Simulation Study on Noise Reduction of Industrial Gas Turbine Silencing Pipeline

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Abstract. The sound field of industrial gas turbine silencing pipeline is simulated and calculated by Virtual Lab, and the acoustic simulation results of the intake pipe show that the performance of the pipes with mufflers decreases slightly, the pressure loss is eligible while the flow uniformity is unqualified. The acoustic properties improve greatly, but the transmission loss in 20~60Hz and 160~460Hz is lower than the standard.

1. Introduction
During the working process of a chemical gas turbine, the changes in the air density and pressure inside the gas pipeline caused by the flow contraction and expansion, and produced pressure waves[1]. In the course of the propagation of the flow waves, it will cause obvious noise pollution, which will affect the physical and mental health of workers and reduce the production efficiency. Under different working conditions, the intensity and distribution of noise and the flow field distribution of gas transmission pipeline will vary according to the working environment. Therefore, the research on the flow field and acoustic performance of the gas turbine intake pipe is of great significance to understand the distribution of the flow field and noise of the intake pipe and to take the measures to reduce the noise.

The material that can absorb and dissipate the energy absorbed on its surface and transform it into other forms of energy, such as heat, is called acoustic absorption material[2]. The principle of the work is: it uses complex pore structure in material and air viscosity resistance, and converts the incoming sound energy into other forms of energy. Figure 1 shows the sound absorption and noise reduction process: The sound wave 1 is incident on the surface of the porous sound absorbing material; Some of the acoustic waves (acoustic wave 2) are reflected back to the air from the surface of the sound absorbing material; another part of the acoustic wave (acoustic wave 3) penetrated into the interior of the material, and a part of the acoustic wave 3 is absorbed by the absorption material, while the other part of the acoustic wave 3 is not absorbed, but the sound energy is further attenuated in the process of the rigid wall of the sound absorbing plate reflected back to the surface of the sound absorbing material. After the repeated reflection and propagation of the incident acoustic wave 1, the attenuation of the sound energy is larger, and only a small amount of sound waves can be re propagated into the air [3].
2. Modeling of Sound Dissipation Pipe

Sound absorption and noise reduction methods are adopted to improve the noise elimination of intake pipes. The 4 wall surfaces are enveloped by a porous plate, and the resistive muffler filled with porous sound absorbing material is installed in the straight pipe section near the outlet; Among them, the material of sound absorbing material is glass fiber cotton, and it is a thermal insulation, light weight, durable, long life and sound quality material [4]. Because the structure size of the muffler and the intake pipe is large; the size of the porous plate small hole is small; the mesh of the finite element is divided according to the real model, which will lead to the large number of grid and the poor quality of the grid. Therefore, this paper simplifies the porous plate structure, that is, no pore structure model is established. The 3D model of muffler intake pipe at assembly site is shown in Figure 2.

3. Acoustic analysis

3.1 Transfer Function

In general, the size and quantity of the holes in the porous plate are small and large. If the finite element mesh is divided according to the real model, the number of grids and the poor quality of the grid will lead to the large amount of computation and the high difficulty in computing. In order to solve the above problems, the acoustic characteristics of the small holes can be simulated by adding and defining the transfer admittance relationship on the meshes on both sides of the porous tube. Figure 3 shows a small hole size for a perforated plate.
When the sound wave passes through the perforated plate, its impedance calculation formula is:

$$Z_p = \frac{\Delta P}{V} = R_p + jX_p$$  \hspace{1cm} (1)$$

For $\Delta P$ is sound pressure difference before and after the sound wave passes through a porous plate; $V$ is vibration volume velocity of a particle in a small hole. When the diameter of the two times small hole is much larger than the thickness of the perforated plate ($l \ll 4a$), the real part of impedance

$$R_p = \frac{1}{\varepsilon} \sqrt{8\omega \eta \rho \left(1 + \frac{l}{2a}\right)}$$  \hspace{1cm} (2)$$

$$X_p = \frac{1}{\varepsilon} \cdot \omega \cdot \rho \cdot (l + 2\Delta l)$$  \hspace{1cm} (3)$$

For $\varepsilon$ is porosity of a porous plate; $\omega$ is frequency, $\omega = 2\pi \cdot f$; $a$ is distance between the core of two adjacent small holes; $\eta$ is dynamic viscosity of a fluid; $\rho$ is density of fluid; $\Delta l$ is the correction term which is related to the radius and arrangement of the holes in the perforated plate and the distance between the holes.

