperature

Q_{ASHP} [kW] = ASHP heating capacity

W_{ASHP} [kW] = compressor power consumption

W_{pump} [kW] = water pump power consumption

W_{indoor} [kW] = indoor terminal power consumption

Experiments and Discussions

To study the effect of water temperature on the energy consumption of air-source heat pump system, a 3-day experiment has been conducted under three different conditions. The outdoor air temperature during experiment is shown in Fig. 19. The average outdoor temperatures are 8.8 °C, 7.3 °C, and 6.4 °C, respectively. The return water temperatures are set to 30 °C, 35 °C, and 40 °C, respectively. The air temperatures of different rooms are all set to 22 °C. The average room air

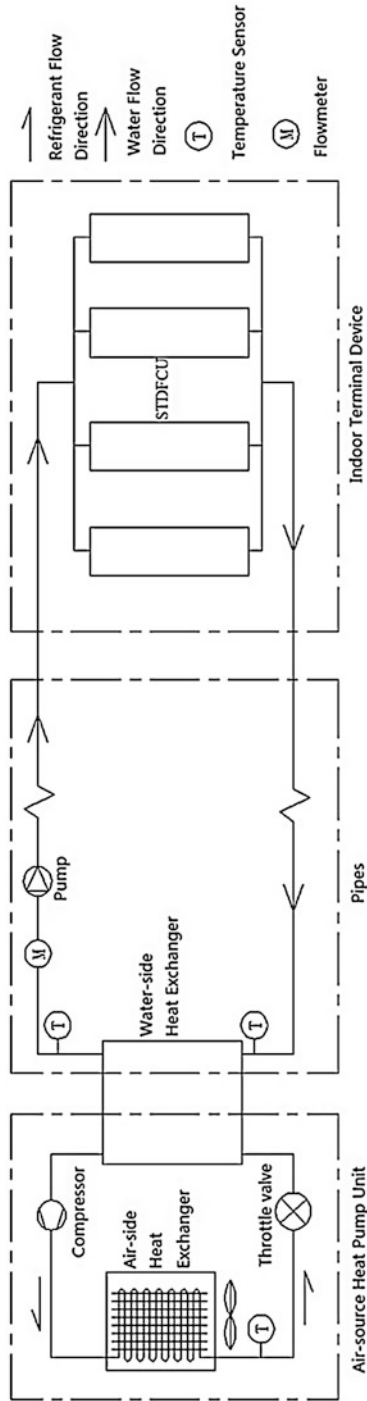


Fig. 17 Schematic of air-source heat pump system

Fig. 18 Photo of outdoor unit

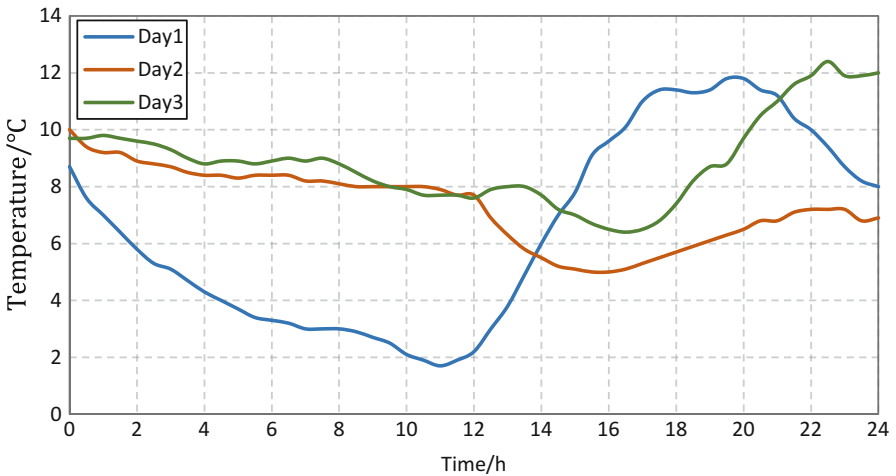
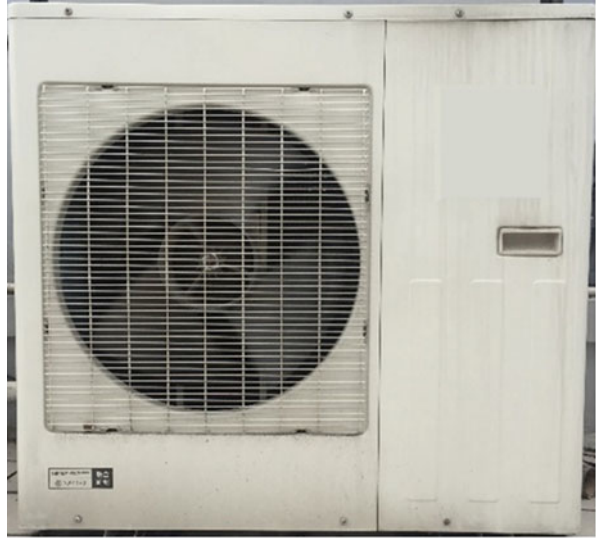


Fig. 19 Outdoor air temperature

temperatures during the test are shown in Table 8. It can be seen from the table that the air temperatures of different rooms are different from each other under different conditions, even though temperatures of these rooms are all set at 22 °C. Temperatures in living room, dining room, and kitchen are lower than the other two bedrooms in the same day because these three rooms have a comparatively higher heating load, and the terminals cannot provide enough heating capacity to keep a 22 °C environment. In addition, room temperatures in day 1 are a little lower than the

Table 8 Average air temperature in different rooms during the test

	Living/°C	Dining/°C	Kitchen/°C	North bed/°C	South bed/°C
Day1	19.5	19.2	18.5	20.1	21.8
Day2	19.9	20.0	19.2	21.3	23.3
Day3	19.8	20.2	19.5	21.5	23.6

other 2 days if statistics of the same room has been made a comparison. This is because the outdoor air temperature on this day is lower, which causes a higher heating load. Nevertheless, the room temperatures during the experiment are all higher than 18 °C which are in people's comfort zone.

Due to the control strategy of air-source heat pump unit, the compressor switches from status on and off regularly. The on-off frequency mainly depends on the relationship between system heating capacity and building load. To analyze the dynamic characteristics of the air-source heat pump system and find out the influence of water temperature on system energy consumption, details in 2 h from 00:00 to 02:00 are shown through the following figures, including water temperature, system power consumption, and energy consumption under three different conditions.

Temperatures of supply and return water during the test are shown in Fig. 20. As illustrated above, the return water temperatures of the system in day1, day2, and day3 are set to 30 °C, 35 °C, and 40 °C, respectively. Figure 20a shows the temperature variation from 00:00 to 02:00 in day1. According to the chart, it can be seen that the return water temperature floats around 30 °C, while the highest supply water temperature is about 37 °C. The air-source heat pump system starts and stops every 15 min. It keeps working for about 7 min and then stays standby for 8 min in a cycle. The on-off frequency is high under this condition, because the water temperature is low and hence the heat transferred to zones is decreased through terminals. As a result, it is easier and sooner for the return water temperature to get to the set point, which means that the compressor operates for a shorter time and consumes lower energy under this condition. Figure 20b shows the temperature variation from 00:00 to 02:00 in day2. The return water temperature floats around 35 °C, while the highest temperature of the supply water is about 42 °C. The air-source heat pump system starts and stops every 15 min. It keeps working for about 10 min and then stays standby for 5 min in a cycle. The compressor operates longer and consumes higher energy under this condition. Figure 20c shows the temperature variation from 00:00 to 02:00 in day3. The return water temperature floats around 40 °C, while the highest supply water temperature is about 47.5 °C. The air-source heat pump system starts and stops every 25 min. It keeps working for about 20 min and stays standby for 5 min afterward in a cycle. The compressor operates for the longest time and consumes the highest energy under this condition.

The system power consumptions under different conditions have been tested, including heat pump compressor, water pump, and terminals. The results of the test are shown in Fig. 21. It can be seen from the chart above that the sum of water pump and terminal power is about 300 W. It keeps constant under different conditions.

