Study on Axial Clearance and Seal of Scroll Compressor

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Abstract. In order to reduce the leakage of the axial clearance, the sealing strip is mounted at the top of the scroll tooth. For achieving the axial displacement compensation seal, it is put forward to open a hole that it is in one side of the high pressure chamber at the bottom of sealing groove. The high pressure chamber gas is introduced into the bottom of the seal. It can form a pressure difference. The article begins with the design of sealing groove structure. The stress analysis was carried out on the sealing strip, and sealing strip was established displacement calculation model of change. Secondly, through the simulation experiment, the friction and wear properties of several kinds of seal materials are compared and analysed. To select PEEK as sealing strip material is the most appropriate. Finally, taking typical parameters of involute scroll compressor as an example, the compressive performance of scroll teeth with sealed structure and non-sealed structure is simulated. The results show that the sealing displacement and axial clearance are basically the same under the two schemes. The sealing scheme is proved to be feasible. This lays the foundation for the further study of the axial seal of the variable section scroll compressor.

1. Introduction

Scroll compressor has the advantages of compact structure, high efficiency, high reliability, low vibration and low cost. A key issue in the development of scroll compressors is clearance leakage, which is an important factor affecting the development of compressors. Many researchers have focused on this subject and done a lot of work.

The radial leakage line of scroll compressor is twice as long as the tangential leakage line. In recent years, researchers at home and abroad have done a lot of work on radial leakage and seal caused by axial clearance. The research on the axial clearance is mainly the theoretical research, and the experimental research is relatively less. The mathematical model of radial clearance leakage of scroll compressor is established by Wang jun. The effect of different clearance on the leakage rate was studied. The results show that the exhaust pressure and temperature increase with the increase of radial clearance. And for the same clearance, the more close to the central part, the relative leakage increased. However, the above conclusions are drawn from the theoretical research, and lack of the corresponding experimental study. Yu Chen et al established a detailed geometric model of the scroll compressor compression process under the ideal conditions of isentropic flow. It gives the expressions of radial leakage area and tangential leakage area. And that the size of the axial gap and radial clearance is related to the size of the compression ratio, and according to the experience of the scroll compressor, The empirical formula for calculating the axial gap size and the size of the tangential gap is given. They also believe reducing the radial or flank leakage gap sizes to zero from the current design does not improve the mass flow significantly.

T. Inaba et al have proposed the method of filling and sealing strip to reduce the axial clearance.
Li haisheng thinks that installing spring at the bottom of sealing groove can solve the problem of wear compensation of sealing strip, and proposes to use truncation average method to calculate axial clearance. Since spring and other elastic elements can generate fixed axial force, leading to large friction and wear at startup and poor adaptability to changes in working conditions, the sealing mechanism is still in the exploration stage and has not been applied in engineering. He has established an experimental system of non-contact measurement for a scroll compressor to analyze the influencing factors of axial clearance. The results show that temperature and rotary speed are inversely proportional to the axial clearance. With the increasing of rotary speed, the axial clearance was gradually decreased. Liu Xingwang [6], in the tooth top to open the labyrinth groove to reduce the axial clearance of the radial leakage. And the simulation experiment platform was established to verify the method can reduce the leakage. However, the processing difficulty of labyrinth groove greatly restricts the popularization and application of the method. Yang Qichao proposed a method of measuring the axial gap by eddy current displacement sensor. The method can accurately measure the gap size and lay the foundation for the accurate calculation of axial leakage[7]. But usually the sealing material is not metal, can not be measured in this way. Therefore, a conductive self-lubricating material is urgently needed as the material for the sealing strip of the end face of scroll compressor.

The above analysis shows that the reasonable sealing structure and sealing material have a direct impact on the compressive performance of the compressor. On the basis of previous studies, this paper designs the structure of the tooth top groove. This paper proposes to replace the spring with a hole at the bottom of the scroll tooth seal groove. PEEK was selected as the sealing material. The axial clearance of dynamic and static scroll teeth was studied by analyzing the stress of sealing strip. According to the calculation model in the literature, the radial leakage amount under the mechanism is calculated, and the feasibility of the sealing method is verified. The results provide a reference for solving the radial leakage problem of axial clearance.

