

Experimental study on the dynamic thermal response of a radiant floor system in an office building

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Abstract. The dynamic thermal performance of radiant terminal plays an important role in the design and control of radiant cooling system, which is shown as the dynamic thermal response of radiant floor system (RFS) under imposed control. In this paper, the field measurement method was used to study the dynamic thermal response of RFS. The RFS was activated in summer and the supply water temperature was regulated in winter to make dynamic change of thermal performance. The floor surface temperature was selected as the characteristic parameter to describe the dynamic heat transfer performance of the system, and response time τ_{95} and time constant τ_{63} were used to quantify the dynamic thermal response. The maximum τ_{95} was 13.5 h and τ_{95}/τ_{63} was greater than or equal to 2 in the cooling mode, while τ_{95} and τ_{63} were both less than 10 h and τ_{95}/τ_{63} was 1.6 in the heating mode. As a result, there was no significant lessening of temperature change rate, and the thermal response of RFS was faster under intermittent control of supply water temperature in winter. Therefore, the study aims at providing reference for making intermittent control strategy by using the dynamic thermal performance of radiant system.

Keywords: radiant floor system, dynamic thermal performance, response time, time constant

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1 Introduction

Since the 21st century, an increasing amount of attention has been given to energy saving due to emission reduction goals, and the rapid development of HVAC technology and radiant cooling/heating air conditioning systems, which represent large portions of building energy budgets, have received an especially high amount attention. Many studies have been conducted on thermal comfort and energy efficiency of radiant cooling/heating systems [1-3]. High-temperature cooling and low-temperature heating systems use lower-grade cold and heat sources [4], this provides favorable conditions for the further promotion and application of radiant terminals. Therefore, the system has great energy saving potential [5]. Radiant floor systems (RFS) transfer heat between the radiant surface and the surroundings via indoor air convective heat transfer and radiative heat transfer. Radiative heat transfer accounts for more than 50% of the total heat transfer in the radiant system [6], increasing the radiative heat transfer with the human body, thus making it central to improving thermal comfort. However, the regulation of radiant systems is more complicated than conventional air systems because the response time of the radiant system is significantly longer [7, 8]. Therefore, accurate control strategies are required to ensure effective operation and meet thermal comfort standards under dynamic weather conditions and indoor heat gain.

Response time τ_{95} and time constant τ_{63} are widely used for evaluating the dynamic thermal performance of radiant systems to reflect if a radiant system response is fast or slow. Ning et al. [9] found that τ_{95} could provide a more consistent metric of long-term thermal response of the full range of radiant system types. Merabtine et al. [10] proposed a simplified analytical model with τ_{63} and delay time for the average floor heating surface temperature. Thermal performance of the radiant terminal can be effectively used by applying an intermittent control strategy combined with controlling supply water temperature and flow rate in radiant panels, this contributes to improve response time. Liu et al. [11] showed that the energy consumption decreased by 3.3% to 7.5% when the different intermittent operation strategies using thermal characteristic of the

building were applied. Hu et al. [12] found that radiant cooling system conserved cooling energy in unoccupied period to balance 9-15% of indoor heat gain in occupied period, and the peak sensible cooling loads of the radiant terminal decreased by 32–39% compared with those using conventional scheduling. Sui et al. [13] observed that increasing the start and stop frequency of the pump during a given cooling period increased the cooling capacity of a tubes-embedded envelope cooling system. Liu et al. [14] proposed intermittent controls based on the minimum outdoor air temperature and the average water supply and return temperature via CFD simulation Tang et al. [15] proposed a novel pulse flow control method that allowed for a 27% reduction of the supply water flow at 50% load operation, and it made the radiant surface temperature distribution more uniform with more accurate control.

Moreover, the unique thermal characteristics of RFS is affected by the indoor environment and outdoor weather conditions, emphasizing the importance of experimental research in the practical applications of RFS. Therefore, it is crucial to test thermal performance of radiant terminal under concrete control in the practical building.

