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ABSTRACT

The twin screw compressor has been widely used in the refrigeration systems due to advantages such as compact structure, stable operation, high efficiency and good adaptability. Intermittent gas flow generates gas pulsation that cause serious problems such as structural vibration and noise in the twin screw refrigeration compressor. Because the mechanical noise can be controlled well with the improvement of machining and assembly accuracy, the aerodynamic noise induced by gas pulsation even has become the main noise source of the twin screw refrigeration compressor. In order to reduce the pressure pulsation, a broadband perforated panel muffler applied in the discharge chamber of the twin screw refrigeration compressor is proposed based on the noise spectrum and flow characteristics of the compressor. In order to obtain the noise spectrum of the twin screw refrigeration compressor, the pressure fluctuation in discharge chamber based on a three-dimensional CFD simulation model is calculated, and the acoustical model is established based on fluid-acoustic coupling method. Then the impacts of different structural parameters on the performance of a perforated panel muffler are investigated, including perforation rate, perforation diameter and panel thickness. Through the optimization of the perforated muffler, a better reduction effect of broadband noise can be achieved. Results of fluid-acoustic coupled analysis can provide guidance on the design and optimization of the perforated muffler and noise reduction of the twin screw refrigeration compressor.

1. INTRODUCTION

Twin-screw compressors have the advantage of a simple structure, less easily damaged parts, excellent running stability, and strong adaptability, and have been widely used in refrigeration and air conditioning fields. Intermittent gas flow generates gas pulsation that cause serious problems such as structural vibration and noise in the twin screw refrigeration compressor. Because the mechanical noise can be controlled well with the improvement of machining and assembly accuracy, the aerodynamic noise induced by gas pulsation even has become the main noise source of the twin screw refrigeration compressor. Fujiwara *et al*. (1986) conducted the first comprehensive measurement of the vibration and noise of a screw compressor. Holmes (2003) analyzed the noise of screw compressors and proposed a method to solve mechanical noise. Mujic *et al*. (2011) conducted a comprehensive analysis of the noise of the screw compressor, and concluded that the gas pulsation is the main cause of screw compressor noise. Therefore, in order to reduce the vibration noise of refrigeration screw compressor, it is necessary to first analyze and attenuate the gas pulsation during operation. J Hauser (2013) predicted the pressure pulsation in the discharge port of a screw compressor using Computational Fluid Dynamics model, which shows great accuracy compared with experimental investigations. Wu *et al*. (2017) constructs a simplified dynamic model of a screw rotor system that uses multibody dynamics to predict the compressor's dynamic response and to determine the reasons for structural vibration. The stimulated vibration frequencies of the proposed model fit well with those measured vibration frequencies of the experimental model. K Matsuo (2013) studied noise and vibration reduction in medium-size oil-flooded twin-screw compressor using the helix modification. Noise reduction by 3dB was achieved under full load conditions. Applications of muffles are effective ways of reducing gas pulsation and absorbing aerodynamic noise of compressors. Kim *et al*. (2016) proposed a suction muffler applied in hermetic reciprocating compressor, which improved both flow performance and acoustic performance. Oh *et al*. (2016a) studied the topology optimization of a suction muffler in a fluid machine to maximize energy efficiency and minimize broadband noise. Oh *et al*. (2016b) also applied a suction muffler on a linear compressor to study the variation of acoustical resonance frequency of duct with orifice depending on periodic motion. However, little literature about the applications and optimization of the perforated panel muffler in twin-screw compressor is available. A fluid-acoustic coupling model of the screw compressor is established in this paper, which provides guidance on the applications and optimization of perforated panel muffler in twin screw compressor to reduce the aerodynamic noise.

2. INTRODUCTION OF FLUID-ACOUSTIC COUPLED MODEL

In order to obtain the aerodynamic noise spectrum of the screw refrigeration compressor, a fluid-acoustic coupling model is established. The pressure pulsation of the discharge section is obtained by calculating the unsteady flow of the screw compressor, which is coupled with the acoustic model of discharge section to calculate the aerodynamic noise of the screw compressor during operation.