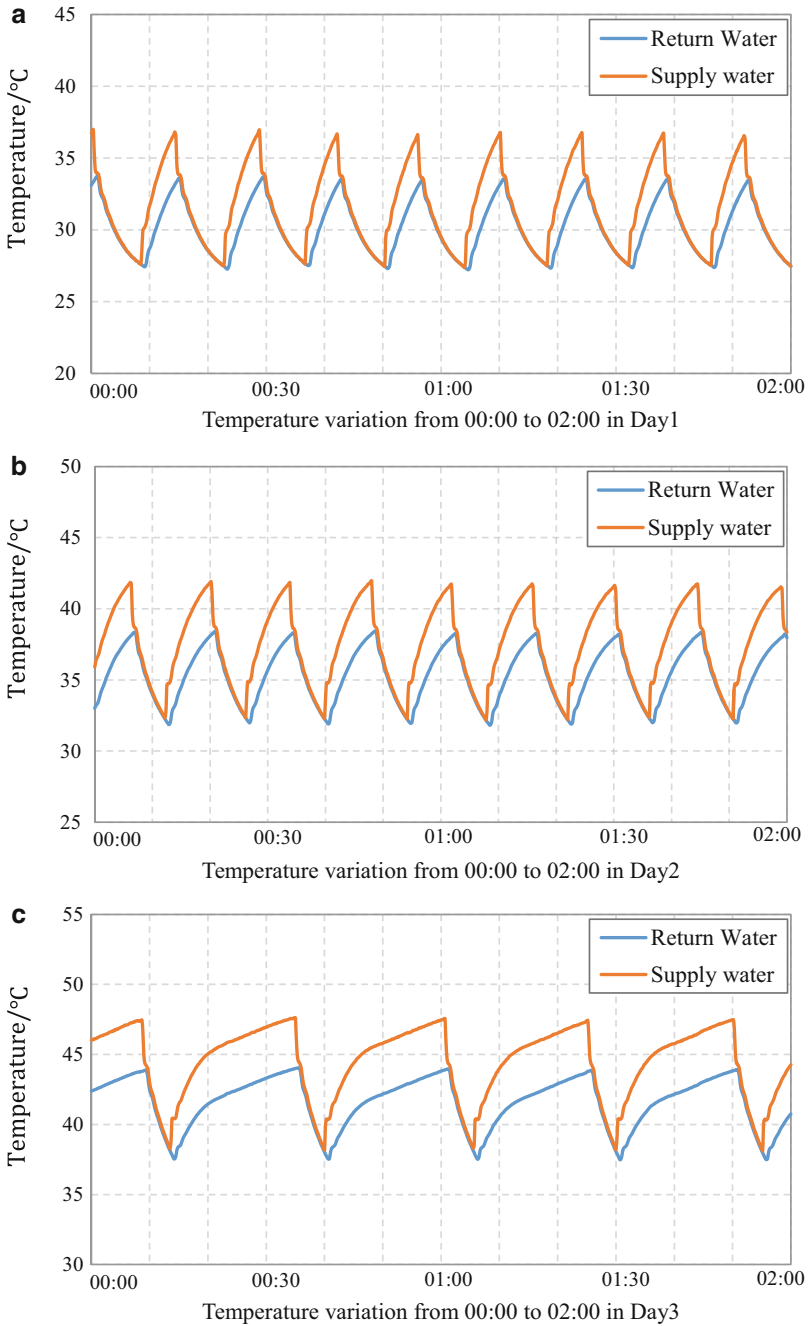


Fig. 20 Condition of supply and return water temperature in 3 days. (a) Temperature variation from 00:00 to 02:00 in day1. (b) Temperature variation from 00:00 to 02:00 in day2. (c) Temperature variation from 00:00 to 02:00 in day3

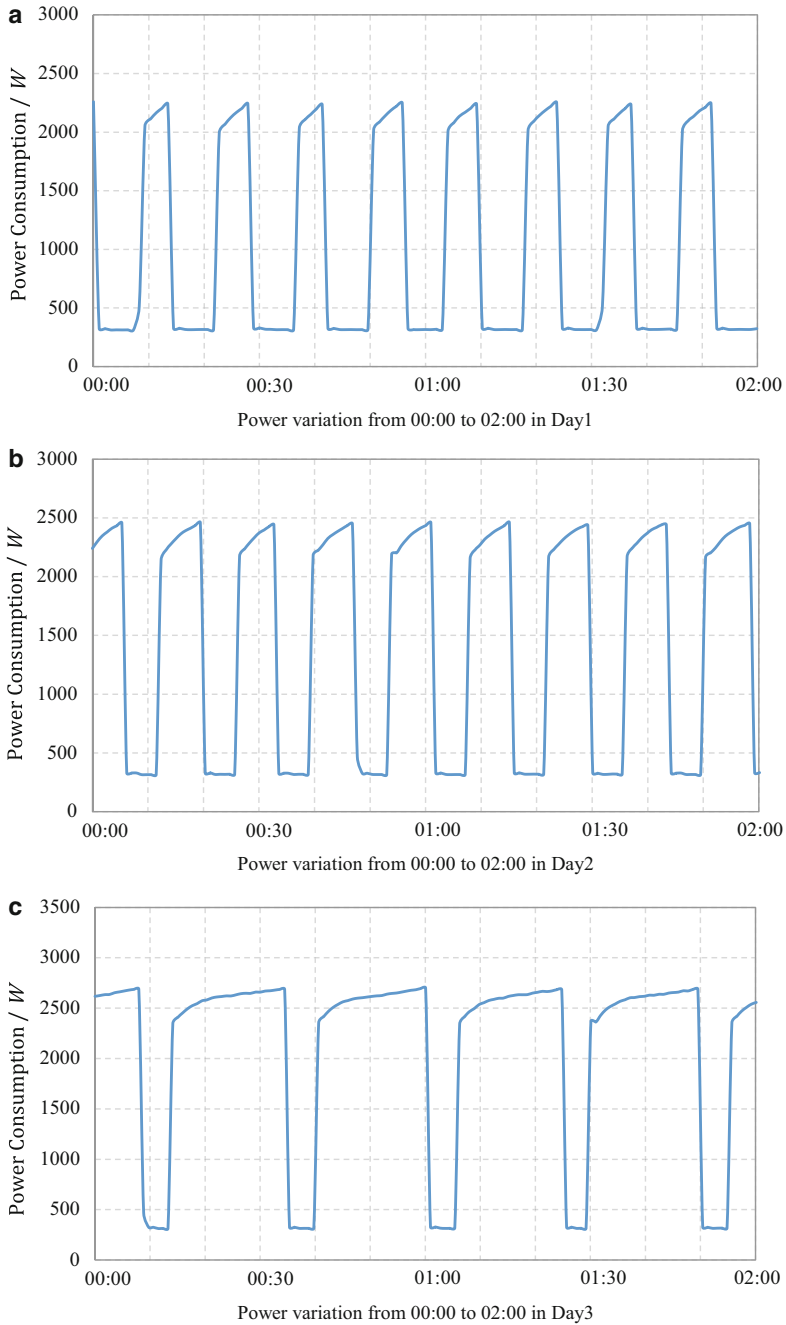


Fig. 21 System power consumption under different conditions. (a) Power variation from 00:00 to 02:00 in day1. (b) Power variation from 00:00 to 02:00 in day2. (c) Power variation from 00:00 to 02:00 in day3

Meanwhile, the compressor power differs a lot from each other under different conditions. The maximum power consumptions of the system under different conditions are 2250 W, 2500 W, and 2700 W, respectively. The maximum system power consumption increases by 20% when the return water temperature is increased from 30 °C to 40 °C. This indicates that the compressor power consumption can be affected greatly by the return water temperature. Moreover, it also implies that it's important to decrease the water temperature as low as possible when the heating capacity is enough to satisfy the building demand. In addition, the system on-off frequency can be easily observed in Fig. 21. According to what is shown from the chart, the ratio of system operating time in an on-off cycle is consistent with what has been illustrated above.

The system heating capacity has also been calculated by testing the inlet and outlet water temperature of air-source heat pump and water mass flow rate. The results are shown in Fig. 22. The maximum heating capacities under different conditions actually are almost the same to each other, which are all near to 9000 W. It indicates that the maximum heating capacity of the system shares no relation with the temperature of the return water. What's more, the system heating capacity also varies between 0 and 9000 W frequently because of the compressor's on-off control strategy. The on-off frequency is determined by the relationship between the heating capacity and the building load, just like what has been mentioned above. It can be seen from Fig. 22 that the total heating capacities during the test are quite different under different conditions. That is to say, the building load is different from each other because of different outdoor temperatures and room air temperatures. Certainly, it's hard to keep the two temperatures exactly the same. Besides, this will increase the cost of experiment. As a result, it's necessary to get a nondimensional number to evaluate the influence of water temperature on the whole system. This nondimensional number is known as COP, and the definition is shown in Eqs. 7 and 8.

The energy consumptions and COP of the air-source heat pump and the whole system under different conditions have been obtained in 24 h. The results are shown in Fig. 23. It can be seen that the energy consumption of heat pump is only 17 kWh when the return water is at the temperature of 30 °C. This is less than half of the energy consumption when the temperature of the return water is 40 °C. It indicates that the return water temperature has a great influence on the energy consumption of air-source heat pump. In addition, the air temperature of the room affects the building load a lot and thus influences the energy consumption indirectly. It can be concluded that a lower water temperature and room air temperature can lead to a great energy saving. The adoption of small temperature difference fan-coil units can realize the former target. The latter one needs to consider the comfort zone of people, which is not in the scope of this chapter. The COP of air-source heat pump is also influenced by water temperature. However, the differences of COP under different conditions are not as great as the energy consumption. It's because that the building loads are different, leading to different heating capacities. Figure b shows the system energy consumption under different conditions in 24 h, including the consumption of heat