In this paper, the field measurement method was used to study the dynamic thermal response of RFS. The experimental room created a real-world scenario in which changes in thermal and humidity parameters in the room were influenced by outdoor weather conditions. The dynamic thermal response of RFS was caused by different imposed control, which were supply water temperature regulation in summer and start-up control in winter respectively. The change of floor surface temperature during the process of heat exchange between radiant floor and buried pipe water was analyzed. In addition, the τ_{95} , τ_{63} and τ_{95}/τ_{63} of the RFS under different imposed controls in winter and summer were compared, and the differences along with the effect of two imposed controls on thermal response were obtained.

2 Methods

2.1 Experimental scenario description

The field test was conducted in a south facing room with a floor area of 50 m² on the fifth (top) floor of an

office building in Jinan. Jinan is located in the north of China, which is in a cold region as shown in Figure 1.



Figure 1 The climate location of Jinan and an outdoor view of the office building.

The heat transfer coefficient of the exterior wall is $0.6 \text{ W}/(\text{m}^2 \cdot \text{K})$, that of the roof is $0.55 \text{ W}/(\text{m}^2 \cdot \text{K})$, that of the floor was $1.03 \text{ W}/(\text{m}^2 \cdot \text{K})$, and that of the external window is $2.4 \text{ W}/(\text{m}^2 \cdot \text{K})$ [16]. The schematic diagram and thermophysical parameters of radiant floor are respectively shown in Figure 2 and Table 1 [14].

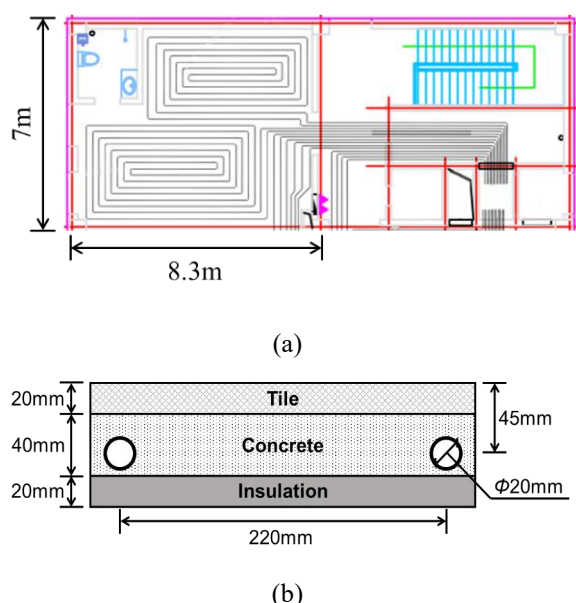


Figure 2 Schematic diagram, (a) distribution of buried pipes in the experimental room, and (b) structure of radiant floor.

Table 1 Thermophysical parameters of layers of radiant floor

Layers	Conductivity ($\text{W}/\text{m} \cdot \text{K}$)	Specific Heat ($\text{J}/\text{kg} \cdot \text{K}$)
Tile	2.1	950
Light concrete	1.45	1000
Pipe embedded	0.4	1000
Insulation	0.022	4600

2.2 Experimental setup

The RFS is depicted in Figure 3. During summer the cooling energy supplied to the radiant terminal comes entirely from the ground heat exchanger and the heat

pump unit was by-passed. Therefore, the underground pipe water was directly supplied to the room without cooling by the heat pump unit. During winter the supply water temperature in the buried pipe was controlled by the heat pump unit. The supply water temperature was heated to the appropriate temperature based on indoor requirement and sent to the room to provide heating for indoor environment.

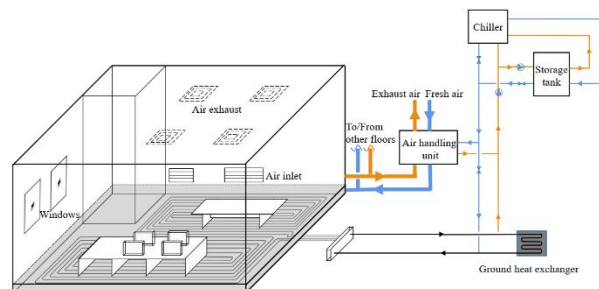


Figure 3 Schematic diagram of the RFS.



