The Application of filled PTFE material in the end sealing of scroll compressor

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Abstract. The feasibility of filling PTFE composite material as the material of the top seal of scroll compressor was studied. Taking a certain type of scroll compressor as an example, had established a mathematical model of sealing strip force, and analyse the change regularity of sealing strip load. The PES, PEEK, the pure PTFE and the Filled PTFE were respectively matched with the HT250. According to the running parameters of scroll compressor, had designed test conditions, and had carried out the simulation experiment in the end wear testing machine. According to the experimental results, analyzed the four materials of the tribological properties. The volume wear rate, friction coefficient and temperature change of PTFE composites were measured. The morphology of the friction surface and the transfer of composites were observed by scanning electron microscopy. The results show that when the experimental conditions are 200N and 2000r/min, the friction coefficient of friction pairs is less than 0.25, and the volume wear rate is $3.328 \times 10^{-6}$. When the temperature increases, the friction coefficient has a rising trend, but it rises slowly. In addition, it is found by scanning electron microscope that when the temperature is above 120 degrees, the material will burn and the wear will be intensified. In summary, compared with the actual working conditions of scroll compressor, the PTFE material is used as the sealing strip material, and its friction and wear properties can basically meet the needs of the work.

1. Introduction
The leakage of scroll tooth end surface of scroll compressor is always the main factor hindering the development of scroll compressor. Because of its long line leakage, the leakage is much larger than other forms of leakage. At present, this problem has become the focus of research. Researchers designed the new scroll lines to reduce the length of the end of the leakage line to reduce the surface leakage. Kitae studied the heat transfer mechanism of scroll compressor. Chiachin studied the influence of temperature on the deformation of the scroll tooth, and the influence of the deformation of the scroll tooth on the leakage of the end face. YuChen established the mathematical model of the seal of scroll compressor, and provided the theoretical basis for the design of the sealing scheme. Liu Xingwang proposed that the labyrinth groove should be machined at the top of the scroll tooth to reduce the gas leakage. Li Haisheng proposed in the scroll tooth top slot filling, sealing strip, the sealing groove is arranged at the bottom of the spring to realize the axial compensation sealing strip,
but the program is still in the exploratory stage, due to the use conditions of the spring force is not suitable for the low reliability of compressor. The tooth top sealing mechanism, although in practice application, but the seal material selection is still the key, because the sealing strip and the scroll plate bottom friction, which requires the production of materials, low coefficient of friction seal wear.

For the study of sealing materials, usually use FEEK, FES, PTFE and other self-lubricating very easy to fail. PTFE has many advantages, such as self lubrication and low friction coefficient, but its application is affected by its large amount of wear. Researchers usually add oxides or metal elements to PTFE to improve their resistance to wear. Document 7 adding different proportions of Al2O3 in PTFE powder to improve friction and wear properties. When Al2O3 is added, the wear rate is the lowest. Studies show that polyimide can increase the bonding strength of the transfer film and the dual component, and reduce the friction coefficient. In document 9, PEEK and SiO2 were used to fill PTFE to improve the friction and wear properties. The experimental results showed that the hardness of the material increased greatly and the wear rate was only 1.5. In literature 10, mullite is filled with 1/530, and its wear rate is about PTFE of pure PTFE. In document 11, the friction characteristics of PTFE filled with water under lubrication condition were studied. The results showed that the surface oxidation of friction pairs could improve the tribological properties of the composites. Document 12 shows that the friction coefficient of the material is decreased by about 17% and the wear resistance is improved by 460 times by adding 5% nanometer SiO2 in PTFE.

In this paper, a scroll compressor parameter is used as an example, the choice of the new combination profile scroll. The tooth top was sealed by the sealing groove and the sealing strip combination seal structure. The sealing load of combining sealing structure on the bearing are calculated by using MATLAB software. According to the results of calculation set up experimental conditions. Simulating of scroll compressor operating conditions, the polyimide PTFE composites filled respectively with HT250 pairs, the friction and wear tester. Through the study of the experiment on the tribological properties of different loads and different speed conditions, validating the filling of PTFE material in the scroll compressor tip seal feasibility application.