2. Tooth top sealing mechanism

Due to design error, machining error and installation error of scroll compressor, axial clearance is formed between moving and static scroll teeth. As shown in Figure 1, when the scroll compressor is working, the gas in the high pressure chamber enters the low pressure chamber through the axial clearance of the tooth tip. Axial clearance is the main factor causing the leakage of scroll compressor. Reducing the leakage of scroll compressor is of great significance to improving the efficiency of scroll compressor. In this paper, we design a kind of tooth top sealing mechanism, by reducing the size of the axial gap, reducing the radial leakage caused by the axial gap. As shown in Figure 2.
As shown in Figure 3, the sealing mechanism is composed of a sealing strip, a sealing groove with a step, and a gas inlet. The sealing strip is made of high molecular self-lubricating material. In order to compensate the axial wear of the seal strip, the bottom of the sealing groove is located on one side of the high-pressure cavity, and the method is used instead of the spring in literature 5. The high pressure gas is introduced into the bottom of the sealing strip, the sealing strip is suspended by high-pressure gas pressure, and the sealing strip is contacted with the bottom of the vortex plate to achieve the sealing effect. In order to ensure sufficient air force at the bottom of the seal, a certain clearance must be set between the sealing strip and the bottom of the sealing groove. Therefore, when the sealing groove is processed, the steps are added at the bottom of the groove to solve the problem. In order to prevent the leakage of gas from the bottom of the groove, the sealing strip is divided into two parts, and the second compression chamber and the compression chamber are drawn, and the sealing groove is provided with a spacer block. The vortex disk is about $\alpha + \frac{17}{6} \pi$.

3. Stress analysis of sealing strip
The sealing strip moves up and down to achieve axial sealing, it is clear that the sealing strip is a key component of the sealing mechanism. However, it is not hard to see, as a result of the compression chamber pressure in a constantly changing, so the displacement and force of the sealing strip, is also changing.

3.1. Stress analysis of sealing strip
Taking the involute scroll as an example, the force on the seal strip includes the axial force of gas, the axial force of gas and the additional axial force generated by overturning torque. The seal strip gravity, the seal strip and the friction force of the vortex disk. The stress analysis is shown in figure 4. Since there is a gap between the sealing belt and the vortex disk, the friction between them can be neglected. The equilibrium equation is as follows.
\[ F_z = p_i A_i + \sum_{i=2}^{N} \pi P_i^2 (2i-1-\frac{\theta}{\pi}) p_i + \left( \frac{h_i}{2} + h_i \right) \sqrt{\left[ p_i P_i \sum_{i=1}^{N} (2i-\frac{\theta}{\pi})(\rho_i - \rho_{in}) \right]^2 + [2rh_i (\rho_i - 1)]^2} / R \]  \hspace{1cm} (1)

among \( \rho_i = \left( \frac{2N-1-\theta_i}{2i-1-\theta_i} \right) \)

\[ F_i(\theta) = (p_i + p_c) s - F_z \cdot mg \]  \hspace{1cm} (2)

\[ s = \left[ \frac{1}{6} r^2 (\phi_2 - \alpha)^3 - \frac{1}{6} r^2 (\phi_2 + \alpha)^3 \right] - \left[ \frac{1}{6} r^2 (\phi_1 - \alpha)^3 - \frac{1}{6} r^2 (\phi_1 + \alpha)^3 \right] \]  \hspace{1cm} (3)

\[ \epsilon = \left( \frac{V_i}{V_c} \right) = \frac{p_i}{p_c} \]  \hspace{1cm} (4)

Due to the compression chamber is connected with the exhaust port, the center of the gas compression chamber is not compressed, and is an exhaust pressure process. Therefore \( p_i = p_c \).

