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Leakage Model of Axial Clearance and Test of Scroll Compressors

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Abstract: Leakage clearance plays an important role in guaranteeing high efficiency of scroll compressors. In view of the shortcomings of the existing leakage models of axial clearance of scroll compressors, a modified Fano flow model and a turbulence model are presented based on the flow characteristics of fluid in the leakage passage under actual working conditions. Firstly, according to the Fano flow energy equation and the turbulence theory, two kinds of leakage models, Fano flow model and turbulence model, of axial clearance are established. Then, the established models are verified through an experimental platform established to measure the axial clearance leakage in the working process of a scroll compressor, and the measured values are compared with the values obtained from the two theoretical models. Finally, the effect of such factors as pressure difference, clearance amount, spindle speed on the axial clearance leakage is analyzed. Results show that the two established models can precisely reflect the variation law of axial clearance leakage of scroll compressor under different working conditions. In particular, the Fano flow model is more suitable for predicting the clearance leakage when the spindle speed is low (less than 3 500 r/min) and the clearance is small (less than 0.025 mm), whereas the turbulence model is suitable for high spindle speed (more than 3 500 r/min) and large clearance (more than 0.025 mm).
Key words: scroll compressor, axial clearance, leakage model, mass flow rate
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0 Introduction

In scroll compressors, gas compression is typically implemented by the meshing of fixed and orbiting scrolls. Therefore, a certain meshing clearance is usually set to reduce the friction loss of the compressors. The leakage caused by the meshing clearance plays an important role in guaranteeing high efficiency of scroll compressors, especially the axial clearance, which causes the most serious leakage due to the long leakage line. The leakage of scroll compressors has also become a hot spot of research topics.

Ishii et al.[1] assumed that the leaking gas in the axial clearance was an incompressible viscous flow, and gave a calculation model of the leakage amount. Through simulation test, the change rule of pressure in the high-pressure cavity and the low-pressure cavity with time was measured in the two kinds of axial clearance. Youn et al.[2] established a calculation model of axial clearance leakage of scroll compressor based on isentropic nozzle flow. In order to simulate the actual working condition, a flow coefficient was introduced, and the numerical value of flow coefficient was obtained through simulation experiments. Bell et al.[3] calculated the amount of axial gap leakage using the modified nozzle model, and compared the settlement results with the unmodified model. The results showed that the change rule of both models was linear. Liu et al.[4] used the N-S equation to establish a laminar flow leakage calculation model of axial clearance, and established a single-phase oil flow model, a steam flow model and an oil flow model, respectively. Liu et al.[5-6] also established a calculation model of leakage amount of axial clearance labyrinth seal. When the friction is not considered in the model, the leaking gas is considered as an isentropic flow. Zha et al.[7] proposed a mathematical model of axial gap leakage based on computational fluid dynamics (CFD), taking the friction, compressible and adiabatic flows of the leaking gas into account. The calculation results showed that the mass flow rate was 1/4 of the calculation results of the isentropic nozzle model. Wu[8] established a three-dimensional unsteady calculation model based on the experimental basis and CFD technology, and compared it with other leakage models. The leakage mass flow calculated by this model is 1/10 of the isentropic nozzle and 1/5 of

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the calculation results in Ref. [3].

From the above literature analysis, in order to simplify the calculation, the existing leakage model of axial clearance usually assumes that the leakage process an isentropic and non-viscous flow, incompressible flow or boundary conditions such as friction between gas and scroll teeth are not considered. However, the leakage process in an actual scroll compressor is a frictional compressible viscosity flow. This makes a large error between the simulation and the actual value of leakage in scroll compressors.

Based on several existing leakage models and the actual working conditions of the scroll compressor, axial clearance leakage prediction models of scroll compressor are established based on Fano flow theory and turbulent flow theory. The models take such factors as the compressibility and the viscosity of the fluid as well as the friction in the leakage process into account, and can well predict the general rule of axial clearance leakage. In order to verify the reliability of the prediction model, a testing platform for axial clearance leakage volume is built. The comparison between experimental results and theoretical calculation results is made.

1 Geometric Model and Working Parameters of Scroll

1.1 Geometric Model

The profile of both orbiting and fixed scroll of scroll compressors is based on the base circle involute. The coordinate equations of the inner and outer walls of the fixed scroll are as follows[9-10].