2.1 CFD Model

The CFD simulation in this paper is based on the following assumptions:

(1) The influence of lubricating oil on the refrigerant is not considered

(2) The discharge process is adiabatic, and the heat exchange between gas and wall, wall and ambient is not considered.

Due to the complex geometry of the compressor's working chamber, the internal fluid domain and the clearance changes continuously with the rotation of the rotor, which made the requirements of the grid extremely demanding. The generated grids of rotors fluid domain are shown in Figure 1.

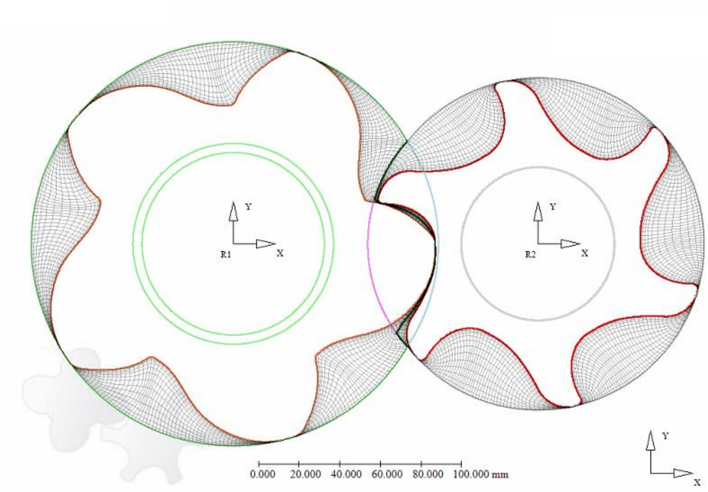


Figure 1: Two-dimensional mesh of rotors fluid domain

In order to ensure the accuracy of simulation of discharge process flow, the corresponding discharge flow channel model is established according to the actual compressor discharge end-seat structure. The simplified suction and discharge basin grid are generated by Ansys Meshing as shown in Figure 2.

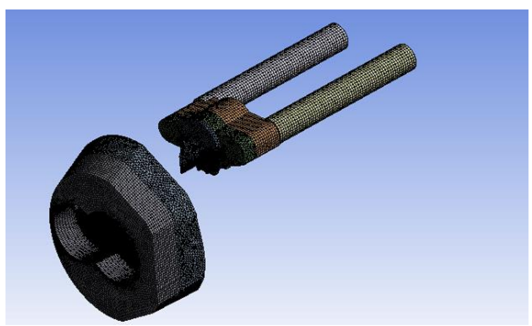


Figure 2: Suction and discharge port fluid domain grid

The refrigerant is R134a, and its physical properties are given by the physical property data fitted by the Peng–Robinson equation in the database. As shown in Table 2, the inlet and outlet temperature and pressure are used as the boundary conditions for the CFD simulation.

Table 2: Boundary conditions for CFD simulation

Parameters	Values
Rotation speed of the male rotor, rpm	3000
Suction pressure, bar	3.50
Suction temperature, K	288.15
Discharge pressure, bar	9.63

2.2 Acoustic Model

The numerical model of acoustic field is a basic control equation which establishes the relationship between sound pressure, density variation and particle velocity, and then solves other acoustic parameters.

$$\begin{cases} \rho_0 \dfrac{\partial v}{\partial t} = -\dfrac{\partial p}{\partial x} \\ \rho_0 \dfrac{\partial v}{\partial x} = -\dfrac{\partial \rho'}{\partial t} \\ p = c_0^2 \rho' \end{cases} \tag{1}$$

where ρ_0 is the static density, ρ' is the density increment, p is the sound pressure, v is the particle velocity, x is the particle displacement and c_0 is the speed of sound.

Assuming that the refrigerant is an ideal fluid, the three-dimensional wave Equation (2) of small amplitude sound wave in a homogeneous ideal fluid medium can be derived from the basic governing Equation (1) of acoustic field under the assumption of static, uniform and no loss of the medium, thus the finite element Equation (3) can be obtained:

$$\nabla^2 p = \frac{1}{c_0^{\ 2}} \frac{\partial^2 p}{\partial t^2} \tag{2}$$

$$\left[K + \omega C - \omega^2 M\right] p = F_A \tag{3}$$

where K is the stiffness matrix, C is the damping matrix, M is the mass matrix, and F_A is the normal velocity.

