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Indoor temperatures for calculating room heat loss and heating capacity of radiant heating systems combined with mechanical ventilation systems

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ABSTRACT

In this study, a typical office room with a radiant heating system and a mechanical ventilation system was selected as the research subject. Indoor temperature formulas for calculating the room heat loss (including transmission heat loss and ventilation heat loss) and heating capacity of the hybrid system were determined according to the principle of heat transfer. A model to predict indoor temperatures in the room was proposed, and it was determined that the predicted indoor temperatures agreed well with the measured data. Qualitative analyses of the effects of heated surface temperature and air change rates on the indoor temperatures were performed using the proposed model. When heated surface temperatures and air change rates were from 21.0 to 29.0 °C and from 0.5 to 4.0 h^{-1}, the indoor temperatures for calculating the transmission heat loss and ventilation heat loss were between 20.0 and 20.3 °C and between 19.6 and 20.5 °C, respectively, and the indoor temperature for calculating the heating capacity of the hybrid system was between 18.2 and 19.8 °C. Accordingly, the relative calculation errors were between 0.3% and 0.5% and between −10.2% and 11.8% for calculating the transmission heat loss and ventilation heat loss, respectively, and between 16.0% and 17.4% for calculating the heating capacity of the hybrid system. Due to large relative calculation errors, it is necessary to consider the effect of heated surface and cool supply air on indoor temperatures for calculating ventilation heat loss and heating capacity of radiant heating systems combined with mechanical ventilation systems.

1. Introduction

Radiant heating systems, such as floor and ceiling heating systems, are now regarded as energy efficient and comfortable heating systems; thus, these systems have been extensively used in buildings of different types [1–5]. Additionally, many energy efficient building technologies with increased thermal insulation and air tightness have also been applied in buildings with radiant heating systems, which may contribute to low building heat loss and reduced energy consumption. Conversely, the application of these energy efficient technologies may cause insufficient fresh air supply due to low infiltration and lead to poor indoor air quality and increased human healthy symptom [6,7]. To avoid this problem, a mechanical ventilation system, such as a mixing ventilation system or a displacement ventilation system, should be installed in buildings for increasing fresh air supply [8–10].

A hybrid system with a radiant heating system and a mechanical ventilation system, which is regarded as an advanced HVAC system, has been applied in many buildings worldwide [11–15]. The radiant heating system was used to control the indoor thermal environment in term of indoor temperature, and the mechanical ventilation system was used to control the indoor air quality in term of outdoor air flow rate. Meanwhile, supply air temperature was kept as constant, normally from 14 °C to 18 °C to avoid uneven air temperature distribution [10,12]. Therefore the radiant heating system and mechanical ventilation system could operate independently and would not fight with each other due to the different control objectives.

Indoor temperatures, such as indoor air temperature and operative temperature, are the key parameters to determine the heat loss of a room and the heating capacity of a radiant heating system.

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Nomenclature

A	area (m^2)
ACH	Air change rate (h^{-1})
c_p	specific heat capacity (J kg^{-1} K^{-1})
D	Diameter (m)
error	Relative error (%)
F	view factor (–)
h	convective heat transfer rate (W m^{-2})
H	Height (m)
L	Length (m)
m	mass flow rate (kg s^{-1})
Q	heat transfer rate (W m^{-2})
t	temperature (°C)
v	velocity (m s^{-1})
W	Width (m)

Greek symbols

ε	emissivity (–)
ρ	density (m^3 kg^{-1})
σ	Stefan–Boltzmann constant (=5.67 × 10^{-8} J s^{-1} m^{-2} K^{-4})

Subscripts

a	space air
e	exhaust
c	ceiling
ew	external window and wall
f	floor
is	internal heat sources
iw	internal wall
r	radiation
s	supply
sys	system
w	wall

When indoor physical parameters are within the thermal comfort range in a room with a radiant heating system without a mechanical ventilation system, the indoor air temperature is approximately equal to the operative temperature and can be used as reference for calculating the room heat loss and the heating capacity of the radiant heating system [16–19]. Compared to a radiant heating system without a mechanical ventilation system, a hybrid system with a radiant heating system and a mechanical ventilation system can create a new situation with regard to indoor air temperature and operative temperature, i.e., the indoor air temperature may not equal the operative temperature, and they may not be merely influenced by the heated surface and also by the cool supply air [20–22]. However, very few studies have considered the effect of heated surface and cool supply air on indoor temperatures for calculating room heat loss and heating capacity of a hybrid system, and irrational indoor temperatures may result in large errors in the sizing of the hybrid system as a result.

In this study, a typical office room with a radiant heating system and a mechanical ventilation system was selected as the research subject. A model that predicts indoor room temperatures will firstly be proposed, and then a qualitative analysis on the effect of heated surface temperature and air change rate on the indoor temperatures using the model are performed. The results in this paper are believed to be beneficial for the design of radiant heating systems combined with mechanical ventilation systems.