When \( F_i \) is greater than zero, it indicates that the sealing strip and the bottom of the vortex plate contact, the mechanism have played a role in sealing. When \( F_i \) is less than zero, the sealing strip separates from the bottom of the scroll disk and generates axial clearance. When \( F \) is greater than zero, the value of \( F_i \) is as small as possible, because the reduction of \( F_i \) can effectively reduce the friction between the rotor and rotor scroll teeth. Therefore, the size of the sealing force is related to the size of the axial gap. Combined solution type (1)-(3) can be found, the sealing force \( F_i \) changes with the angle of the seal \( \theta \). Since the turning angle \( \theta \) is a periodic variation, it is possible to deduce the relationship between the \( F_i \) and the time.

3.2. The relationship between the displacement of the seal and the rotation angle

The equivalent vibration model is established by using the seal strip as the research object. Because the seal strip and the side face of the sealing groove are less friction. The motion of the seal is equivalent to the forced vibration of a single degree of freedom system under arbitrary excitation. Using the impulse response function method to solve the equation of motion of the seal in the initial system.

On the basis of \( \theta = \omega \tau \), \( F_i(\tau) \) can be converted to \( F_i(\theta) \), the same as \( \delta_i(\tau) \) can be converted to \( \delta_i(\theta) \).

\[ \delta_i(\tau) = \frac{1}{m \omega_i} \int_0^{\tau} F_i(\tau) \sin \omega_i (\tau - \tau) d\tau \]  \hspace{1cm} (5)

4. Study on properties of sealing strip material

Due to the requirement of oil free lubrication, the friction pairs of the top gear teeth need to be used as a self-lubricating material, the basic requirement is that the friction coefficient is low and the wear resistance is good. Some kinds of non-metallic self-lubrication materials presented in literature 13 and 14 are presented in this paper.
It have polytetrafluoroethylene (PTFE) composites, polyetheretherketone (PEEK) composites, and PES, The performance is shown in Table 1.

| material | density (g/cm³) | elongation (%) | water absorption (%) | distortion temperature (℃) | Coefficient of linear expansion (1/℃) | tensile strength (MPa) |
|----------|-----------------|----------------|----------------------|-----------------------------|----------------------------------------|-----------------------|
| PES      | 1.38            | 60             | <0.4                 | 210                        | 2.3*10⁻⁸                               | 8.5                   |
| PEEK     | 1.34            | 50             | 0.5                  | 326                        | 4.7*10⁻⁵                               | 92.2                  |
| PTFE     | 2.15            | 30             | 0.01                 | 120                        | 10.3*10⁻⁵                              | 27.6                  |

The HT250 material is a commonly used material for making scroll. The above three materials, respectively and HT250 to composition friction pair. The simulation experiment was carried out on the MMU-10G screen display material end face high temperature friction and wear test machine, as shown in Figure 5. The experiment was carried out by orthogonal test method. Experimental condition: Using disk and disk friction. Speed is set to:1200min/r,1500min/r,1800min/r,2000 min/r,2500 min/r. Test time is set to 4 hours. Test force is 100N. Data are collected every 10s. Acquired real time friction coefficient and friction temperature. After the experiment, the sample was cleaned by ultrasonic wave and then dried. The samples in acetone solution before and after the test are cleaned by ultrasonic wave for about 10 minutes. After drying in the oven, the quality of the sample was measured by PING FA-1004 HANG (precision 0.1mg) electronic analysis balance. Using optical reading microscope to measure the width of the surface grinding mark. Each material is made of 5 samples, and finally the average friction coefficient, average wear rate, average temperature. Variation trend of friction coefficient at different speeds as shown in Figure 6. Variation trend of friction temperature at different speeds as shown in Figure 7.

![Fig. 5 MMU-10G Friction sample machine and working principle diagram](image1)

![Fig. 6 Relationship between spindle speed and average friction coefficient and wear](image2)
Fig. 7 Relationship between spindle speed and average friction temperature

From the above analysis can be clearly seen, PEEK wear, friction coefficient, the average temperature is the smallest of these three materials, therefore, the choice of PEEK for the seal of the material is the best choice.

