FIELD MEASUREMENT OF A RESIDENTIAL FLOOR COOLING SYSTEM AND EVALUATION OF HUMAN THERMAL COMFORT

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Abstract:

A residential floor cooling system was tested, and the indoor thermal environmental parameters during the stable operation period of the system were measured. Based on the measured data, the indoor thermal environmental characteristics were analyzed, and the human thermal comfort was also evaluated from two aspects including overall thermal comfort and local thermal discomfort. The results show that during the stable operating period of the system, the PMV index can be controlled with a range of -0.32 – 0.35 during a test day cycle through the on-off control of the chiller despite a large variation of outdoor air temperature. Due to a small supply of air volume, the local thermal discomfort caused by draft can be avoided. Because of the large thermal inertia and large storage coolness capacity of radiant floor and other envelopes, the indoor temperature fluctuates little with load variation so that there are no discomforts due to temperature fluctuations with time. Because residential building load is relatively small and thus the cooling floor bears only a small cooling load, the cooling floor surface temperature is not too low to cause local discomfort of the human feet. Floor cooling increases the vertical gradient of air temperature, and thus causes local discomfort of the human body. The results can provide a reference for the application and promotion of a residential radiant floor cooling system in similar buildings.

1 Introduction

Located around the Yangtze River, with special architectural climate characteristics due to their unique geographical location, China's hot summer and cold winter zones are hot and fuggy in summer, clammy in winter, and have a long humidity climate period. In addition, these regions have a large population density, and their economy and culture are both well developed. Unique architectural climate features, higher level of economic and cultural development make the summer and cold

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winter regions' requirements for healthy and comfortable living environment particularly prominent. At present, popular summer air-conditioning devices are mainly dominated by convection air-conditioners, which make room temperature uniform, poor comfort, and high energy consumption. Due to the shortcomings of traditional residential air-conditioning systems, there is an urgent need for a new air-conditioning system to replace the traditional air-conditioning system. A radiant cooling system is an effective choice, and has received a lot of concerns by a growing number of HVAC engineers and occupants. Studies have shown that radiant cooling systems have the following advantages over traditional air-conditioning systems by convection [1-7]: (1) Reducing indoor vertical temperature gradient, almost no air flow, reducing local discomfort. (2) The smaller air supply volume can reduce cooling feeling due to cold air, and thus it can reduce discomfort caused by draft. (3) The radiant terminal device bears all or part of the sensible heat load, and the air supply terminal device only bears the latent heat load and the residual sensible heat load, and meets occupants' healthy needs for fresh air. The smaller air volume can reduce air transport energy and then achieve energy-saving effect. (4) Reducing indoor noise. Radiant cooling systems have been widely used in Europe, mainly in commercial and office buildings. In recent years, radiant cooling systems have received more and more attention so that they have been increasingly used in China, and gradually also in residential buildings. Currently, domestic radiant cooling cases/systems are mainly applicable to the technical systems of radiant cooling combined with ground-source heat pump, and the district buildings are centralized cooled. However, centralized cooling cannot achieve occupants' flexible adjustments neither meet their individual needs.

In view of the current problems existing in indoor living environment in hot summer and cold winter zones of China, and a wide range of needs for energy efficient control equipment of heat and moisture environment in this zones, supported by the Key Project of the National Eleventh-Five Year Research Program of China 'Research and Demonstration on Low-Power Control Technology of Indoor Thermal Environment in Residential Buildings around the Yangtze River Valley', based on the climate characteristics in these regions, this research studied the combination of active and passive air-conditioning technology system which could compatibly meet heating, cooling, dehumidification and ventilation requirements, and developed a residential environmental control machine [8-11], which used the air source heat pump as the hot and cold sources, radiant floor or radiant ceiling for heating and cooling, and also employed the outdoor air supply system to meet both indoor health requirements and dehumidification needs.