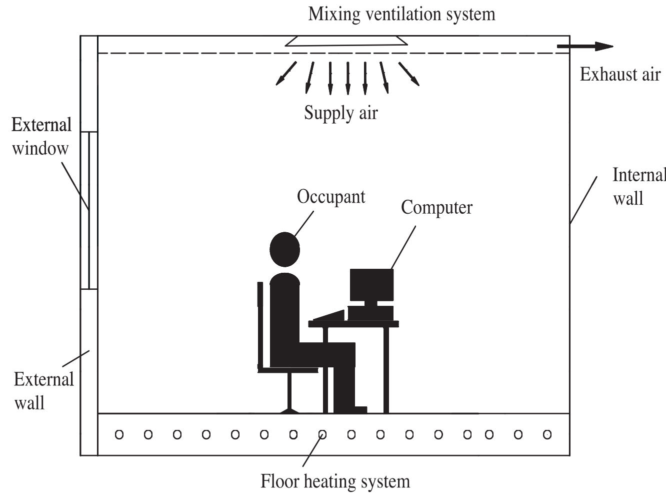

Fig. 1. A typical office room with floor heating system and mixing ventilation system.

2. Establishment of the indoor temperature prediction model

A hybrid system with a radiant heating system and a mechanical ventilation system normally includes at least one of the following cases: (1) a floor heating system combined with a mixing ventilation system, (2) a floor heating system combined with a displacement ventilation system, (3) a ceiling heating system combined with a mixing ventilation system or (4) a ceiling heating system combined with a displacement ventilation system. In this study, a typical office room with a floor heating system and a mixing ventilation system was selected as the research subject, as shown in Fig. 1.

The high velocity cool air is supplied from the mixing ventilation system outlets and enters the occupied zone, where it is mixed with the updraft air caused by the internal heat sources present in the room, such as computers or occupants (as shown in Fig. 1). To establish a model that predicts indoor temperature in the room, some assumptions regarding to building heat transfer must be made, as seen in Ref. [22]. In this study, the only difference from Ref. [22] is the assumption of vertical distribution of indoor air temperature, which is not linear but uniform due to the mixing ventilation in the room.

Based on the above assumptions, the mathematical description of the heat transfer processing the room with floor heating and mixing ventilation can be obtained, which consists of five energy balance equations, as shown in Eq. (1). It should be noted that the heat storing effect at the floor surface mass was ignored due to the steady heat transfer processes in the room.

$$\begin{cases} Q_{cf} + Q_{rf} + Q_{rsf} + Q_f = 0 \\ Q_{cc} + Q_{rc} + Q_{rsc} = 0 \\ Q_{cew} + Q_{rew} + Q_{rsew} + Q_{ew} = 0 \\ Q_{ciw} + Q_{riw} + Q_{rsiw} = 0 \\ Q_{cs} - Q_{cf} - Q_{cc} - Q_{cew} - Q_{ciw} - Q_s = 0 \end{cases} \quad (1)$$

where Q_{cf}, Q_{cc}, Q_{cew} and Q_{ciw} are the convective heat transfer rate between the envelope surface and the indoor air (as shown in Eq. (2)), Q_{rf}, Q_{rc}, Q_{rew} and Q_{riw} are the radiant heat transfer rate between any two envelope surfaces (as shown in Eq. (3)), Q_{rsf}, Q_{rsc}, Q_{rsew} and Q_{rsiw} are the radiation incident on the envelope surface from the internal heat sources (as shown in Eq. (4)), Q_f is the upward heat flow from the floor heating system (which is somewhat less than the total floor heating capacity), Q_{ew} is the rate of heat loss through the window and the external wall (as shown in Eq. (5)), Q_{cs} is the convective heat gain to the indoor air from the internal heat sources (as shown in Eq. (8)) and Q_s is the heat deduction through the supply air flow (as shown in Eq. (9)).

$$Q_{ci} = A_i h_{ci} (t_a - t_i) \tag{2}$$

where A_i, t_a and t_i are the envelope surface area, indoor air temperature and envelope surface temperature and h_{ci} is the convective heat transfer coefficient between the envelope surface and the indoor air and can be found in Refs. [23,24].

$$Q_{ri} = \sum_{j=1}^{3} A_i F_{i-j} h_r \left(t_j - t_i\right) \tag{3}$$

where F_{i-j} is the view factors between any two envelope surfaces and h_r is the radiant heat transfer coefficient between any two envelope surfaces, generally equal to 5.7 W m^{-2} K^{-1} [25,26].

$$Q_{rsi} = \mathrm{rsi} Q_i \tag{4}$$

where rsi is the percentage of radiation from the internal heat sources that is incident on the envelope surface and Q_i is the total heat gain from the internal heat sources, such as computers and occupants.

$$Q_{ew} = A_{ew} h_{ex} (t_{ex} - t_{n,ew}) \tag{5}$$

where t_{ex} is the external air temperature, h_{ex} is the average heat transfer coefficient of the window/wall surfaces and $t_{n,ew}$ is the indoor temperature for calculating the transmission heat loss, which can be observed in the following equation:

$$t_{n,ew} = \frac{h_{cew} t_a + h_{rew} t_{r,ew}}{h_{cew} + h_{rew}} \tag{6}$$

where $t_{r,ew}$ is the equivalent mean radiant temperature for calculating radiant heat transfer rate between the inside surfaces of the window and the external wall and the other inside surfaces, which can be expressed as Eq. (7) according to Eq. (3).

$$t_{r,ew} = \sum_{j=1}^{3} F_{ew-j} t_j \tag{7}$$

$$Q_{cs} = \left(1 - \sum \mathrm{rsi}\right) Q_i \tag{8}$$

$$Q_s = c_p m_s (t_a - t_s) \tag{9}$$

where t_s and m_s are the supply air temperature and flow rate.