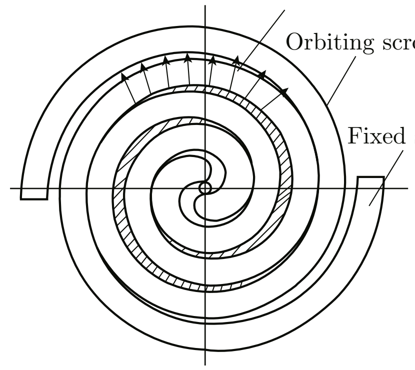
$$\left.\begin{aligned} x_{\mathrm{I}} &= a[\cos(\varphi_{\mathrm{I}}+\alpha)+\sin(\varphi_{\mathrm{I}}+\alpha)] \\ y_{\mathrm{I}} &= a[\sin(\varphi_{\mathrm{I}}+\alpha)-\cos(\varphi_{\mathrm{I}}+\alpha)] \\ x_{\mathrm{O}} &= a[\cos(\varphi_{\mathrm{O}}-\alpha)+\varphi_{\mathrm{O}}\sin(\varphi_{\mathrm{O}}-\alpha)] \\ y_{\mathrm{O}} &= a[\sin(\varphi_{\mathrm{O}}-\alpha)-\varphi_{\mathrm{O}}\cos(\varphi_{\mathrm{O}}-\alpha)] \end{aligned}\right\}. \tag{1}$$

The length of axial clearance leakage line and the volume of compression chamber at any spindle angle are[11]

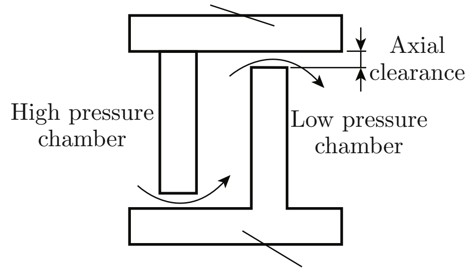
$$L_i(\theta) = 2\pi^2 a\left(2i - \frac{\theta}{180^\circ}\right), \tag{2}$$

$$V_i(\theta) = 4\pi^2 a h(\pi a - t_{\mathrm{c}})\left(2i - 1 - \frac{\theta}{180^\circ}\right), \tag{3}$$

where, the symbol of subscript I represents the parameters of the inner wall; the subscript O represents the parameters of the outer wall; L_i is the radial leakage line length; a is the radius of the base circle; φ is the spreading angle of the base circle involute; α is the initial angle of the involute; i is the compression chamber; h and t_{c} are the height and the thickness of the scroll teeth; θ is the spindle angle. The tooth tip of scrolls is modified by double circular arc (seen in Ref. [1] for details). The orbiting scroll is obtained by rotating 180° of the fixed scroll. The schematic diagram of axial clearance leakage of scroll compressor can be seen in Fig. 1.



(a) Diagram of radial leakage in axial clearance



(b) Diagram of axial clearance

Fig. 1 Schematic diagram of axial clearance leakage of scroll compressor

1.2 Working Parameter

Taking a base involute variable-frequency scroll compressor as an example, the radial leakage of the axial gap between two adjacent compression chambers is studied. The geometric and working parameters are $a = 5.04\,\mathrm{mm}$, $t_{\mathrm{c}} = 10\,\mathrm{mm}$, $\alpha = 56.9^\circ$ and $h = 40\,\mathrm{mm}$. The initial exhaust angle $\theta^* = 89^\circ$. The number of working cavity is 3, and the medium is air. The suction temperature is 300 K. The exhaust pressure is 0.5 MPa, and the rotating speed of the spindle can be adjusted.

2 Prediction Model of Axial Clearance Leakage

2.1 Fano Flow Model

Based on the nozzle model, the Fano flow model considers the influence of viscosity on fluid. The process of gas leakage is considered as the viscous gas flowing through a long and narrow leakage channel. Meanwhile, the influence of spindle speed is introduced in the model, which is more consistent with the actual leakage of scroll compressor.

The axial clearance leakage line of scroll compressor is long, thus the axial clearance leakage channel is equivalent to a straight pipe according to the method given in Ref. [12]. Axial clearance leakage channel of scroll teeth is shown in Fig. 2, and the mass leakage of axial clearance is calculated according to Fano flow theory[12].