The transmission loss (*TL*) of the muffler can be calculated by Eq. (4) based on the sound pressure nephogram of sound field inside muffler obtained by Equation (3).

$$TL = 10\lg(\frac{W_{in}}{W_{out}}) = 10\lg(\frac{p_{in}{}^2 S_{in}}{p_{out}{}^2 S_{out}}) \tag{4}$$

where W_{in} and W_{out} are input and output sound power of the muffler, p_{in} and p_{out} are the incident and transmitted sound pressure of the muffler, and S_{in} and S_{out} are the area of the inlet and outlet of the muffler.

Generally, it is assumed that there are 6 acoustic units in the minimum wavelength of sound wave, that is, the side length of the largest element should be less than 1/6 of the shortest wavelength, as the expression shown:

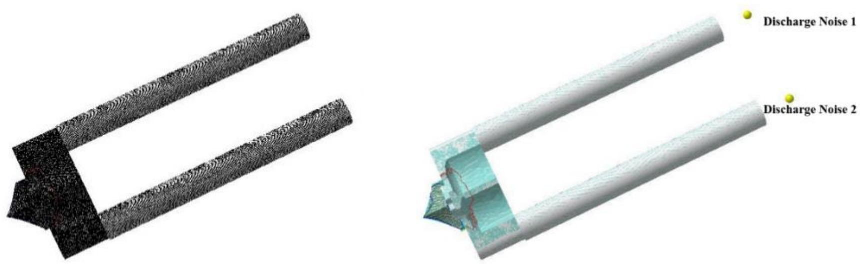
$$L \le \frac{c_0}{6 f_{\max}} \tag{5}$$

where c_0 is the propagation velocity of sound wave in fluid medium and L is the linear element length.

In order to improve the simulation accuracy and quality of the grid, the maximum side length of the whole grid cell can be controlled below 1/12 of the shortest wavelength. The surface boundary of the acoustic model is set as a rigid wall, and the boundary conditions of constant velocity excitation at the entrance and nonreflection at the exit are used for the acoustic finite element calculation.

2.3 Fluid-Acoustic Coupled Model

In order to further study the internal relationship between air flow pulsation and aerodynamic noise and mechanism of aerodynamic noise induced by air flow pulsation, the unsteady CFD results are mapped to the acoustic field model. The fluid-acoustic coupling model of the screw compressor discharge section is established, as shown in Figure 3. There are two noise monitors at a distance of 100mm from the discharge pipe outlets.



(a) Unsteady CFD results (b) Fluid-acoustic coupling model

Figure 3: Fluid-acoustic coupling model of the screw compressor discharge section

3. RESULTS OF FLUID-ACOUSTIC COUPLED SIMULATION

3.1 Results of CFD Simulation

Figure 4 shows the pressure distribution and temperature distribution of the suction and discharge port fluid domain and the surface of the rotor when the discharge ports are connected, which are obtained by CFD simulation. The rotor surface pressure and temperature gradually increase from the suction end to the discharge end.

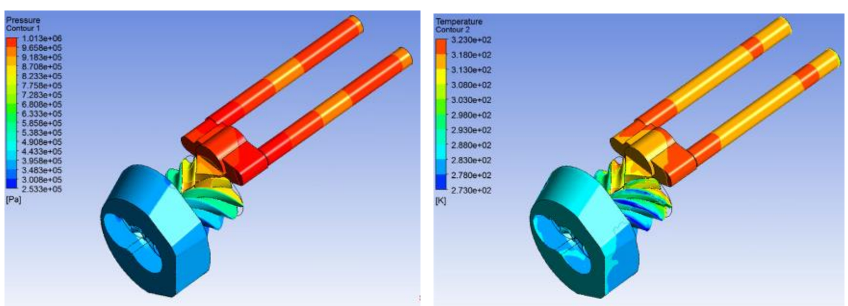


Figure 4: Pressure and temperature distribution when the discharge port is connected

Figure 5 shows the pressure pulsation monitored by the monitoring point in the working chamber of the compressor. From the low-pressure suction section to the high-pressure discharge section, the pulsation energy increases sharply, especially in the final compression chamber. The data shows that the final compression pressure when the radial discharge port is connected is about 1153kPa, which is much higher than the 963kPa set as the discharge pressure. After the axial discharge port opens, the compression pressure continues to decrease to 767kPa. There is an intense pressure pulsation in the compression chamber of the discharge section, while the pressure pulsation is much lower in the discharge pipe due to the large volume which provides buffer and attenuation functions.