As shown in Eq. (1), the equation group for predicting indoor temperature consists of five energy balance equations and five unknown boundary parameters, which include the supply air temperature and flow rate, the heated surface temperature, the total heat gain from internal heat sources and the external air temperature. When the boundary parameters are known, the indoor temperatures can be calculated by solving these energy balance equations using the boundary parameters as inputs.

3. Validation of the proposed indoor temperatures prediction model

3.1. Indoor temperatures formulas for calculating room heat loss and heating capacity of the hybrid system

The heat loss in a room normally includes the transmission heat loss and the ventilation heat loss [16,17]. The former represents the rate of heat loss through the window and external wall, and the latter represents the heat deduction through the supply airflow.

Table 1
Boundary parameters for different experimental cases.

Cases	t_s (°C)	t_f (°C)	ACH (h^{-1})	Q_i (W)	t_{ex} (°C)
1	14.2	24.2	0.5	100	−5
2	14.2	22.9	0.5	100	5
3	14.1	24.5	1.0	100	−5
4	14.1	23.2	1.0	100	5
5	18.0	22.8	1.0	200	−5
6	17.8	22.2	1.0	200	5
7	17.8	23.0	2.0	200	−5
8	17.8	22.3	2.0	200	5

The indoor temperature for calculating the ventilation heat loss is equal to the indoor air temperature, as shown in Eq. (9).

The concept of the indoor temperature for calculating the transmission heat loss is more like to find the operative temperature, which is to measure the human body sensible heat loss at the skin surface. The human body sensible heat loss is the heat conduction from inside body to outside surface through skin, which is equal to the convective and radiant heat exchange between the skin surface and the indoor environment. Similarly, the transmission heat loss is the heat conduction from inside surface to outside surface through envelope, which is equal to the convective and radiant heat exchange between the inside surface and the indoor environment. Hence, according to the definition of operative temperature, the indoor temperature for calculating the transmission heat loss can be expressed as the weighted average of the indoor air temperature and the equivalent mean radiant temperature, as shown in Eq. (6).

The heating capacity of the hybrid system represents the rate of heat transfer through the floor surface, and the concept of the indoor temperature for calculating the rate of heat transfer through the floor surface is also more like to find the operative temperature. The rate of heat transfer through the floor surface is the heat conduction from inside pipe to outside surface through floor, which is equal to the convective and radiant heat exchange between the floor surface and the indoor environment. Hence, according to the definition of operative temperature, the indoor temperature for calculating the rate of heat transfer through the floor surface can be expressed as the weighted average of the indoor air temperature and the equivalent mean radiant temperature, as shown in the following equation:

$$t_{n,f} = \frac{h_{cf} t_a + h_{rf} t_{r,f}}{h_{cf} + h_{rf}} \tag{10}$$

where $t_{r,f}$ is the equivalent mean radiant temperature for calculating the radiant heat transfer rate between the floor surface and the other inside surfaces, which can be expressed as Eq. (11) according to Eq. (3).

$$t_{r,f} = \sum_{j=1}^{3} F_{f-j} t_j \tag{11}$$

3.2. Boundary parameters

To validate the proposed model, measured data [10,27] in a simulated office (4.2 × 4.0 × 2.4 m) are used in this paper to compare with the predicted results, as shown in Table 1, where t_s is the supply air temperature, t_f is the heated surface temperature, ACH is the air change rate, Q_i is the total heat gain from internal heat sources and t_{ex} is the external air temperature. The measurement accuracy for the air temperature is ±0.3 °C, and the measurement accuracy for the air velocity is ±(0.02 + 0.02x) m s^{-1}, where x is the measured value of air velocity. The total heat transfer coefficient of the window and the external wall is equal to 2.0 W m^{-2} K^{-1} in these experiments.

Table 2
Measured and predicted indoor temperatures for calculating room heat loss and heating capacity of the hybrid system.

Cases	Measured $t_{n,ew}$ (°C)	Predicted $t_{n,ew}$ (°C)	Relative error (%)	Measured t_a (°C)	Predicted t_a (°C)	Relative error (%)	Measured $t_{n,f}$ (°C)	Predicted $t_{n,f}$ (°C)	Relative error (%)
1	22.0	21.9	0.5	21.3	21.8	2.3	21.2	21.2	0.0
2	22.0	21.6	1.8	21.6	21.5	0.5	21.6	21.2	1.9
3	22.3	21.9	1.8	21.7	21.7	0.0	21.6	21.2	1.9
4	22.2	21.6	2.7	21.8	21.4	1.8	21.8	21.2	2.8
5	22.1	21.1	4.5	21.5	21.1	1.9	21.5	20.5	4.7
6	22.1	21.4	3.2	21.8	21.4	1.8	21.8	21.1	3.2
7	22.1	21.2	4.1	21.5	21.3	0.9	21.5	20.6	4.2
8	22.3	21.9	1.8	21.9	21.6	1.4	21.9	21.2	3.2

As shown in Table 1, the measured heated surface temperatures and air change rates were between 22 and 24 °C and from 0.5 to 2.0 h^{-1}, respectively. Due to the limitation of the experimental conditions, the upper limit of these parameters are limited and can be up to 29 °C and 4.0 h^{-1} according to ISO 7730 [28] and EN 13251 [29].