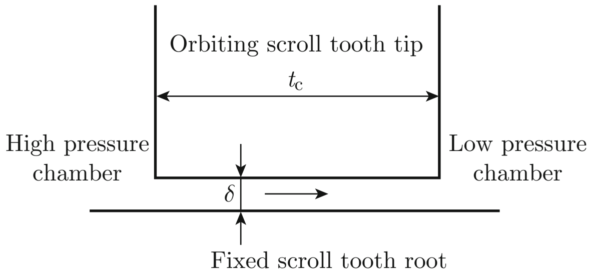


Fig. 2 Leakage channel of axial clearance of scroll tooth

Due to the small tooth thickness of scroll teeth, the leaking gas is considered as an isothermal flow with friction. According to the Fano flow energy equation, the following equations can be written:

$$\frac{1}{\rho}\frac{\mathrm{d}p}{\mathrm{d}x}+v\frac{\mathrm{d}v}{\mathrm{d}x}+f\frac{1}{\mathrm{d}x}\frac{v^2}{2D}=0, \tag{4}$$

$$D=\frac{L_i(\theta)\delta}{2[L_i(\theta)+\delta]}. \tag{5}$$

According to the ideal gas theorem, it is known that

$$\rho=\frac{p}{RT} \tag{6}$$

Based on the continuity equation, it is known that

$$v=\frac{\dot{m}}{\rho A}, \tag{7}$$

$$A=L_i(\theta)\delta. \tag{8}$$

Substitute Eqs. (5)—(8) with Eq. (4), and integrate the velocity along the tooth thickness, then

$$\mathrm{d}\theta=\frac{\pi n}{30}\mathrm{d}t.$$

The mass flow rate of radial leakage can be obtained as

$$\dot{m}=\frac{30}{\pi n}\sqrt{\frac{A^2(p_1^2-p_2^2)}{RT\left(f\frac{t_c}{D}+2\ln\frac{p_1}{p_2}\right)}}, \tag{9}$$

where, ρ is the leakage gas density; p is the gas pressure; v is the average velocity; D is the radial mass leakage equivalent diameter; δ is the axial clearance; R is the gas constant; T is the working chamber temperature; $\dot{m}$ is the radial mass leakage; A is the area of the leakage channel; n is the spindle speed; p_1 and p_2 are the pressure values of high and low pressure chambers; f is the resistance coefficient and is regarded as a constant.

2.2 Turbulent Flow Model

Because the scroll compressor is generally working at a high speed, leakage gas Reynolds number is large. This conforms to the turbulent flow theory. Meanwhile, the effects of fluid viscosity and friction between orbiting and fixed scroll tooth on the leakage are also considered in line with the actual working conditions of scroll compressor. A volume element in the axial clearance is taken as the research object for force analysis, as shown in Fig. 3, and the equilibrium equation can be listed as[2]

$$2\tau\mathrm{d}x=p\delta-\delta\left(p+\frac{\partial p}{\partial x}\mathrm{d}x\right). \tag{10}$$

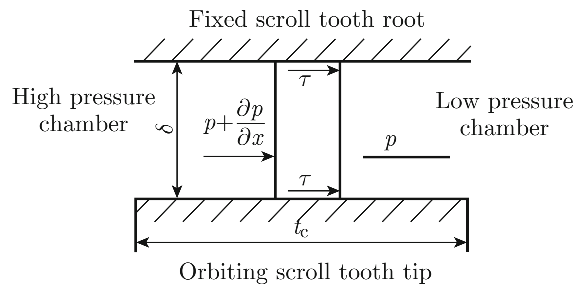


Fig. 3 Gas volume elements in axial clearance

The above equation can be simplified as

$$\frac{\mathrm{d}p}{\mathrm{d}x}=-2\frac{\tau}{\delta}. \tag{11}$$

According to Clausius' empirical formula, the friction force per unit volume between the gas and the scroll is

$$\tau=0.5f\rho v^2. \tag{12}$$

Substitute Eqs. (2), (3), (11) and (12) into Eq. (10) to get

$$p\mathrm{d}p=-\frac{0.378\,4\mu^{1/4}\dot{m}^{7/4}}{\delta^3\rho L_i^{7/4}(\theta)}. \tag{13}$$

