RESEARCH ON THE INFLUENCE OF VENTILATION MODE ON THE CONDENSATION RISK AT THE WALL RADIANT TERMINAL

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Abstract. The separate control of indoor heat and humidity in radiant air conditioning has more energy-saving advantages than convection air conditioning that use the same cold source to handle heat and humidity loads, which is in line with the international trend of carbon reduction and emission reduction. However, the surface of the radiation terminal is prone to condensation, which poses a health and safety hazard. In order to reduce the risk of condensation at the wall radiant terminal, this paper uses CFD to study the effect of ventilation mode on the risk of condensation at the wall radiant terminal, and evaluates the risk of condensation by using the term of the minimum supercooling condensation temperature. The simulation results show that the thermal boundary layer on the surface of the radiant terminal under different ventilation velocity is different, which affects the minimum supercooling condensation temperature, and makes the condensation risk of the radiant air conditioning system different. When the wind speed of the attached jet is 1.07 m/s, the thickness of the thermal boundary layer is 0.045 m, and the minimum supercooling condensation temperature at the radiation terminal is 1.79 K. By further increasing the wind speed and reducing the thickness of the thermal boundary layer, the minimum supercooling condensation temperature can be further increased to 2.2 K, reducing the risk of condensation.

1 introduction

Since the global attention to climate change has increased, in order to cope with the severe carbon reduction situation and implement the "Paris Agreement", countries have set carbon peaking and carbon neutrality goals one after another. Among them, from the perspective of China, the carbon emission of the construction sector accounts for 51.2% of the national carbon emission [1]. Therefore, it is imperative to save energy and reduce emissions in the building sector. At present, the common forms of air conditioning in buildings include convection air conditioning and radiant air conditioning. Radiant air conditioning, as an emerging form of air conditioning, has demonstrated its energy-saving effect through a series of experiments and simulations [2, 3]. J. Miriel et al. [4] found that radiant air conditioners can save 10% energy compared with convection air conditioning through TRANSYS simulation and experimental cabin measurements. Corina S [5, 6] conducted experiments in an office in the United States, and the results show that the radiant cooling system can save 30% energy compared with the traditional all-air system. Therefore, radiant air conditioning has application potential in the current background of energy saving and carbon reduction.

However, the radiant panel is limited by condensation when it is used [7]. If the droplets condensed on the surface of the radiant panel are not removed, the indoor bacteria and other microorganisms will grow over time with mildew and other phenomena, which will affect the indoor air quality and pose potential health and safety hazards[8].

The most common way to reduce the risk of condensation in a radiant air-conditioned room is to use a supply air system to reduce indoor humidity. Fernandez F et al. [9] found through experiments and simulations that when the floor air supply and the floor radiant panel are coupled, the cold air avoids the contact between the humid air and the low temperature floor, and prevents water from condensing on the floor. Ding Y et al. [10] found that for the radiant ceiling panel, the mode of up supply and up return can better prevent the occurrence of condensation. Kyu-Nam, RHEE et al. [11] showed through experiments that when the air supply speed of the ceiling radiant panel increased to 2 m/s, the cooling capacity of the radiant ceiling panel was increased, and the temperature difference between the indoor air and the surface of the radiant panel was enlarged, which reduces the risk of condensation. However, the existing studies are all focused on the radiation from the floor or the ceiling, and there is a lack of research on the risk of condensation on the wall radiant panels.

In view of the condensation risk of the wall radiant panels, starting from the condensation conditions of side wall radiant panels, this paper analyzes the effect of the air supply velocity on the condensation in different ventilation modes by using CFD for simulation, and explores ventilation mode that reduce the risk of condensation.

2 Condensation conditions on the wall radiant panels

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In the widely accepted theory of "condensation nucleation hypothesis", the thermodynamic method is used to analyze the mechanism of condensation droplets. By balancing the internal and external balance of spherical and spherical droplets with a radius of r, it can be concluded that the pits with a radius of r_c on the wall require the minimum supercooling condensation temperature to occur, and the theoretical calculation formula of the minimum supercooling condensation temperature can be obtained [12].

However, this theoretical formula is only applicable to droplets whose shape is spherical. Condensed droplets appearing on a radiant panel are subjected to gravity and surface tension. When the radiation panel is laid on the side wall, for the droplets with diameters of microns and below, the gravity is negligible compared with the surface tension, so the droplets are closer to the spherical shape, but when the droplet size gradually increases, the influence of gravity cannot be ignored, the droplet deviates from the thermodynamic equilibrium state, and the phenomenon of contact angle hysteresis occurs [13]. At this time, the shape of the droplet is not spherical. Therefore, the theoretical derivation of the conditions for generating the condensation droplet on the side wall needs to be carried out again.