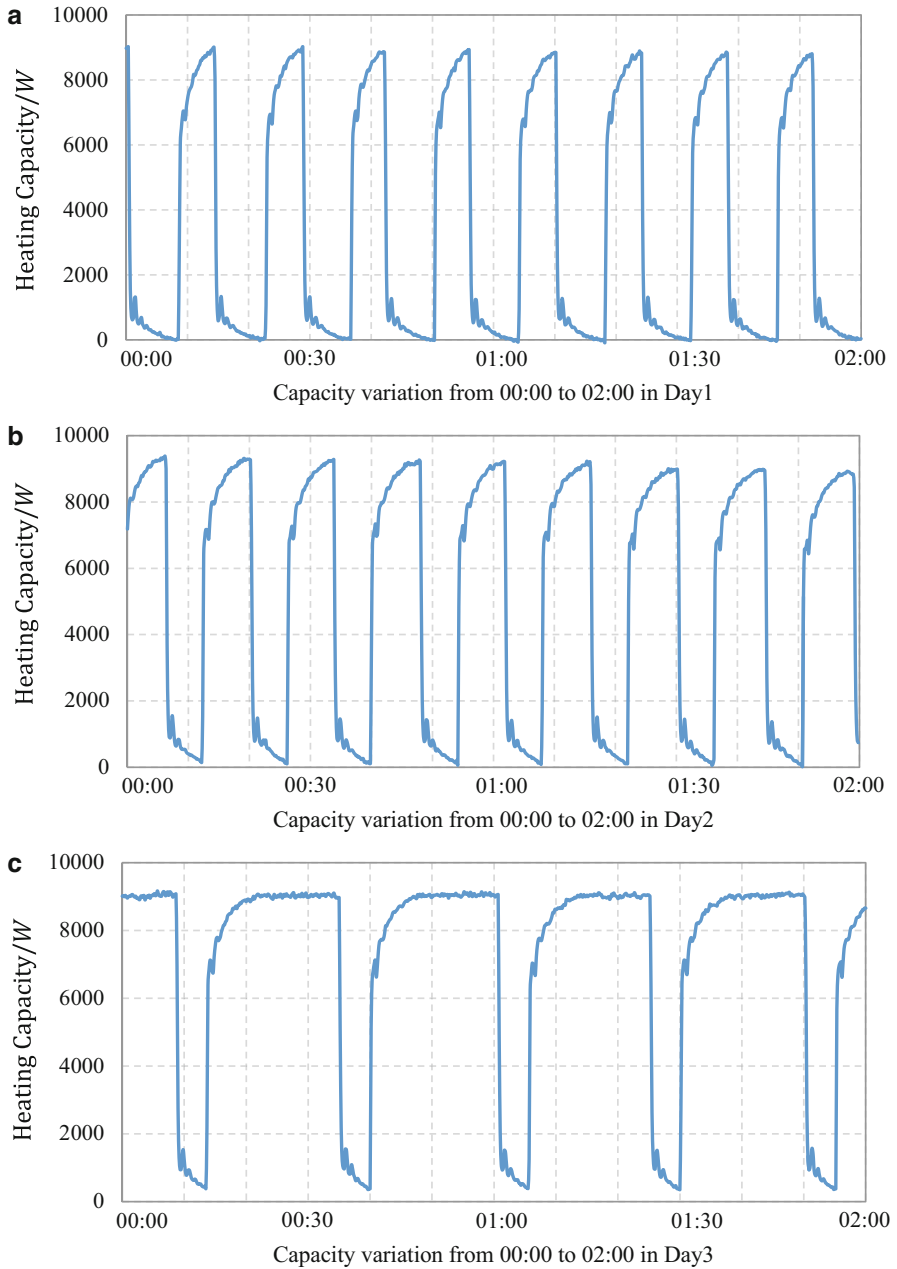


Fig. 22 System heating capacity under different conditions. (a) Capacity variation from 00:00 to 02:00 in day1. (b) Capacity variation from 00:00 to 02:00 in day2. (c) Capacity variation from 00:00 to 02:00 in day3

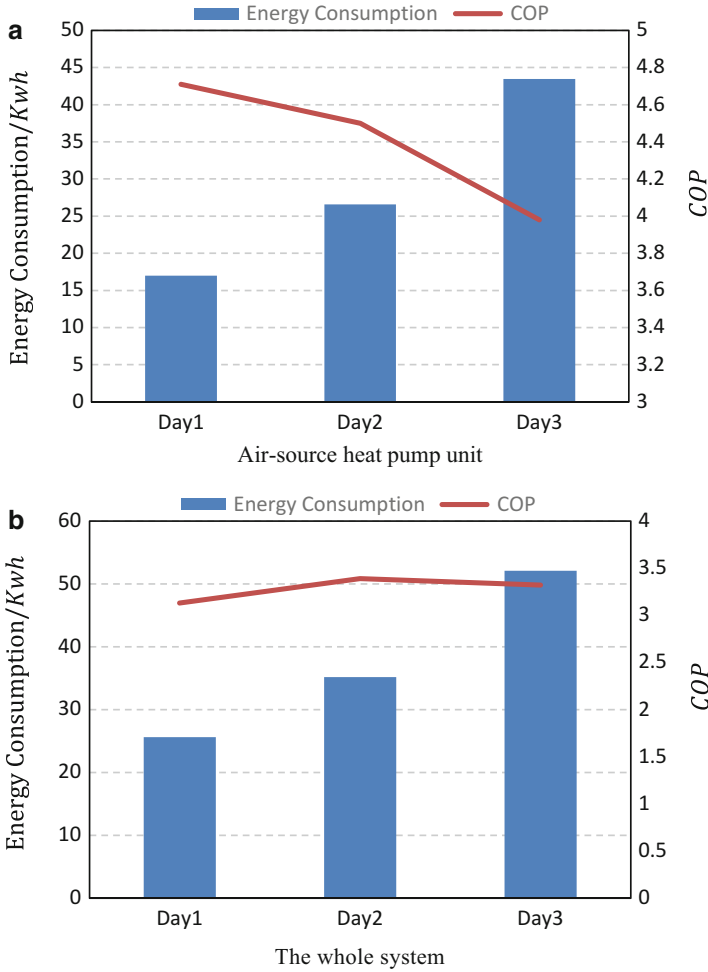


Fig. 23 Energy consumption and COP under different conditions. (a) Air-source heat pump unit. (b) The whole system

pump compressor, water pump, and terminals. Due to the extra constant energy consumption, the ratio of system energy saving is lower. However, the system energy consumption when the temperature of the return water is 30 °C is also almost half of the consumption if the temperature of the return water is 40 °C. However, the COP of the whole system is almost the same under different conditions due to the energy consumption of water pump, which keeps running in the whole 24 h and is not so energy efficient. If the water pump is replaced with a variable frequency one, the system performance can be improved. Besides, this is also going to increase the system COP a lot by decreasing the supply water temperature.

Fig. 24 Outdoor unit and expansion tank



ASHP System for Public Building

Four small temperature difference fan-coil units were adopted in a 100 m² apartment to explore the influence of water temperature on system energy consumption in its application to residence in the part above. This part presents an analysis of a larger air-source heat pump system's application to a public building. Besides, this part also presents an analysis over the performances of STDFCU, NFCU, and FHC which are analyzed including energy consumption and zone comfort.

System Description

An air-source heat pump system is installed in the Sino-Italy green energy lab located in Shanghai Jiao Tong University. The total air-conditioning area is 292 m², including one room on the first floor and four rooms on the second floor. The heating load is 34.1 kW, while the heating capacity of the ASHP unit is 39.5 kW, with a nominal power of 12.2 kW. The outdoor unit and expansion tank are shown in Fig. 24. The whole system consists of an air-cooled heat pump unit, water pump, water pipes, and indoor terminals. Figure 25 presents the schematic diagram of the experimental system. Most indoor terminals are normal fan-coil units, while three kinds of indoor terminals are installed in the room on the first floor and hence to compare the influence of different indoor terminals on system's energy efficiency and indoor thermal comfort. Figure 26 shows the detailed layout of this room on the first floor. The test time is 8:00 am to 5:00 pm every weekday in November and December 2014 and January 2015.