According to Ref. [13], the relationship between viscosity and temperature is

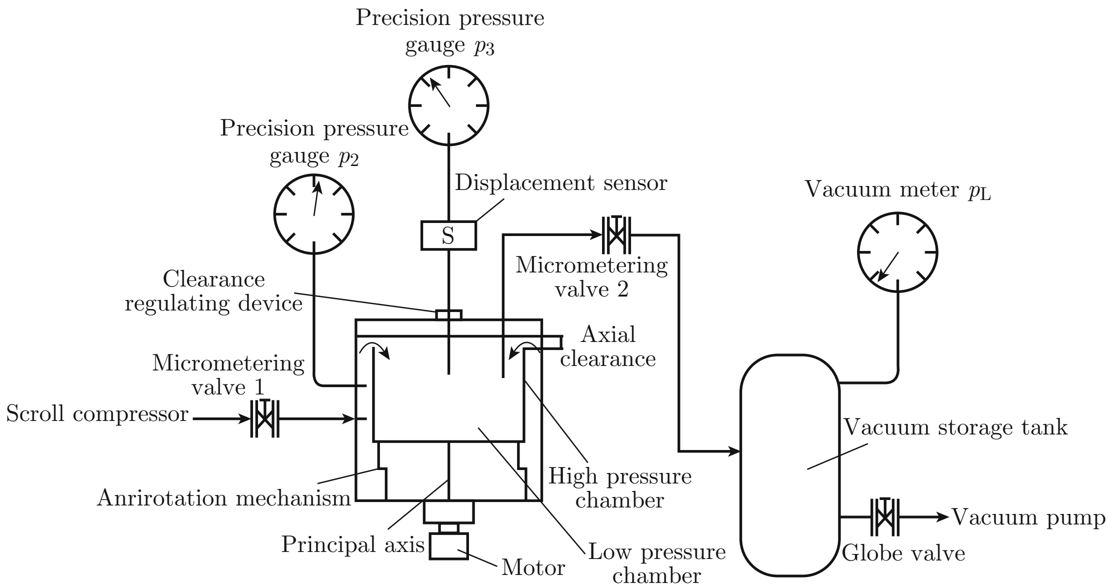
$$\mu=5.023\times10^{-7}T^{0.647}.$$

Equation (13) is integrated along the tooth thickness direction of scroll tooth, and the mass flow rate of clearance leakage of turbulent flow considering viscosity is expressed as

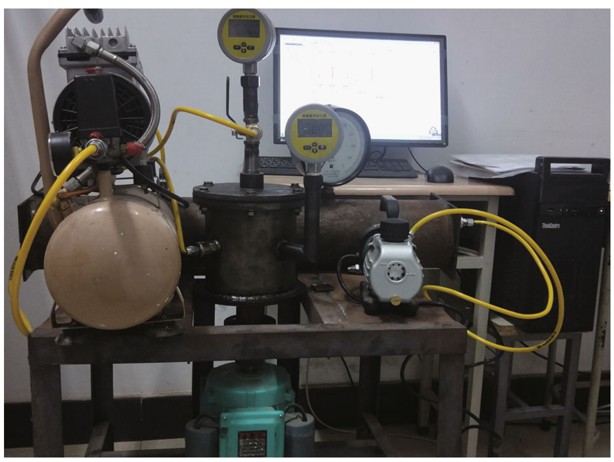
$$\dot{m}=\frac{30L_i(\theta)}{\pi n}\left[\frac{(p_1^2-p_2^2)\delta^3}{0.020\,6Rt_cT\mu^{1/4}}\right]^{4/7}. \tag{14}$$

3 Mass Flow Measurement of Axial Clearance Leakage

The volume of the second and the third working chambers is simulated when the scroll compressor begins the discharge process. The mass flow of axial clearance leakage within the volume is measured by the method of tank filling[5]. Figure 4(a) is the leakage measurement plan, and Fig. 4(b) is the photo of the leakage test platform.



(a) Diagram leakage flow test plan



(b) Photo of leak flow test platform

Fig. 4 Test bench for axial clearance leakage

Error analysis of experimental measurement shows:

(1) The established mass flow models are calculated under condition that the leakage gap is a constant. However, in the experiment, the moving scroll will tilt to some extent due to the motor speed fluctuation and the centrifugal force, which leads to the change of axial clearance. Therefore, the results will be somewhat different from the actual results.

(2) The outer wall's linear velocity direction of the orbiting tooth is opposite to that of the inner wall of the fixed scroll tooth, which has a reversed transport effect on the leaking gas and can reduce the leakage to some extent. In this experiment, because the rotating radius of the moving scroll is small, the reversed transport effect can be ignored.

