Effect of Top Ventilation on Cooling Characteristics of Passenger Cabin

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Research Article

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Effect of Top Ventilation on Cooling Characteristics of Passenger Cabin
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Abstract
In order to solve the hot soak effect of car during summer parking, the CFD numerical method was used to simulate the cooling law of passenger cabin under different inlet area, inlet velocity and inlet angle at the top vent. From the two dimensions of cooling rate and cooling effect, the influence of top natural ventilation and top forced ventilation on the cooling characteristics of passenger cabin were studied. The results show that under the condition of top natural ventilation, the cabin can reach thermal balance at about 10 min under different vent area conditions, and the cooling rate is the largest when the vent is fully opened, and the minimum temperature can reach about 45 °C within 4 min, which is 10 °C higher than the ambient temperature. Under forced ventilation, the inlet velocity has a great influence on the cooling rate and cooling effect of cabin. When the inlet angle is 90° and the inlet velocity is 5 m / s, the cooling rate is the largest and the cooling effect is the best.

1 Introduction
Taking the typical summer climate in southern China as an example, after the outdoor cars are exposed to the sun, the hot soak effect generated in the cabin will not only make the drivers feel uncomfortable, but also greatly affect the restart performance of the automobile air conditioning. At the same time, the hot soak effect leads to the high temperature environment in the cabin, which will release a large number of harmful substances from the plastic parts in the cabin, and seriously affect the health of the crew[1],[2]. Thus, it is particularly important to study the ventilation and cooling characteristics of cars in summer.

In order to reduce the temperature in the passenger cabin in summer, domestic and foreign scholars have done in-depth research on improving the thermal environment of the passenger cabin. Wei et al. It was very necessary to solve the high temperature environment in passenger cabin in summer[3]. Grundstein et al. The temperature in the passenger cabin of automobile in different seasons and different weather conditions is collected through experiments, they were found that the temperature in the passenger cabin can still reach the lethal temperature even in the cloudy environment[4]. Zhang et al. Through the comparison of experiment and simulation, the accuracy of the prediction model is verified, and the influence of cabin temperature field and velocity field on occupant thermal comfort and energy consumption is analyzed[5]. Kilic et al. The influence of three different inlet forms on the thermal environment of the cabin was studied by numerical simulation[6]. Turnow et al. Numerical simulation method was used to study the flow and temperature distribution in the cabin under three different intake modes, and the influence of different flow structures on human skin temperature was discussed[7]. Sabora et al. Effects of three ventilation schemes on thermal comfort of occupants, the results of the numerical simulations showed that human thermal comfort in the passenger car cabin and energy efficiency is influenced by the applied ventilation variant[8]. Parrino et al. The natural ventilation precooling is used to reduce the temperature in the passenger compartment before starting the automobile air conditioning, the actual influence of ventilation precooling on the thermal environment of the passenger cabin and the optimal ventilation time and ventilation air volume are analyzed[9]. Yoon et al. The effects of different inlet angles and
ventilation modes on cabin environment were studied\textsuperscript{[10]}. Han et al. Influence factors of air cooling in passenger cabin. The effect of a 50% reduction in interior thermal mass showed positive effects on the AC performance during the early stage of the cool-down process, in the first 6 minutes of cooling, the cooling rate is the fastest, after 30 minutes of cooling, the cabin temperature reaches a stable state\textsuperscript{[11]}. Zhang et al. studied the relationship between the cabin temperature variation characteristics and the window glass parameters, and found that the appropriate window glass transmittance can greatly reduce the cabin temperature\textsuperscript{[12],[13]}. Chen et al. studied the effects of different air outlet positions and inlet parameters of the air conditioning on the thermal environment in the cabin\textsuperscript{[14]}. Huang et al. studied the effects of different air flow rates of the air conditioning on the temperature change of the cabin, and their results showed that when the air flow rate was 25L–30L/s, the temperature difference between the inside and outside of the cabin could be controlled at 7\textdegree–9\textdegree C\textsuperscript{[15]}.