3.3. Comparison of the predicted indoor temperatures with the measured data

A comparison of the predicted indoor temperatures for calculating room heat loss and the heating capacity of the hybrid system with the measured data is shown in Table 2. The relative error is the calculation difference between the measured and the predicted data.

As shown in Table 2, the maximum relative error between the predicted indoor temperatures for calculating transmission heat loss and ventilation heat loss and the measured data are 4.5% and 2.3%, respectively. Additionally, the maximum relative error between the predicted indoor temperatures for calculating the heating capacity of the hybrid system and the measured data is 4.7%. Because the maximum relative errors between the predicted indoor temperatures and the measured data were all less than 5%, the proposed indoor temperature prediction model is rational and valid.

4. Calculation results and discussion

4.1. Calculation condition

An indoor thermal environment should be acceptable and comfortable in a room with a radiant heating system and a mechanical ventilation system. Normally, the indoor operative temperature is regarded as the index of thermal comfort for a standard office worker in the room, which can be calculated as following according to ASHRAE 55 [30].

$$t_{op} = At_a + (1 - A)t_r \quad (12)$$

where A is a function of air speed and is approximately equal to 0.5 for air speeds less than 0.2 m s^{-1}, and t_r is the indoor mean radiant temperature, as shown in Eq. (13).

$$t_r = \sum_{j=1}^{4} F_{P-j} t_j \quad (13)$$

where F_{p-j} is the view factors between the occupant and the envelope surfaces and can be calculated according to ISO 7726 [31].

According to ISO 7730 and ASHRAE 55, the indoor operative temperature should be equal to 20 °C in the winter to maintain comfortable thermal conditions for a standard office worker in a room with a radiant heating system and a mechanical ventilation system. Hence, in this study, the indoor operative temperature is kept at 20 °C as the calculation condition.

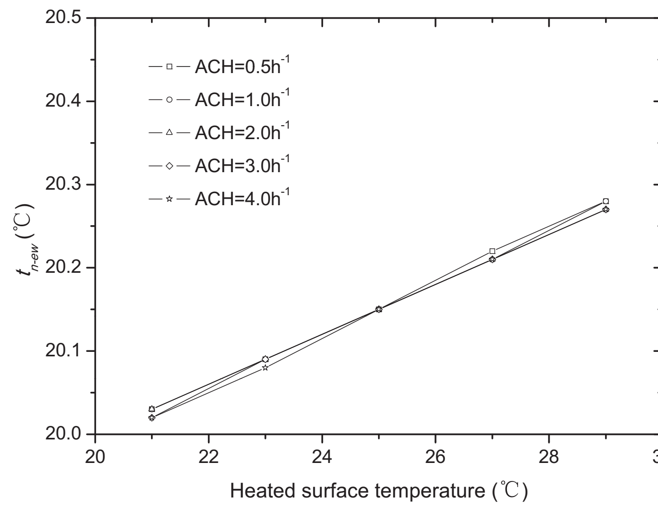

Fig. 2. Effect of heated surface temperature on indoor temperature for calculating transmission heat loss.

4.2. Results and discussion

To quantitatively analyze the effect of heated surface temperature and air change rate on the indoor temperatures for calculating the room heat loss and heating capacity of the hybrid system, the heated surface temperatures and air change rates are set between 21 to 29 °C and 0.5 and 4.0 h^{-1}, respectively. Additionally, the supply air temperature was kept at 16 °C to avoid draft.

4.2.1. Effect of heated surface temperature and air change rate on indoor temperatures for calculating room heat loss

(1) Effect of heated surface temperature and air change rate on indoor temperature for calculating transmission heat loss.

The effect of heated surface temperature and air change rate on indoor temperature for calculating transmission heat loss can be observed in Figs. 2 and 3.

The indoor temperature for calculating transmission heat loss is changed from 20.0 to 20.3 °C when the heated surface temperature is changed from 21.0 to 29.0 °C (as shown in Fig. 2). It can be observed that the heated surface temperature has a slightly positive impact on the indoor temperature for calculating the transmission heat loss. This is mainly because increased heated surface temperature will slightly increase the inside surface temperature of the internal walls; thus, the indoor temperature for calculating the transmission heat loss will slightly increase according to Eq. (6).