Assuming that the droplet shape on the wall radiant panel is shown in Fig.1, the shape of the contact line is a combination of two ellipses, that is, the upper and lower half contours of the contact line on the horizontal axis x are both half ellipses [14]. From the side view, the highest height of the droplet is h, and the droplet tends to slide down, so the lower contact angle is the advancing contact angle θ_a, and the upper contact angle is the receding contact angle θ_R.

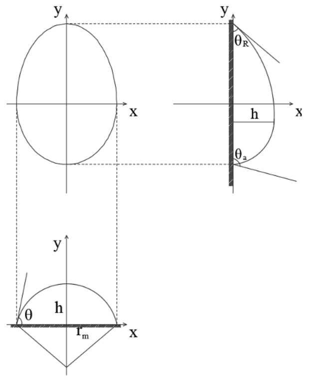


Fig. 1. Contact line shape model

According to the Laplace equation, Clausius-Carabellon equation, and assuming that the steam is an ideal gas, the degree of subcooling necessary to maintain the equilibrium droplet in the form shown in Figure 1 can be obtained.

$$\Delta T_{wi} = T_s - T_w = (T_s - T_\infty) + (T_\infty - T_s + \frac{2\sigma}{h} A \cdot \frac{v_l T_s}{h_{fg}}) \cdot (\frac{\delta_t}{\delta_t - h}) \quad (1)$$

$$A = sin^2\left(\frac{\theta_R}{2}\right) + sin^2\left(\frac{\theta_a}{2}\right) \quad (2)$$

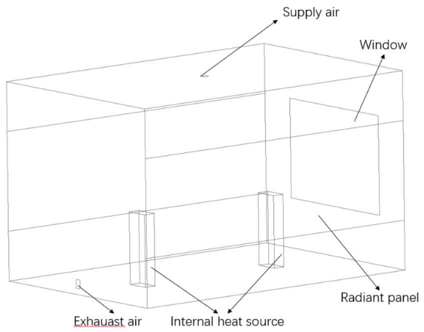
$$h = \frac{1 - cos\theta}{sin\theta} \cdot r_m \quad (3)$$

Among them, ΔT_{wi} is the minimum supercooling condensation temperature on the wall, which is the difference between the dew temperature and the wall temperature, K; T_s is the saturation temperature under steam pressure, namely air dew point temperature, K;T_w is the Wall temperature, K;T_∞ is the air temperature in the mainstream area, K; σ is the surface tension, N/m, take 0.0725 N/m; θ is the angle between the liquid and the solid-liquid interface, that is, the equilibrium contact angle, ° ; r_m is the mouth radius of the conical pit on the wall, m; θ_a、 θ_R is the contact angle before and after the rolling direction of the droplet, namely the advancing contact angle and the receding contact angle, ° ; v_l is the Liquid specific volume, m^3/kg, take 0.001 m^3/kg; h_{fg} is the latent heat of vaporization of water, kJ/kg, take 2257.6 kJ/kg; δ_t is the thickness of the temperature boundary layer, m;

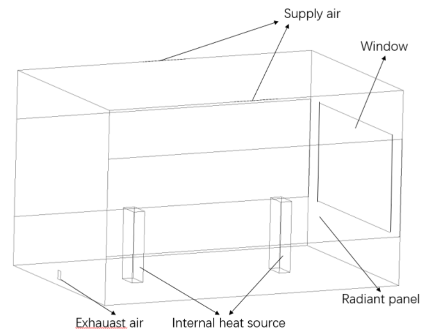
According to the calculation formula (1) of the minimum supercooling condensation temperature at the wall, the factors that actually affect the minimum supercooling condensation temperature are θ, θ_a, θ_R, r_m and δ_t. In order to expand the minimum subcooling condensation temperature, these factors can be changed to reduce the risk of condensation caused by the dew temperature rising rapidly above the surface temperature of the radiant panel when the humidity in the room increases.

3 Simulation process

In order to study the effect of air supply velocity on the temperature boundary layer and even the minimum supercooling condensation temperature in the ventilation mode, the physical model of the coupling wall radiant terminal of the two air supply systems of the up supply and down return and the attached jet is established in CFD, as shown in Fig.2. below.