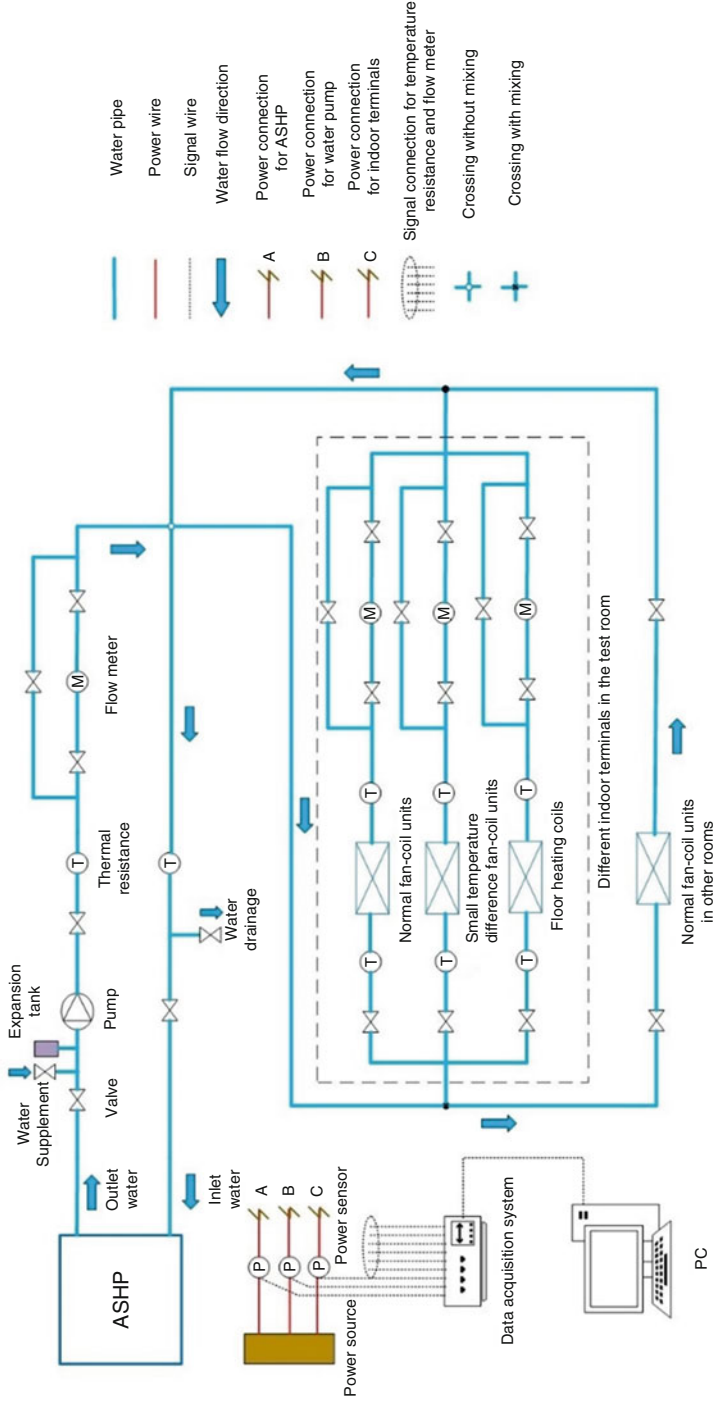


Fig. 25 Schematic diagram of the experimental system

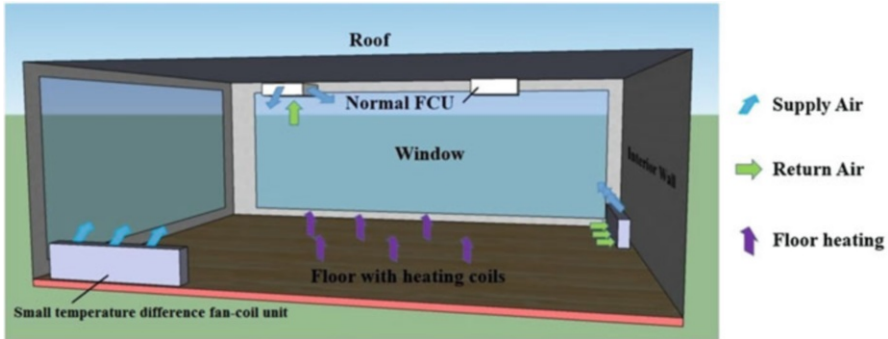


Fig. 26 Detailed layout of different indoor terminals in the ground floor

Table 9 Specifications of the sensors used in experimental system

Name	Number	Measurement range	Measurement uncertainty
Current transformer	3	0–50 A	5%
Flow meters	4	0–40 m ³ /h	5%
Thermal resistance	8	0–100 °C	3%
T&RH logger	20	T: 0–100 °C; RH: 0–100%	T: 5%; RH: 5%

As shown in Fig. 25, several kinds of sensors are used in the experimental system to monitor the operation parameter of the system. First of all, in order to get the energy consumption of different components in the system, current transformers are used in the power wires to gather power consumption data of ASHP unit, pump, and indoor terminal. Apart from the installation of transformers, flow meters are installed in four locations in the system to obtain the flow rate of ASHP total supply water and also the rate of three different indoor terminals. Thermal resistance Pt1000 are immersed into different locations of water pipe to get the inlet and outlet temperature of ASHP unit and different indoor terminals. Specifications of the sensors are shown in Table 9, while photos of them are shown in Fig. 27.

As for the temperature and humidity measurement of supply air, return air, and indoor environment, a portable temperature and relative humidity logger (T&RH Logger) is used to record the temperature change of different locations. T&RH Loggers for supply air and return air are placed in the middle of air inlet and outlet of the FCUs. Besides, T&RH Loggers for indoor environment are placed at five different spots in the room. T&RH Loggers at these spots are placed at five different heights that are 0.5 m, 1 m, 1.5 m, 2 m, and 2.5 m, respectively. T&RH Loggers’ horizontal and vertical layouts are shown in Fig. 28.

The system performance under different operation condition can be evaluated through the operation parameters obtained from the test system. The heating capacity of the ASHP Q_{ASHP} is calculated by Eq. 6. The COPs of heat pump units and the whole system (including ASHP, water pump, and indoor units) are calculated by Eqs. 7 and 8.

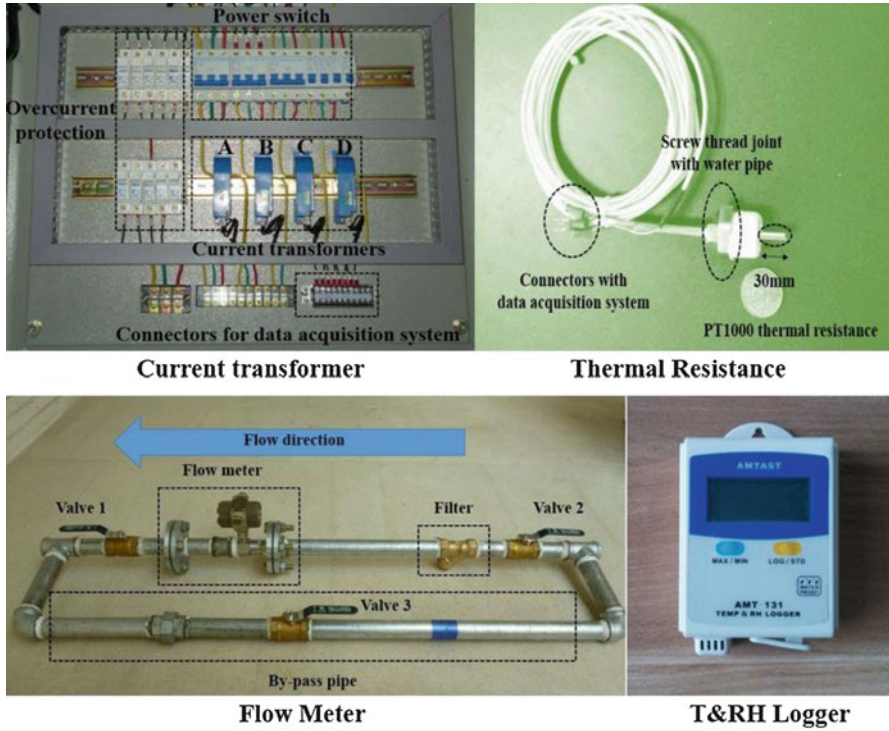


Fig. 27 Photos of different sensors used in experimental system

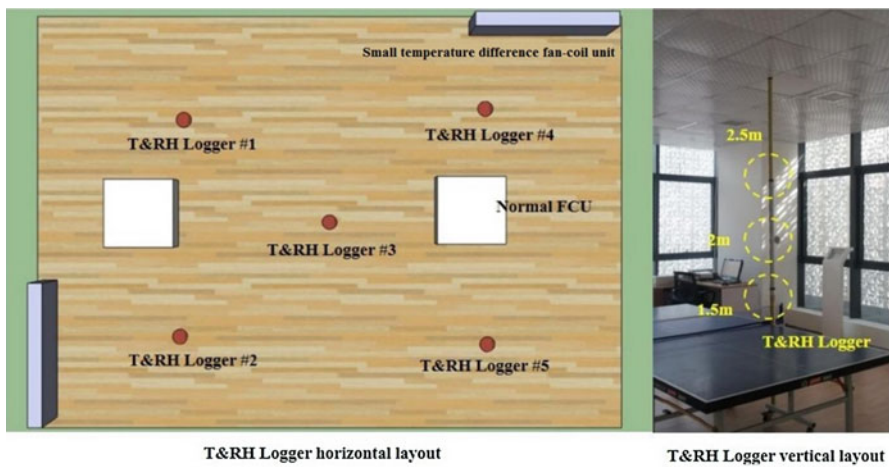


Fig. 28 Temperature and humidity loggers layout in the tested room

The uncertainties of Q and COP are calculated by Eq. 9.

$$\varepsilon(Q) = \sqrt{\left(\frac{\partial Q}{\partial m} \cdot \varepsilon(m)\right)^2 + \left(\frac{\partial Q}{\partial c} \cdot \varepsilon(c)\right)^2 + \left(\frac{\partial Q}{\partial T_{in}} \cdot \varepsilon(T_{in})\right)^2 + \left(\frac{\partial Q}{\partial T_{out}} \cdot \varepsilon(T_{out})\right)^2} \quad (9)$$

In this equation, $\varepsilon(m)$, $\varepsilon(c)$, $\varepsilon(T_{in})$, and $\varepsilon(T_{out})$ represent the uncertainty of flow rate, specific heat capacity, water inlet temperature, and water outlet temperature, which can be referred to Table 8. The uncertainty of Q and COP is 7.8% and 18%, respectively.

Cooling Mode in Summer

In order to make the results more comprehensible, a brief introduction of the ASHP unit's operation control strategy is presented first. The unit has two compressors, which can start and stop separately. The on and off of ASHP unit is controlled by inlet water temperature, while the number of working compressor depends on the building load. There are four typical operation modes in this test, just as shown in Fig. 29. For instance, if the inlet water temperature is set to 12 °C, then ASHP will turn off when inlet water temperature stabilizes at 12 °C. Otherwise, it will keep on working. The numbers of the working compressors also depends on the weather condition because different weather conditions will result in different cooling load.

In operation mode 1, the inlet water temperature decreases to the set point quickly after the ASHP is running for a short period of time. Afterward, it doesn't change much any longer. This mode usually happens in a very cool day when the building's cooling load is small. In operation mode 4, ASHP unit has always been working with two compressors. The ASHP unit usually has this mode in a very hot day or when there are much more occupancies than the design condition. Therefore, the inlet water temperatures are always higher than set point. In operation mode 2, the ASHP unit works between one compressor and off state. It usually has this mode in a relatively cool day when one compressor is enough to decrease the inlet temperature to the set point. Similarly, operation mode 3 happens in a relatively hot day, switching between one compressor and three compressors. This can be better observed from Figs. 30 and 31.

Energy consumption and system COP for different indoor terminals under different modes can be compared. It makes no sense to compare the two different systems in mode 1, because both are not working while the comparison results between mode 2 and mode 3 are similar in basic fundamentals, so the two different systems in mode 2 and mode 4 are compared.