5. Calculation model of leakage

The axial clearance leakage of the scroll compressor is as follows: the gas in the high pressure chamber flows through the axial gap, and flows into the tangential gap. This paper focuses on the calculation of axial clearance leakage. Due to the existence of the original error of the installation error and the tooth top processing error, according to the results of literature4. It is assumed that the axial initial gap is 20μm. However, in practical application, the axial clearance were changed.

In order to calculate the leakage of axial clearance, and equivalent physical model were established. As shown in figure 8. The leakage of high pressure chamber gas to the axial clearance were equivalent to the contraction nozzle. The axial clearance leakage is equivalent to a straight pipe. The viscosity and friction cannot be ignored in the whole process of leakage.

![Figure 8](image)

Fig. 8 Equivalent calculation model of axial clearance leakage

The pressure loss of the high pressure chamber and the low pressure chamber could be expressed as the adiabatic, incompressible and viscous flow:

$$\frac{p_i - p_f}{\rho g} = \int \frac{L_2 v_i^2}{d^2 g}$$

\(L_{ni}\) — The length of radial leakage line of in the \(i\) compression chamber.

The influence of the leakage line length caused by the correction of the model line were ignored. It could be seen from the above formula that the gas leakage was related to the axial clearance and rotation angle.

6. Example analysis

Taking the basic parameters of air scroll compressor as an example. The base circle radius is 4.3 mm. The initial angle is 40°, and wall thickness is 6mm. It turning radius is 7.5mm, and tooth depth is 45mm. The rated working speed of scroll compressor is 3000r/min. It suction pressure is 0.1MPa, and discharge pressure is 0.4MPa. The sealing groove depth is 3mm, and width is 4mm. The base radius of sealing strip is 4.3mm, and width is 3.5mm, and height is 5.5mm. The density of PEEK is 1.29*10^{-3} g/mm³.
6.1. Calculation of axial clearance sealing force of seal strip
The above values are brought into the formulas (1)-(5). Using matlab software to calculate the size of seal axial clearance force. The results were shown in Figure 10. The size of the sealing force varies with the spindle angle.

![Graph showing relationship between seal force and rotation angle](image)

**Fig. 9** Relationship between seal force and rotation angle

It can be seen from the figure 9, The minimum sealing force was 33.7N, the maximum was 57.9N. When the angle between 0-1.7π, sealing force and rotation angle was approximately proportional to the relationship. When the angle was 1.7π, seal force reaches the maximum. When the angle was between1.7π-2π, sealing force and angle was inversely proportional. Since the first compression chamber suction has just ended, the bottom pressure of the sealing groove was almost zero. When the turning angle was from 0-1.7π, the gas enters into the compression chamber, the gas pressure in the compression chamber was gradually increased, and the bottom of the sealing groove was caused by the pressure difference between the high pressure chamber and the low pressure chamber, so that the sealing strip was suspended and the sealing force was increased gradually. During this period the sealing force appears the maximum value. Through the analysis, when the angle \( \theta \) is \( \Theta \), the sealing force reached the maximum value. At the same time, the sealing strip was close to the bottom of the vortex plate, and the sealing effect was the best. When the rotation angle was greater than 1.7π, the scroll compressor entered the exhaust phase, and the seal force was gradually reduced.

6.2. Simulation calculation of axial clearance of sealing strip
In this paper, it was assumed that the installation error and machining error are zero. The above values are brought into the formulas 8. Using matlab software to calculate the axial gap size. The results were shown in Figure 10. The curve of axial clearance with the change of spindle angle was obtained.

![Graph showing relationship between axial clearance and rotation angle](image)

**Fig. 10** Relationship between axial clearance and rotation angle

It can be seen from Figure 11, the distribution law of the end play was directly related to the sealing force of the seal. The size of the axial clearance was approximately parabola distribution with the spindle rotation angle. In a period of time, due to the change of pressure difference, the seal force changed, so that the sealing strip moved up and down, resulting in displacement. When the angle was \( \pi \), the maximum axial clearance. At this time, the compressor starts to exhaust, the axial load was maximum, the sealing force was reduced, and the sealing strip had downward movement. In the whole
process, the sealing force was always greater than zero, so the axial gap was very small, the maximum value was about 1.6μm.

