Modelling of the predicted thermal comfort of the metro passengers under different crowd densities

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Abstract: The aim of this study is to find out the nature of the metro passenger's thermal comfort under different crowd densities and different ambient environment factors. Taking the human factors, garment factors and ambient environmental factors into consideration, a sub-model human thermal perception was investigated based on the Fanger's heat balance equation. Also, a steady thermal sub-model of the B-type metro vehicle was built based on the heat conservation among the passengers, the outside-vehicle environment and the air of human-vehicle system. Then, the human thermal perception and its alternation were quantified by applying the PMV index. Simulation results show that the passenger's thermal perception cannot only variates with the airflow velocity, the relative humidity and the volume of the new air, but also with the crowd density.

Keywords: thermal comfort; metro; crowd density; thermal balance; predicted mean vote; PMV.

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1 Introduction

City rail transit (metro, monorail, light rail, tram, etc.) is an essential part of public transportation in big cities in China. With the development of the economy, the living standard of citizens has been enhanced dramatically. Citizens not only concern the safety, convenience and cost performance of public transportation but also pay close attention to the travelling experience and comfort. So, the comfort of public transportation is becoming a more and more significant issue in designing and operation process of municipal transits.

Metros are the universal ways of city rail transit in China, and it has become the preferred selection for commuters in sizable cities (Bao, 2018). However, the metro in China is always crowded, and the crowded passengers may significantly change the environment of the metro. For example, the temperature of the in-coach air will be affected by the heat emitted from the passengers, while the humidity will be changed by the water vapour emitted via human's respiration and evaporation from the skin, and consequently, the thermal perception of the passengers' may be altered. Thus, to investigate the human thermal comfort under different crowd densities is of great importance.

Meanwhile, there is nearly no literature reported investigating the inter-affecting of human thermal behaviour and vehicle thermal behaviour, as well as thermal comfort under different crowd densities in China's metro coaches. The studies are mainly focused on the thermal comfort in detailed environments and their focuses are mainly about the thermal comfort of the platform rather than metro coaches. Assimakopoulos and Katavoutas (2017) studied the thermal comfort at the platforms of Athens metro. Fukuyo et al. (2002), Tian et al. (2018) and Liu et al. (2017) also studied the thermal comfort of the metro platforms. However, there is fewer literature reported about the inner-coach thermal comfort of the metro. Abbaspour et al. (2008) used the relative warmth index to evaluate the thermal comfort in Tehran metro. Ordódy (2000) investigated the thermal comfort in the passenger areas of the Budapest metro. Jiang et al. (2007) examined the thermal comfort in Shanghai metro in China and the suggestions for improvement of the thermal comfort are put forward. Liao and Zou (2016) studied the thermal comfort and air quality of coaches of Shanghai metro in winter. However, these studies did not take the crowd density into consideration. Thus, the outputs of these studies may be not acceptable when the crowd density is relatively high and the ability of the air condition system is limited.

The effect of crowd density on thermal comfort is often studied in elevators in China. Wu et al. (2006) studied the air flow velocity on thermal comfort of the elevator and drew the conclusion that the thermal comfort would be optimised when the air flow velocity is higher when the elevator is crowded. Hu and Xu (2009) studied the increment of the PMV index under different crowd densities and concluded that the crowd density is the dominant factor affecting passengers' thermal perception. The situation of the metro coach is more or less similar to the elevator, which is of the feature that the inner space is relatively small and the crowd density is usually high. However, the metro has some characteristics different from the elevator, so there is needed to study the thermal comfort of the metro based on the crowd density.

2 Sub-model of human thermal behaviour

According to the first law of thermodynamics, the heat change rate between the passengers and the vehicle should equal to the heat production rate subtracts the heat storage rate.

The heat change between the passengers and the ambient environment can be in three main ways: radiation, convection and evaporation. In the 1970s, the heat balance equation was concluded by a Danish researcher, Fanger (1972), namely,

$$M - W = R + C + E + S \tag{1}$$

where M means the metabolic rate, W means the mechanical power of human activity, R, C and E mean the heat transfer rate via radiation, convection and evaporation, respectively, and S means the heat storage rate.

And, if the heat storage rate equals to zero or nearly zero, the human perception of the thermal environment will be nearly comfort.

Taking the real situation that the current Chinese metro is, the mechanical power will be not considered because there will be nearly no one who will do heavy physical work in the metro carriage. Thus, the equation can be simplified as

$$M = P + S \tag{2}$$

where P is the total heat change rate, apparently,

$$P = R + C + E \tag{3}$$

However, the method of obtaining the metabolic rate, the heat change rate is not directly given in Fanger's equation. Also, the heat storage rate appears to be non-intuitive to show the human's thermal perception to the ambient environment.

2.1 Factors affecting human thermal perception

Human thermal perception can be influenced by many factors, all of these factors can be concluded into three groups: human factors, garment factors and ambient environmental factors.

2.1.1 Human factors

Metabolic rate

In Chinese metro, the majority of the passengers will be standing without doing nothing, so the mean metabolic rate of a standing person can be taken as the mean metabolic rate of a metro passenger (ISO, 2004), namely,

$$M_H = 70\, W/\mathrm{m}^2 \tag{4}$$

Skin temperature

According to the Olesen and Fanger's (1973) study, the mean skin temperature is

$$T_H = 35.7 - 0.028\, M_H \tag{5}$$

where T_H is the temperature of the skin surface, and M_H is the metabolic rate.

Surface area

From the statistical data (National Health and Family Planning Commission of the People's Republic of China, 2015), the mean height of Chinese adult male is 1.671 m and that of an adult female is 1.558 m, and the mean weight of the adult male and adult female are 66.2 kg and 57.3 kg, respectively. According to the Stevenson's equation universally applied in China (Li, 2012), the surface area of the passenger is

$$A_H = 0.61\, H_H + 0.0128\, W_H - 0.1529 \tag{6}$$

where H_H is the height of the passenger, W_H is the weight of the passenger and A_H is the surface area of the skin. Thus, the surface area of the male passenger and female passenger are 1.71377 m^2 and 1.53092 m^2, respectively. Assuming that the percentage of the male and the female passenger are both nearly 50%, the surface area of the passenger can be 1.622 m^2, on average.