Since the STDFCUs have comparatively stronger cooling capacity than the NFCUs, it means the STDFCUs can maintain the same indoor temperature with higher inlet water temperature in the same building under the same weather condition. Thus the ASHP inlet water temperature can be adjusted to a higher set point,

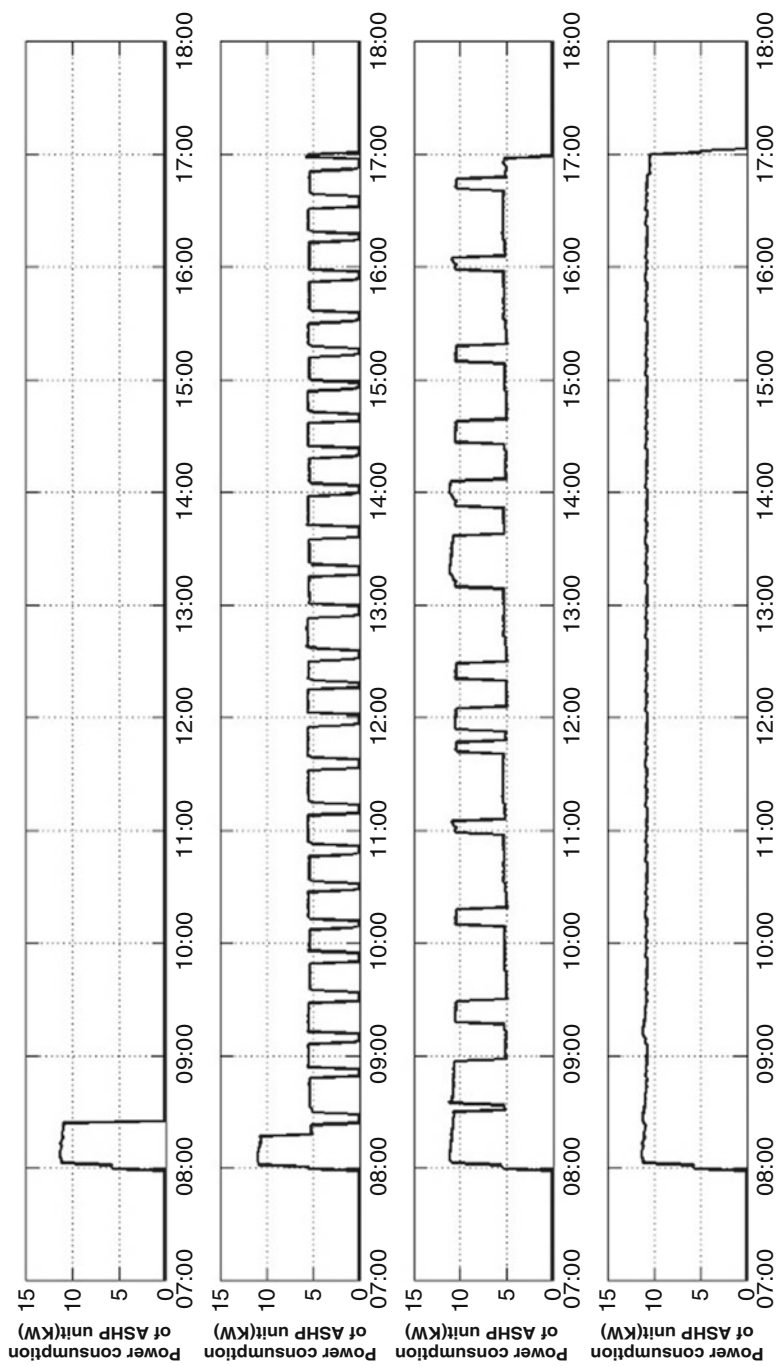


Fig. 29 Power consumption of ASHP unit in four different operation modes

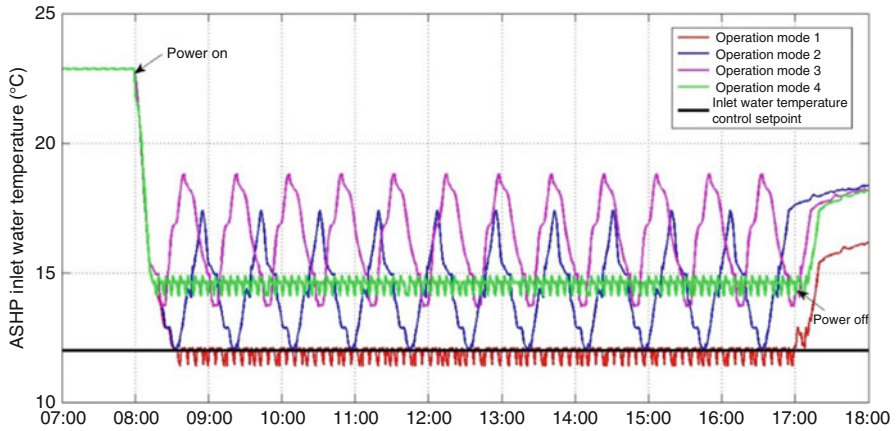


Fig. 30 ASHP inlet water temperature change in different modes during a day

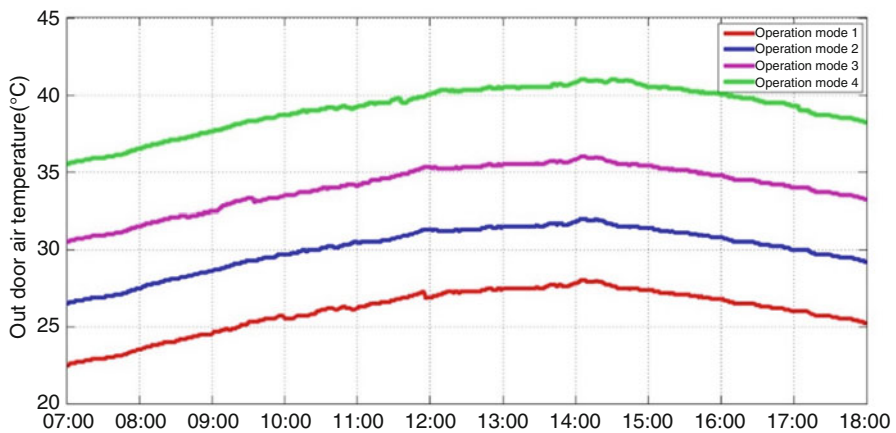


Fig. 31 Typical outdoor air temperature for different operation modes during a day

which can then have positive effect on the ASHP unit power consumption, just as what it is shown in Figs. 32 and 33.

In Fig. 32, it can be seen that ASHP + STDFCU system has a much shorter running period than that of the ASHP + NFCU system. And it will lead to a considerable energy saving which can be seen in Fig. 35. Also in Fig. 33, it can be seen that ASHP + STDFCU system has lower power consumption than ASHP + NFCU system. The reason for measuring this difference can be referred to Fig. 34. It shows that due to bigger cooling capacity of STDFCU, the inlet water temperature can be adjusted to a higher level, which makes the ASHP unit run in a better condition, thus reducing its power consumption.

In Fig. 34, it can be seen that the average outlet water temperature is 6.9 °C in the same building under same weather condition when ASHP is combined with NFCUs.

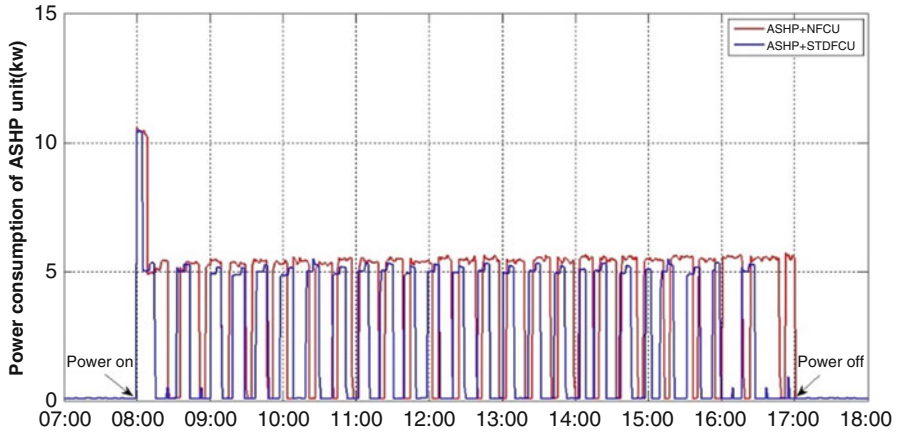


Fig. 32 Power consumption of ASHP unit with different indoor terminals during a day in operation mode 2

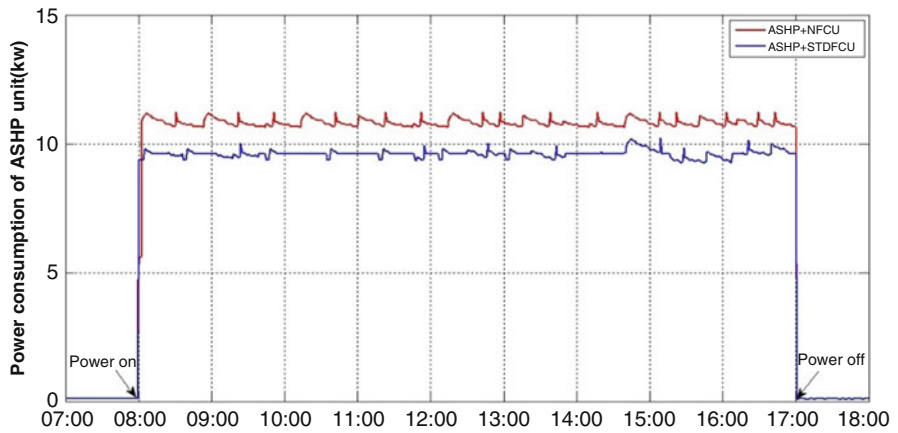


Fig. 33 Power consumption of ASHP unit with different indoor terminals during a day in operation mode 4

When it is combined with STDFCUs, the average outlet water temperature is 10.5 °C. Thus the operation mode of ASHP is greatly optimized, especially under extreme weather conditions. That's also the greatest advantage of using STDFCU instead of NFCU.