There are mainly two patterns used in radiant terminal devices: (1) radiant floor for heating in winter and radiant ceiling for cooling in summer; (2) uniform radiant terminal device such as radiant floor or radiant ceiling for both heating and cooling. To the second pattern, summer cooling function is added on the basis of the traditional floor heating system. It can reduce the initial investment thanks to using only a set of air-conditioning terminal device. The technology of floor heating in residential buildings is relatively mature, and its advantages in terms of thermal comfort have been widely recognized. However, there are few literatures about the application of floor cooling in Chinese residential buildings. This paper aims to evaluate the application of a residential radiant floor cooling system combined with an outdoor air handling unit applied in hot summer and cold winter zones of China and based on the field measurement. The tested system was located in a villa in Shanghai City in China. The indoor thermal environment parameters in stable operating conditions were both measured and analyzed. Also, the human thermal comfort was evaluated from two aspects including overall (thermal comfort) and local thermal comfort. The results can provide a reference to the current application and promotion of residential radiant floor cooling system in similar buildings.

2 Introduction of tested system

2.1 Introduction of the tested residential building

The tested residential building is a three-layer linking row villa, whose floor area is approximately 170 square meters long. The 30 mm expanded polystyrene (EPS) board was taken as the thermal insulation layer for its external wall, and the 40mm extruded polystyrene (XPS) board as the thermal insulation layer for its roof. The thermal transmittance values for building envelopes are shown in Table 1. Contrasted and compared with the limits of the standard ‘design standard for energy efficiency of residential buildings in hot summer and cold winter zones of China’ [12], all the coefficients shown in Table 1 are within the limits, and meet the energy efficiency design requirements. For HVAC system design in Shanghai, outdoor design dry-bulb temperature (DBT) in summer is 34 °C, and the design outdoor wet-bulb temperature (WBT) is 28.2 °C. Outdoor design DBT in winter is -4 °C. In conventional HVAC system design for residential buildings, many engineers use 26 °C DBT and 50% relative humidity (RH) in summer, and 18 °C DBT and 50% RH in winter as a target space condition [12]. However, according to the literature [13, 14], if radiant cooling is used for space cooling and radiant heating is used for space heating, 1–2 °C higher design space temperature in summer and 1–2 °C lower design space temperature in winter can be used without significant negative impact on thermal comfort. Therefore, the target space condition in the research was set at 28 °C DBT and 50 % RH in summer, and 16 °C DBT and 50 % RH in winter. Design cooling load of the tested building is 4652 W, and the cooling load per unit area is 33.7 W/m^2. Design heating load is 4077 W, and the heating load per unit area is 29.5 W/m^2.

The selected test room in this research is one bedroom on the third floor of the tested villa, whose construction area is about 15.6 m^2. It has a south exterior wall with an exterior window. The other three walls are all interior walls. During the testing period, 1 or 2 persons occupied the tested room, and the internal electrical equipment was testing instruments and a computer. Internal lights were turned on at night if needed.

2.2 Introduction of a tested air-conditioning system

The tested system employed an air source heat pump as the cooling and heating source. In summer, one part of chilled water is provided for the outdoor air handling unit, another part is provided for the radiant cooling floor after being mixed with the return water. In winter, one part of heated water is provided for the outdoor air handling unit, another part is provided for the radiant heating floor. For the air-side system, the outdoor air is handled by a total heat recovery unit before it has entered into the air handling unit for recovering part of the exhaust air energy. The Fig. 1 shows the schematic diagram of the water system of floor cooling combined with outdoor air supply system. It provides an intuitive conformation form of the water system. Traditional air-source heat pump can only provide low-temperature chilled water. High-temperature chilled water was made for chilled floor by mixing supply water with one part of return water in the study. The tested water system has three electromagnetic three-way valves, matching three temperature sensors measuring points. The first electromagnetic three-way valve is used for controling the mixed water volume in summer by sensing the return water temperature of the radiant floor, so the water supply temperature of the radiant floor can be controlled within a certain range without maintaining condensation on the cooling surface.

The second electromagnetic three-way valve is used for regulating water supply volume of the radiant floor in response to sensible load change.

Table 1.Thermal parameters of the tested building envelopes.