At present, scholars mainly study the ventilation and cooling characteristics based on air conditioning vents or window gaps, while there are few studies on the ventilation and cooling through the top of the car (especially the sunroof). However, for the passenger cabin with hot soak effect, the internal air temperature shows obvious stratification. The air temperature near the top of the passenger cabin is the highest, and the temperature near the foot is the lowest. The maximum temperature difference can reach 9\textdegree C\textsuperscript{[13]}. Therefore, it is particularly important to study the cooling characteristics of the passenger cabin through the top ventilation of the car. In this paper, a certain type of vehicle is taken as the research object. FLUENT19.0 software is used to simulate the two cases of natural ventilation and forced ventilation at the top of the vehicle\textsuperscript{[16]}. The effects of different ventilation areas, different wind speeds and different inlet angles on the cooling characteristics of the passenger cabin are studied respectively, which provides certain reference for the study of the thermal comfort of the car.

2 Passenger Cabin Geometry Model and Simulation Settings

2.1 Establishment of cabin geometric model

SolidWorks was used to carry out 1:1 geometric modeling of the passenger cabin of a certain type of car. In order to reduce the number of grids while ensuring the quality of grids, the model was simplified, with the rearview mirror, dashboard buttons and door trims ignored. The geometric model is shown in Fig.1.

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{figure1.png}
\caption{Position of key monitoring points in passenger cabin}
\end{figure}

2.2 Mathematical model and simulation settings

Without considering a small amount of gas exchange with the outside through the door gap, it is considered that the indoor air tightness is good. Assuming that the air in the cabin is incompressible and consistent with the Boussinesq assumption, the change of gas density is only related to the temperature change, and only the change of fluid density is considered when calculating buoyancy. The dominance equations for the numerical analysis in this study are as follows:
\[ \rho = \rho_0 [1 - \beta (T - T_0)] \]  

(1)

Where \( T \) and \( T_0 \) are air temperature and ambient temperature; \( \rho_0 \) is the air density at temperature \( T_0 \); \( \beta \) is the thermal expansion coefficient.

The accuracy of CFD calculation depends largely on the quality of the grid. In order to analyze the problem more accurately, the seat area grid is encrypted. The unstructured tetrahedral grid is used in the whole calculation area. The maximum size of the body grid and the car surface is 20mm, and the maximum size of the seat surface is 10mm. FLUENT19.0 finite element analysis software was used for numerical simulations. The fluid dynamics equation is solved by the finite volume method. The turbulence model is chosen as the realizable \( k-\varepsilon \) model and solved by the following equations (2) and (3)\(^{[17]} \). This model can accurately simulate the diffusion velocity of plane or circular jets such as air conditioning outlets.

\[
\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_i} \left[ \left( u_i + \frac{u_i}{\sigma_k} \right) \frac{\partial k}{\partial x_i} \right] + G_k + G_b - \rho \varepsilon
\]

(2)

\[
\frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial (\rho \varepsilon u_i)}{\partial x_i} = \frac{\partial}{\partial x_i} \left[ \left( u_i + \frac{u_i}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_i} \right] + \rho C_1 S_{\varepsilon} - \rho C_2 \varepsilon \left( \frac{\varepsilon^2}{k + \sqrt{\varepsilon}} + \frac{C_{1\varepsilon}}{k} \right) + C_{1\varepsilon} \varepsilon 
\]

(3)

Formula (2) and formula (3) are \( k \) equation and \( \varepsilon \) equation respectively, where \( G_k \) is a turbulent kinetic energy term produced by laminar velocity gradient; \( G_b \) is a turbulent kinetic energy term produced by buoyancy; and \( C_2 \) are constants; \( k \) and \( \varepsilon \) are turbulence Prandtl numbers of \( k \) equation and \( \varepsilon \) equation, respectively. The expression of parameters and the empirical value of constants are shown in formula (4).

\[
\begin{align*}
C_1 &= \max \left[ 0.43 \frac{\eta}{\eta + 5} \right], \quad \eta = S \frac{k}{\varepsilon} \\
C_{1\varepsilon} &= 1.44; C_2 = 1.92; \sigma_k = 1; \sigma_\varepsilon = 1.2
\end{align*}
\]

(4)

The near wall is treated by the standard wall function, and the momentum and turbulence equations are discretized by the second-order upwind scheme. The SIMPLEC algorithm is used to solve the incompressible flow field. The inlet adopts the velocity inlet condition; Outlet is the pressure export condition; The relevant wall is set as wall, and the default value of surface roughness is adopted. The thermal radiation parameter of wall boundary condition is set as BC Type glass to be semi-transparent wall, and other walls are set as opaque wall. In the simulation analysis, the radiation heat transfer involved is calculated using the Do model. The sunshine factor is set to 1. Namely, sunny weather and the solar radiation at 13 noon in summer in southern China is simulated.