Figure 35 illustrates the daily energy consumption of ASHP system with STDFCUs and NFCUs under similar weather condition. Both ASHP + STDFCU system and ASHP + NFCU system's power consumption increases as the outdoor air temperature goes up. However, under every weather conditions, ASHP + STDFCU

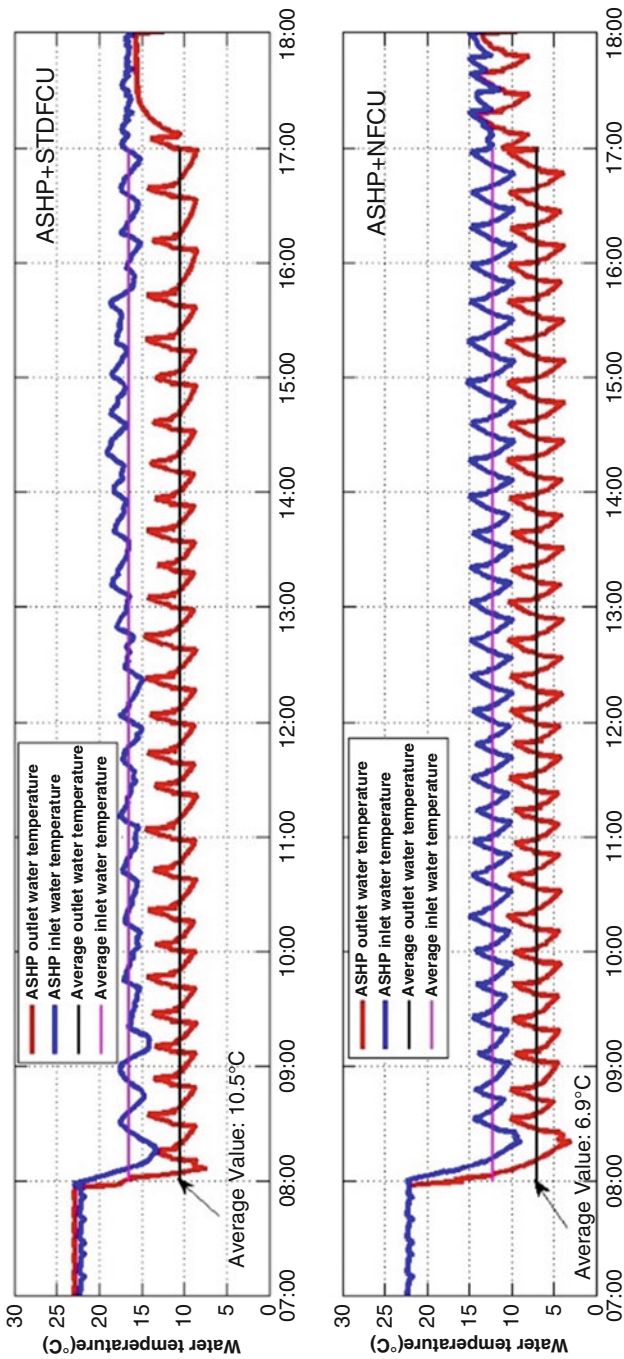


Fig. 34 ASHP inlet water temperature change for ASHP with STDFCUs and NFCUs

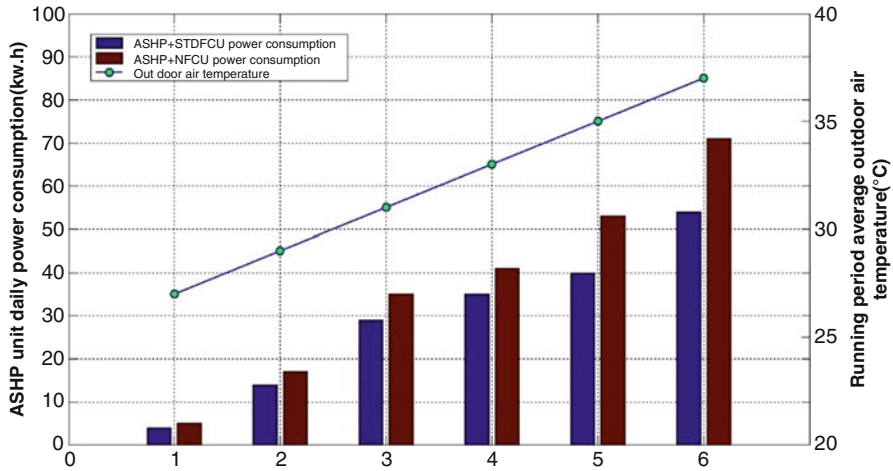


Fig. 35 Daily energy consumption of ASHP with STDFCUs and NFCUs under different weather conditions

consumes 15–22% less energy than ASHP+NFCU, depending on the particular weather condition.

COP performance of both ASHP+STDFCU and ASHP+NFCU can be compared during the test period as well. It is noteworthy here that since ASHP unit may be at the off-state status during particular hours, comparing the hourly COP of the system will be meaningless. Because the COP will be infinite for ASHP unit at off state. So here the daily energy consumption of ASHP with STDFCUs and NFCUs are compared under different weather conditions. Figure 36 illustrates that ASHP+STDFCU is 13–24% higher in COP compared with that of ASHP+NFCU under similar weather condition.

The indoor temperature distribution for different indoor terminal locations serves as another important aspect of the investigation. Here only the temperature distribution at different horizontal and vertical locations are discussed when the indoor air temperature is steady.

Figure 37 shows the vertical temperature distribution at different heights for STDFCU and NFCU at different measurement locations. For all the locations, the indoor air temperature has a slight increase as the height increases. This distribution is reasonable, considering the buoyancy effect of hot air.

Yet, the temperature distribution at a working area height of 1.5 m is more important, so the horizontal temperature distributions at this height are compared in Fig. 38. From Fig. 38, it can be seen that for STDFCUs installed in the lower part of a room, measurement point #5 has the lowest temperature, while point #1 has the highest temperature. For NFCUs that are installed in the ceiling of a room, measurement point #3 has the lowest temperature, and other points have similar temperature. For STDFCUs, the average temperature at 1.5 m heights is higher than NFCUs.

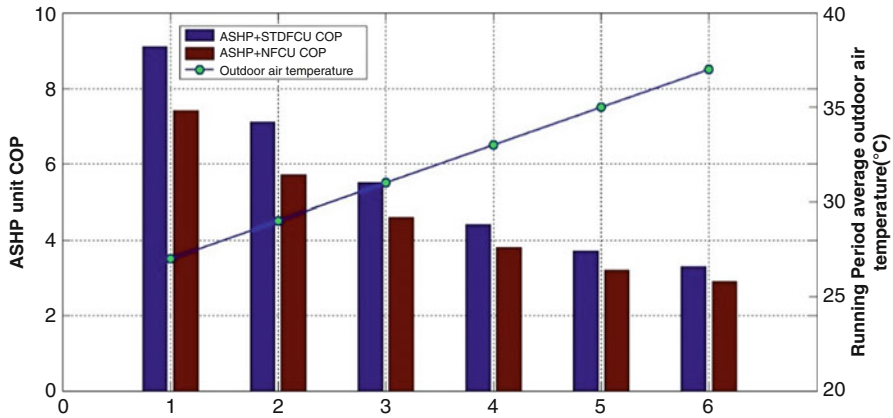


Fig. 36 Daily COP performance of ASHP with STDFCUs and NFCUs under different weather conditions

From this case study, it can be concluded that the NFCUs seem to have a better indoor air distribution than STDFCUs do in the cooling mode. However, this conclusion cannot be applied into every case, because the installation location of FCUs differs from case to case. The indoor air distribution in heating mode will be discussed later.

Heating Mode in Winter

Similar to the cooling conditions test, the basic idea of the heating conditions test is to compare the effect of different types of terminals on the system energy efficiency under similar outdoor temperature conditions. Different from the cooling test, not only normal fan-coil units and small temperature difference fan-coil units are adopted but also floor-heating coil is added to the comparison during the heating test (Fig. 39). The structure of floor-heating system is listed below:

Five centimeter thick EPS polystyrene foam board + single-layer aluminum foil + three heat-resistant polyethylene pipe PERT loop (pipe diameter 2 cm, pipe space 10 cm) + concrete 4 cm + composite wood floor 12 mm.

Turns out, there are three combinations of air-source heat pump unit and terminal under winter heating conditions. The temperatures of the return water in the unit are set to different values and hence to compare the water temperature demand of different terminals. In addition, this can be also used to compare the influence of different terminals on the energy consumption of system and indoor thermal comfort.

In this part, 12 days in winter are chosen as the experiment days and divided into four groups (marked as group 1, group 2, group 3, and group 4 in the following figures). Figure 40 shows the outdoor air temperatures of these selected 12 days.

