Natural Ventilation Performance Simulation and Analysis of New Type Sound Insulation Window

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Abstract: To evaluate the ventilation performance of new type sound insulation window, a window model with streamline-structure micro-perforated panel absorber (MPA) was simulated based on computational fluid dynamics (CFD). Finite element model of the insulation window was built to analyze the flow field in muffler channel, velocity in vent and pressure loss of the channel. To contrast with the simulation model, an insulation window artifact was produced and tested in ventilation quantity detection. The results show that the dominant pressure loss is local hydraulic loss that comes from MPA structure. The total pressure loss increased by 4.3 Pa under 0.5 m/s inlet wind velocity. The difference of pressure loss between simulation and actual test is 6.3%. CFD simulation is an effective and low-cost method to evaluate the ventilation performance with visual results. The results provide valuable reference for optimal design of insulation window.

1. Introduction

In recent years, transformer and traffic noise pollution are becoming more and more serious with the rapid development of substation. The buildings around substation are seriously affected by noise. Therefore, ventilation and sound insulation window has become the most commonly used noise reduction measures in noise-sensitive buildings. At present, sound insulation windows can be divided into two types: mechanical ventilation and natural ventilation. For example, F. Asdrubali [1] install sound absorbing materials such as asbestos fibers and the anechoic box composed of fan, which can realize the sound insulation capacity of 33 to 38 dB and the ventilation capacity of 90 m²/h. ZhaiGuoqing [2] designed a mechanical ventilation and sound insulation window with double-layer hollow glass, which has the characteristics of sound insulation, ventilation and lighting. Yu Wuzhou [3] and others designed a new type of natural ventilation and sound insulation window with micro-perforated panels. The sound insulation capacity within 400-4000Hz is higher than 25dB, and the ventilation capacity of a single unit is higher than 28m³/h. However, due to the structural limitations of the micro perforated structure, the wind resistance is still relatively high. Ventilation louver structure is widely used in the design of ventilation vent because of its small flow resistance and certain role of air guide plate [4]. This article combines the micro-perforated plate with the streamlined louver structure to obtain the sound insulation window structure with good natural ventilation effect.

The natural ventilation effect is easily affected by the change of meteorological conditions [5]. Different microperforated panel anechoic units installed in the ventilation channel of sound insulation window will cause a certain ventilation pressure loss. At present, field measurement methods are often used to evaluate the ventilation performance of sound insulation windows, such as simulating the...
natural ventilation effect by using the pressure difference between the inside and outside of the room formed by the "blast door"[6], and measuring the room ventilation volume by using the tracer gas method [7]. Field measurements can evaluate the actual ventilation effect of sound insulation windows, but the experimental cost is high, and the experiment is limited by the weather conditions. In this paper, the CFD method based on finite element analysis is used to simulate the flow field in the anechoic ventilation passage of a new type of sound-proof window. The pressure loss and the velocity distribution in the anechoic passage are calculated, which provides a certain reference value for the evaluation of the ventilation performance of the sound-proof window and the optimization design of the anechoic passage.

2. Establishment of flow field model

The new ventilation and sound insulation window studied in this paper has the advantages of noise reduction, ventilation and lighting. When the outer window is opened, the noise reduction is 28dB, and when the outer window is completely closed, the noise reduction is 33dB. Its structure is shown in Figure 1, consisting of a double deck upper suspension window and a single bottom suspension window. The distance between the two windows of the hanging window is about 10 cm. Streamline micro-perforated anechoic unit is installed in the middle to form the anechoic ventilation channel. When the main window is closed, the outer window of the hanging window is opened, and the outdoor air enters the silencing passage through the outer window. The air flows through the micro perforated panel area and enters the room from the vent.

![Fig.1 Structure of ventilation sound insulation window](image)

2.1 geometric model

The size of the object's form is 1500mm×1200mm, and the thickness is 10mm. Therefore, removing the volume occupied by aluminum alloy window frames, the actual size of the noise elimination channel is 1100mm×400mm×100mm. Streamlined anechoic unit is composed of four micro-perforated plates bended into ellipse, which are evenly distributed in the anechoic ventilation passage at 45 degrees inclination. The thickness of the micro perforated plate is 1mm, the perforation diameter is 1mm, and the perforation rate is 1%. The geometric model of the simulation object can be simplified as the cavity in the anechoic channel. The entrance of the silencing passageway takes up the condition of the opening of the suspension window at 45 degrees [8][9]. In this study, the channel models with streamlined anechoic units and without anechoic units are established respectively.
Fig. 2 Geometrical model of 2 structures

In order to obtain the velocity distribution inside the flow field, four velocity distribution sampling lines (Line 1-4) are created in the model. Each line takes 10 to 20 sampling points to calculate the velocity vector at this point. The completed geometric model is shown in Figure 3.