2. Geometric model of scroll disk and calculation of working load

2.1 Geometric model of scroll disk

1) involute equation of internal circle

\[
\begin{align*}
x_i &= a_i \cos \phi + (a_i \pi \phi/180 + d) \sin \phi \\
y_i &= a_i \sin \phi - (a_i \pi \phi/180 + d) \cos \phi \\
0.75\pi & \leq \phi \leq 2\pi
\end{align*}
\]

(1)

2) interior high order curve equation

\[
\begin{align*}
x_2 &= a_{s1} \cos \phi + (a_{s2} + d) \sin \phi \\
y_2 &= a_{s1} \sin \phi - (a_{s2} + d) \cos \phi \\
2\pi & \leq \phi \leq 4.5\pi
\end{align*}
\]

among them: 

\[
a_{s1} = C_1 + 2C_2 (\phi - \pi/2) + 3C_3 (\phi - \pi/2)^2
\]

\[
a_{s2} = C_0 + C_1 (\phi - \pi/2) + C_2 (\phi - \pi/2)^2 + C_3 (\phi - \pi)^3
\]

(2)

3) involute equation of excircle

\[
\begin{align*}
x_i &= -a_i \cos \phi - (a_i \pi \phi/180 - d) \sin \phi \\
y_i &= -a_i \sin \phi + (a_i \pi \phi/180 - d) \cos \phi \\
0.75\pi & \leq \phi \leq 2\pi
\end{align*}
\]

(3)

4) exterior higher order curve equation

\[
\begin{align*}
x_4 &= -a_{s1} \cos \phi - (a_{s2} - 2d) \sin \phi \\
y_4 &= -a_{s1} \sin \phi + (a_{s2} - 2d) \cos \phi \\
2\pi & \leq \phi \leq 4.5\pi
\end{align*}
\]

5) exterior arc equation:
\[
\begin{align*}
    x_3 &= -(R_3 + D - d) \cos(\phi - 4.5\pi) + D \\
    y_3 &= -(R_3 + D - d) \sin(\phi - 4.5\pi)
\end{align*}
\]

In accordance with the double arc correction method, \( C_0, C_1, C_2 \) and \( C_3 \) are obtained by the method in document 13:

- \( \alpha_1 = 2.25 \)  
- \( \phi_1 = 2\pi \)  
- \( \phi_2 = 4.5\pi \)  
- \( R_3 = 44.0142 \)  
- \( d = 1.3 \)  
- \( D = 6.375 \)

by using double arc correction, the large arc equation is corrected as follows:

\[
\begin{align*}
    x_\Delta &= 4.4279 \cos(\phi) - 0.0546 \\
    y_\Delta &= 4.4279 \sin(\phi) + 3.1274
\end{align*}
\]

\(-0.5\pi \leq \phi \leq 0.25\pi\)

modified small arc equation:

\[
\begin{align*}
    x_\Delta &= 1.8279 \cos(\phi) + 0.0546 \\
    y_\Delta &= 1.8279 \sin(\phi) - 3.1274
\end{align*}
\]

\(-1.5\pi \leq \phi \leq 0.75\pi\)

among: \( \phi \) - Show angle; \( \theta \) - shaft rotation angle; \( d \) - offset distance;

- \( D \) - Adjust the parameters of the arc radius;
- \( \alpha_1 \) - base radius; \( R_3 \) - Large arc radius.

The equations (1) - (7) generate the coordinates in the Excel software, and import the CAD and Creo2.0 software to generate the two-dimensional and 3D models of the scroll disk, as shown in figure 1.

![Fig. 1 scroll disk model](image-url)

2.2 Design of sealing structure and calculation of seal strip load

2.2.1 Sealing structure design

The sealing structure adopts the opening at one side of the high pressure cavity at the bottom of the groove of the tooth seal groove, and the gas of the high-pressure cavity is introduced into the bottom of the sealing strip, so as to realize the axial compensation of the sealing strip. In order to ensure sufficient gas force at the bottom of the sealing strip, the lower side of the sealing groove is provided with a step on the side of the low pressure chamber. The sealing structure and the force analysis of the sealing strip are shown in figure 2.

2.3 Seal load calculation

The main load on the sealing strip is the pressing force between the sealing strip and the bottom of the scroll plate. The friction force is the main factor that causes the failure of the sealing strip because the pressing force is a friction pair between the sealing strip and the scroll disk. This paper mainly calculates the pressing force, which is the resultant force of axial gas force, side friction and the gas force at the bottom of the seal.
2.3.1 Calculation of axial gas force