The maximum wall temperature can reach 72°C\(^{[17]} \) when the sun directs the seat surface, and the seat temperature has an important influence on the thermal comfort of the passenger. Therefore, the seat temperature measured by the sunburn test is used as the initial temperature of the passenger cabin. The head of the vehicle is southward. The physical parameters and surface optical properties of the numerical model are shown in table 1.
Table 1. Related physical parameters and surface optical properties

| Automobile Components | Material | Density $\rho$ (kg/m$^3$) | Thermal conductivity $\lambda$ (W/m·°C) | Isobaric heat capacity $c_p$ (J/kg·k) | Absorptivity |
|-----------------------|----------|---------------------------|-----------------------------------|-----------------------------------|--------------|
| Car body              | steel    | 7785                       | 100                               | 448                               | 0.12         |
| Windshield            | glass    | 2529                       | 1.17                              | 754                               | 0.16         |
| Instrument panel      | ABS      | 996                        | 0.17                              | 1464                              | 0.9          |
| Seat                  | PU       | 1030                       | 0.25                              | 1700                              | 0.8          |

3. Effect of natural ventilation on cooling characteristics of passenger cabin

In order to simulate the effect of natural ventilation on the cooling characteristics of passenger cabin when a car is parking in summer, an appropriate computational domain is set up outside the vehicle model. The calculation domain is: the length of the front and rear ends of the car is 2 times of the length of the car, the width of the left and right sides of the car is 1.5 times of the width of the car, and the height of the roof is 2 times of the height of the car. The calculation domain and the inlet and outlet positions are shown in Fig. 2.

![Fig. 2. Natural ventilation calculation domain](image)

In order to simulate the effect of natural vent area on the cooling characteristics of passenger cabin, the car sunroof (710 mm * 340 mm) was used as the top vent. Under the same other conditions, the top vent was opened, 1/2 area opened, 1/3 area opened and 1/4 area opened. The light wind state of the external environment with wind speed of 2 m/s and wind direction of horizontal direction was simulated, and the external environment temperature was set at 35°C.

3.1 Analysis of cooling rate of natural ventilation

Through the simulation calculation, the temperature variation of six key monitoring points, including front head, front abdomen, front foot, rear head, rear abdomen and rear foot, can be obtained under four different working conditions in 20 minutes, as shown in fig.3.

![Figures showing temperature variation](image)
It can be seen from Fig. 3 that the cooling rate of the six key monitoring points is the fastest under the condition of full opening of the vent area, and the cooling curve shows a change rule of “first drop and then rise”. At the first 4 min after natural ventilation, the six observation points can be reduced to 44.4°C ~ 46.6°C. Six key monitoring points in the vent opening 1/4 area when the cooling rate is the slowest, especially in the rear foot measuring point. The temperature is 54.6°C in the 4 minute of cooling, and there is a temperature difference of 9.5°C compared with the full area condition of the vent. Each observation point will reach the thermal balance state in about 10 min for all of the four different conditions.

3.2 Effect of natural ventilation on cooling effect of passenger cabin

In order to more accurately observe the cooling effect of the cabin under natural ventilation conditions, the real-time temperature of each monitoring point in the cabin is selected after 10 min cooling under the full opening condition of the vent, and the 3D Map is drawn. The results are shown in Fig. 4.
It can be seen from Fig. 4 that at the same time, the temperature of the front head observation point is more than that of the rear head observation point, and the temperature of the front abdomen observation point is more than that of the rear abdomen observation point, and the temperature of the front row foot observation point is more than that of the rear foot observation point. Thus, under the condition of top natural ventilation, the air temperature in the front row of the cabin is high, while the air temperature in the rear row is low. The average temperature of each observation point after 10 min is taken as the thermal balance temperature, as shown in Table 2.