Fig. 3 Geometrical model of muffle channel

2.2 control equation and calculation model

Air in anechoic duct can be regarded as incompressible viscous fluid, and its flow needs to satisfy the law of conservation of mass, momentum and energy. The basic control equation in the flow field is the mathematical description of the above three laws.

Mass Conservation Equation: The increment of mass in a fluid unit in a unit time is equal to the net mass flowing into the unit from outside at the same time. The mathematical expression is as follows:

\[ \frac{\partial \rho}{\partial t} + \frac{\partial (\rho u)}{\partial x} + \frac{\partial (\rho v)}{\partial y} + \frac{\partial (\rho w)}{\partial z} = 0 \]  

\( \rho \)—density; \( t \)—time; \( u, v, w \)—The component of velocity vector in X, y and Z directions.

Momentum Conservation Equation: N-S Equation, the rate of change of momentum in one direction per volume of fluid to time equals the sum of the net inflow rate of momentum in that direction and the external force acting on that direction.

\[ \frac{\partial (\rho u)}{\partial t} + \text{div} (\rho u U) = -\frac{\partial p}{\partial x} + \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{xy}}{\partial y} + \frac{\partial \tau_{xz}}{\partial z} + F_x \]  

\[ \frac{\partial (\rho v)}{\partial t} + \text{div} (\rho v U) = -\frac{\partial p}{\partial y} + \frac{\partial \tau_{yx}}{\partial x} + \frac{\partial \tau_{yy}}{\partial y} + \frac{\partial \tau_{yz}}{\partial z} + F_y \]
\[\frac{\partial (\rho_w)}{\partial t} + \text{div} (\rho_w U) = \frac{\partial p}{\partial z} + \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + \frac{\partial \tau_{xz}}{\partial z} + F_x \tag{4}\]

\(\text{div}\) — Vector symbol, \(\text{div} = \frac{\partial \phi}{\partial x} + \frac{\partial \phi}{\partial y} + \frac{\partial \phi}{\partial z}\); \(U\) — velocity vector; \(P\) — Pressure on fluid unit; \(F_x, F_y, F_z\) — Forces acting on fluid units; \(\tau_{xx}, \tau_{yz}, \tau_{xz}\) — The component of viscous stress in different directions.

Energy Conservation Equation: First Law of Thermodynamics, the rate of increase of energy in a fluid unit is equal to the sum of the net heat flux entering the unit and the work done by the external force on the unit.

\[\frac{\partial (\rho T)}{\partial t} + \text{div} (\rho U T) = \text{div} \left( \frac{k}{C_p} \text{grad} T \right) + S_T \tag{5}\]

\(T\) — temperature; \(C_p\) — Specific heat capacity; \(\text{grad}\) — gradient, \(\text{grad} T = \frac{\partial T}{\partial x} + \frac{\partial T}{\partial y} + \frac{\partial T}{\partial z}\); \(k\) — Fluid heat transfer coefficient; \(S_T\) — The internal heat source of the fluid and the part of the mechanical energy converted to heat by viscous action.

2.3 boundary condition handling

In natural ventilation, the velocity of air flow at the inlet of the muffler passage depends on the meteorological conditions inside and outside the room. According to the survey, the average annual wind speed in Shanghai urban area is 2.5 m/s to 3.0 m/s [10]. In order to obtain the inlet boundary velocity of sound insulation window ventilation, a three-dimensional flow field model of a building is established under this environment, and the pressure and velocity distributions near the building surface are calculated. According to the relevant research [11], assuming that the building occupies an area of 20 m × 10 m and is 20 m high, the calculation area of the flow field simulation is determined as follows: the upstream area of the building is four times the width of the building, the downstream area is 12 times the width of the building, and the width is six times; the height is four times the height of the building, and the space size is 170 m × 120 m × 80 m. Finally, the comprehensive flow velocity near the windward surface of the building is calculated to be 0.5 m/s, and the air flow velocity at the entrance of the anechoic passage of the ventilation and sound insulation window is calculated.