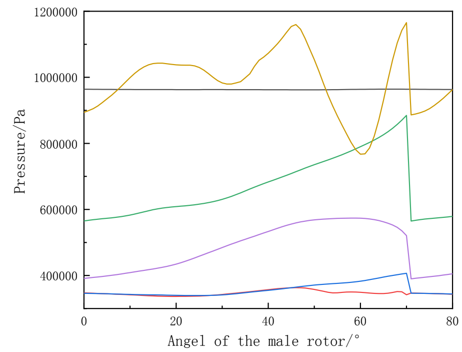


Figure 5: Pressure pulsation in the working chamber

3.2 Noise Characteristics of the compressor

Figure 6 shows the calculation results of two monitoring points of discharge aerodynamic noise of screw refrigeration compressor. The discharge aerodynamic noise of the screw refrigeration compressor has significant harmonic characteristics. The noise values at the fundamental frequency, the second frequency and the third frequency of the engagement frequency of the male and female rotors are particularly significant, which need to be improved. Therefore, the perforated panel muffler is designed to reduce the discharge noise aimed at the noise of the fundamental frequency, double frequency and triple frequency of the screw compressor.

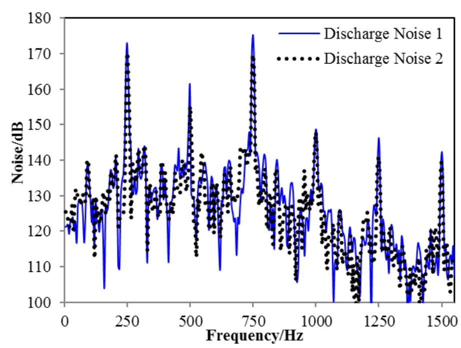


Figure 6: Discharge noise spectrum of screw compressor at the speed of 3000rpm

4. DESIGN AND OPTIMIZATION OF PERFORATED MUFFLER

In order to reduce the noise of screw refrigeration compressor, a perforated panel muffler suitable for screw refrigeration compressor is proposed. Its typical structure is shown in Figure 7. The perforated panel muffler is mainly composed of the perforated panel and the cavity behind it. l is the panel thickness, a is the perforation radius, d is the distance between the centers of the neighboring holes, and D is the depth of the cavity behind the perforated panel. There are many factors that affect the performance of the perforated muffler, such as the perforation rate, perforation diameter, panel thickness and resonator volume. In the application of screw compressor noise reduction, the volume of perforated panel muffler is not easy to change and the depth of cavity is limited due to the compact structure of compressor, so the perforation rate, perforation diameter and panel thickness are particularly important for the optimization of acoustic performance of perforated panel muffler.

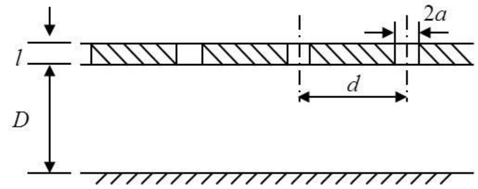


Figure 7: Typical structure of a perforated panel muffler

4.1 Analysis of Structural Characteristic

4.1.1 Perforation diameters: The transmission loss of the perforated panel muffler with different perforation diameters is shown in Figure 8, when cavity depth of the perforated panel is 16.65mm, the number of perforations is 36, and the thickness of the perforated panel is 2mm. As the perforation diameter decreases, the resonance peak of the perforated tube muffler shifts to low frequency, and the sound absorption effect of the resonance peak increases first and then decreases. With the decrease of the perforation diameter, the damping and friction effect of the perforation wall on the acoustic wave is enhanced, the acoustic energy of the acoustic wave is attenuated, the acoustic energy of the acoustic wave is reduced, and the sound absorption performance of the muffler is improved. When the perforation diameter continues to decrease, the number of acoustic waves passing through the perforation diameter is reduced, resulting in the decrease of the acoustic wave energy of the attenuation, and the noise reduction effect is reduced accordingly. As a result, the resonant frequency of the perforated panel muffler can be reduced by reducing the perforation diameter properly, and the sound absorption performance of the perforated panel muffler can be improved at the same time.