The first compression chamber is surrounded by a modified arc curve and a circle curve of a round involute curve. According to the calculation method of document 14, the axial projection area of the first compression cavity is:

\[ A_1 = \frac{a^2}{6} \left\{ \left( \frac{5}{2} \pi - \alpha - \theta \right) - \left( \frac{3}{2} \pi + \alpha - \theta \right) \right\} - S_i, \quad 0 \leq \theta \leq \theta^* \]

\[ A_1 = \frac{a^2}{6} \left\{ \left( \frac{9}{2} \pi - \alpha - \theta \right) - \left( \frac{5}{2} \pi + \alpha - \theta \right) \right\} - S_i, \quad \theta^* \leq \theta \leq 2\pi \]

(8)

The volume of the first compression chamber is:

\[ V_1 = 2A_1H \]

Second the compression chamber is surrounded by the curve of the inner curve of the static scroll tooth \( \frac{5}{2} \pi - \alpha - \theta, \frac{9}{2} \pi - \alpha - \theta \) and surrounded by the curve of the outer wall of the moving scroll tooth \( \frac{3}{2} \pi + \alpha - \theta, \frac{7}{2} \pi + \alpha - \theta \) the dynamic and static vortex disk profiles are high order curves. The axial projection area \( A_2 \) of the second compression chamber is:

\[ A_2 = \frac{1}{2} \int_{\frac{5}{2} \pi - \alpha - \theta}^{\frac{9}{2} \pi - \alpha - \theta} \rho_s^2 d\phi - \frac{1}{2} \int_{\frac{3}{2} \pi + \alpha - \theta}^{\frac{7}{2} \pi + \alpha - \theta} \rho_s^2 d\phi \]

(9)

Second the volume of the compression chamber is:

\[ V_2 = 2A_2H \]

among: \( A \) - Sealing strip axial projection area; \( V_i \) - suction pressure.

Third the compression chamber is surrounded by the curve of the inner curve of the static scroll tooth \( \frac{5}{2} \pi - \alpha - \theta, \frac{9}{2} \pi - \alpha - \theta \) and the curve curve of the outer curve of the moving scroll tooth \( \frac{7}{2} \pi + \alpha - \theta, 4.5\pi \) the dynamic and static vortex disk profiles are high order curves. The axial projection area \( A_3 \) of the second compression chamber is:

\[ A_3 = \frac{1}{2} \int_{\frac{5}{2} \pi - \alpha - \theta}^{\frac{9}{2} \pi - \alpha - \theta} \rho_s^2 d\phi - \frac{1}{2} \int_{\frac{7}{2} \pi + \alpha - \theta}^{4.5\pi} \rho_s^2 d\phi \]

(10)

Third the volume of the compression chamber is:

\[ V_3 = 2A_3H \]

The axial force is applied to the end face of the sealing strip, and the axial force is expressed as a vortex disk:

\[ F_s = A \sum \left( \frac{V_i}{V_f} \right) \rho_s \]

(11)
\( \kappa \)-isentropic index; \( \theta^* \)-Start off angle.

2.3.2 Lateral friction calculation

The sealing strip is in the sealing groove and is generated by the axial force and the pressure difference between the high and the bottom compression chamber, and the friction force is generated between the sealing wall and the groove wall. The sealing strip adopts the equal wall thickness vortex curve, as shown in Figure 3, the outer curve of the inner curve of the vortex disc is moved outwards, and the shape line equation of the sealing strip is obtained. In order to avoid the leakage of high pressure gas along the bottom of the groove, the sealing groove is provided with three sections, each section is separated and closed, and the sealing strip is correspondingly divided into three sections and is arranged in the three section groove. First section \([0.75\pi, 1.75\pi]\), second section \([1.75\pi, 3.75\pi]\), third section \([3.75\pi, 4.5\pi]\). The size of the friction force at each side of the sealing strip is expressed as:

\[
F_{if} = fA_{ai}(p_{i} - p_{i+1}) \quad [3.75\pi, 4.5\pi]
\]

among \( A_{ai} = h_{i}L_{ai} \); \( L_{ai} = \frac{\theta_{ai} - \theta_{a}}{2} \); \( \rho = \sqrt{(x_{i}^{2} + y_{i}^{2})} \);

\( f \)-coefficient of friction; \( p_{i} \)-High pressure chamber gas pressure;

\( p_{i+1} \)-Low pressure chamber gas pressure; \( L_{ai} \)-Expansion length of sealing strip;

\( F_{if} \)-Friction force of sealing strip and sealing groove sidewall;

\( A_{ai} \)-The sealing strip extends out of the side area of the sealing groove;

\( h_{i} \)-The sealing strip extends out of the sealing groove;

![Figure 3 sealing strip structure](image)