3. Analysis of calculation results

3.1 flow field analysis

Fig. 4 Velocity distributions on Line 1, 2 and 4

Figure. 4 is the velocity distribution of the sound insulation window model on different sampling lines. Line 1 and Line 2 represents the velocity distribution in the Y direction of the left cavity of the anechoic channel and the left connecting the right channel; Line 4 represents the velocity distribution.
in the X direction at the outlet of the ventilation channel, and the average and maximum wind speeds are shown in Table 1. From the data of Line 1 and Line 2, it can be seen that the velocity near the wall is close to 0 m/s. In the middle passage, the window frame leads to the obvious acceleration of the fluid, and the average velocity is about 3 m/s. The velocity distribution in the vertical direction of the silencing unit is more uniform; the air velocity in the upper part of the window is higher, and the difference between the air velocity in the left cavity and the middle passage is 2 m/s. According to the data of Line 4, the average velocity increases to about 4m/s because the area of ventilation outlet is only 0.015m², which is much smaller than the average cross-section area of the anechoic channel.

| Model | Sample Line | Average Velocity (m/s) | Max Velocity (m/s) |
|-------|-------------|------------------------|-------------------|
| Case.1| Line 1      | 0.81                   | 1.31              |
|       | Line 2      | 2.87                   | 3.77              |
|       | Line 4      | 3.81                   | 5.44              |
|       | Line 1      | 0.80                   | 1.15              |
| Case.2| Line 2      | 3.06                   | 3.62              |
|       | Line 4      | 4.09                   | 5.23              |

Figure 5 shows the velocity distribution on the slanting line (Line3) in the right side of the anechoic channel. The difference of velocity distribution before and after adding silencing unit is obvious. When there is no anechoic unit, the air flow basically conforms to the characteristics of the air under the wall restriction, and the velocity is the greatest in the middle part far from the wall. Under the influence of the outlet, the wind velocity in the vertical part is higher than that in the upper part. The addition of the anechoic unit partitions the flow of air, reduces the flow area and the anechoic unit, so there is a higher flow distribution anechoic unit between the anechoic units.
Fig. 6 Velocity components on Line1, Line2 and Line3

Figure 6 shows the distribution of \( U, U_x, U_y \) and \( U_z \) in the three directions of X, Y and Z on Line1, 2, 3. It can be seen that the flow in the X (horizontal) direction is the dominant air flow direction in the anechoic duct and the main contributing factor of the air flow. On the left side, due to the influence of the half-open window, there is also a strong upward flow in the Y (vertical) direction, while there is also a Z (lateral) direction flow, indicating that there is a large number of swirling air on the left side. In the middle passage, the flow in the direction of Y and Z decreases and the swirling flow decreases. After entering the right channel, the air flow in the vertical direction is obviously divergent under the action of the diversion of the streamline anechoic unit structure. There is a large lateral swirling flow between the two muffler units, while the other parts have little lateral flow.

3.2 Pressure field analysis

Fig. 7 Total pressure distribution contour on XY (Z=50mm) and ZX (Y=200mm) plane
Figure 7 is the total pressure distribution on the XY plane (Z=50mm) and the ZX plane (Y=200mm, the middle section of the channel). The pressure loss is mainly caused by the airflow obstructing the lower passage in the window frame, the narrowing and expansion of the outlet and the flow direction being restricted by the window. After adding silencers, the pressure loss is mainly due to the obstruction of the muffler unit to the airflow in the X direction. Between the two anechoic units, the loss along the flow line is small because of the larger fluid space. Because there is almost no flow in the muffler unit, the pressure is smaller than other areas. The total pressure at the inlet of the anechoic unit is higher than that at the inlet of the non-anechoic unit because of the obstruction of the flow by the anechoic unit. The pressure loss of the anechoic passage is calculated by the mean total pressure at the inlet and the mean total pressure at the outlet, as shown in Table 2. After adding the silencer unit, the pressure loss is higher than that of the anechoic unit.

Tab.2 Pressure loss of different models (under 0.5m/s inlet wind velocity)

| Model  | Average Total Pressure of Inlet (Pa) | Average Total Pressure of Outlet (Pa) | Pressure Loss (Pa) |
|--------|--------------------------------------|---------------------------------------|-------------------|
| Case1  | 30.03                                | 14.98                                 | 15.05             |
| Case2  | 22.76                                | 11.81                                 | 10.95             |

4. Conclusion

A new flow-line microperforated panel anechoic channel model of ventilation and sound insulation windows was established by using finite element method. The airflow in the anechoic channel under natural ventilation was simulated numerically, and the velocity and velocity in the channel were obtained. The pressure data and distribution map accurately and intuitively reflect the flow field distribution under the influence of sound absorption structure. The local pressure loss of the anechoic channel is increased by the setting of the microperforated panel anechoic unit, and the total pressure loss is higher than that of the non-anechoic unit structure by 4.1Pa at the inlet wind speed of 0.5m/s. In summary, the CFD numerical simulation method can effectively evaluate the ventilation performance of sound insulation window, and provide valuable reference for the optimization design of sound insulation window silencing channel.

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