As shown in Fig. 3, the indoor temperature for calculating transmission heat loss remained relatively constant when the air change rate is changed from 0.5 to 4.0 h^{-1}, which indicated that the air change rate has nearly no influence on the indoor temperature for calculating transmission heat loss. This occurs

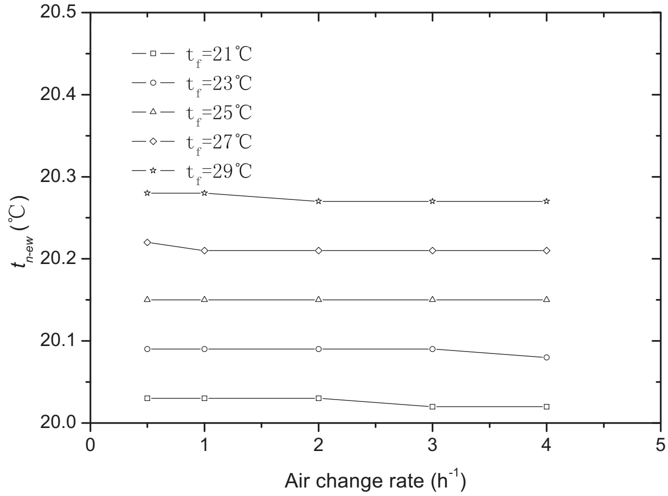

Fig. 3. Effect of air change rate on indoor temperature for calculating transmission heat loss.

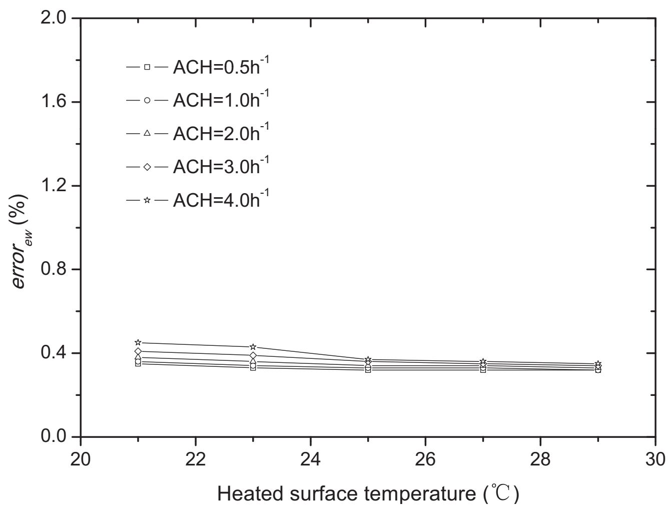

Fig. 4. Effect of heated surface temperature on the relative calculation error for transmission heat loss.

because the increased air change rate reduces the indoor air temperature, while the heated surface temperature and inside surface temperature of the internal walls should be increased to maintain an indoor operative temperature at 20 °C. As a result, the indoor temperature for calculating the transmission heat loss will remain relatively constant according to Eq. (6).

The effect of heated surface temperature and air change rate on the relative calculation error for transmission heat loss can be observed in Figs. 4 and 5, where the relative calculation error for transmission heat loss is the difference between the calculated transmission heat loss using t_{n-ew} and that using t_{op} (=20 °C), as shown in the following equation:

$$\text{error}_{ew} = \frac{(t_{n,ew} - t_{ex}) - \left(t_{op} - t_{ex}\right)}{t_{n,ew} - t_{ex}} = \frac{t_{n,ew} - t_{op}}{t_{n,ew} - t_{ex}} \quad (14)$$

As shown in Figs. 4 and 5, the relative calculation errors for transmission heat loss remains nearly constant when the heated surface temperature and air change rate are changed from 21.0 to 29.0 °C and from 0.5 to 4.0 h^{-1}, which indicates that both the heated surface and the cool supply air have nearly no impact on the relative calculation errors for transmission heat loss. This is mainly due to the rather small change of indoor

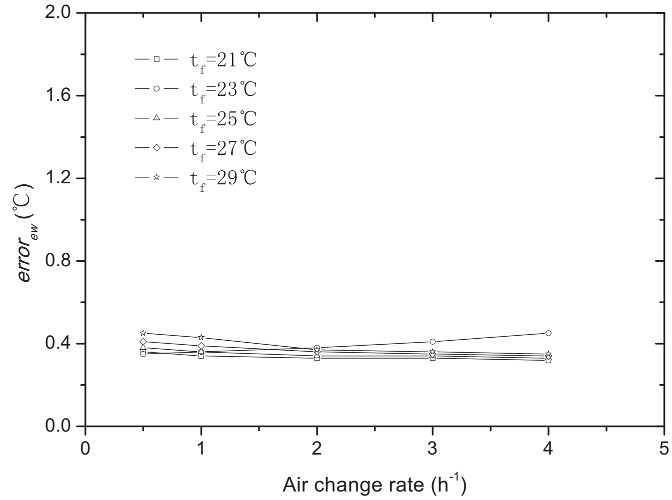

Fig. 5. Effect of air change rate on the relative calculation error for transmission heat loss.