Envelope type	Thermal transmittance *W* [m^2·°C]	Limit of the energy efficiency standard
Exterior wall	0.75	1.5
Exterior window	3.2	3.2
Exterior door	2.5	3.0
Interior window	2.68	\
Roof	0.57	1.0
Interior wall	0.72	2.0

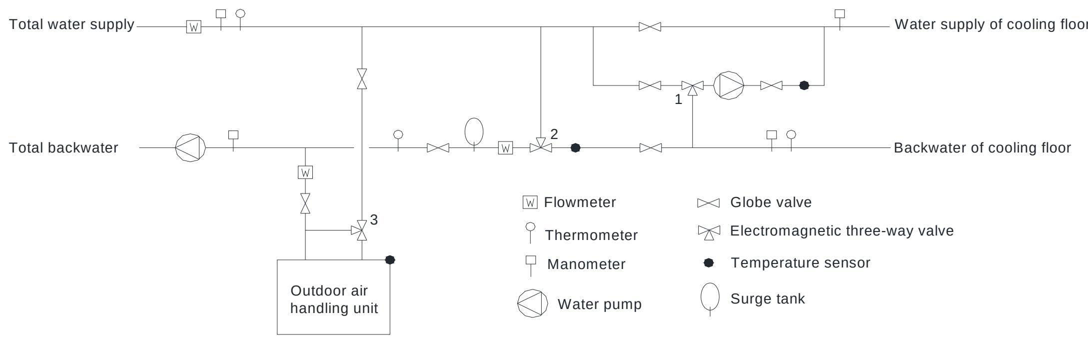

Figure 1. Water system of floor cooling combined with outdoor air supply system.

When the return water temperature of the radiant floor exceeds a certain limit, the electromagnetic three-way valve opens up, and one part of water supply of the radiant floor comes into the return water pipe directly. The third electromagnetic three-way valve is used to adjust the water supply temperature of the outdoor air handling unit.

As for the air-side ventilation strategy, all outdoor air ventilation strategy was employed in the research. The air supply terminal device employed the minimum outdoor air volume, which bore not only all the latent heat loads but also a small part of sensible heat loads, and played a functional role of diluting indoor pollutants. The outdoor air ventilation rate was set at one air change per hour. As for the air-distribution, underfloor air distribution (UFAD) system was used. In the test room, the air was supplied approximately 0.2 m above floor level and exhausted 0.2 m below the ceiling level.

2.3 Measurement equipment

The testing period was August 20, 2008 to September 15, during which the outdoor temperature was 17-34 °C. A typical day with a maximum outdoor air temperature close to design condition (34 °C) was selected for the analysis in this paper. The test data of the typical day during the system stable operating time was analyzed. Some parameters such as indoor temperature and humidity, indoor air speed, outdoor temperature, cooling floor surface temperature, inside surface temperature of envelope, water supply and return temperatures, were measured. Copper-constantan thermocouples were used as temperature sensors, all the data was observed and automatically recorded by a data acquisition system in real time, relative humidity in the tested room was automatically recorded by a temperature and humidity meter. All measurement devices and instruments are presented in Table 2. The experimental error sources included measurement equipment errors, calibration errors, and data processing errors, of which the measurement equipment errors played a major role. By calibration of the measuring equipment, as well as insulation of the connecting pipe, the test error could be controlled less than 5 %. The layout of indoor temperature measuring points is shown in Fig. 2.

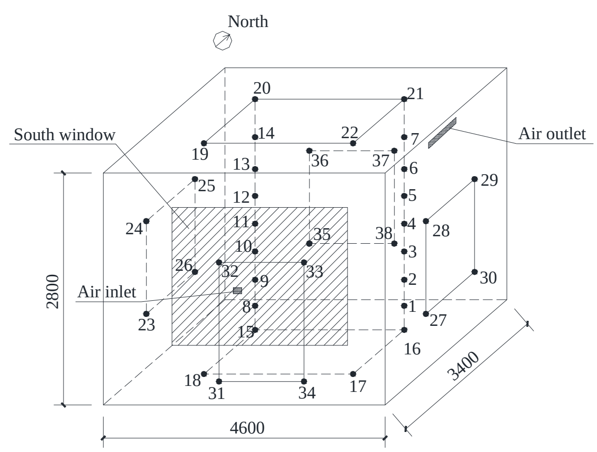

Figure 2.Temperature measurement points layout.