2.1.2 Garment factors

Thermal resistance

The thermal resistance is an index to show the ability of heat transferring from the ambient environment to the human body. Usually, the thermal resistance of the clothes is expressed in the way of 'clo' value and 1 clo equals 0.155 m^2 · °C/W (ISO, 2007). Namely,

$$\theta_c = 0.155\, R_c \tag{7}$$

where θ_c is the thermal resistance expressed in m^2 · °C/W while R_c is that expressed in clo.

Garment area factor

Usually, a dressed person has a larger radiation area than that of a naked person. And the rate between them can be described as the garment area factor. For a light-dressed person, the garment area factor can be predicted by an equation that Fanger (Huang and Zhang, 2011) suggested, namely,

$$f_c = 1 + 1.29\ R_c \quad \left(R_c < 0.078\ \text{m}^2 \bullet {}^\circ\text{C/W}\right) \tag{8}$$

while for the heavy-dressed one, the equation should be

$$f_c = 1.05 + 0.645\ R_c \quad \left(R_c \geq 0.078\ \text{m}^2 \bullet {}^\circ\text{C/W}\right) \tag{9}$$

2.1.3 Ambient environmental factors

Air temperature and mean radiation temperature

Usually, the metro vehicle is running underground in China (Ministry of Housing and Urban-Rural Development of the People's Republic of China, 2017), where the impact of solar radiation is non-significant, so the mean radiation temperature approximately equals to the air temperature, namely,

$$T_{amrt} = T_a \tag{10}$$

Air humidity

The partial pressure of saturated water vapour varies with the temperature change. Table 1 (Dai, 1999) shows the air temperature and its corresponding partial pressure of saturated water vapour at the temperature range of 0~44°C.

Table 1 The air temperature (T_a) and its corresponding partial pressure of saturated water vapour (P_a)

T_a (°C)	P_a (P_a)	T_a (°C)	P_a (P_a)	T_a (°C)	P_a (P_a)	T_a (°C)	P_a (P_a)	T_a (°C)	P_a (P_a)
0	610.5	9	1,146.4	18	2,063.5	27	3,564.4	36	5,939.8
1	655.8	10	1,227.7	19	2,196.8	28	3,779.1	37	6,274.4
2	705.2	11	1,310.3	20	2,338.1	29	4,004.3	38	6,623.7
3	757.1	12	1,402.3	21	2,486.0	30	4,241.6	39	6,990.3
4	811.8	13	1,497.0	22	2,643.3	31	4,492.2	40	7,374.2
5	871.8	14	1,598.3	23	2,806.6	32	4,753.5	41	7,776.7
6	934.4	15	1,704.9	24	2,983.3	33	5,032.1	42	8,198.0
7	1,001.1	16	1,816.9	25	3,167.2	34	5,318.7	43	8,637.8
8	1,071.7	17	1,936.8	26	3,360.5	35	5,622.6	44	9,099.1

To calculate the partial pressure of saturated water vapour for any T_a between 0~44°C, the polynomial regression was utilised to build the relationship between T_a and P_a. Table 2 shows the R-square value of the fitting by different orders of the polynomial.

Table 2 The R-square of different orders of the polynomial

Order(s) of the polynomial	*R-square value*
1	0.906
2	0.997
3	1.000
4	1.000
5	1.000

From Table 2, the three-order polynomial was selected, and the regression equation is,

$$P_a = 0.06688\, T_a^3 + 0.1046\, T_a^2 + 59.07\, T_a + 596.6 \tag{11}$$

If the relative humidity is taken into consideration, the partial pressure of water vapour is,

$$P_{a\varphi} = P_a \varphi_a \tag{12}$$

where φ_a is the relative humidity.

Airflow velocity around the human body

The airflow velocity around the human body can be measured by the hot wire anemometers in experiments.

2.2 Heat transfer coefficients

To study the thermal transfer among the human body, clothes and ambient environment, the index of heat transferring coefficients is needed to show the ability of heat transfer.

According to the Stefan-Boltzmann's law, the heat change rate of radiation from the surface with the temperature T_1 to the surface of temperature T_2 can be expressed as

$$E_R = \sigma\left(T_1^4 - T_2^4\right) \tag{13}$$

where σ is the Stefan-Boltzmann's constant and the value of it is 5.67×10^{-8} W/(m^2 · K^4), the unit of the temperature is Kelvin (K). And the definition of the radiation transfer coefficient is

$$h_r = E_R / \left(T_1 - T_2\right) \tag{14}$$

Thus, the radiation transfer coefficient can be expressed as

$$h_r = \frac{\sigma\left(T_1^4 - T_2^4\right)}{T_1 - T_2} \tag{15}$$

However, the radiation area of the human body does not equal to the surface area of the human body, and not all the heat the clothes received can be emitted, taking the clothes into consideration, the radiation heat transfer coefficient from the surface of clothes and the ambient atmosphere can be expressed as

$$h_r = \sigma \varepsilon f_{eff} \frac{\left(T_{cl} + 273.15\right)^4 - \left(T_{amrt} + 273.15\right)^4}{T_{cl} - T_{amrt}} \tag{16}$$

where ε is the emittance, for human skin, the value is nearly 1 while for clothes the value is 0.95; f_{eff} is the percentage of the valid radiation area of human body values 0.77 for standing person while it is 0.7 for the seated person (Huang and Zhang, 2011); T_{cl} and T_{amrt} are the temperatures of garment and that of the mean radiation temperature of ambient atmosphere and their unit are degree centigrade.