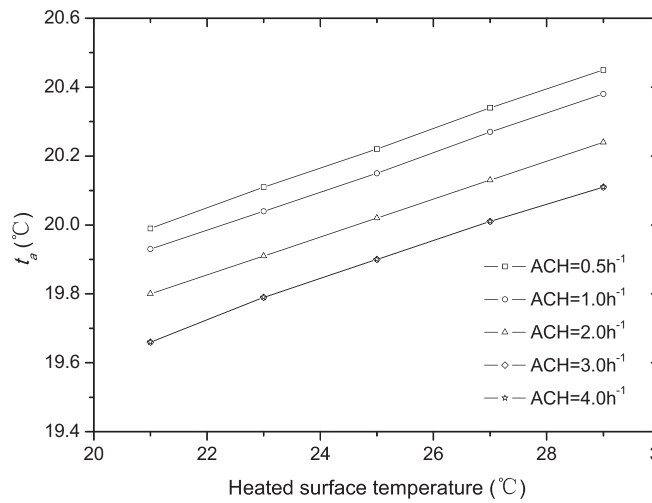

Fig. 6. Effect of heated surface temperature on indoor temperature for calculating ventilation heat loss.

temperature for calculating transmission heat loss as the heated surface temperature and air change rate is changed, as shown in Figs. 2 and 3.

Very small relative calculation errors occur (between 0.3% and 0.5%), so the effect of the heated surface and cool supply air on the indoor temperature for calculating transmission heat loss can be neglected.

(2) Effect of heated surface temperature and air change rate on indoor temperature for calculating ventilation heat loss.

Effect of heated surface temperature and air change rate on indoor temperature for calculating ventilation heat loss can be observed in Figs. 6 and 7.

The indoor temperature for calculating ventilation heat loss is changed from 19.6 to 20.5 °C when the heated surface temperature is changed from 21.0 to 29.0 °C (as shown in Fig. 6), which means that the heated surface temperature has a significantly positive impact on the indoor temperature for calculating ventilation heat loss. This is mainly because the increased heated surface temperatures will increase inside the surface temperature of the internal walls and thus increases the indoor air temperature.

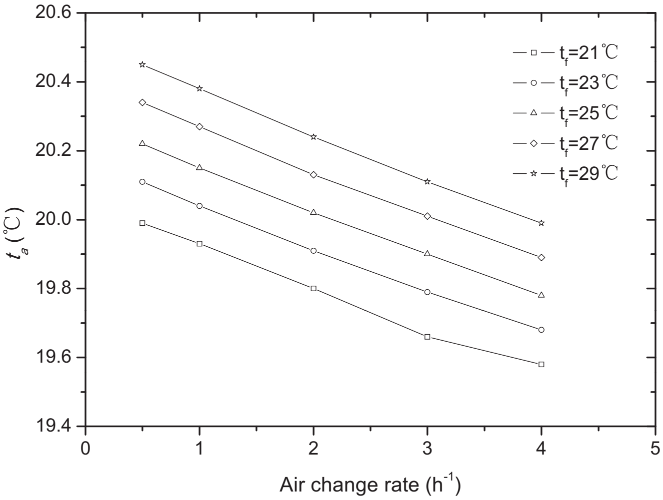

Fig. 7. Effect of air change rate on indoor temperature for calculating ventilation heat loss.

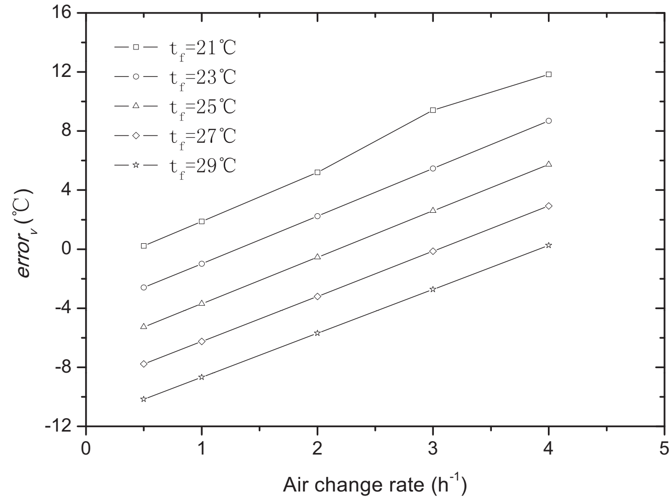

Fig. 9. Effect of air change rate on the relative calculation error for ventilation heat loss.

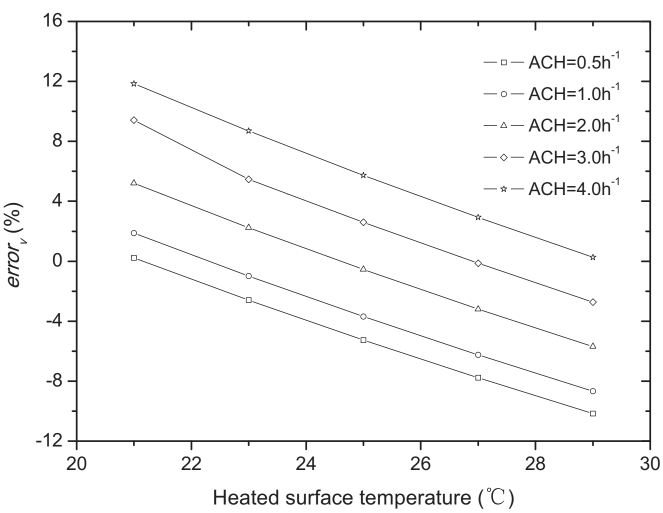

Fig. 8. Effect of heated surface temperature on the relative calculation error for ventilation heat loss.