Table 2. Measurement equipment.

Instrument	Type (specifications)	Measurement uncertainties	Function
Thermocouple	T type	±0.5 °C	Temperature sensor
Data acquisition system	FLUKE	±0.36 °C	Data collection
Computer	Lenovo	\	Save and record data automatically
Anemometer	Hot ball anemometer (QDF-2A)	±0.01 m/s	Test air speed in room
Temperature and humidity meter	Digital (handheld) (HM34C)	±0.3 °C; ±2% RH	Test air temperature and humidity in room
Thermometer	Infrared thermometer (testo 830-T1)	±1.5 °C	Test surface temperature of envelope
Temperature and humidity recorder	Automatic temperature and humidity recorder (HOBO Pro)	±0.2 °C; ±2.5% RH	Test temperature and humidity in room

3 Indoor thermal environment evaluation

3.1 Overall thermal comfort evaluation

The evaluation of overall thermal comfort of the human body is to be based on a thermal comfort model. The Predicted Mean Vote/Predicted Percentage Dissatisfied (PMV/PPD) model was selected in the paper. The PMV/PPD model is the most widely used single-node model at present [15, 16]. Based on a human thermal equilibrium theory, it was mainly used for predicting human thermal responses and evaluating the overall thermal sensation in steady-state thermal environment controlled by air-conditioning systems. The PMV/PPD model has been validated by many laboratories and field studies around the world since many years ago [17-19]. Besides, early researchers conducted a large number of experimental studies on thermal comfort in radiant cooled environment and proposed that the PMV/PPD model was also applicable to radiant cooled environment [20]. Factors that affect the PMV are metabolic rate, clothing insulation, air temperature, mean radiant temperature, air speed and relative humidity. PPD is an index expressing the thermal comfort level as a percentage of thermally dissatisfied people, and is directly determined from PMV. Much more details including calculation methods of PMV and PPD are described in ISO 7730 [21]. It recommends the range of the PMV/PPD indices to be adopted: $-0.5 \leq \text{PMV} \leq 0.5$; $\text{PPD} \leq 10\%$.

Taking the central location of the tested room as the research location, human thermal comfort in room which employed the radiant cooling system combined with an outdoor air handling unit was evaluated with PMV/PPD indices. For radiant cooling environment, the mean radiant temperature is an important thermal micro-climate parameter affecting human thermal comfort. To calculate its value, the first step is calculating the angle factor. The angle factor between the body (denoted as *p*) and plane (denoted as *j*) can be calculated by the following formula [22]:

$$F_{\mathrm{p-j}} = \frac{1}{4\pi}\left[\frac{X}{\sqrt{1+X^2}}\tan^{-1}\left(\frac{Y}{\sqrt{1+X^2}}\right)+\frac{Y}{\sqrt{1+Y^2}}\tan^{-1}\left(\frac{X}{\sqrt{1+Y^2}}\right)\right] \tag{1}$$

The mean radiant temperature can be calculated by the following formula:

$$T_{\mathrm{mrt,p}}^4 = \sum_{j=1}^{N} F_{\mathrm{p-j}} T_{\mathrm{j}}^4 \ , \tag{2}$$

where, $X = a/1.8c$, $Y = b/1.8c$; meanings of a, b, c are defined in ASHARE 55-2010 [23]; T_{j} is the plane temperature, °C.

By MATLAB program, the human thermal comfort index can be calculated with the PMV/PPD model. For residential buildings, indoor occupants' activities usually are sitting and rest, the body metabolic rate is 1 met. In summer, occupants generally used to wear shorts, shirt, pajamas, slippers and other household clothing in room, whose thermal resistance is low with a value between 0.25 clo to 0.65 clo. In this study, the closing thermal resistance is set to 0.5 clo, just the same as that value of summer conditions in

ASHARE 55-2010. It can represent the closing thermal resistance of the majority of people in residential buildings.