To simplify the calculation of the radiation heat transfer coefficient, the value of the ε is taken as 0.97. And considering that a large percentage of passenger will stand while taking the metro, f_{eff} is considered as 0.75. Then the equation (16) can be simplified as

$$h_r = 4.125 \times 10^{-8} \times \frac{\left(T_{cl} + 273.15\right)^4 - \left(T_{amrt} + 273.15\right)^4}{T_{cl} - T_{amrt}} \tag{17}$$

The convection is also a dominant way of heat change. There are two types of convection, natural convection and forced convection. The natural convection transfer coefficient is a function of the difference in the temperature (Nielsen and Pedersen, 1952), usually,

$$h_{cl} = 2.38\left(T_{cl} - T_a\right)^{0.25} \tag{18}$$

where T_{cl} is the temperature of the garment surface and T_a is the temperature of the ambient atmosphere. Meanwhile, the forced convection transfer coefficient is a function of air flow velocity (Winslow et al., 1939), usually,

$$h_{cf} = 12.1\sqrt{V} \tag{19}$$

where V is the air flow velocity. And the convection transfer coefficient is the maximum one of h_{cl} and h_{cf}, namely,

$$h_c = \max\left\{h_{cn}, h_{cf}\right\} \tag{20}$$

2.3 *Calculation of the surface temperature of the clothes*

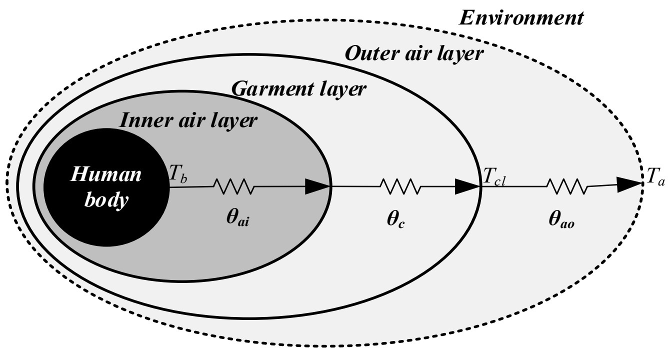
As shown in Figure 1, the three-layer model of the human-clothes-environment system is usually applied to analyse the thermal behaviour of the clothes.

The first layer is the inner air layer, which is a thin air layer between the human body and clothes. The heat will transfer via conduction and radiation. Thus, the thermal resistance of the layer is (Arslanoglu and Yigit, 2016),

$$\theta_{ai} = \frac{1}{h_r + k_a / x} \tag{21}$$

where k_a is the thermal conductivity of the air and looks on as a constant with the value of 24 W/(mm • K), and x is the average distance between the human body and clothes.

Figure 1 The human-clothes-environment system



The second layer is the garment layer, and the thermal resistance of the layer is θ_c.

The third layer, or outer air layer, however, is an imaginary one, which is regarded as a thermal resistance between the surface of the clothes and the ambient environment. Usually, the thermal conduction is not the main route of heat transfer. Thus, the thermal resistance of the outer air layer is (Arslanoglu and Yigit, 2016),

$$\theta_{ao} = \frac{1}{h_r + h_c} \tag{22}$$

An assumption is made that the heat emitting from the clothes equals to the heat gaining from the human body, namely,

$$\frac{T_b - T_{cl}}{\theta_{ai} + \theta_c} = \frac{T_{cl} - T_a}{\theta_{ao}} \tag{23}$$

Thus, the surface temperature of clothes is,

$$T_{cl} = \frac{\theta_{ao} T_b + (\theta_{ai} + \theta_{cl}) T_a}{\theta_{ai} + \theta_{cl} + \theta_{ao}} \tag{24}$$

To calculate the T_{cl}, the h_r and h_c are needed. However, the calculation equations of h_r and h_c are also the function of T_{cl}. Moreover, it arises out of some difficulty to induce the T_{cl} by simultaneous solution process. So, a new approach of calculating T_{cl} should be developed. Considering that the value of T_{cl} is between that of T_a and T_b due to the heat conservation, the T_{cl} can be calculated in the following steps,

1 Create a vector $\mathbf{x} = [x_1, x_2, \ldots, x_n]^T$ from T_a to T_b and the intervals are 0.001°C.

2 For each x_i (i = 1, 2, …, n), calculate the index

$$D_i = \left| \frac{\varepsilon m_i T_b + (n_i + \theta_{cl}) T_a}{n_i + \theta_{cl} + \varepsilon m_i} - x_i \right| \tag{25}$$

with

$$m_i = \frac{1}{u_i + v_i} \tag{26}$$

$$n_i = \frac{1}{u_i + k_a/x} \tag{27}$$

$$u_i = 4.125\frac{(x_i + 273.15)^4 - (T_{mrt} + 273.15)^4}{x_i - T_{mrt}} \tag{28}$$

$$v_i = \max\left\{2.38(x_i - T_a)^{0.25}, 12.1\sqrt{V}\right\} \tag{29}$$

3 The x_i which makes that the D_i is the minimum one, then xi is the calculated approximation of T_{cl}, namely,

$$T_{cl} = x_k\big|_{D_k = \min D_i\,(i=1,2,\ldots n)} \tag{30}$$

2.4 Calculation of the heat emission and the heat storage

From the fundamental theories of heat transfer, the heat emission rate via radiation can be calculated by following equations,

$$R = f_{cl}h_r\left(T_{cl} - T_{amrt}\right) \tag{31}$$

And the heat emission rate via convection can be calculated by Newton's law of cooling, namely,

$$C = h_c\left(T_{cl} - T_a\right) \tag{32}$$

The evaporation heat change can be obtained by the experimental formula built by ASHRAE (1993). Namely,

$$E_{res} = 0.0173\,M\left(5.87 - P_{a\varphi}\right) + 0.0014\,M\left(34 - t_a\right) \tag{33}$$

$$E_{pers} = \frac{\left(0.06 + 0.94W_{rsw}\right)\cdot 16.7h_c\cdot\left(P_{sk}^* - P_{a\varphi}\right)}{1 + 0.143\,h_c f_c \theta_c} \tag{34}$$

where E_{res} and E_{pers} are the heat transfer rate of the evaporation via respiration and perspiration; P_{sk}^* is the partial pressure of saturated water vapour under the skin temperature; W_{rsw} is the sweating rate of the skin.

Thus, the total heat change rate and the heat storage rate can be calculated, namely,

$$P = R + C + E_{res} + E_{pers} \tag{35}$$

$$S = M - P \tag{36}$$

2.5 Build and validation of the human thermal perception model

The predicted mean vote or PMV index (ISO, 2005) is an index of human thermal perception in Fanger's (1972) research. The definition of the *PMV* is

$$PMV = \left(0.303e^{-0.036M} + 0.0275\right)S \tag{37}$$

The meaning of the PMV is shown in Table 3.