In practical engineering applications, the holes in the porous plates are generally arranged in two ways: square arrangement and regular hexagonal arrangement [5]. The square arrangement, as shown in Figure 4 (a), is shown in the arrangement of regular hexagons as shown in Figure 4 (b). Among them, figure 4 (a) small hole perforation rate formula is:

$$\varepsilon = \frac{\pi a^2}{d^2}$$  \hspace{1cm} (4)$$

The formula for the correction term is as follows:

$$\Delta l = \begin{cases} 0.86a \left(1 - 2.34 \frac{a}{d}\right) & 0 < \frac{a}{d} \leq 0.25 \\ 0.68a \left(1 - 1.9 \frac{a}{d}\right) & 0.25 < \frac{a}{d} \leq 0.5 \end{cases}$$  \hspace{1cm} (5)$$
Among them, figure 4 (b) small hole perforation rate formula is:

$$\varepsilon = \frac{6\pi a^2}{5\sqrt{3}d^2}$$  \hspace{1cm} (6)

The formula for the correction term is as follows:

$$\Delta l = \begin{cases} 
0.85a \left(1 - 2.52 \frac{a}{d}\right) & 0 < \frac{a}{d} \leq 0.25 \\
0.68a \left(1 - 2.0 \frac{a}{d}\right) & 0.25 < \frac{a}{d} \leq 0.5
\end{cases}$$  \hspace{1cm} (7)

The linear relationship between the vibration velocity and the sound pressure on the two sides of the porous plate can be created with the transfer admittance relation. The expression of transfer admittance relation is:

$$[v_{n1}] = \begin{bmatrix} a_1 & a_2 \\ a_3 & a_6 \end{bmatrix} [p_1] + [a_4]$$  \hspace{1cm} (8)

For $v_{n1}$ and $v_{n2}$ are normal vibration velocity on both sides of a porous plate; $p_1$ and $p_2$ are sound pressure on both sides of a porous plate; $a_1$, $a_2$, $a_3$, $a_4$, $a_5$ and $a_6$ are transfer admittance coefficient; $a_3$ and $a_6$ depend on the sound source coefficient, in the acoustic calculation of mufflers, $a_3 = a_6 = 0$. Normally, $a_1$, $a_2$, $a_3$, $a_4$ and $a_5$ are determined through physical experiments, if the holes on the perforated plates are arranged in the square of Figure 4 (a) or hexagons in Fig. 4 (b). The transfer admittance of perforated plates in LMS Virtual Lab can be expressed by defining the following matrix:

$$\begin{bmatrix} \beta & -\beta \\ -K\beta & K\beta \end{bmatrix}$$  \hspace{1cm} (9)

3.2 Acoustic Finite Element Modeling

In this paper, the acoustical finite element model of the anechoic tube, as shown in Figure 5, is built by ICEM, which is enveloped by a porous plate and filled with sound absorbing materials. The blue part is air finite element mesh, which is given air properties, and the green is a sound absorbing material finite element mesh, which is endowed with the properties of sound absorbing materials. The grid type is a linear unstructured mesh with a maximum size of 50mm.

![Figure 5. Acoustic Finite Element Model for Acoustic Pipeline](image)

The arrangement of the holes in the perforated plates is square arrangement. According to the calculation formula of the perforation rate in the transfer admittance:

$$\varepsilon = \frac{\pi a^2}{d^2} = 0.28$$  \hspace{1cm} (10)
And the Determinant condition is \( \frac{a}{d} = \frac{1.5}{5} = 0.3 \), so amendment is: \( \Delta l = 0.6 \Delta a \left( 1 - 1.9 \Delta d \right) = 0.439 \). The transfer admittance parameters of porous metal plates are shown in Table 1.

| Attribute                              | Parameter |
|----------------------------------------|-----------|
| Thickness of Porous Plate Wall(mm)     | 1         |
| Hole Cores Distance(mm)                | 5         |
| Radius of Circular Hole(mm)            | 1.5       |
| Fluid Density \( (kg/m^3) \)           | 1.225     |
| Kinetic Viscosity \( (Pa\cdot s) \)     | \( 1.79 \times 10^{-5} \) |
| Inner Diameter of Circular Tube(mm)    | 890       |
| Outer Diameter of Circular Tube(mm)    | 900       |
| Perforation Rate                        | 0.283     |
| Determinant Condition                   | 0.3       |
| Amendment \( \Delta l \)               | 0.439     |
| Coefficient                             | 0.989     |

According to the calculation formula, the real and imaginary parts of the impedance under different frequencies are calculated for different acoustic frequencies. For the convenience of calculation, the real and imaginary values of the impedance in each frequency range (20~1000Hz, step length 20Hz) are obtained.

3.3 Acoustic Simulation Results

Through calculation, the transmission loss curve of the muffler tube is obtained, which is compared with the transmission loss of the intake pipe into the Origin to draw the transmission loss of the two, as shown in Figure 6.

![Figure 6. Loss Comparison Between Intake Pipe and Anechoic Tube](image)

Figure 6 shows that, on the whole, the muffling amount of the pipeline assembled with the muffler is obviously higher than the transmission loss of the intake pipe, but the noise reduction of 20~60Hz and 160~460Hz is less than 50dB, which is not satisfied with the requirements of the construction site. The transmission loss value of the frequency band needs to be improved; the transmission loss curve of the 460~1000Hz is much fluctuant; the assembly silencer is assembled. There are several noise peaks and valleys around the pipes, especially in 20~400Hz, where an anechoic Valley and a peak of noise elimination are interchanged.
4. Conclusion
Virtual Lab is used to simulate the internal sound field of the noise reduction pipe, and the transmission loss is calculated. The results of the simulation are compared with the acoustic simulation results of the intake pipe. The results show that the noise reduction of the pipe is greatly improved after the installation of the muffler, but the noise reduction of 20–60Hz and 160–460Hz is lower than that in the field. The transmission loss value of the frequency band needs to be improved.

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