Fig. 3 shows the variation trends of the mean radiant temperature and indoor temperature during a day cycle. From Fig. 3, it can be seen that the indoor mean radiant temperature is lower than the indoor air temperature, and the difference is about 3 °C. This is due to the existence of radiant cooling floor as well as its cooling influence to other envelopes. Fig. 4 shows the variation trend of the PMV occupants in sitting posture during a test day.

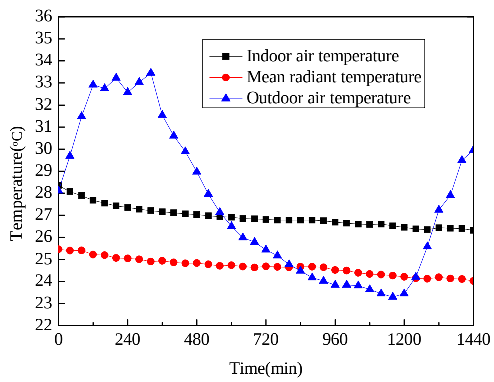

Figure 3. Mean radiant temperature and indoor temperature variation trends during a day cycle.

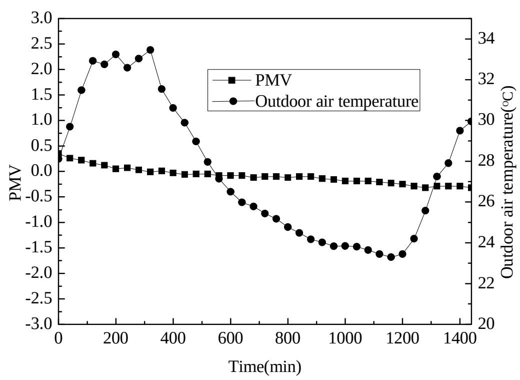

Figure 4. PMV index variation trends during a day.

During a testing day cycle, the outdoor temperature changes largely with a range of 23 °C – 33.2 °C, the PMV index can be controlled in the range of -0.32-0.35, which means that the PPD index can be controlled within 10 %. These two indexes both meet the comfort standard ISO 7730. In this study, when the outdoor temperature varies largely, the control of indoor thermal environment parameters is achieved by stop-opening regulation of the chiller.

3.2 Local thermal discomfort evaluation

3.2.1 Temperature distribution in the vertical direction

The temperature measuring points were set at the height of 0, 0.2, 0.4, 0.8, 1.6, 2.2, 2.4, 2.6, 2.8 m, respectively. For radiant cooling environment, the indoor air flow field is formed due to the combined effects of temperature difference caused from floor radiation, natural convection caused from indoor thermal load, and forced convection caused from outdoor air supply. The temperature distribution along the vertical direction is shown in Fig. 5. Two sections in the center location of the room were selected as the study objects, which were named X section and Y section, respectively.

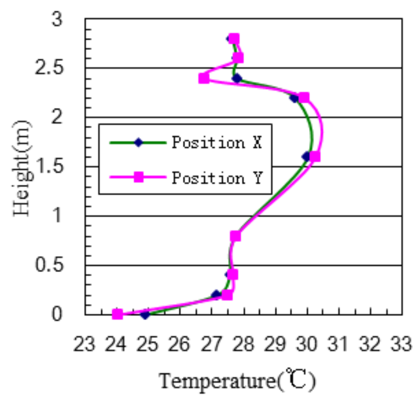

Figure 5. Indoor air temperature variation trends in the vertical direction.