Table 3 The meaning of the PMV

PMV	*Meaning (human perception of the thermal environment)*
–3	Very cold
–2	Cold
–1	Cool
0	Comfortable
+1	Warm
+2	Hot
+3	Very hot

Thus, if the PMV value is calculated, the mean human perception of the thermal environment can be estimated.

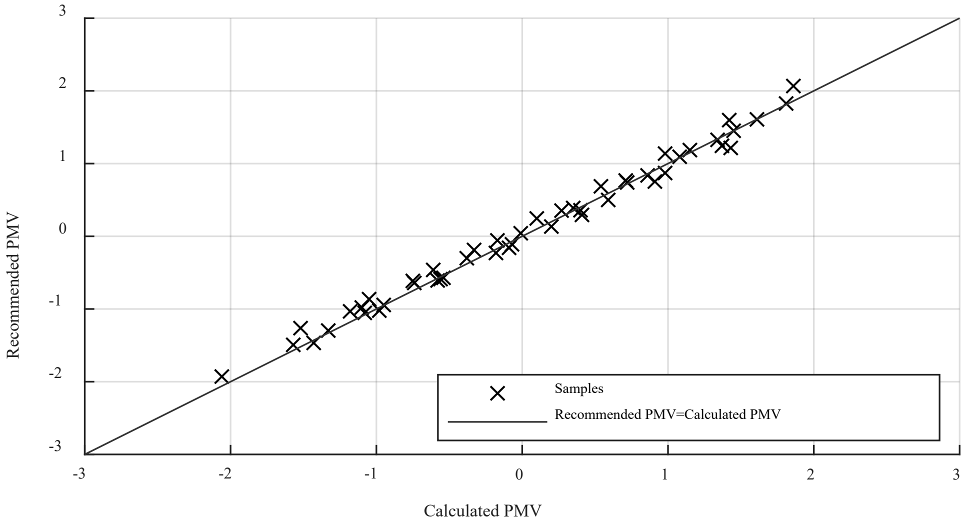
Table 4 shows PMV value of some specific environments with the humidity of 50% recommended by ISO (2005).

Table 4 The PMV value recommended by ISO 7730

Condition	θ_{cl} *(clo)*	*M* *(W/m^2)*	*V* *(m/s)*	T_a *(°C)*	*PMV*	*Condition*	θ_{cl} *(clo)*	*M* *(W/m^2)*	*V* *(m/s)*	T_a *(°C)*	*PMV*
1	0.25	58.15	0.1	24	–1.52	25	0.5	58.15	0.15	23	–1.33
2	0.25	58.15	0.1	25	–1.05	26	0.5	58.15	0.15	24	–0.95
3	0.25	58.15	0.1	26	–0.61	27	0.5	58.15	0.15	25	–0.56
4	0.25	58.15	0.1	27	–0.17	28	0.5	58.15	0.15	26	–0.18
5	0.25	58.15	0.1	28	0.27	29	0.5	58.15	0.15	27	0.20
6	0.25	58.15	0.1	29	0.71	30	0.5	58.15	0.15	28	0.59
7	0.25	58.15	0.1	30	1.15	31	0.5	58.15	0.15	29	0.98
8	0.25	58.15	0.1	31	1.61	32	0.5	58.15	0.15	30	1.37
9	0.25	58.15	0.2	24	–2.06	33	1	69.60	0.1	16	–1.18
10	0.25	58.15	0.2	25	–1.57	34	1	69.60	0.1	18	–0.75
11	0.25	58.15	0.2	26	–1.08	35	1	69.60	0.1	20	–0.33
12	0.25	58.15	0.2	27	–0.58	36	1	69.60	0.1	22	0.10
13	0.25	58.15	0.2	28	–0.09	37	1	69.60	0.1	24	0.54
14	0.25	58.15	0.2	29	0.41	38	1	69.60	0.1	26	0.98
15	0.25	58.15	0.2	30	0.91	39	1	69.60	0.1	28	1.42
16	0.25	58.15	0.2	31	1.43	40	1	69.60	0.1	30	1.86
17	0.5	58.15	0.1	23	–1.10	41	1	69.60	0.2	16	–1.43
18	0.5	58.15	0.1	24	–0.74	42	1	69.60	0.2	18	–0.98
19	0.5	58.15	0.1	25	–0.38	43	1	69.60	0.2	20	–0.54
20	0.5	58.15	0.1	26	–0.01	44	1	69.60	0.2	22	–0.07
21	0.5	58.15	0.1	27	0.35	45	1	69.60	0.2	24	0.40
22	0.5	58.15	0.1	28	0.72	46	1	69.60	0.2	26	0.86
23	0.5	58.15	0.1	29	1.08	47	1	69.60	0.2	28	1.34
24	0.5	58.15	0.1	30	1.45	48	1	69.60	0.2	30	1.81

Figure 2 shows the validation results.

Figure 2 Validation results



The calculated PMV_1 is highly correlated with the recommended PMV_2 (with R = 0.99). Assume that,

1 H_0: $PMV_1 = PMV_2$

2 H_1: $PMV_1 \neq PMV_2$.

Then, selecting the level of significance $\alpha = 0.05$, and the degree of the freedom is 94, the T-statistic (Walpole et al., 2012) is

$$t = \frac{\bar{x}_1 - \bar{x}_2}{\sqrt{\frac{47s_1^2 + 47s_2^2}{94}} \bullet \sqrt{\frac{1}{47} + \frac{1}{47}}} \tag{38}$$

where $\bar{x}_1, \bar{x}_2$ are the average value of PMV_1 and PMV_2; s_1, s_2 are the standard deviation of PMV_1 and PMV_2.

Then $t = -0.1565$, and $t_0 = 1.9855$, then $-t_0 < t < t_0$, the assumption H_0 is accepted. That is, the calculated PMV from equation (37) is valid at the significance of 0.05.