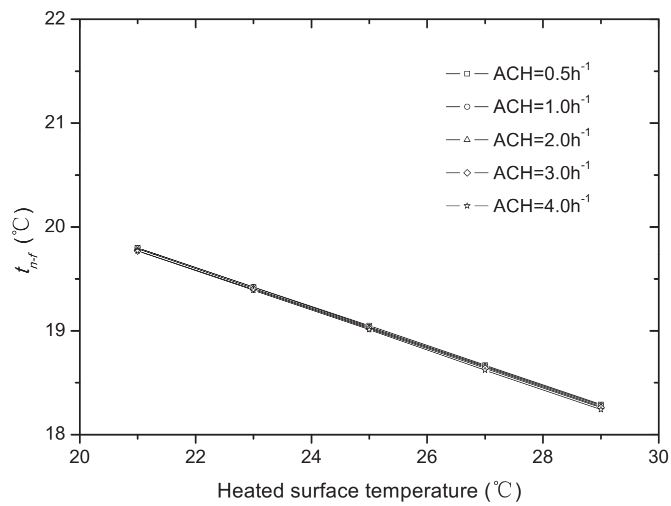

Fig. 10. Effect of heated surface temperature on indoor temperature for calculating heating capacity of the hybrid system.

As shown in Fig. 7, the indoor temperature for calculating ventilation heat loss is changed from 20.5 to 19.6 °C when the air change rate is changed from 0.5 to 4.0 h^{-1}. These results indicated that the air change rate has a significantly negative impact on the indoor temperature for calculating ventilation heat loss; primarily due to the increase in the air change rate that significantly reduces the indoor air temperature.

The effect of heated surface temperature and air change rate on the relative calculation error for ventilation heat loss can be observed in Figs. 8 and 9, where the relative calculation error for ventilation heat loss is the difference between the calculated ventilation heat loss using t_a and that using t_{op} (=20 °C), as shown in the following equation:

$$\mathrm{error}_v = \frac{(t_a - t_s) - (t_{op} - t_s)}{t_a - t_s} = \frac{t_a - t_{op}}{t_a - t_s} \quad (15)$$

As shown in Figs. 8 and 9, the relative calculation errors for ventilation heat loss are between −10.2% and 11.8% when the heated surface temperature and air change rate are changed from 21.0 to 29.0 °C and from 0.5 to 4.0 h^{-1}, so both the heated surface temperature and air change rate have a large impact on the relative calculation errors for ventilation heat loss. This is mainly due to the large indoor temperature change for calculating ventilation heat loss as the heated surface temperature and air change rate are changed, as shown in Figs. 6 and 7.

Large relative calculation errors (up to 11.8%) occur, so it is necessary to consider the effect of the heated surface and the cool supply air on the indoor temperature for calculating ventilation heat loss.

4.2.2. Effect of heated surface temperature and air change rate on indoor temperature for calculating heating capacity of the hybrid system

The effect of heated surface temperature and air change rate on the indoor temperature for calculating heating capacity of the hybrid system can be observed in Figs. 10 and 11.

As shown in Fig. 10, the indoor temperature for calculating the heating capacity of the hybrid system is changed from 19.8 to 18.2 °C when the heated surface temperature is changed from 21.0 to 29.0 °C, so the heated surface has a significantly positive impact on the indoor temperature for calculating the heating capacity of the hybrid system. These results occur mainly because the increased heated surface temperature will reduce the inside surface temperature of the external wall and windows to keep the

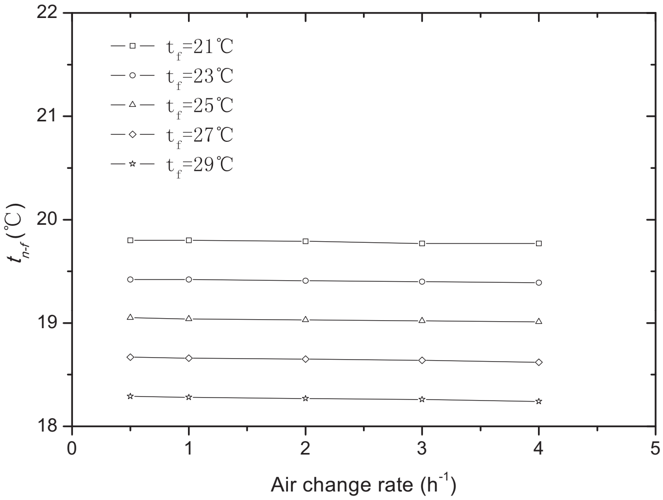

Fig. 11. Effect of air change rate on indoor temperature for calculating heating capacity of the hybrid system.