The figure shows indoor temperature distribution in the vertical direction divided into three layers. The first layer is near the ground surface with a height under 0.4 m. Due to the strong convection heat transfer near the radiant cooling floor surface, there is a large vertical temperature gradient in this layer. The maximum temperature difference at the X position and Y position is 2.8 °C and 3.7 °C, respectively. The second layer is between the heights of 0.4 m and 2.4 m. In this layer, the temperature firstly increases with height due to the rise of hot air, and reaches the maximum value at the height of 2 m, and then begins to decrease. The third layer is near the ceiling with a height above 2.4

m. In this layer, the air temperature is more evenly distributed and close to the ceiling temperature. According to the comfort standard ISO 7730, the vertical temperature gradient from between ankles and head should be less than 3 °C/m. The temperature gradient of 0.1–1.1 m (distance from ankles to head for sit-in person) at X position is about 2.47 °C /m, and that at Y position is about 3.23 °C/m. The temperature gradient of 0.1 – 1.7 m (distance from ankles to head for standing person) at X position is about 2.68 °C /m, and that at Y position is about 3 °C/m. It is obvious that the temperature gradient at Y position does not meet the requirement of the comfort standard.

The forming of the indoor vertical temperature gradient is mainly due to the integrated effect of underfloor air distribution and radiant cooling floor. Floor cooling increases the vertical gradient of air temperature. The temperature gradient in space of 0.1–1.1 m at Y position exceeds only 0.23 °C/m more than the limit of the standard so that the local discomfort is not serious most of the time. The method of reducing the air supply temperature difference and improving the air supply velocity is suggested to be used at the same time so as to reduce the vertical indoor temperature gradient.

3.2.2 Local thermal discomfort caused by draft

In air-conditioning environment by convection, draft caused by air movement is one of the main factors which lead to unwanted local discomfort of the body. Previous studies show that discomfort caused by draft is significant for people taking minor physical activities such as sitting quietly with the standard dress. Therefore, it is necessary to analyze local discomfort caused by draft in residential air-conditioning environment. Professor Fanger's research shows that predicted percentage of people dissatisfied due to draft (PD) depends on air speed, air temperature, turbulence intensity of air. Its calculation formula is [23]:

$$PD = (34 - T_a)(V - 0.05)^{0.62}(0.37VTu + 3.14), \quad (3)$$

where T_a is local air temperature, °C; V is local mean air speed, m/s; T_u is local turbulence intensity, %.

Four points of 0.1 m, 0.3 m, 0.5 m, 0.7 m height are selected as the research points, which have a distance of 0.5 m from the air inlet, named by A, B, C, D, respectively. According to ASHRAE 55-2004, the turbulence intensity of mixing ventilation is around 35 %. In this research, the air supply strategy is underfloor air supply, and the air supply speed is relatively high with a value of 2 m/s, therefore, the turbulence intensity in this section is also set to the value of 35 %.The predicted percentages of people dissatisfied due to draft of these four points are calculated, and the results are shown in Table 3. As can be seen from the data in the table, indoor air speed is very low and all the values of PD are less than 15 %, which shows that the radiant cooling system has good thermal comfort and avoids unwanted local discomfort due to draft. The main reason is that the air supply terminal device employed the minimum outdoor air volume. The air supply volume is much smaller than that of traditional all-air systems, and thus local thermal discomfort caused by draft can be avoided.

3.2.3 Limit of floor surface temperature

Occupants may feel uncomfortable due to contact with floor surface that is too warm or too cool. According to ASHRAE 55-2010 standard, the acceptable floor temperature range for people wearing lightweight indoor shoes is 19-29 °C within the comfortable requirements. Fig. 6 shows the floor temperature variation trends during the test day.

Table 3. Predicted percentage of people dissatisfied due to draft.

Height [m]	Testing point	Average air speed [m/s]	PD [%]
0.1	A	0.16	7.17
0.3	B	0.14	6.33
0.5	C	0.17	7.58
0.7	D	0.18	7.96

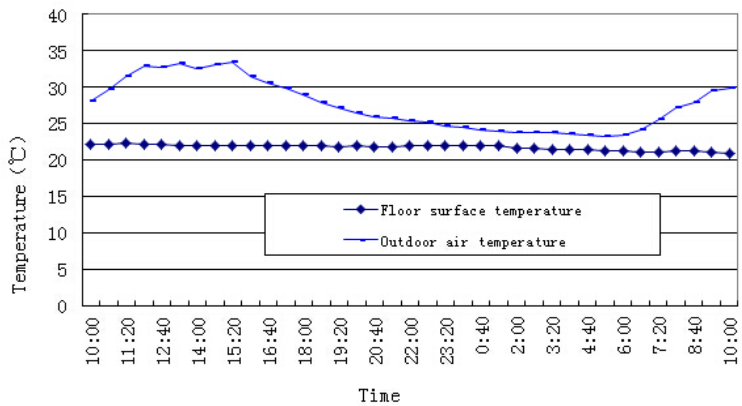

Figure 6. Floor surface temperature variation trends.