3 Sub-model of the vehicle thermal behaviour

The model of the vehicle was built via the first law of thermodynamics, namely, when the thermal state is steady, for the air inside the vehicle,

$$\sum_{i=0}^{m} Q_i = 0 \tag{39}$$

And it can be deduced that

$$\sum_{i=1}^{m} P_i = \sum_{i=1}^{m} \frac{Q_i}{A\Delta t} = 0 \tag{40}$$

Namely, when the sum of heat transfer rate is almost zero, the state is the thermal steady state.

Assuming that,

1 the vehicle is a B-type metro in China
2 the vehicle is running with all the doors and windows closed
3 the heat from the sun and the vehicle equipment is ignored, only the heat from passengers, from the walls, floor and ceiling and from the air conditioning devices is taken into account.

Thus, if the parameters of the B-type metro and the heat change of the vehicle are known, the thermal model can be established to simulate the thermal behaviour of the vehicle.

3.1 Parameters of the B-type metro

The vehicle model is based on the B-type metro in China, the inner space of the vehicle is 19,000 mm × 2,800 mm × 2,300 mm (SAC, 2016), and the estimated volume of the air inside the vehicle is about 120 m^3 without any passengers. If there are n passengers inside the train, the air volume is,

$$Vol_a = 120 - \sum_{i=1}^{n} Vol_{hi} \tag{41}$$

where Vol_{hi} is the volume of the passengers. Usually, the density of human tissue is about 1,000 kg/m^3 (Heymsfield, 1989), and if the average weight of passengers is taken, then equation (41) can be simplified as

$$Vol_a = 120 - \frac{nW_H}{1{,}000} \tag{42}$$

The heat transfer area of the vehicle surface is around 200 m^2 without taking the heat change between the coach and the connected coach. The heat conductivity is about 2.3 $W/m^2 \cdot °C$ (Wang et al., 2013). And the maximum capacity of the vehicle is around 240 persons per car (SAC, 2013).

3.2 The thermal model of the vehicle

Passengers affect the air temperature by means of heat radiation, convection and evaporation. The heat transfer of a single passenger has been discussed in Section 2 of this paper. The heat transfer rate of the crowd of passengers is the summation of each passenger's, namely,

$$P_H = \sum_{i=1}^{n} P_i \tag{43}$$

where *Pi* is the heat change rate of a single passenger. If taking the parameters of each passenger are similar, the equation (43) can be simplified as,

$$P_H = nP \tag{44}$$

Also, the heat can transfer from the inner space and out-train atmosphere via the walls, floor and ceiling can be calculated via equation (45).

$$P_o = k_v A_v \left(T_{out} - T_a\right) \tag{45}$$

where k_v is the heat conductivity of the vehicle, and A_v is the valid heat transfer area of the vehicle, which values are 2.3 W/m^2 · °C and 200 m^2 as mentioned.

And the heat transfers by the air conditioner is

$$P_{ac} = \frac{\rho qc}{3{,}600}\left(T_{ac} - T_a\right) \tag{46}$$

with

$$\frac{\rho qc}{3{,}600}\left|T_{out} - T_{ac}\right| < P_m \tag{47}$$

where ρ is the air density, q is the volume of the fresh air per hour or the flow rate of the fresh air, c is the heat capacity of the air at constant volume, T_{ac} is the new air temperature, T_{out} is the temperature outside the train, P_m is the maximum power of the air conditioning devices.

Then, the heat change rate of the vehicle is,

$$P_v = P_H + P_o + P_{ac} \tag{48}$$

when $P_v > 0$, it means the air is heated and vice versa; and if $P_v = 0$, there will be a uniform thermal environment between the vehicle and the passengers.

Then, the temperature change rate of the air inside the metro can be derived.

$$\Delta T = \left.\frac{P_v}{\text{cm}}\right|_{\Delta t = 1s} \tag{49}$$

with

$$m = \rho Vol_a \tag{50}$$

4 Simulations and applications

Two sub-models are coupled and can be utilised to simulate the thermal behaviour of the vehicle and the passengers to study the thermal perception of the passengers under different seasons, different crowd densities and different ambient environments.

The estimation of the mean thermal comfort can be given by the PMV value, when the thermal environment is steady. However, the limited ability of the air conditioning devices makes that the control of the air temperature would fail. So, there are two questions need to be discussed before estimation, those are,

1 Can the temperature be comfortable when the steady state?

2 If the temperature is not comfortable at the steady state, what is the mean thermal perception of the passengers?

Firstly, the air best temperature that passengers will feel most comfortable can be defined as the temperature with $PMV = 0$, namely,

$$T_{opt} = \{T_i \,|PMV_i = 0, i = 1, 2, \ldots, n\} \tag{51}$$

For instance, if the metabolic rate is 58.15 W/m^2, the thermal resistance of the clothes is 0.25 clo, the relative humidity is 50% and the air velocity is 0.2 m/s, the best air temperature is 28.34°C, and from Table 4, the best air temperature is 28.18°C by linear interpolation.

Next, the mean thermal perception will be estimated in the following way, if the needed human factors, garment factors and environmental factors are given.

1 Calculate the T_{opt} with $PMV = 0$ as the set temperature.

2 Assume the initial value air temperature and the new air temperature is T_{set}, namely,

$$\begin{cases} T_a(1) = T_{opt} \\ T_{ac}(1) = T_{opt} \end{cases} \tag{52}$$

3 Calculate the temperature change $\Delta T_a(i)$ by equation (49).

4 Refresh the air temperature inside the metro, namely,

$$T_a(i+1) = T_a(i) + \Delta T_a(i) \tag{53}$$

5 If $T_a(i + 1) > T_{opt}$ then $T_{ac}(i + 1) = T_{ac}(i) - 0.1$ and if $T_a(i + 1) < T_{opt}$ then $T_{ac}(i + 1) = T_{ac}(i) + 0.1$ while

$$\frac{\rho qc}{3{,}600}\left|T_{out} - T_{ac}(i)\right| < P_m \tag{54}$$

or $T_{ac}(i + 1) = T_{ac}(i)$ if equation (54) is not satisfied.