The expression for pressing force of sealing strip is:

\[
F = A_{d} \sum p_{i} - mg - \sum F_{i}
\]

numerical calculation is carried out by using MATLAB software. The calculation results are shown in Figure 4. It could be seen that the angle was zero, the maximum compression chamber gas pressure, so the pressing force was maximum. When the angle was about 180 degrees, the compression chamber gas pressure was minimum. The sealing strip and the scroll from the bottom and then produce clearance under the action of gravity, and the sealed pressure was zero. However, the axial load varies periodically with the rotation angle of the spindle, and the average value of axial load is about 200N in one cycle. This calculation result is used as a reference for load setting in the simulation experiment.
Fig. 4 Variation of the clamping force with the rotation angle of the spindle

3. Experiment

3.1 Sample material
The choice of sample material is consistent with the static vortex disk material, HT250; the sample material is FP-2, PTFE is filled, and the main filler is polyimide. The sample preparation process was prepared under reference 9 and 15 methods. The physical properties of the upper and lower sample materials are shown in Table 1.

| Sample       | Material | Density (g/cm³) | Hardness | Tensile Strength (MPa) | Tensile Strain (%) |
|--------------|----------|-----------------|----------|------------------------|--------------------|
| Lower sample | HT250    | 7.1             | 180HB    | 275                    | -                  |
| Upper sample | FP-2     | 1.7             | 50HD     | 15                     | 220                |

3.2 Experimental methods and experimental equipment
In order to more accurately simulate the scroll and the seal working conditions by using MMU-2 high speed face Ji'nan Yihua production of friction wear tester, roughness of 0.3-0.5 disc specimens under static eddy simulation using a sample size, the seal for the 54*5 ring is simulated according to table 2 experimental conditions of 60min friction test in the testing machine. The quality wear was measured by BSA224S-CW electronic balance, and the volume wear rate was calculated according to formula (12) [15].

$$K = \frac{\Delta m}{\rho Fl} \ (mm^3/l(N\cdot m))$$

Among: $\Delta m$ is the mass difference before and after wear; $F$ is load for experiment; $l$ is friction distance; $\rho$ is sample density.

In the formula, the quality difference before and after the sample wear is the load added to the experiment; the friction distance is the sample density. Repeated friction experiments were performed 3 times for each specimen, and the average friction coefficient and average volume wear rate were taken. The surface of the friction surface was analyzed by JSM-5600LV low vacuum scanning electron microscope of Japan electron light corporation.
Table 2 experimental conditions of friction

| NO | Load/N | Speed/n/min | NO | Load/N | Speed/n/min |
|----|--------|-------------|----|--------|-------------|
| 1  | 100    | 500         | 10 | 150    | 1000        |
| 2  | 100    | 1000        | 11 | 150    | 1500        |
| 3  | 100    | 1500        | 12 | 150    | 2000        |
| 4  | 100    | 2000        | 13 | 300    | 500         |
| 5  | 200    | 500         | 14 | 300    | 1000        |
| 6  | 200    | 1000        | 15 | 300    | 1500        |
| 7  | 200    | 1500        | 16 | 300    | 2000        |
| 8  | 200    | 2000        | 17 | 350    | 2000        |
| 9  | 150    | 500         |    |        |             |

4. Experimental results and analysis

4.1 Analysis of friction and wear under different experimental conditions

The end face seal of scroll compressor requires low friction coefficient and little wear. By changing the load size and spindle speed, the friction and wear of the filled PTFE are analyzed. Figure 5 shows the change of friction coefficient under different loads and different speeds. As a whole, load and speed are directly proportional to friction coefficient. As shown in Figure 4, the axial average load of scroll compressor is about 200N. The load curve of 200N in Figure 5 is analyzed. When the speed is 500 r/min, the friction coefficient is 0.164, and when the speed is 2000 r/min, the friction coefficient is 0.245. When the speed is more than 1500 r/min, the friction coefficient tends to be stable. When the speed is 2000r/min, the change of the load shows that the friction coefficient is from 0.204 to 0.273, which proves that the influence of the load on the friction coefficient is not obvious. From the above analysis, it can be seen that filling PTFE meets the requirements of low friction coefficient of scroll compressor end face seal material.