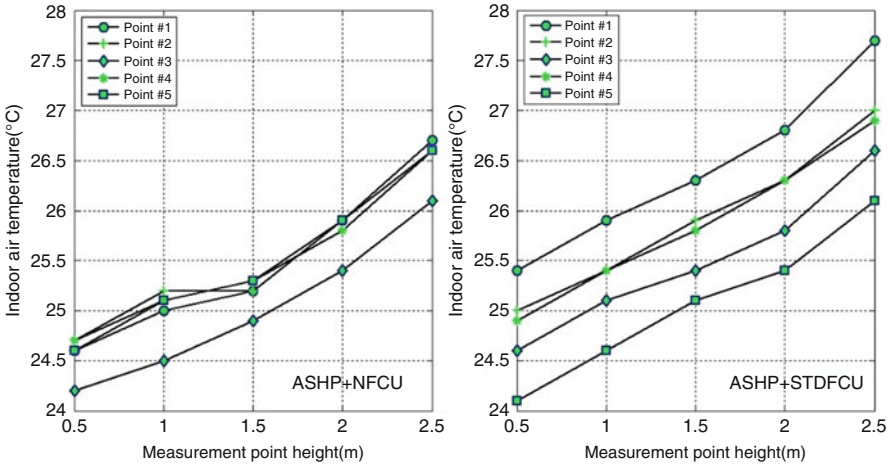


Fig. 37 Vertical indoor air temperature distribution at different locations

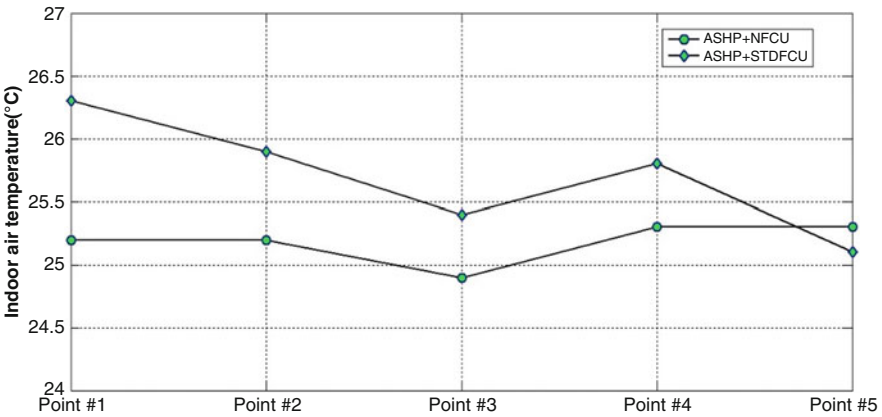


Fig. 38 Horizontal indoor air temperature distribution at 1.5 m heights

Figure 41 presents the comparison of the ASHP unit’s power consumption. The power consumptions in three different days are obtained. It can be seen that system with STDFCU and NFCU almost operates with one compressor, while the system with FHC operates with one or two compressors and stops for a long time. This is because radiant system has a higher thermal inertia compared with fan-coil system. In addition, system with STDFCU has lower power consumption and operates for a shorter time than that with NFCU. Tracing its reason, it can be found that STDFCU has a higher heat transfer performance and needs a lower-temperature water.

Fig. 39 floor-heating coils used in the test room

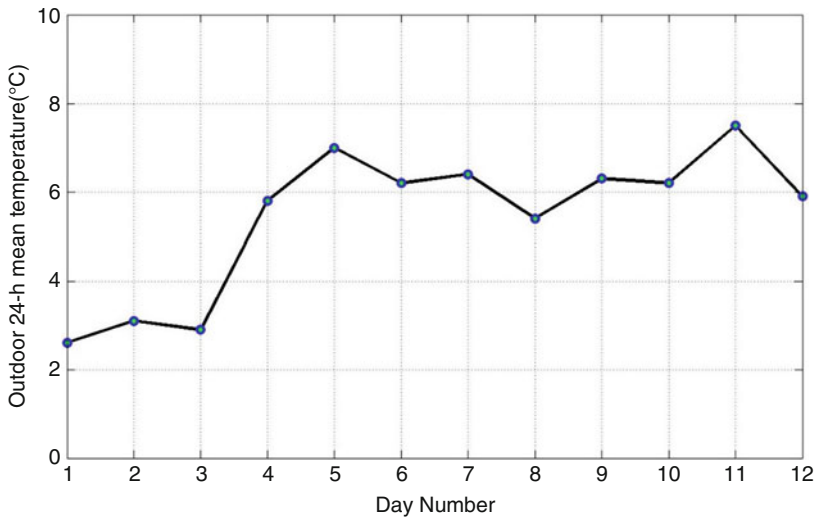


Fig. 40 Outdoor air temperature during the test days

In Fig. 42 presents the indoor air temperature change curve shown by three different kinds of terminals. Similarly, the system is turned on at 8:00 and turned off at 17:00. Compared with FCUs, FHC has a very obvious feature, which is its long response time. When FCU is turned on, the indoor air temperature will increase rapidly and then reaches the set point (20 °C) in no time, while it takes FHC comparatively to reach steady state, which is about 90 min. This response time also depends on various conditions including floor-heating coils construction, building construction, etc. However, floor-heating coils have a longer response time than fan-coil units in most cases.

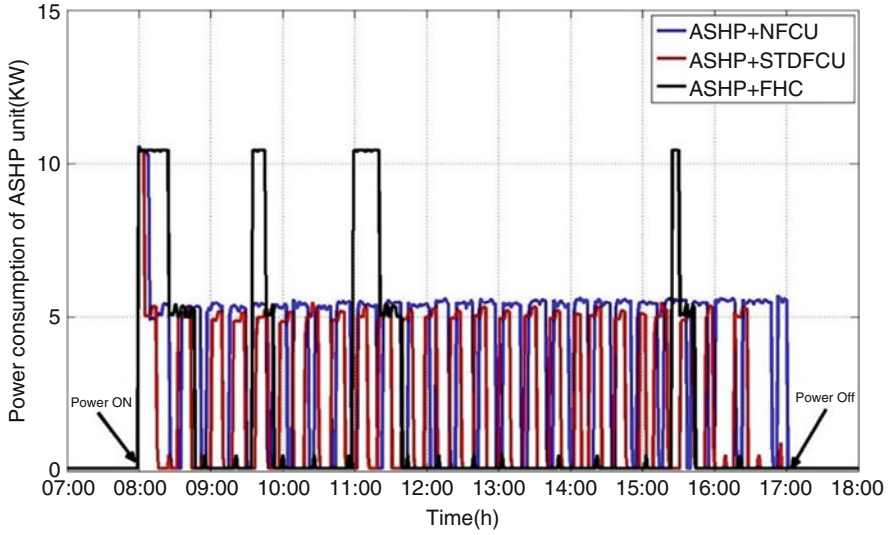


Fig. 41 ASHP system power consumption with different kinds of terminals

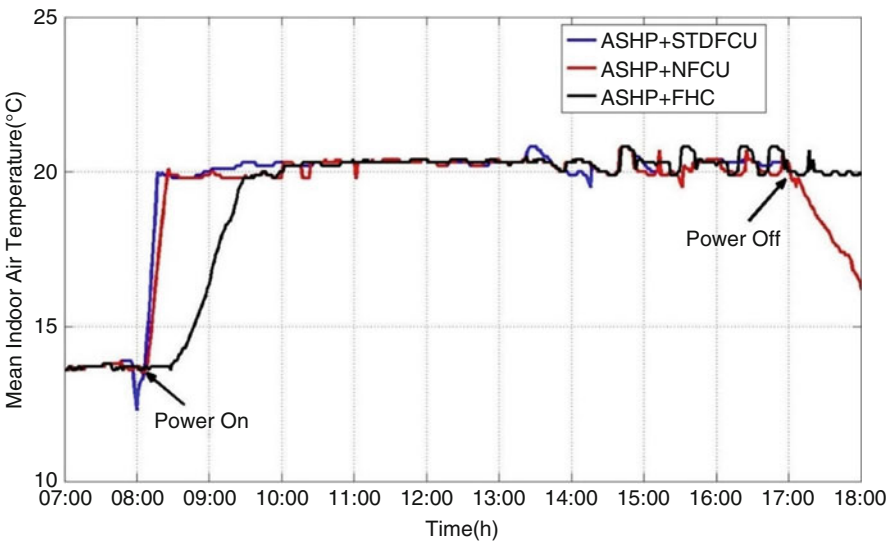


Fig. 42 Response time of three different terminals

Figure 43 presents the comparison of the daily energy consumption of ASHP system. The energy consumption in group 1 is higher than that in other groups because the average outdoor temperature is 3 °C in group 1 and 6 °C in other groups, just as what is shown above. Besides, a lower outdoor temperature will lead to a higher heating load and energy consumption. What's more, the system with NFCU

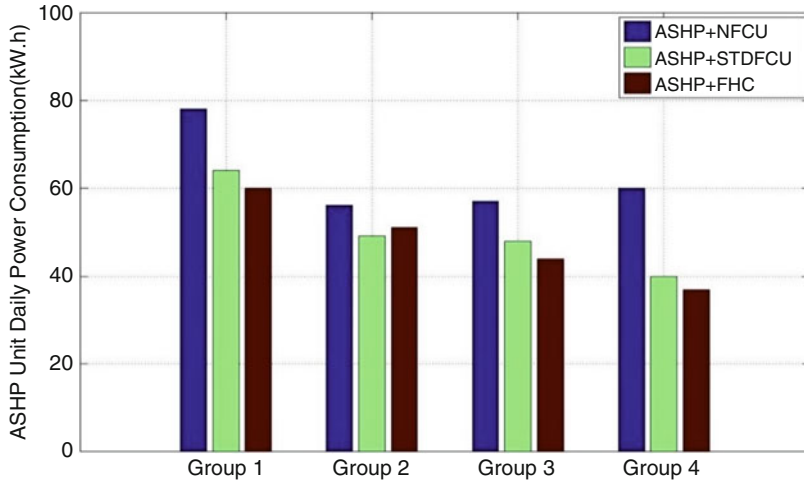


Fig. 43 Energy consumption of ASHP system with different terminals

consumes more energy. The energy consumptions of other two systems are close to each other. Taking group 1, for example, the daily energy consumption of system with NFCU is close to 80 kWh, while other two systems consume only 60 kWh. That is to say, the other two systems consume 25% less energy. This is because of the difference of their supply water temperatures that are illustrated above. The inlet and outlet temperature of NFCU and STDFCU are shown in Fig. 44. It can be seen that the NFCU has an average outlet temperature of 42 °C, which is 5 °C higher than that of STDFCU.

The COP of ASHP system with different terminals is shown in Fig. 45. Compared to NFCU, using STDFCU and FHC can effectively increase the COP of system. When outdoor air temperature is around 3 °C, the COP of system with NFCU is around 2.7, while COP of system with STDFCU and FHC is above 3. The difference in COP can further prove the advantage of using STDFCU or FHC.