Figure 4 The field test environment in the office room.

The field test environment is shown in Figure 4. The measurement parameters included supply and return water temperatures and flow rates, floor surface temperature, wall and ceiling temperatures, room air temperature. The layout of surface temperature and supply and return water measuring points is shown in Figure 5. The four floor surface temperature measuring points were placed near walls to the east, west, north, and south of the room in summer. Other six floor surface temperature measuring points (red) added were placed along the flow direction of supply water in the middle of the room in winter. The wall temperature measuring points were centered on their respective walls, and ceiling measuring points were uniformly distributed on the ceiling. One indoor air parameter measuring point was placed at the vertical height of 1.1 m in the center of the room, and other six measuring points at vertical heights of 0.1 m were uniformly distributed in the room. The supply and return water temperature measuring points were placed at manifolds on the supply and return water pipes. The detailed parameters of experimental equipment are shown in

63% of the total temperature change, which can be used to measure the time required for radiant terminal from the beginning to stable heat transfer [17]. However, τ_{63} can only describe dynamic thermal performance of the radiant terminal at the early period, but cannot reflect the whole process from one stable condition to another. Therefore, it is required to combine τ_{95} which is the time it takes for radiant surface temperature of a radiant system to reach 95% of the difference between its final and initial values[8]. The time that achieving 95% of the total temperature change is very close to the final steady state. Thus, τ_{95} can be used to well describe thermal response of radiant terminal from one stable to another.

Moreover, variable heat transfer between radiant floor and buried pipe water was also used to show dynamic change of thermal performance of radiant terminal, which could be calculated based on the temperature difference and thermal resistance between radiant surface and buried pipe water. Chilled/hot water circulated in embedded surface system provide cooling/heating for radiant terminal, and then heat is transferred between radiant terminal and indoor environment via convective heat transfer and radiative heat transfer. Cooling/heating provided by buried pipe water can be calculated by Eq (1).

$$Q = |T_s - T_w|/R - q_{stored} \dots \quad (1)$$

where, Q is cooling/heating provided by buried pipe water, W/m^2 ; T_s is radiant surface temperature, $^{\circ}C$, T_w is supply and return water temperature, $^{\circ}C$, R is thermal resistance between radiant surface and buried pipe water, $(K \cdot m^2)/W$, and q_{store} is the cooling energy used to remove the heat accumulated in radiant floor, W/m^2 .

3 Results and discussion

3.1 Thermal response under imposed control in the cooling mode

Figure 7 shows the changes of T_f , T_{in} and Q_c after the start-up of RFS in summer. In the beginning when chilled water started to be supplied, T_f was the highest due to the heat storage in radiant terminal and indoor heat accumulation, along with the maximum heat transfer between the floor and water supply because of the greatest temperature difference between the two. With the continuous cooling supplied by chilled water,

T_f and heat removed by chilled water gradually decreased, while the decreasing gradient of T_f tended to lessen. The decreasing rate of T_f after τ_{63} was smaller than that during τ_{63} .