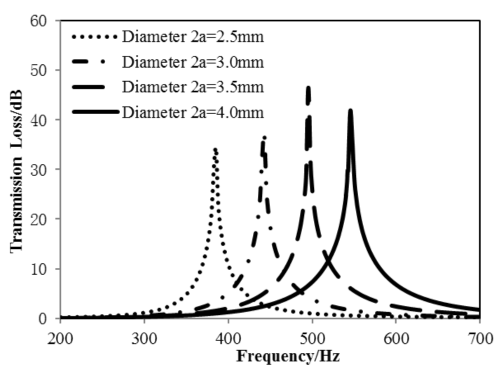


Figure 8: Acoustic absorption performance with different perforation diameters

4.1.2 Perforation rate: The acoustic absorption performance of the perforated panel muffler with different perforation rate is shown in Figure 9, when cavity depth of the perforated panel is 16.65mm, the perforation diameter is 3.5mm, and the thickness of the perforated panel is 2mm. With the increase of perforation rate, the resonance peak of the perforated panel muffler shifts to high frequency, and the absorption performance of perforated plate muffler increases first and then decreases. Although the damping and friction effect of each perforation wall on acoustic wave is unchanged, with the increase of perforation rate of perforated panel and the number of perforations, the total damping and friction effect of all perforation walls increases, and the total energy of acoustic wave attenuation increases, so the sound absorption performance of the perforated panel muffler is improved. When the number of perforations continues to increase, due to the decrease of the spacing between neighboring holes, they will interfere with each other and affect the final sound absorption performance. As a result, with the increase of the perforation rate of the perforated panel muffler, the resonant frequency gradually shifts to the high frequency, and the absorption performance of the formant increases first and then decreases.

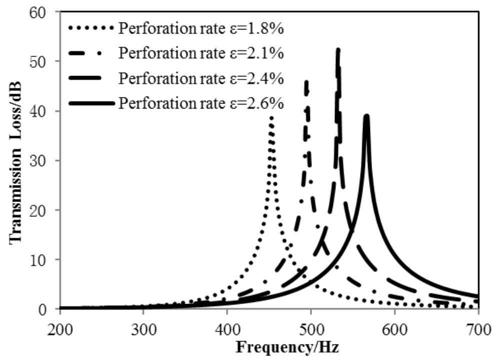


Figure 9: Acoustic absorption performance with different perforation rate

4.1.3 Panel thickness: The acoustic absorption performance of the perforated panel muffler with different panel thickness is shown in Figure 10, when cavity depth of the perforated panel is 16.65mm, the perforation diameter is 3.5mm, and the number of perforations is 36. With the increase of panel thickness, the resonant frequency of perforated panel muffler shifts to low frequency, and the absorption performance of formant increases first and then decreases. With the increase of the panel thickness, the depth of the acoustic wave passing through the perforation wall increases, and the damping and friction effect in the propagation process increases. Therefore, the energy of the acoustic wave attenuated on the wall increases, which leads to the improvement of the sound absorption performance of the perforated panel muffler. When the panel depth continues to increase, the increasing the damping and friction of the wall may reduce the number of sound waves passing through the perforated panel, thus affecting the sound absorption performance. As a result, the resonant frequency of the perforated panel can be reduced and the acoustic absorption performance of the perforated panel muffler can be improved by properly increasing the panel thickness.

To sum up, the structure parameters of perforated panel muffler such as perforation rate, perforation diameter and panel thickness have significant impact on the resonance frequency and absorption performance, and the analysis of structure parameters can provide guidance on the design and optimization of the perforated muffler.