|                | Front head | Front abdomen | Front foot | Rear head | Rear abdomen | Rear foot |
|----------------|------------|---------------|------------|-----------|--------------|-----------|
| Full area      | 50.6       | 50            | 49.9       | 48.9      | 47.7         | 46.4      |
| 1/2 area       | 51.3       | 51.8          | 49.1       | 48.6      | 47.7         | 46.3      |
| 1/3 area       | 48.7       | 49.5          | 49.9       | 48.7      | 48.3         | 46.5      |
| 1/4 area       | 50.1       | 48.9          | 51         | 48.3      | 48.3         | 48        |

From Table 2, it can be seen that the difference of thermal balance temperature under four conditions is small, and the cooling effect is similar. When the cabin reaches steady state, the front head position temperature is still about 50°C, which is higher than the ambient temperature of 15°C, affecting the comfort of the occupants.

4 Effect of forced ventilation on cooling characteristics of passenger cabin

In this paper, a forced ventilation scheme at the top of the car is proposed. Due to the characteristics of large wind pressure and high efficiency of turbine fan, a turbine fan is arranged at the inlet. Two cross flow fans, which have the characteristics of large air volume, low noise and stable outlet air flow, are arranged at the outlet, as shown in Fig. 5.

The inlet radius is 150mm, while the outlet size is 310mm*55mm, and the outlet pressure is 14.7pa, and other conditions are unchanged. The effects of different inlet angles and different inlet speeds on the cooling characteristics of the cabin are studied.

4.1 Effects of Different Inlet Angles on Cooling Characteristics of Cabin

Setting the vehicle moving direction as X-axis positive direction and vertical downward as Y-axis positive direction, the angle of the air inlet angle is X-axis and the direction of the air inlet flow. The air inlet velocity is 3m/s, and other conditions remain unchanged. The influence curves of inlet angles of 30°, 60°, 90°, 120° and 150° on cabin temperature are simulated, as shown in Fig. 6.
It can be seen from Fig. 6 that under each ventilation angle condition, the six key monitoring points all cool significantly, and reach the thermal balance in about 5 min. Before reaching the thermal balance, when the inlet angle is 90°, the cooling rate of the key monitoring points in the front row is the largest. When the inlet angle is 120°, the cooling rate of the key monitoring points in the rear row is the largest, and the temperature variation range is large when the thermal balance of the foot of the front row is reached.

The thermal balance temperatures of the key monitoring points with different inlet angles are shown in table 3.

Table 3. Thermal balance temperature under different inlet angles (°C)

| Front head | Front abdomen | Front foot | Rear head | Rear abdomen | Rear foot |
|------------|---------------|------------|-----------|--------------|-----------|
|            | 30°           | 60°        | 90°       | 120°         | 150°      |
|            | 30°           | 60°        | 90°       | 120°         | 150°      |
|            | 30°           | 60°        | 90°       | 120°         | 150°      |
|            | 30°           | 60°        | 90°       | 120°         | 150°      |
|            | 30°           | 60°        | 90°       | 120°         | 150°      |
|            | 30°           | 60°        | 90°       | 120°         | 150°      |
It can be seen from Table 3 that the cooling effect of the front row is the best when the inlet angle is 90°, which can reach the heat balance at 37.9°C. The second best is the inlet angle of 60°, which can reach the heat balance at 38.6°C. The cooling effect of the rear row is the best when the inlet angle is 120°, and it can reach the heat balance at 37°C. Secondly, the heat balance can be achieved at 37.7°C when the inlet angle is 90°.

4.2 Effects of Different Inlet Velocity on Cooling Characteristics of Passenger Cabin
Simulation cabin temperature curve results of 1m/s, 2m/s, 3m/s, 4m/s, and 5m/s inlet velocity respectively when inlet angle is 90° are shown in Fig. 7.

Fig. 7. Influence curve of inlet velocity on cabin temperature
It can be seen from Fig. 7 that under different inlet velocity conditions, the six key monitoring points are obviously cooled, and the heat balance can be achieved in about 5 min. The cooling rate of each key monitoring point is the largest under the inlet velocity of 5 m/s.