(3) Due to the sealing error of the experimental device, the vibration caused by the high-speed rotation of the moving scroll will lead to a small amount gas leakage in some locations.

(4) Assuming that the working chamber temperature is constant.

In the experiment, in order to simulate the second and the third compression chambers when the spindle angle of the scroll compressor is 0°, the radial clearance sealing model is machined. So the wall thickness of the moving scroll is equal to the thickness of the scroll teeth of the scroll compressor; the circumference of the moving scroll is equal to the leakage line length of the second compression cavity; and the volume of the high pressure cavity is equal to the volume of the second compression cavity at the angle of 0°; the volume of the low pressure cavity is equal to the volume of the third compression cavity at the angle of 0°. The axial clearance is simulated and adjusted by the clearance adjusting device. Leakage clearance is found at the top of the two chambers. The axial clearance is adjusted according to the displacement sensor reading. The end face of the moving scroll and the end face of the end cover of the moving scroll are ground to assure the roughness of the end face is 1.6 μm. The rotation radius of the machined scroll is equal to the theoretical radius of rotation, to approximate the actual working condition of the scroll compressor. Before each experiment, sealing test of the device is carried out to ensure that there is no external leakage.

The measurement process of axial clearance leakage is as follows: close trim valve 2, open the globe valve, close trim valve 1 after vacuum tank is evacuated by vacuum pump. Open the scroll compressor, and start the motor and open the fine-tuning valve 1 when the pressure rises to a certain value. Adjust the fine tuning valve 1 until the pressure of p_2 and p_3 is equal to that of the second and the third compression chambers under a given spindle angle. Close the fine tuning valve 1 and the stop valve, open the fine tuning valve 2. Measure the pressure increasement in the vacuum tank in time period t, and then calculate the mass flow of leakage in the high-pressure chamber by[6]

$$\dot{m}_{\mathrm{s}} = \frac{V_{\mathrm{L}}\gamma}{RTt}\Delta p_{\mathrm{L}}, \tag{15}$$

where, $\dot{m}_{\mathrm{s}}$ is the calculate quality leakage; Δp_{L} is the measurement of gas pressure difference in vacuum vessel before and after measurement; T is the gas temperature value of vacuum vessel; V_{L} is the volume of vacuum vessel; γ is the air molecular weight; t is the leakage test time.

In order to study the influence of spindle speed and axial clearance values on leakage mass flow rate, variable frequency motor is used to obtain different spindle speeds. In this study, five rotational speeds, 2 500, 3 000, 3 500, 4 000, 4 500 r/min, are used to measure the leakage flow rate. The axial clearance values are set to be 0.020, 0.025, 0.030, 0.035 and 0.040 mm. The leakage mass flow rate under different rotational speeds and different axial clearances is thus measured by the above measurement methods.

4 Results and Discussions

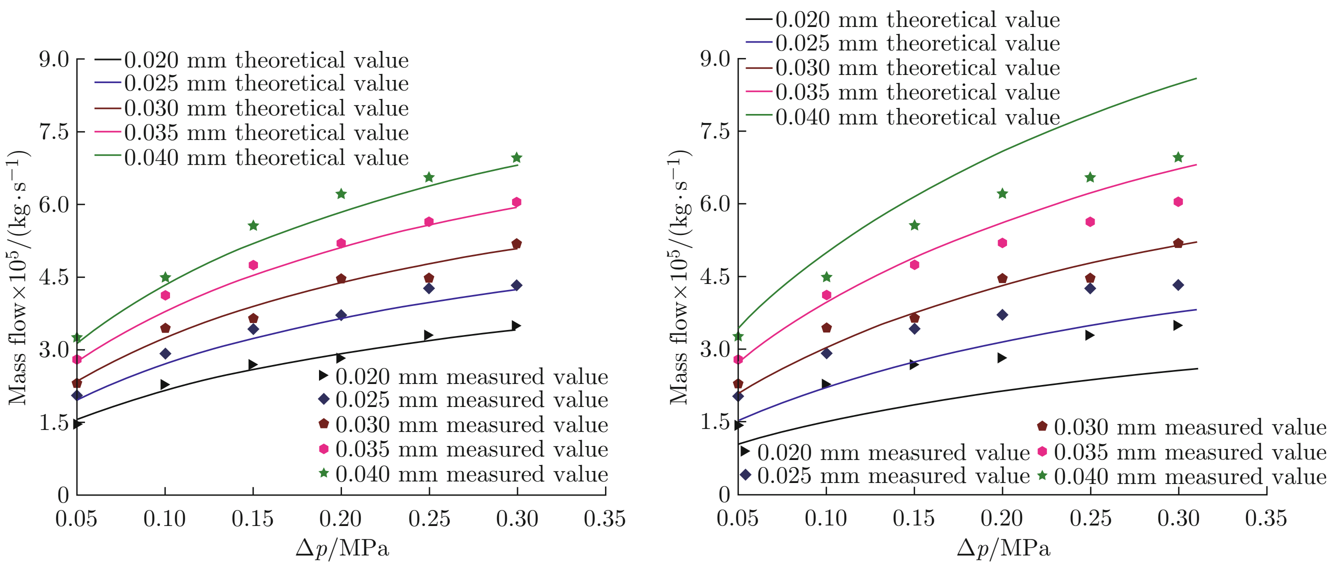
4.1 Influence of Axial Clearance on Leakage Flow