6 If existing the integer N for all $i > N$ the $\Delta T_a(i)$ is small enough or i reaches the giving integer M, the calculation process should be stopped. And define the steady temperature,

$$T_{ess} = T_a(k) \tag{55}$$

where $T_a(k)$ is the value of $T_a(i)$ when calculation stops; or re-do the steps from 3 to 6.

7 Predict the mean thermal perception of passengers by the PMV value calculated by Tess, the thermal perception can be expressed in the words listed in Table 3.

For instance, if the metabolic rate is 69.60 W/m^2, the height and the weight of the passenger are 1.6145 m and 61.75 kg, respectively, the thermal resistance of the clothes is

0.5 clo, the relative humidity is 50%, the air velocity is 0.2 m/s, and the outside temperature is 32°C, the crowd density is 100 persons per coach and the new air flow rate is 5,400 m^3/h. If the power of the air conditioner is limited in the range of 15 kW, the estimated temperature of the air inside at the steady state the train is 28.28°C and the estimated PMV is 1.0041. Thus, the predicted thermal perception is 'warm'.

However, the mean thermal perception is not enough to describe the comfort of the passengers. So, the PPD index is implemented to estimate the percentage of passengers who will feel uncomfortable. The PPD can be calculated as follows (ISO, 2005),

$$PPD = 100 - 95e^{-0.03353 PMV^4 - 0.2179 PMV^2} \tag{56}$$

In the case, the PPD value is 26.29%, which shows that there will be 26.29% of passengers are predicted to be uncomfortable.

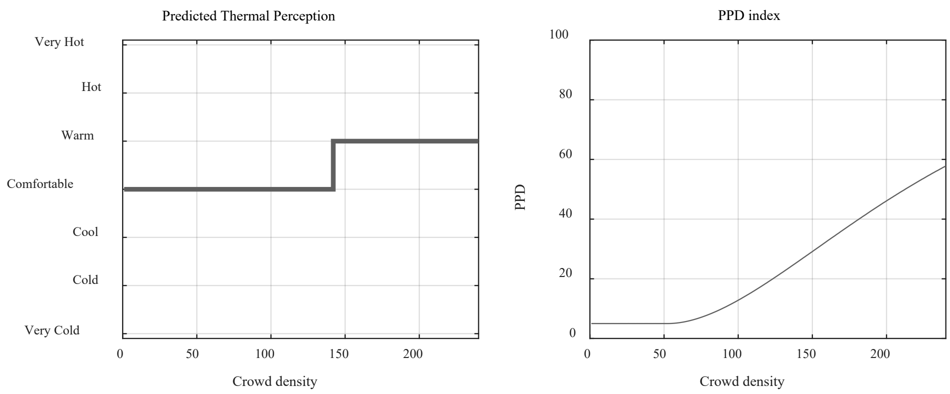
4.1 The estimation of the thermal comfort in summer in the air-conditioned metro

Assuming that the temperature of the tunnel is 30°C, the air flow velocity inside the train is 0.25 m/s, the relative humidity is 60%, the thermal resistance of the clothes is 0.5 clo, the new air flow rate is 5,400 m^3/h and the maximum power of the air conditioner is 15 kW and the metabolic rate, the height and the weight of the passenger are 70 W/m^2, 1.6145 m and 61.75 kg, respectively, the best temperature with $PMV = 0$ is calculated, namely

$$T_{set} = 25°C \tag{57}$$

Then, the interaction process can be applied and the PMV index can be in application to estimate the thermal perception of the passengers under different conditions. Figure 3 shows the estimated thermal perception of passengers under different crowd densities from 1 person per car to 240 persons per car. As shown in Figure 3, if the crowd density is less than 141 persons per car, passengers will feel comfortable, on average.

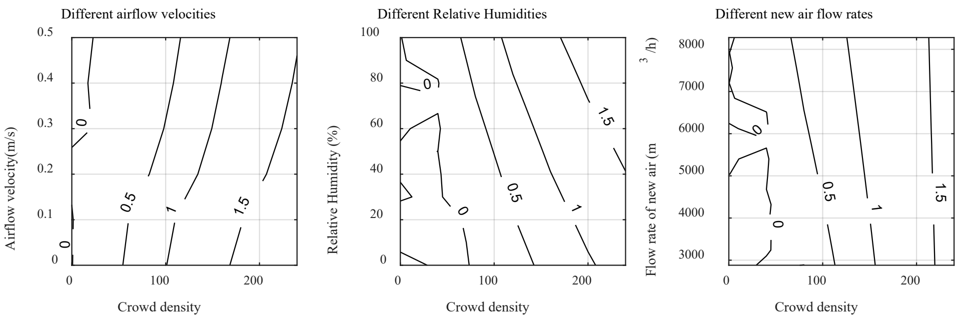
Figure 3 The estimated thermal comfort of the passengers under different crowd density in summer



And the value of PPD index is shown in Figure 3. For example, if the crowd density is 124 persons per car, the estimated thermal perception is 'comfortable' while there will be 20% of passengers will feel uncomfortable.

If the air flow velocity varies from 0 m/s to 0.5 m/s, the relative humidity varies from 0% to 100%, or the new air flow rate changes from 3,000 m^3/h to 8,000 m^3/h, respectively, without changing other factors, the PMV index will be shown as Figure 4.

Figure 4 The PMV under different crowd densities in summer



4.2 *The estimation of the thermal comfort in spring or autumn in the air-conditioned metro*

Assuming that the temperature of the tunnel is 20°C, the air flow velocity inside the train is 0.2 m/s, the relative humidity is 50%, the thermal resistance of the clothes is 0.75 clo, the new air flow rate is 5,400 m^3/h and the maximum power of the air conditioner is 15 kW and the metabolic rate, the height and the weight of the passenger are 70 W/m^2, 1.6145 m and 61.75 kg, respectively, the best temperature with $PMV = 0$ is calculated, namely

$$T_{set} = 23.8°C \tag{58}$$

Then, the interaction process can be applied and the PMV index can be in application to estimate the thermal perception of the passengers under different conditions. Figure 5 shows the estimated thermal perception of passengers under different crowd densities from 1 person per car to 240 persons per car. As shown in Figure 5, no matter how the crowd density is, passengers will feel comfortable, on average. And the value of PPD index is shown in Figure 5.