(a) up supply and down return air supply system



(b) attached jet system

Fig. 2. Room physical model

The object of the simulation is a standard office located in an ultra-low energy building in Shanghai, China. The size of the room is 5.4 m*3.6 m*3.3 m, including a 1.8 m*2.3 m north window. The indoor temperature is set to 26 °C and the humidity is 60 %. In this study, it is considered that the office is located in the middle of the floor, with only one north wall as the outer wall, and the surrounding rooms are all air-conditioned rooms, so the influence of heat transfer from the inner envelope structure is ignored. Since the exterior windows are north windows, the influence of direct sunlight on indoor heat exchange is ignored. Therefore, the indoor cooling load includes heat transfer from exterior walls and windows, as well as heat dissipation from indoor personnel, equipment, and lighting, and simplify the indoor heat source into two cuboids with 0.4 m*0.2 m*1.2 m. The air supply speed can be changed by changing the size of the air supply outlet. The specific cases are shown in Table 1. The thermal boundary layer thickness on the surface of the radiant panel under different air supply speeds can be obtained through numerical simulation.

Table 1. Simulation case

Case for simulation	Supply air temperature is 296 K, radiant panel temperature is 293 K		
	Air outlet size	wind speed	Air supply system
Case1	0.25*0.2 m	0.33 m/s	up supply and down return
Case2	0.2*0.2 m	0.42 m/s	
Case3	0.15*0.1 m	1.11 m/s	
Case4	0.8*0.01 m	1.07 m/s	attached jet

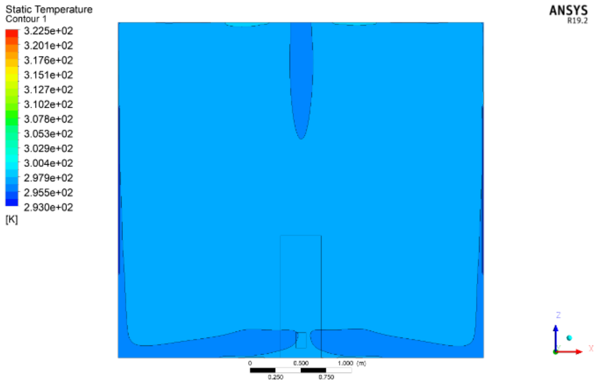
Therefore, the boundary condition settings in CFD can be summarized in Table 2 below, and the energy equation, standard R-ε turbulence model and DO radiation model are enabled at the same time.

Table 2. Boundary Condition Setting

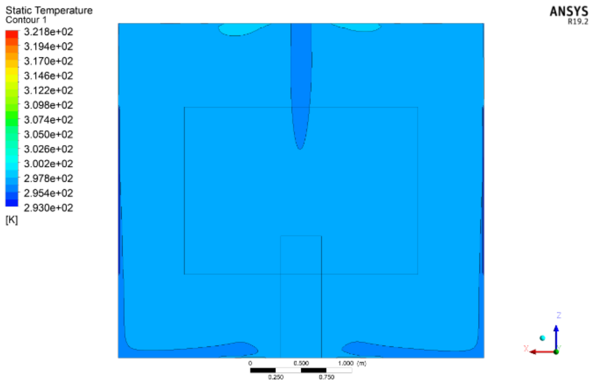
Boundary	Boundary conditions	
North outer wall	Heat transfer coefficient	0.4 W/(m^2·K)
	External free airflow temperature	307.6 K
North outer window	Heat transfer coefficient	2.1 W/(m^2·K)
	External free airflow temperature	307.6 K
Inner envelope	Heat flow	0 W/m^2
Radiant panel	Surface temperature	293 K
Air supply outlet	Speed exit	
AIR exhaust	Pressure outlet	
Energy equation source term	Evenly distribute the cooling load generated by the heat source of irregular objects such as people and lamps to each side of the heat source in the room	

4 Results analysis

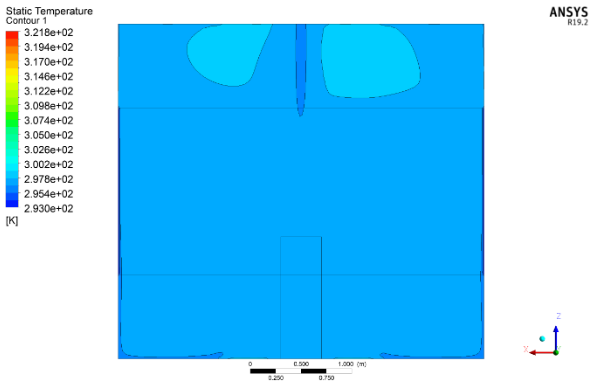
The temperature profile of the cross-section of the room is shown below.