Figure 5 shows the variation of leakage mass flow under different axial clearance values when the spindle speed is 3 000 r/min (Δp is differential pressure). Figure 5(a) is the result of the comparison between calculated and measured values of Fano flow model. Figure 5(b) is the comparison between calculated and measured turbulence model values. It can be seen that leakage mass flow increases with the pressure difference of two adjacent chambers, and the larger the axial clearance, the greater the flow change curve slope. The theoretical curve calculated from the established models and the measured results almost has the same changing trend. The value from Fano flow model is closer to the measured value, and all the results are slightly smaller than the measured ones. Whereas in the turbulence model, the values calculated from turbulence model is larger than the measured value when axial clearance is more than 0.025 mm; the values from turbulence model is smaller than the measured value when axial clearance is less than 0.025 mm. The reason for the above phenomenon is that the leak path is equivalent to a straight pipe while calculating according to the Fano flow theory. But in the actual working conditions, the axial clearance is changing while the orbiting scroll in motion is subject to the action of upsetting moment. As a result, the leakage channel in some position is no longer a straight pipe, but rather gradually reducing and expanding pipe, which leads to lager measured values.

The reason why the turbulence model result is greater than the actual measured value is as follows. According to Ref. [13], when the Reynolds number is greater than 4 000, the clearance flow is a turbulent flow. Therefore, in actual working conditions, the leakage velocity may be small and the flow may change into laminar flow, which leads to the larger calculation result of the turbulent flow model. In addition, the friction between gas and wall surface of scroll teeth will decrease with increasing axial clearance, leading to the acceleration of leakage speed. The reason why the turbulence calculation result is smaller than the measured value, the friction on the top tooth between the orbiting and the fixed scroll intensifies due to small axial clearance, and the friction factor of the friction surface will change. Thus, the micro-gap leakage becomes obvious. By contrast, in the theoretical calculation, the friction factor is regarded as a fixed value, therefore, the measured values are greater than the theoretical ones.