If the air flow velocity varies from 0 m/s to 0.5 m/s, the relative humidity varies from 0% to 100%, or the new air flow rate changes from 3,000 m^3/h to 8,000 m^3/h, respectively, without changing other factors, the PMV index will be shown as Figure 6.

Figure 5 The estimated thermal comfort of the passengers under different crowd density in spring or autumn

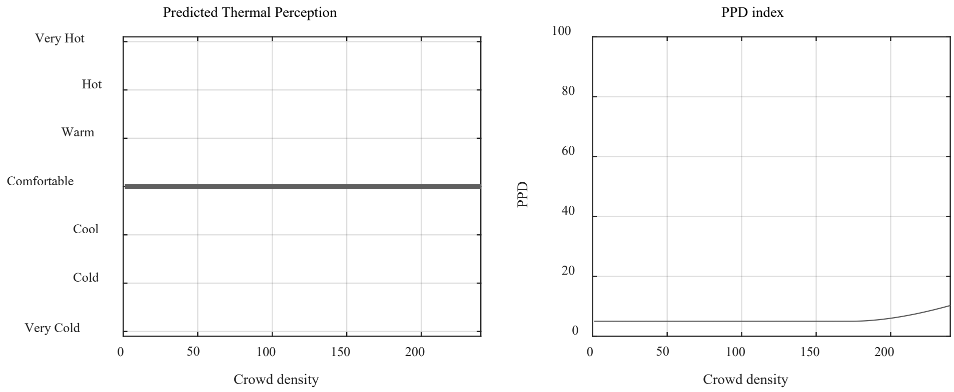
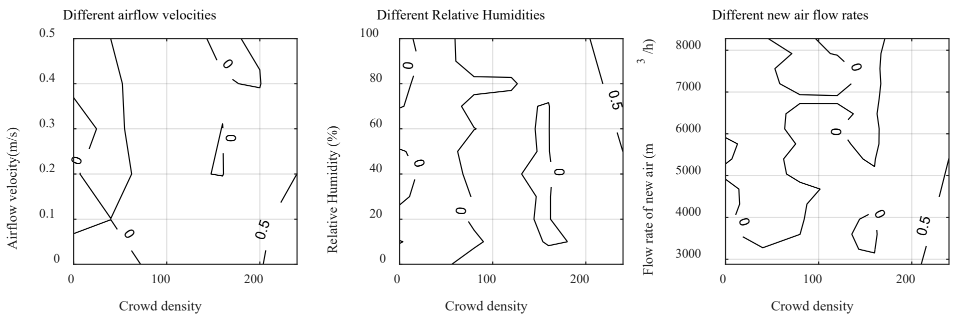


Figure 6 The PMV under different crowd densities in spring or autumn



4.3 The estimation of the thermal comfort in winter in the air-conditioned metro

Assuming that the temperature of the tunnel is 5°C, the air flow velocity inside the train is 0.15 m/s, the relative humidity is 40%, the thermal resistance of the clothes is 1.5 clo, the new air flow rate is 5,400 m^3/h and the maximum power of the air conditioner is 15 kW and the metabolic rate, the height and the weight of the passenger are 70 W/m^2, 1.6145 m and 61.75 kg, respectively, the best temperature with $PMV = 0$ is calculated, namely

$$T_{set} = 19.16^{o}\text{C} \tag{59}$$

Then, the interaction process can be applied and the PMV index can be in application to estimate the thermal perception of the passengers under different conditions. Figure 7 shows the estimated thermal perception of passengers under different crowd densities from 1 person per car to 240 persons per car. As shown in Figure 7 shows that in winter, passengers will feel comfortable under almost all the crowd density. And the value of PPD index is shown in Figure 7.

If the air flow velocity varies from 0 m/s to 0.5 m/s, the relative humidity varies from 0% to 100%, or the new air flow rate changes from 3,000 m^3/h to 8,000 m^3/h, respectively, without changing other factors, the PMV index will be shown as Figure 8.

Figure 7 The estimated thermal comfort of the passengers under different crowd density in winter

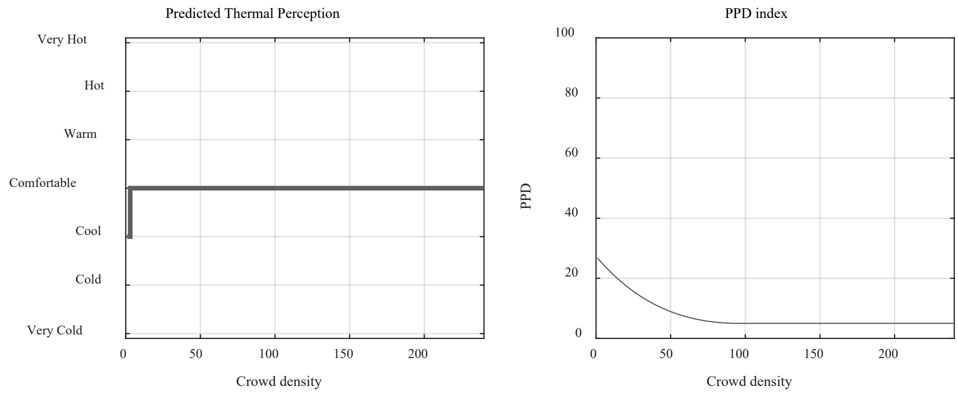
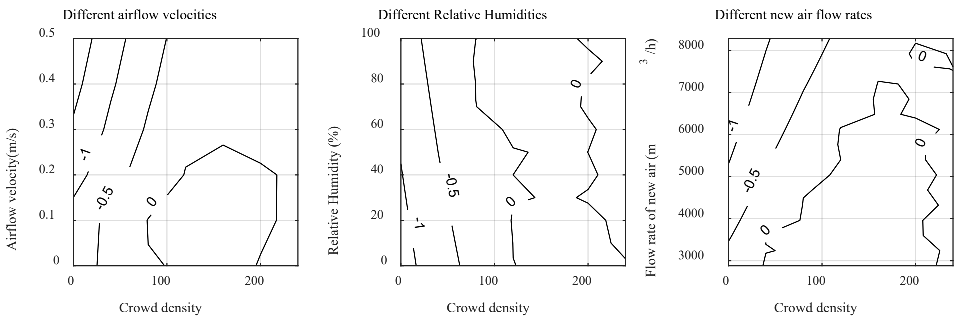