During a testing day cycle, the floor temperature is relatively constant and about 22 °C, which exceeds 3 °C compared to the lowest limit of the thermal comfort standard. Mainly due to the large thermal inertia of the radiant cooling floor, its floor surface temperature will not change dramatically with the on-off control of the chiller. On the other hand, because the residential building load is relatively small and thus the cooling floor only bears a small cooling load, the cooling floor surface temperature is not too low to cause local thermal discomfort of the human feet.

3.2.4 Temperature fluctuations with time

Previous studies show that fluctuations of both air temperature and mean radiant temperature caused by indirect self-regulation of the individual occupant may affect the thermal comfort response of building occupants. ASHRAE 55-2010 standard shows that the permissible operating temperature fluctuation range is 1.1 °C/15min. The standard also specifies the maximum change in operative temperature allowed during a period of time, as shown in Table 4.

Fig. 7 shows the indoor air temperature and wall temperature variation trends during a test day. It can be seen from the figure, in stable operating conditions, the cooling floor surface temperature fluctuates little with a range from 21.0 °C to 22.4 °C. During the test period, in a day cycle, the outdoor temperature varies largely with a range from 23.3 °C to 33.5 °C, and the average indoor temperature varies between 26.3 °C and 28.4 °C. The fluctuation of it is little and a maximum temperature difference within 24 hours of a day is only 2.1 °C. During the test day, the minimum time interval corresponding to the operative temperature change of 1 °C is 7.3h. It is obvious that the maximum temperature difference is within the limit of temperature fluctuation limit as shown in Table 4. It can clearly be seen that there is a small indoor temperature fluctuation in continuously cooled room despite the fact that the outdoor temperature varies largely in a day cycle, which meets the limit of the thermal comfort standard.

There are two reasons why the indoor temperature can remain stable. On the one hand, the control system is flexible and can regulate the chiller timely according to the outdoor temperature. The tested system in the study used on-off control for controlling and regulating the temperature of the room. On the other hand, the thermal inertia of the cooling floor and other envelopes is large so that the indoor temperature fluctuation is very small. Their delay characteristics make indoor temperature maintain stable.

Table 4. Limits on temperature drifts and ramps.

Time period	0.25 h	0.5 h	1 h	2 h	4 h
Maximum operative temperature change allowed (°C)	1.1	1.7	2.2	2.8	3.3

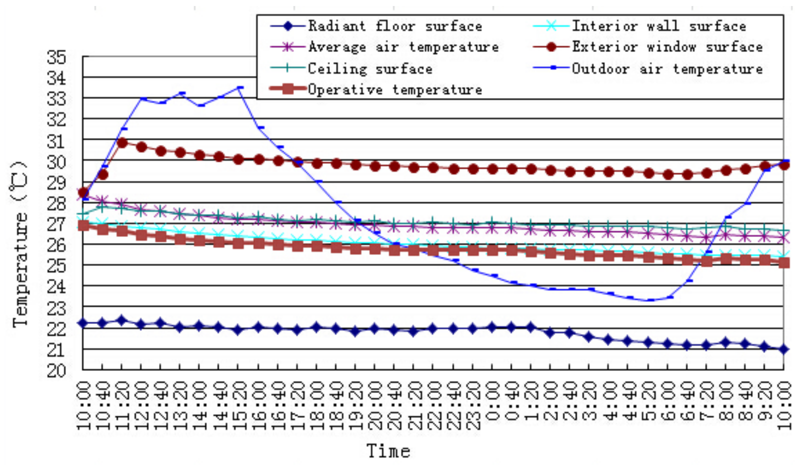

Figure 7. Indoor air temperature and wall temperature variation trends during a day.