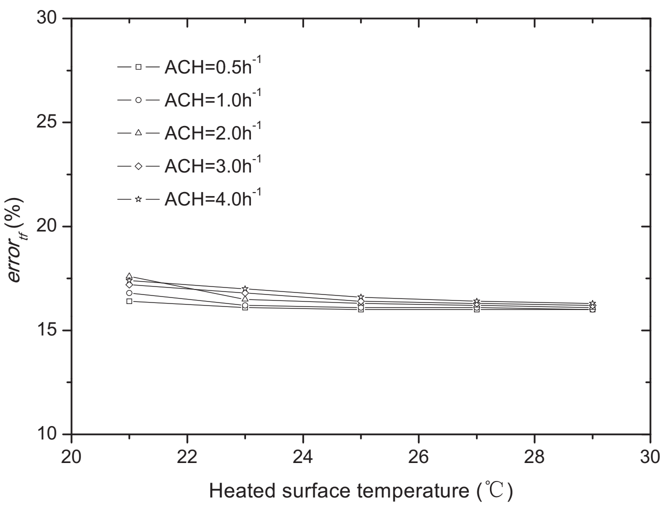

Fig. 12. Effect of heated surface temperature on the relative calculation error for heating capacity of the hybrid system.

indoor operative temperature at 20 °C; then, the indoor temperature for calculating the heating capacity of the hybrid system will decrease according to Eq. (10).

As shown in Fig. 11, the indoor temperature for calculating the heating capacity of the hybrid system remains nearly constant when the air change rate is changed from 0.5 to 4.0 h^{-1}, so the air change rate has a nearly no impact on the indoor temperature for calculating the heating capacity of the hybrid system. These results are mainly due to the increased air change rate that will reduce the indoor air temperature, and the heated surface temperature and the inside surface temperature of the internal walls should be increased to keep the indoor operative temperature at 20 °C. Then, the indoor temperature for calculating the heating capacity of the hybrid system will not change according to Eq. (10).

The effect of the heated surface temperature and air change rate on the relative calculation error for the heating capacity of the hybrid system can be observed in Figs. 12 and 13. The relative calculation error for the heating capacity of the hybrid system is the difference between the calculated heating capacity of the hybrid system using t_{n-f} and that using t_{op} (=20 °C), as shown in the following equation:

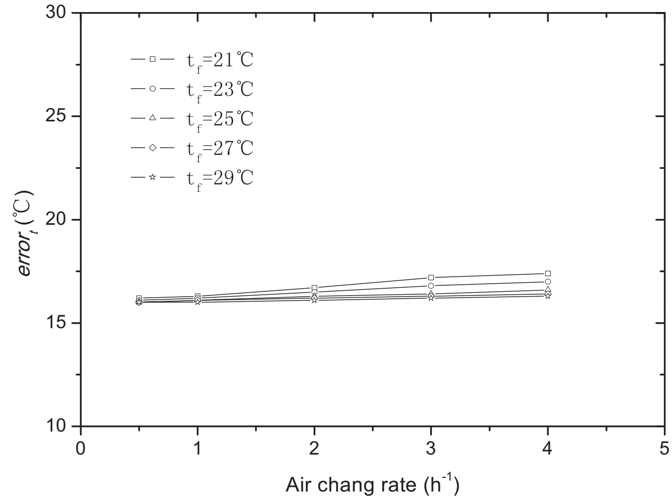

Fig. 13. Effect of air change rate on the relative calculation error for heating capacity of the hybrid system.

$$\text{error}_{tf} = \frac{\left(t_f - t_{n,f}\right) - \left(t_f - t_{op}\right)}{t_f - t_{n,f}} = \frac{t_{op} - t_{n,f}}{t_f - t_{n,f}} \quad (16)$$

As shown in Figs. 12 and 13, the relative calculation errors for the heating capacity of the hybrid system remains relatively constant when the heated surface temperature and the air change rate are changed from 21.0 to 29.0 °C and from 0.5 to 4.0 h^{-1}, respectively, which indicates that both the heated surface and the cool supply air have nearly no impact on the relative calculation errors for the heating capacity of the hybrid system. This is mainly due to the difference between the heated surface temperature and the indoor operative temperature that increased with the difference between the heated surface temperature and the indoor temperature for calculating the heating capacity of the hybrid system (as shown in Fig. 10). The small change of the indoor temperature for calculating the heating capacity of the hybrid system also impacted the results as air change rate changed (as shown in Fig. 11).

Large relative calculation errors (up to 17.4%) occurred, thus it is necessary to consider the effect of the heated surface and the cool supply air on the indoor temperature for calculating the heating capacity of the hybrid system.

5. Conclusions

A valid model to predict indoor temperatures in a typical office room with a radiant heating system and a mechanical ventilation system was developed. Qualitative analyses of the effect of heated surfaces and cool supply air on indoor temperatures for calculating room heat loss and the heating capacity of the hybrid system were performed using the proposed model. The conclusions of the study are as follows:

(1) Both the heated surfaces and the cool supply air had nearly no influence on the indoor temperature for calculating transmission heat loss, and their impact on indoor temperature for calculating transmission heat loss can be neglected due to very small relative calculation errors.
(2) Both the heated surfaces and the cool supply air had a large impact on the indoor temperature for calculating ventilation heat loss, and their impact on indoor temperature for calculating ventilation heat loss should be considered due to large relative calculation errors.
(3) Heated surfaces had a large impact on the indoor temperature for calculating the heating capacity of the hybrid system, and

cool supply air had nearly no influence on the indoor temperature for calculating the heating capacity of the hybrid system. The effect of the heated surfaces and the cool supply air on the indoor temperature for calculating the heating capacity of the hybrid system should be considered due to large relative calculation errors.

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