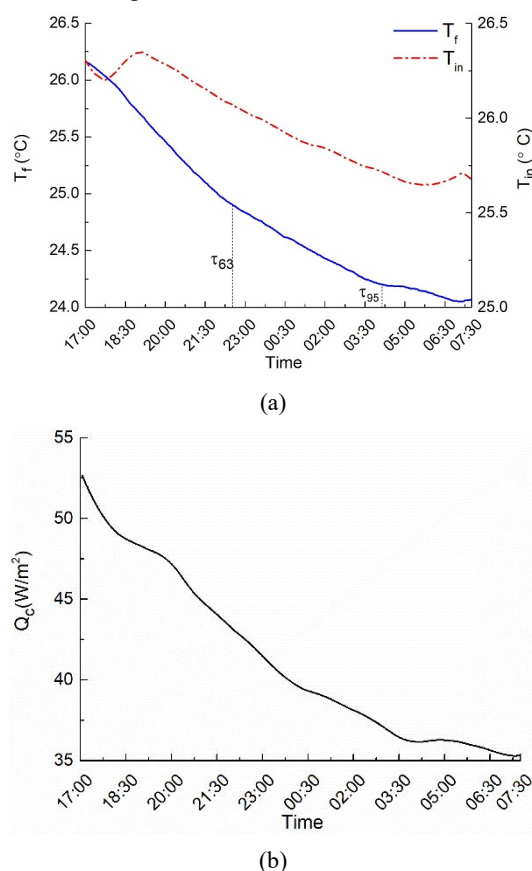


Figure 7 Thermal response when activating RFS (a) T_f and T_{in} reduction process when activating RFS, and (b) Q_c decaying process when activating RFS.

When approaching 95% of the total temperature change, T_f decreased slowly, and essentially stabilized after a small decrease. The RFS was combined with displacement ventilation system for cooling in summer. After the RFS was started, T_f was higher due to shutdown of displacement ventilation system so that the insufficient cooling caused T_{in} to increase in a short time. When T_f decreased by about $1^{\circ}C$, cooling capacity of RFS could counter indoor heat gain and achieve cooling effect, and therefore T_{in} began to decrease. With the decrease of heat transfer between chilled water and radiant floor, T_{in} gradually remained a stable level.

3.2 Thermal response under imposed control in the heating mode

The changes of T_f , T_{in} and Q_h after T_{sa} of RFS increased in winter are illustrated in Figure 8. T_{sw} decreased by $2 \sim 3^{\circ}C$ after working period, and

meanwhile heat stored in radiant terminal gradually released. When T_{sw} increased to 28.4°C, T_f was at the lowest level, and the temperature difference between the two was the greatest, so the heat transfer from hot water was maximum at initial time.

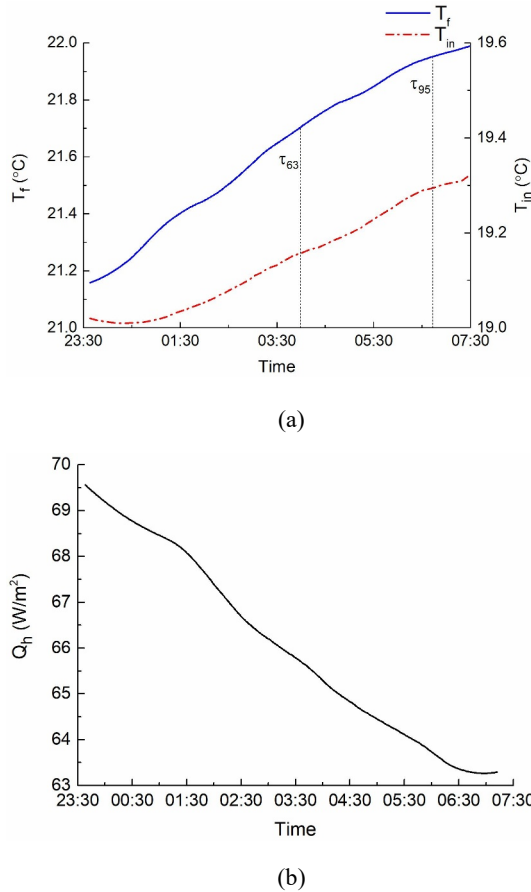


Figure 8 Thermal response when increasing T_{sw} , (a) T_f and T_{in} increasing process when increasing T_{sw} , and (b) Q_h decaying process when increasing T_{sw} .

Affected by hot water flowing in the radiant terminal, T_f increased continuously, heating provided by hot water decreased gradually since the temperature difference between radiant floor and hot water kept decreasing, and the increasing trend of T_f slowed down. τ_{63} was approximately half of the total T_f change time, along with the similar decreasing gradient of T_f before and after this time. Heating provided by hot water became stable, and the temperature changed more slightly after τ_{95} . While T_{in} did not increase immediately with increase of T_f until the RFS operated with high T_f for half an hour. As the heating transfer tended to balance, T_{in} increased slightly.