3.3 Analysis of condensation risk

The condensation problem of the radiant cooling surface in summer is an important issue that currently impedes the widespread application of this system. Condensation can not only damage the system's normal cooling capacity, but also lead to indoor microbial growth, thus resulting in health problems. Fig. 8 shows the variation trends of indoor average relative humidity during a testing day cycle. From the figure can be seen that the indoor average RH varies between 50 % and 60 % with an average value of 56.9 %. In addition, during the testing period, the indoor DBT is around 27.3 °C, and the dew point temperature corresponding to the average relative humidity is 18.4 °C. The cooling floor surface temperature is 21.4 °C, obviously higher than the dew point temperature so that there is no condensation risk during the testing day.

4 Conclusions

This paper presents an evaluation on the indoor thermal environment of a residential building with a radiant floor cooling system based on field measurement. The main conclusions are drawn as follows:

1) In the stable operation period of the system, although the outdoor temperature varies largely during a testing day cycle, the PMV index can be controlled with a range of -0.32-0.35 and the PPD index can be controlled within 8 %. These two indices meet the comfort standard ISO 7730. This is because the start-stop control of the heat pump was conducted under part load conditions so as to control the thermal comfort level of indoor thermal environment.

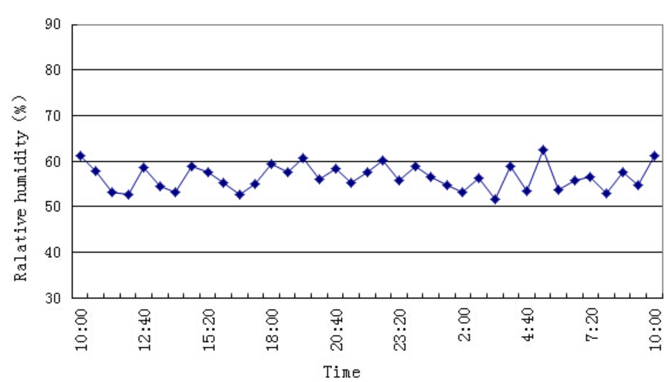

Figure 8. Indoor average relative humidity variation trends.

2) The air temperature gradients of representative measurement points at the vertical direction are 2.5 °C/m and 3.2 °C/m, respectively, which does not completely meet the comfort standard ISO 7730. Floor cooling increases the vertical gradient of air temperature, and thus causes local discomfort of the human body. Using the method of reducing the air supply temperature difference and improving the air supply velocity at the same time can reduce the vertical indoor temperature gradient.
3) The predicted percentages of people dissatisfied due to draft of representative measurement points are less than 15 %. Evidently, local discomfort due to draft caused by traditional convection air-conditioning system has been eliminated due to the supply of air volume of the radiant cooling system that is commonly the minimum outdoor air volume and also much smaller compared to conventional all-air system. The indoor air velocity is between 0.1 m/s and 0.2 m/s, which meets the permitted range of the thermal comfort standard.
4) As the residential building load is relatively small, and thus, the cooling floor bears only a small cooling load, the cooling floor surface temperature is not too low to cause local thermal discomfort. During a testing day cycle, the floor temperature is kept relatively constant and about 22 °C, which exceeds 3 °C compared to the lowest limit of the thermal comfort standard.
5) As the cooling floor and other envelopes have a large inertia, the indoor air temperature and wall temperature of the continuously cooled room fluctuate little in spite of a large cyclic variation of the outdoor temperature in a day cycle. So, there are no discomforts due to temperature fluctuations with time. The maximum indoor temperature fluctuation range is only 2.1 °C during a test day, which can meet the permitted range of the thermal comfort standard.
6) During the stable operation of the system, the indoor relative humidity varies between 50% and 60 %. The condensation risk on radiant cooling floor surface does not exist. The prerequisites for the room without condensation are as follows: reasonable design, dehumidification unit matching, normal system operation, users' proper use of the system, and a well sealed room.

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