Thermal balance temperatures of key monitoring points at different inlet velocities are shown in Table 4.

| Inlet Velocity | Front Head | Front Abdomen | Front Foot | Rear Head | Rear Abdomen | Rear Foot |
|---------------|------------|---------------|------------|-----------|--------------|-----------|
| 1m/s          | 39.3       | 38.8          | 39.8       | 38.9      | 39.2         | 38.3      |
| 2m/s          | 38.2       | 38.5          | 39.1       | 38.9      | 38.7         | 38.3      |
| 3m/s          | 38.2       | 37.9          | 39.6       | 38.2      | 37.9         | 37.8      |
| 4m/s          | 37.6       | 37.8          | 38.4       | 37.7      | 37.5         | 37.8      |
| 5m/s          | 37.4       | 37.5          | 38.2       | 37.5      | 37.5         | 37.7      |

It can be seen from Table 4 that the thermal balance temperature of each key measuring point is the lowest at the inlet speed of 5m/s, and the cooling effect is the best. The head temperature of the front row is 37.4℃, which is only 2.4℃ higher than the ambient temperature.

Based on the above analysis, the inlet velocity and the inlet angle were selected as 5m/s and 90°, and other conditions were set unchanged. The temperature change in the cabin within 10～30 min under the forced ventilation condition was simulated, with the results shown in Fig. 8.

It can be seen from Fig. 8 that at the same time, the temperature of the front head observation point < the temperature of the front abdomen observation point < the temperature of the front foot observation point; The temperature fluctuation of the observation point in the rear row is large, and the temperature of the observation point in the rear abdomen is the lowest compared with other observation points in the rear row. Compared to Fig. 4, it can be seen that the temperature fluctuation of each observation point under the top forced ventilation condition is significantly reduced, and the temperature values of each key monitoring point are significantly reduced. Therefore, it can be seen that the top forced ventilation condition significantly improves the cabin hot soak effect.

5. Conclusion

In order to solve the thermal comfort problem of car during summer parking, the CFD numerical method was used to study the cooling characteristics of passenger cabin with different inlet area, inlet velocity and inlet angle from two aspects of top natural ventilation and top forced ventilation. The conclusions are as follows:
1) Under the top natural ventilation condition, the opening area of the vent has little effect on the cooling effect of the cabin. No matter how large the opening area of the vent is, the heat balance can be achieved in the passenger cabin at about 10 min. The thermal balance temperature of front head position is about 50°C, which is still 15°C higher than the ambient temperature, which is not conducive to the comfort experience of drivers and passengers.

2) Under the top natural ventilation condition, the opening area of the vent has a great influence on the cooling rate of the cabin. Before the heat balance, the cooling rate of the full area of the vent is the fastest, and the cooling curve shows a change law of ‘first drop and then rise’. The cooling rate of the 1/4 area of the vent is the slowest, especially at the rear foot measuring point. The temperature is 54.62°C in the 4 minute of cooling. Compared with the full area condition of the vent, the temperature difference is 9.5°C.

3) The cooling effect of forced ventilation at the top of the cabin is obvious. Under the condition of 3m/s inlet wind speed, the cabin can reach the thermal balance in 5 min no matter how large the inlet angle is. The thermal balance temperature at the head is about 37.4°C, which is only 2.4°C higher than the ambient temperature, which can significantly improve the hot soak effect of the crew.

4) Under the top forced ventilation condition, the inlet angle has a great influence on the cooling effect and cooling efficiency of cabin. The cooling effect of the front row is the best when the inlet angle is 90°, and it can reach the heat balance at 37.9°C. The second is the inlet angle of 60°, which can reach the heat balance at 38.6°C. The cooling effect of the rear row is the best when the inlet angle is 120°, and it can reach the heat balance at 37°C. Secondly, when the inlet angle is 90°, the heat balance can be reached at 37.7°C.

5) The inlet velocity of top forced ventilation has great influence on the cooling rate of passenger cabin. When the inlet angle is 90°, the cooling rate is the largest and the cooling effect is the best when the inlet velocity is 5m/s.

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**Author contributions**

S-F.C. Proposed the scheme of ventilation and cooling at the top of cabin and Z-W.M. did numerical simulations of different schemes, Q-P.L. and F.H discussing the research content and checking and modifying the manuscript.

**Competing interests**

The author declare no competing interests.

**Additional information**

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