Figure 8 The PMV under different new air flow rates and crowd densities in winter



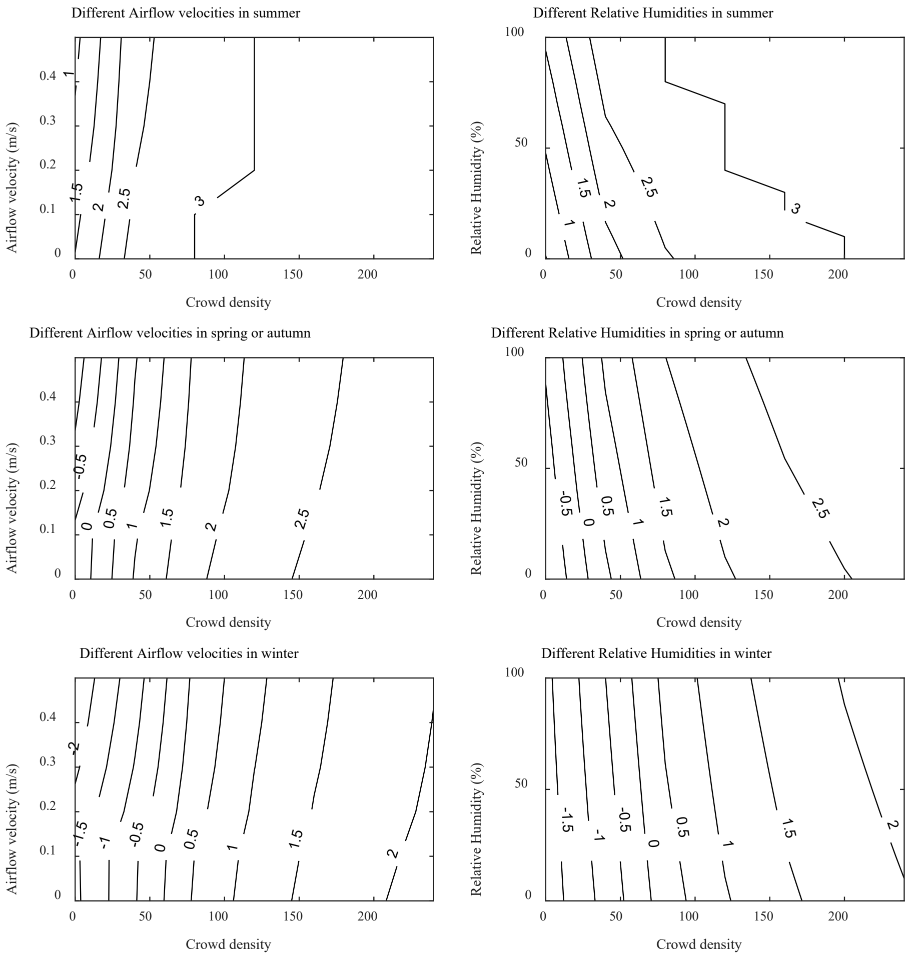
4.4 *The estimation of the thermal comfort of the metro without air conditioning system*

The PMV index under different air flow velocities and different relative humidity in summer, spring and autumn and winter when the air conditioner is invalid is shown in 0. The conditions are shown in Table 5.

Table 5 Simulation conditions

	Summer	*Spring and autumn*	*Winter*
Outside temperature (°C)	30	20	5
Thermal resistance of clothes (clo)	0.5	0.75	1.5
Metabolic rate (W/m^2)	70	70	70
Average height (m)	1.6145	1.6145	1.6145
Average weight (kg)	61.75	61.75	61.75

Figure 9 The PMV index without air conditioning devices running



5 Conclusions

Based on the Fanger's heat balance equation and the law of conservation of the energy, the sub-model of human thermal behaviour and vehicle thermal behaviour are established, and the applications of utilising of the coupled model are discussed. Then, the model is applied to estimate the mean thermal comfort in different seasons under different crowd densities, different air flow velocities, different relative humidities and different new air flow rates. And from the estimation of the thermal comfort in summer, spring and autumn and winter, some conclusions can be drawn.

1 In summer or the ambient temperature is much higher than the temperature of $PMV = 0$, the air conditioner can more or less prevent passengers from uncomfortable experience, however, if the higher the crowd density is, the more difficult to handle the temperature. And a higher airflow velocity, a lower humidity

and a lower rate of the new air can be of benefit in managing the air temperature in the range that is comfortable for passengers.

2 In spring and autumn or the ambient temperature is near the temperature of *PMV* = 0, the function of the air conditioner is to keep the neutral thermal condition. As the thermal load is relatively low in this condition, the temperature is relatively easy to handle.

3 In winter or the ambient temperature is much lower than the temperature of *PMV* = 0, the air conditioner can prevent passengers from uncomfortable experience when the crowd density is low. And the heat emits from passengers can be helpful in decreasing the heat load of the air conditioning devices when the crowd density is moderate. And the adequately lower airflow velocity and the lower new air flow rate can be helpful in meeting the needs of passengers.

4 If the air conditioning devices or systems are out of function, passengers will feel hot or very hot when the crowd density is extremely high. Thus, the impact of the crowded passengers on the thermal comfort is significant.

However, there are some questions leaving need to be further studied. First, the sweat is not taken into consideration, as a result, the thermal perception in the summer under the relatively high air flow velocity may be worse than the real situation because the perspiration will help human to lose heat. Second, cold and wet air flowing between the clothes and human body via convection and penetration through the clothes in winter is not taken into consideration, thus, the prediction of the thermal perception under the high relative humidity may be not the case. Third, heat from equipment and devices is not taken into consideration. Fourth, water vapour emitted by passengers are not taken into account. Fifth, dynamic change and the spatial distribution of the factors are not studied and need future simulation by CFD or other methods. And finally, the quantitative estimation methods of the optimum crowd density, optimum airflow velocity, optimum relative humidity and an optimum new air flow rate are not formed.

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