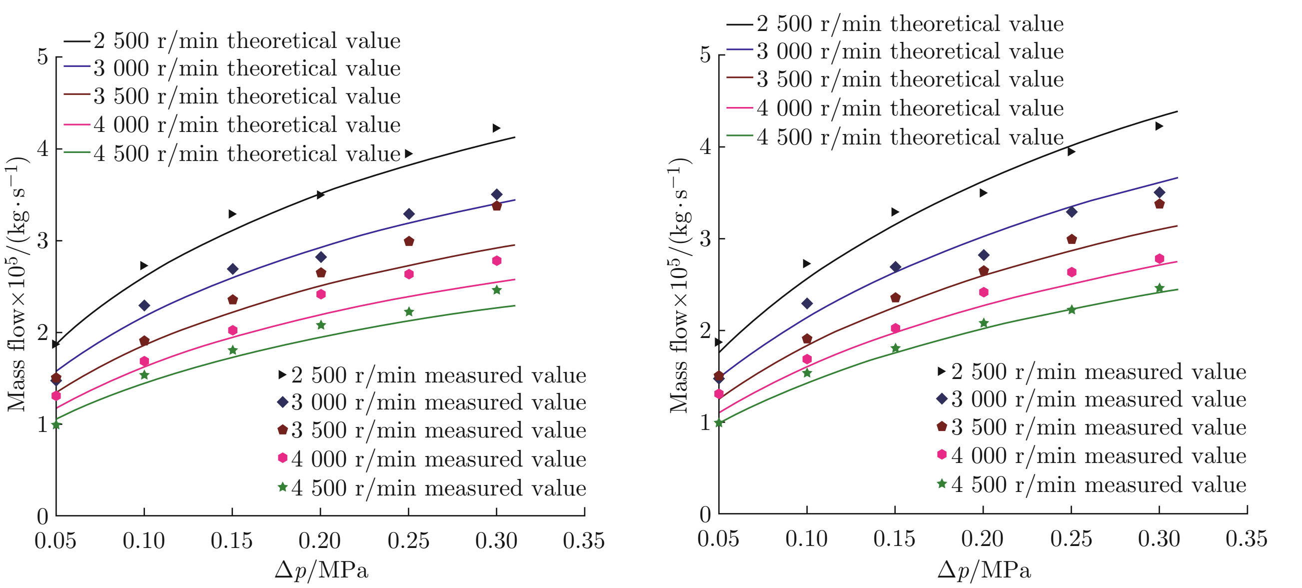
4.2 Influence of Spindle Speed on Clearance Leakage

Figure 6 demonstrates the variation of leakage mass flow at different spindle speeds when the axial clearance is 0.020 mm. Figure 6(a) compares the measured values



(a) Comparison of Fano flow model and measured value (b) Comparison of turbulence flow model and measured value

Fig. 5 Curve of different axial clearance leakage flow



(a) Comparison of Fano flow model and measured value (b) Comparison of turbulence flow model and measured value

Fig. 6 Curve of different speeds of main shaft leakage flow

with theoretical ones calculated from Fano flow model. Figure 6(b) shows the comparison between theoretical turbulence model values and measured ones.

It can be seen that the leakage mass flow decreases with the increase of the spindle speed. When the speed exceeds 3 000 r/min, the leakage mass flow decreases significantly as the pressure difference increases. This is because the motion of the orbiting scroll accelerates the velocity of gas in the high-pressure chamber, leading to the decrease of pressure within the chamber. Under the action of centrifugal force, the gas leakage through the axial clearance decreases with the increase of rotating speed. Figure 6(a) illustrates that the deviation between the theoretical and measured values gradually increases as the rotational speed increases, due to the increased leakage variation caused by overturning torque which increases with the rotational speed in practical work. As can be seen from Fig. 6(b), the higher the rotating speed, the closer the calculated result based the turbulence model to the measured ones, which proves that the gas in the chamber subjects to the turbulent flow rule more closely with the increase in the spindle. Therefore, the turbulent flow model can accurately predict the axial clearance leakage of the scroll compressor at high rotating speed. The reason why the turbulence calculation result is smaller than the measured value is that the friction between the orbiting scroll and the fixed scroll on the tooth top intensifies when the clearance is too small. Therefore, the friction factor on the friction surface will change, and the micro-gap leakage becomes obvious. While in the theoretical calculation, the friction factor is regarded as a fixed value, so the measured value is greater than the theoretical value.

5 Conclusion

The paper presents a model of axial clearance leakage, which takes the influence of rotational speed, viscosity, clearance value and other parameters on the leakage into account, aimed at developing an clearance leakage prediction that can simulate the actual working condition of scroll compressor. The Fano flow model and the turbulent flow model are established respectively based on the actual working conditions of scroll compressor. Experiments are carried out on leakage mass flow test platform to verify the established axial leakage models.

(1) By comparing the theoretical calculation results with the experimental ones, it shows that both theoretical models have the same variation trend as the measured values and can provide a theoretical reference for predicting the change law of an axial clearance leakage in a scroll compressor.

(2) Both models are calculated on the basis that the leakage clearance is a constant. In fact, the actual axial clearance will change due to a certain inclination of the orbiting scroll caused by overturning moment. Therefore, there are some differences between the measured results and the actual ones.

(3) Through a comparative analysis of the two models, the Fano flow model is more suitable for predicting the clearance leakage when the spindle speed is low (less than 3 500 r/min) and the clearance is small (less than 0.025 mm). Whereas the turbulence model is suitable for high spindle speed (more than 3 500 r/min) and large clearance (more than 0.025 mm).

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