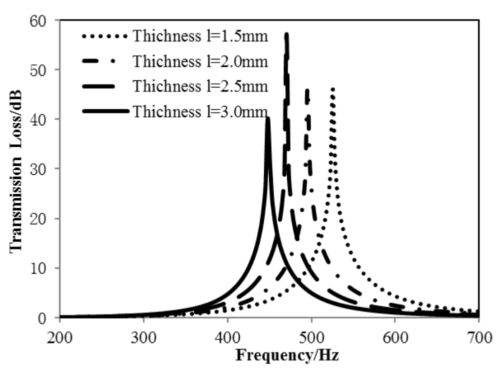


Figure 10: Acoustic absorption performance with different panel thickness

4.2 Optimization of Acoustic Characteristic

Based on the noise characteristics of screw refrigeration compressor, it can be seen that the first three times frequency noise of the engagement fundamental frequency of the male and female rotors of screw compressor is particularly significant, which needs to be improved. In order to reduce the noise of the screw refrigeration compressor, based on the influence of the structural parameters of the perforated panel muffler on the performance of the muffler, a broadband perforated panel muffler is designed in series with multiple chambers, and then its structure is optimized to improve its performance, as shown in Figure 11.

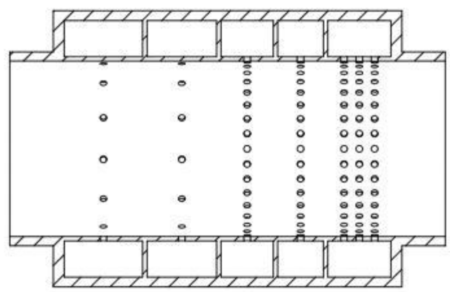


Figure 11: Sectional view of a multi-chamber broadband perforated panel muffler

Figure 12 shows the acoustic absorption performance of the optimized multi-chamber broadband perforated panel muffler. As shown in the Figure 12, the perforated panel muffler has remarkable acoustic absorption performance at the fundamental frequency, second frequency and third frequency of the discharge noise of the screw refrigeration compressor, which can effectively reduce the discharge noise of the compressor.

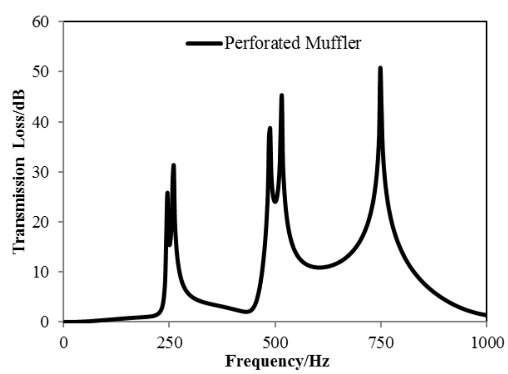


Figure 12: Acoustic absorption performance of the perforated panel muffler

5. CONCLUSIONS

A fluid-acoustic coupling model of a twin screw refrigeration compressor is established. The pressure pulsation and noise spectrum of the discharge section are obtained. In order to reduce the noise of screw refrigeration compressor, a perforated panel muffler suitable for screw refrigeration compressor is proposed. The impacts of different structural parameters on the performance of a perforated panel muffler are investigated, including perforation rate, perforation diameter and panel thickness. The conclusions are as follows:

- Results of CFD simulation show that there is an intense pressure pulsation in the compression chamber of the discharge section, while the pressure pulsation is much lower in the discharge pipe due to the large volume which provides buffer and attenuation functions.
- The resonant frequency of the perforated panel muffler can be reduced by reducing the perforation diameter properly, and the sound absorption performance of the perforated panel muffler can be improved with proper diameter.
- With the increase of the perforation rate of the perforated panel muffler, the resonant frequency gradually shifts to the high frequency, and the absorption performance of the formant increases first and then decreases.
- The resonant frequency of the perforated panel can be reduced and the acoustic absorption performance of the perforated panel muffler can be improved by properly increasing the panel thickness.
- The optimized multi-chamber broadband perforated panel muffler has remarkable acoustic absorption performance at the fundamental frequency, second frequency and third frequency of the discharge noise of the screw refrigeration compressor

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