As for the indoor thermal comfort, the indoor air temperature distribution in both vertical and horizontal direction is tested. The indoor air temperature difference in vertical direction is shown in Fig. 46. It is rather obvious that the temperature gradient in vertical direction is the largest with NFCU working in the room. The maximum temperature difference is 3.5 °C. As for the STDFCU, the temperature gradient is lower, while the maximum temperature difference is about 2 °C. It's because that the STDFCUs are installed on the floor, and the air distribution in the room is more reasonable than that with NFCU installed on the ceiling. The temperature distribution in vertical direction is the most uniform when the indoor terminal is FHC. The temperatures are almost the same, which is the advantage of radiant terminal under heating conditions.

Indoor air temperature distribution in horizontal direction is shown in Fig. 47. The measurement point is at a height of 1.5 m, which is also the height of thermostat.

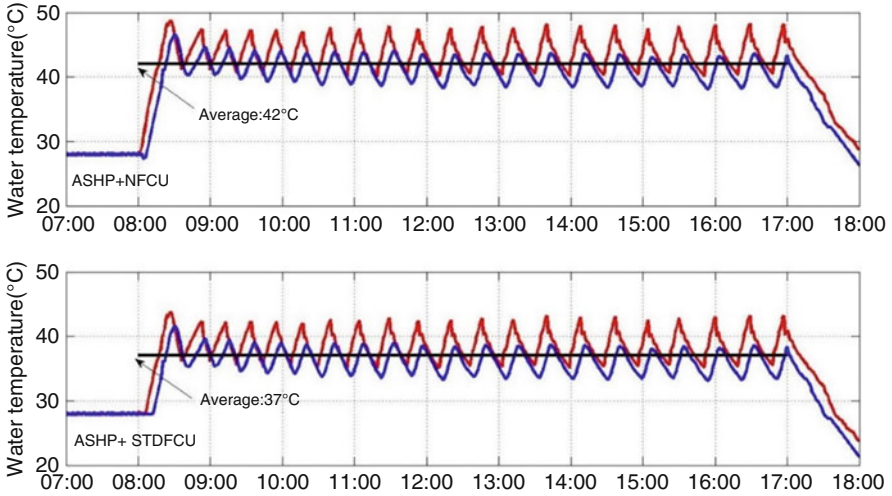


Fig. 44 Inlet and outlet water temperature of ASHP with different terminals

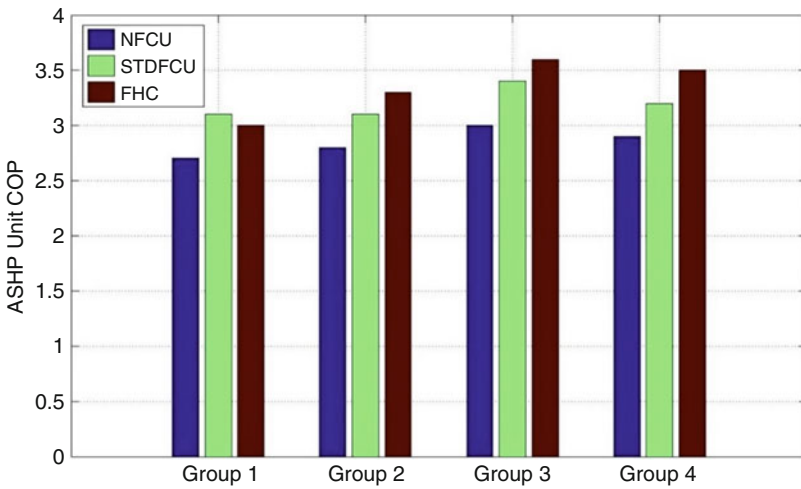


Fig. 45 COP of ASHP system with different terminals

It can be seen that the temperatures at this height are almost the same. Only temperature distribution with NFCU is slightly nonuniform due to the terminal install location. It indicates that the temperatures are all controlled well to the set point, and the comparison of energy consumption and COP is convincing.

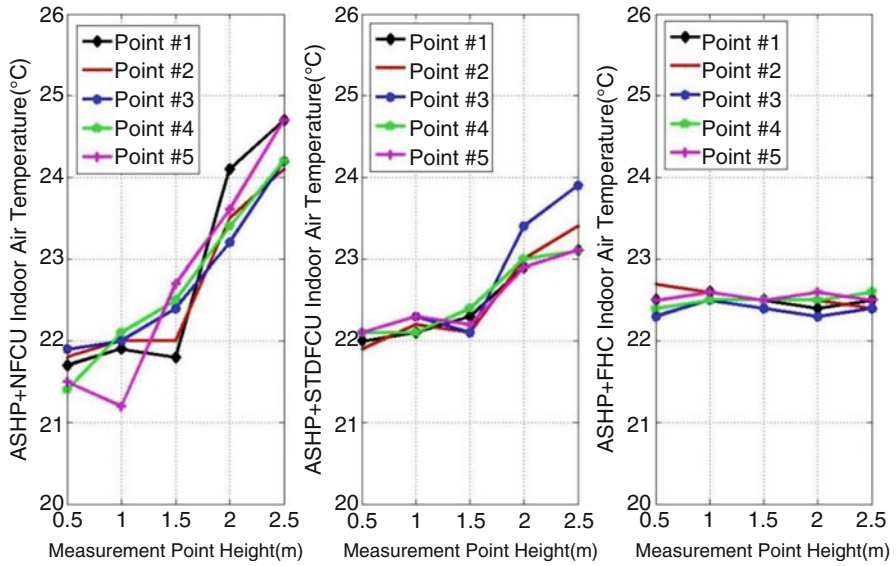


Fig. 46 Indoor air temperature distribution in vertical direction

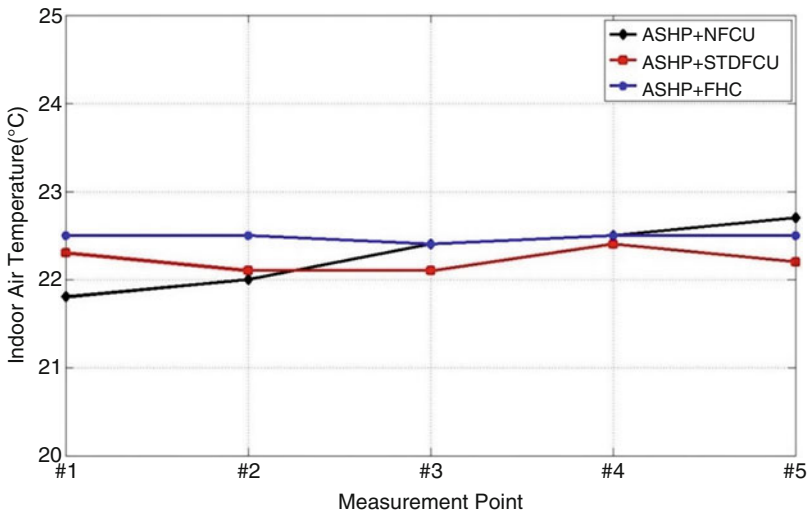


Fig. 47 Indoor air temperature distribution in horizontal direction

Conclusion

This chapter has introduced and studied small temperature difference terminals. The small temperature difference fan-coil units, which are composed of fin-tube heat exchanger, cross-flow fan, brushless DC motor, and a condensate plate, are introduced first. The concept of small temperature difference fan-coil unit makes a compromise between initial cost and operation cost. With the development of air-source heat pump system, it becomes normal that air-source heat pump is used for both heating and cooling in the whole year in hot summer and cold winter climate regions. This illustrates that the system operates longer than before. Besides, the performance of it not only in summer but also in winter should be fully considered. The proposal of small temperature difference fan-coil unit means that more heat transfer enhancement technologies are applied to the terminal and more heat transfer areas are adopted. That is to say, the initial cost needs to be increased to make the terminal and air-source heat pump system more energy efficient, which is a trend around the world. Then, another kind of small temperature difference terminal floor-heating coil is introduced. It transfers heat mainly through radiation. Besides, it has some advantages in heating such as making a better indoor thermal comfort, decreasing the indoor air temperature demand, and occupying little indoor space. However, the floor-heating system isn't suitable for the retrofit of existing building and can't be used for cooling. After that, a small temperature difference fan-coil unit is designed and manufactured. Moreover, the heat transfer performance is tested under different conditions. The results show that the terminal has a heating capacity of 2090 W under the condition of 45 °C water, which can satisfy the design goal. What's more, the designed prototype is improved in its appearance to be a real product and applied to a 100 m² apartment with an air-source heat pump system. The system performance is tested under different conditions of different return water temperature. According to the result, it shows that the small temperature difference fan-coil unit can decrease its demand water temperature due to its better heat transfer performance. The system energy consumption can be reduced a half if return water temperature is decreased from 40 to 30 °C while the room air temperature is still in a comfort level. Finally, a larger air-source heat pump system is applied to a small office. The performances of STDFCU and NFCU in cooling mode are analyzed, including its energy consumption and indoor temperature distribution in summer. Meanwhile, the performances of STDFCU, NFCU, and FHC in heating mode are analyzed from aspects of energy consumption and zone comfort in winter. The result shows that ASHP + STDFCU system can save 15–22% energy compared to traditional ASHP + NFCU system. Thus, it has a 13–24% higher COP than ASHP + NFCU system depending on the weather condition in summer. What's more, STDFCU and FHC can reduce system 20% of the energy consumption in the heating mode. Besides, the air distribution in the room is more uniform in winter. The FHC is only suitable for heating condition and can't operate in summer, and one more system is needed for cooling which will increase the initial cost. In a word, the STDFCU is the most suitable terminal in those climate regions that have demands of both heating and cooling.

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