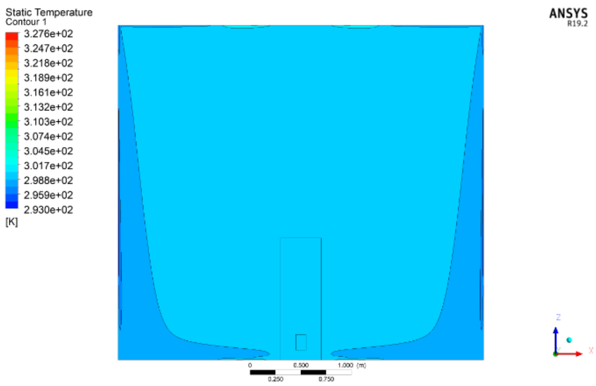
(a) Case1(v=0.33 m/s)



(b) Case2(v=0.42 m/s)



(c) Case3(v=1.11 m/s)



(d) Case4(v=1.07 m/s)

Fig. 3. Temperature cloud map

It can be seen from the temperature distribution cloud diagram that there is a significant temperature change near the radiant panel. The thickness of the thermal boundary layer is defined as the distance to the wall when the excess temperature of the fluid defined by the wall temperature reaches 99% of the excess temperature, that is, $t - t_w = 0.99(t_\infty - t_w)$. By calculating the temperature near the radiant panel under the three working conditions, the thickness of the temperature boundary layer under each working condition can be obtained. When the indoor temperature is 26℃, the humidity is 60%, the dew temperature is 17.66℃, the contact angle θ of the radiant panel is 50°, the advancing contact angle θ_a is 70°, the receding contact angle θ_R is 40°, and r_m is 0.0001 m, by substituting different thermal boundary layer thicknesses into formula (1), the minimum supercooling condensation temperature under different wind speeds can be obtained.

Table 4. Temperature boundary layer thickness

Case	Boundary Layer δ_t Thickness	the minimum supercooling condensation temperature
Case1	0.0750 m	1.78696 K
Case2	0.0608 m	1.78710 K
Case3	0.0575 m	1.78715 K
Case4	0.0453 m	1.78737 K
Case5	0.01 m	1.79105 K
Case6	0.001 m	1.83377 K
Case7	0.0001 m	2.28162 K

It can be seen from the results in the table 4. that the thickness δ_t of the thermal boundary layer gradually decreases with the increase of the wind speed at the upper and lower return air outlets. Comparing case 3 and case 4, although the wind speed of the attached jet air outlet is lower than that of the upper and lower return, the thickness δ_t is still smaller than that of case 3, because the airflow of the attached jet is close to the wall radiant panel, and the wind speed on the surface is close to air outlet speed. The supply outlet for the upper and lower return is in the center of the ceiling, and the wind speed reaching the wall radiant panel is much smaller than the wind speed of the supply outlet. That is, a smaller thermal boundary layer thickness and a larger minimum supercooling condensation temperature can be obtained at the same wind speed by using the attached jet.

Comparing the minimum supercooling condensation temperature in the former four cases, it can be found that the change is not large, and if the wind speed continues to increase, the thickness δ_t will continue to decrease, such as case5 to case7, it can be seen that the minimum supercooling condensation temperature significantly increased. Therefore, in order to reduce the risk of condensation, the attached jet can be used to supply air at a high wind speed close to the radiant panel to increase the minimum supercooling condensation temperature. Therefore, the radiant panel in the room has a higher possibility of not reaching the condensation condition in a high humidity environment, and more difficult to dew condensation.

5 Conclusion

Based on the condensation nucleation theory, this paper theoretically deduces the occurrence conditions of condensation on the wall radiant terminal. When condensation droplets appear on the wall radiant terminal, the difference between the dew temperature and the wall surface temperature must be greater than the minimum supercooling condensation temperature. The magnitude of the difference is related to the contact angle and surface roughness of the radiant panel, as well as the airflow organization of the air conditioning system.

In order to increase the minimum supercooling condensation temperature and reduce the risk of condensation, the attached jet can be used. When the air velocity of the attached jet outlet is 1.07 m/s, the minimum supercooling condensation temperature on the surface can reach 1.79 K.

Compared with the air supply form of upper and lower return, the attached jet can reduce the thermal boundary layer on the surface of the radiant panel. It is considered that the attached jet with high wind speed can be preferentially used for coupling with the radiant terminal, which provides suggestions for the design of the radiant air conditioning system. .

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