![Friction coefficient vs Speed](image)

Fig. 5 Variation of friction coefficient under different load and speed

Figure 6 shows the volume wear rate at different speeds when the load is 200N. The results show that the higher the speed is, the greater the volumetric wear rate is. When the speed is 500r/min, the wear rate is 0.903, the speed is 2000r/min, the wear rate is 17.830. In addition, the higher the speed is, the higher the surface temperature of the sample is.
Fig. 7 is the result of wear under different loads at a speed of 2000r/min. The results show that when the load is less than 300N, the wear degree of the material is close. When the load is 350N, the wear rate increases rapidly, reaches 42.017, and the friction coefficient reaches 0.304. It is found that the load has a greater influence on the wear rate. When the PTFE is filled with a layer of transfer film on the surface of the cast iron, the friction temperature is not high and the friction coefficient is stable at about 0.2 when the load is less than 300N. When the load is 350N, the friction temperature increases, the sample temperature is as high as 142.3 degrees, near the softening temperature of filling PTFE, the transfer film starts to be burned, and the amount of wear will increase under dry friction condition. It should be explained that the experiment speed is higher. For safety, the experiment is carried out in a closed environment, which is not conducive to heat dissipation, so the experimental results are larger than the actual results.

Figure 8 (a) 100N and 2000r/min, SEM surface of composite wear surface. It can be seen that the surface formed a uniform transfer film, in addition to minor scratches, proved to have a slight abrasive wear. According to figure 5 analysis, it is the uniform transfer film on the friction surface that makes the friction coefficient low and the wear rate low. When the load is increased to 200N and 2000r/min, the mass of the friction surface is observed, as shown in Fig. 8b. This is due to the increase of load size at high speed, the increase of the friction surface temperature, the tendency of the transfer film to destroy, and the phenomenon of partial burning. To better understand the relationship between load, temperature and friction, the load continued to increase to 350N, 2000r/min, but the experiment only carried out 40min. It can be seen from the figure 8C that the transfer film of the friction surface has been sintered in large area and produced a large number of pores. In order to see the distribution and size of the pores and observe them with ultra depth of field microscopy, as shown in Fig. 8D, a hole with a diameter of about 1.043mm can be clearly seen. The reason for pore formation is that when the temperature is too high, some of the elements in the composite melt and form voids.

4.2 Friction profile analysis

Figure 8 (a) 100N and 2000r/min, SEM surface of composite wear surface. It can be seen that the surface formed a uniform transfer film, in addition to minor scratches, proved to have a slight abrasive wear. According to figure 5 analysis, it is the uniform transfer film on the friction surface that makes the friction coefficient low and the wear rate low. When the load is increased to 200N and 2000r/min, the mass of the friction surface is observed, as shown in Fig. 8b. This is due to the increase of load size at high speed, the increase of the friction surface temperature, the tendency of the transfer film to destroy, and the phenomenon of partial burning. To better understand the relationship between load, temperature and friction, the load continued to increase to 350N, 2000r/min, but the experiment only carried out 40min. It can be seen from the figure 8C that the transfer film of the friction surface has been sintered in large area and produced a large number of pores. In order to see the distribution and size of the pores and observe them with ultra depth of field microscopy, as shown in Fig. 8D, a hole with a diameter of about 1.043mm can be clearly seen. The reason for pore formation is that when the temperature is too high, some of the elements in the composite melt and form voids.
Figure 8 the surface morphology after PTFE friction experiment

Figure 9 is the surface morphology of the dual material. In 100N, 2000r/min under the condition of friction pair surface composite transfer film of uniform distribution, and a slight furrow marks, shown in Figure 9a). As shown in Fig. 9b, the transfer film starts to fall off under 200N and 2000r/min conditions. As the surface temperature of the metal increases, the transfer film begins to peel off. As shown in Fig. 9C, it can be seen that under the condition of 350N and 2000r/min, the transfer film of the metal surface is sintered together with metal grinding, and the friction coefficient rises linearly and the wear is intensified.

5. Conclusion

(1) The top scroll tooth make use of combined sealing technology. Through the stress analysis of sealing strip, it was related to the rotation angle.

(2) Through analysis of test results showed that the speed and load of PTFE composites filled with great influence on the life length. However, By simulating the actual working condition of scroll compressor, It was found that the composite can meet the requirements of wear.

(3) By scanning electron microscopy analysis showed that the temperature had quality influenced for the friction surface of PTFE composites filled . When the temperature build-up would cause the composite burning, increasing friction effect of life. So in the actual application process, the temperature field must be considered.

(4) To sum up, The polyimide filled PTFE composites with HT250 pairs, and simulated of scroll compressor friction test conditions. The results showed that PTFE could satisfy the filling material of scroll compressor seal working requirements.

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