3.3 Comparison of thermal response in the cooling and heating mode

T_f at different stage and parameters reflecting the thermal response of the RFS in winter and summer are listed in Table 4. The dynamic thermal response of RFS in winter and summer was caused by different controls including increasing T_{sa} and initiating water supply. It was inferred that τ_{95}/τ_{63} was greater than or equal to 2 in the cooling mode and 1.6 in the heating mode. According to the radiant system categories in the literature [8], the radiant floor of the experimental room was classified as type A. τ_{95} of the RFS in the cooling mode was close to or even surpassed the range from 3 to 13h under different impact factors obtained from the literature [8]. It was related to the fact that the RFS used underground pipe water for cooling in summer, and higher T_{sa} had less effect on the thermal performance of the radiant terminal. While τ_{95} of the RFS in the heating mode was at higher level within range of 2.9 ~ 11.8h provided by the literature [8]. It could be concluded that T_{sa} regulation in winter can lead to the faster change of T_f , making the thermal response of RFS faster.

Table 4 T_f at different stage and thermal response parameters under imposed controls

Modes	Date	T_0 (°C)	τ_{63} (h)	T_{63} (°C)	τ_{95} (h)	T_{95} (°C)	T_{100} (°C)
Cooling mode	Aug. 16	26.2	5.5	24.9	11.2	24.2	24.1
	Aug. 23	25.7	6	24.4	13.5	23.7	23.6
Heating mode	Jan. 13	21.1	4.5	21.7	7.2	21.9	22
	Jan. 14	22	5	22.6	8	22.8	22.9

4 Discussion

The RFS operated 24h during weekdays entirely depending on the underground pipe water for cooling in summer. Therefore, lack of intermittent control led to few cases of thermal response of the radiant terminal, and the heat exchange was small between the radiant floor and buried pipe water, which caused longer response time of radiant terminal. The thermal response of radiant floor is that the radiant floor changes from one stable condition to another under the control signal, and the time constant is used to describe the response rate of the system. Intermittent heat transfer caused the floor surface temperature to change. The change rate of

floor surface temperature gradually decreased with continuous heat transfer between supply water and radiant floor. This period was the main part of the thermal response, which had a great influence on the time constant. The floor surface temperature was eventually stable, and maintained a small temperature difference of relatively stability with supply water temperature, achieving heat transfer balance. At this period, radiant floor was in a stable condition, and the small temperature difference impacted little on the time constant. In the future, it is necessary to use the dynamic thermal performance of RFS to determine the intermittent operation parameters, so as to ensure the indoor thermal environment meet the requirements and achieve the energy-saving operation control.

5 Conclusion

Under the start-up control of RFS in the cooling mode and T_{sa} regulation in the heating mode, thermal performance of radiant terminal dynamically changed.

(1) With the decrease of the temperature difference between pipe water and radiant floor, the heat exchange lessened, and the change of the T_f came to stability. While T_{in} began to decrease/increase with the effect of cooling/heating 0.5 ~ 1 h after imposing control to RFS.

(2) In the cooling mode, τ_{95}/τ_{63} was greater than or equal to 2, and the maximum τ_{95} was 13.5h. In the heating mode, τ_{95}/τ_{63} was 1.6, and the maximum τ_{95} was 8h. Both indicated the long response time of RFS.

(3) Compared with the start-up control in summer, T_{sa} regulation in winter had a greater impact on the thermal performance of the RFS, which made the thermal response faster.

Conflicts of interest

The authors declare no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

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Nomenclature

Q	Total heat transfer (W/m^2)
S_n	measuring point
T	temperature ($^{\circ}C$)
V	flow rate

Subscripts

0	initial temperature of the temperature change
100	final temperature of the temperature change
in	indoor air
f	floor surface
s	surface of the walls and ceiling
sw	supply water

Symbols

τ_{63}	time to achieve 63% of the temperature change (h)
τ_{95}	time to achieve 95% of the temperature change (h)

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