Compare this result with the experimental results of literature 5. The results of the two methods were similar. In theory, the sealing structure can replace the spring to achieve the axial compensation of the seal. The non-spring seal mechanism used gas pressure difference to create a sealing force. It provided the power for the axial compensation of the sealing strip. In addition, the utility model had the advantages of simple structure, and could overcome the sealing failure caused by the failure of the spring.

6.3. Calculation of axial clearance leakage rate of sealing strip and non sealing strip
When it had weather strip to seal, the friction coefficient was 0.2, and the gap width was 3.5mm, The axial clearance was calculated by formula 7, and the leakage rate $Q_s$ could be calculated by 15 and 16 formula.

$$Q_s(\theta) = 2.33 \times \delta_s \sqrt{\frac{\delta_s}{(4\pi-\theta)(\delta_s+3.5)}}$$

According to the research results of literature 14, the recommended value of no seal axial gap was 20μm, the coefficient of friction of the sealing material was less than 0.15, and the gap width was 6mm. the leakage rate $Q$ was obtained.

$$Q(\theta) = 10^{-4} \times \sqrt{\frac{4.7}{4\pi-\theta}}$$

Formula 7 and Formula 8 were plotted by MATLAB software. The figure 11 shows the relationship between leakage and rotation angle. As can be seen from the graph, the leakage rate of packingless was proportional with the angle of the spindle. It was because in the process of suction and exhaust of the scroll compressor, the gas pressure in the compression chamber was changing, so the gas leakage was also changing. The leakage rate of the seal is proportional to the change of the axial clearance, and the parabola distribution was similar to the rotation angle of the spindle. When the gap was maximum, the maximum amount of leakage, which was when the angle of π, the maximum leakage rate. Compared to the size of $Q_s$ and $Q$, it is clear that the leakage of the seal is much less than the leakage of the seal. It shows that the sealing structure can reduce the leakage of the medium and improve the compression performance to some extent.

Fig. 11  Relationship between leakage and rotation angle

7. Conclusion
(1) The friction and wear test was carried out by simulating the working conditions of the vortex plate and the seal. The properties of several kinds of self-lubricating materials are compared. It is concluded that the performance of PEEK is the best. The raw material of the sealing strip is favorable for prolonging the service life of the sealing strip and improving the use efficiency of the scroll compressor.

(2) The tooth end face sealing mechanism used the compression force between the compression
chamber to cause the sealing strip to generate displacement, and the axial clearance of the scroll tooth was effectively reduced during the floating process of the seal strip, thereby reducing the radial leakage of the axial clearance. This method was not only simple in structure, but using is safe and reliable.

(3) Due to the sealing strip was a non-metallic material, the eddy current displacement sensor cannot be used to measure the axial dynamic clearance. So how to measure the axial clearance of the scroll compressor of the seal device made of non-metal and it would be the key to our next step to solve the problem.

Nomenclature

\( v_m \) —Mean velocity; mm/s  
\( d \) —Radial mass leakage equivalent diameter; mm  
\( r \) —base radius; mm  
\( p_1 \) —Gas pressure in a central compression chamber; MPa  
\( t \) —Tooth thickness; mm  
\( p_r \) —Pressure of gas in an arbitrary compression chamber; MPa  
\( R \) —gas constant;  
\( p \) —Pitch of circular involute; mm  
\( f \) —friction coefficient;  
\( \kappa \) —ratio of specific heat capacities;  
\( h \) —The dynamic vortex plate to drive the distance of tooth root; mm  
\( \theta \) —angle of axis; rad  
\( F_z \) —Axial gas force; N  
\( F_s \) —Sealing force; N  
\( F_d \) —Seal bottom gas force; N  
\( V_i \) —Inspiratory chamber volume; mm\(^3\)  
\( V_c \) —Compression chamber volume; mm\(^3\)  
\( \varepsilon \) —Compression ratio under ideal condition  
\( k \) —Adiabatic compression index;  
\( \delta_1 \) —Tangential gap; mm  
\( \delta_2 \) —Axial clearance; mm  
\( h \) —Scroll tooth height; mm

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