Research on structure characteristics of the helium-lubricated spiral grooved thrust bearing for helium turbine expander

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Abstract: The helium-lubricated spiral grooved thrust bearing of helium turbo expander was studied by numerically simulating the steady-state non-linear Reynolds equation, the Reynolds equation of the pressure was solved by finite difference method (FDM), the film pressure distribution is calculated iteratively by MATLAB program, the structural parameters including the spiral angle, the groove length ratio, the groove width ratio influenced on the static characteristics were analyzed. The results show that: The bearing loading capacity of helium lubricated spiral grooved thrust gas bearing is lower than that of air lubricated bearing under the same conditions; The viscosity of the lubricant only affects the gas carrying capacity of the thrust disc when the spiral groove is on the bearing, while the density of the lubricant has a greater influence on the bearing loading capacity; When the spiral groove is designed on the rotor thrust disc, the bearing loading capacity of the spiral grooved thrust bearing increases by 6% on average compared with that of the spiral groove on the bearing.

1. Introduction
Gas lubricated bearing is a kind of sliding bearing with gas as lubricant. It has a series of advantages such as low friction power consumption, high motion accuracy, small vibration, pollution-free, high and low temperature resistance and stable operation in special environment [1]. Therefore, it has been widely applied in high-speed turbine expander, precision instruments and nuclear engineering [2].

The main problem in the development of spiral groove thrust gas bearing is that the working clearance is too small, so it is very difficult to obtain the gas film pressure field by measuring method [2]. Therefore, numerical simulation is a convenient and fast method, which can study the formation mechanism of gas film pressure and help to design excellent structure of spiral grooved thrust gas bearing [3].
In the large-scale cryogenic liquid helium refrigerant system, helium turbo expander is a core part of the system. In order to improve the purity of liquid helium, the bearing gas of helium turbo expander must use helium. Compared with air turbine expander, the physical properties of helium are different from air, for example, the density of helium is less than air. It is a more challenging task to design helium lubricated spiral grooved thrust gas bearing which can produce the same bearing loading capacity as the bearing lubricated by air.\(^4\)\(^-\)\(^6\).

2. The Model of the Spiral Grooved Thrust Gas Bearing

2.1 Basic equation of spiral grooved thrust gas bearing

The rotor thrust disc will bring the gas into the spiral groove due to the action of viscous lubrication gas, and the viscous gas will be compressed to form the axial bearing loading capacity\(^7\), as shown in Figure 1-2. In polar coordinates, the Reynolds equation of compressible steady-state gas lubricated spiral grooved thrust gas bearings is:

\[
\frac{\partial}{\partial r} \left( \frac{rP}{\mu} \frac{\partial \Phi}{\partial r} \right) + \frac{\partial}{\partial \theta} \left( \frac{P}{\mu} \frac{\partial \phi}{\partial \theta} \right) = 6\omega r \frac{\partial \phi}{\partial \theta}
\]

(1)

\[\text{Figure 1. Structural diagram of bearing-rotor.}\]

\[\text{Figure 2. Structural diagram of bearing thrust surface.}\]

Defining dimensionless parameters:

\[H = \frac{h}{h_0}, R = \frac{r}{r_0}, P = \frac{P}{P_a}, A = \frac{6 \mu \omega r^2}{P_h h_0^2}\]

In the formula:

- \(P\) — the dimensionless film pressure
- \(H\) — the dimensionless film thickness
- \(\zeta\) — the dimensionless radius coordinates
- \(A\) — the number of dimensionless bearings
- \(P\) — the film pressure
- \(P_a\) — the environmental pressure
- \(h_0\) — the minimum film thickness
- \(\omega\) — the rotor angular velocity
- \(\mu\) — the aerodynamic viscosity
- \(r\) — the rotor radius
- \(\theta\) — the circumferential coordinates
- \(\alpha/\beta\) — the spiral angle
- \(r_0\) — the radius of base circle
- \(r_1\) — the radius of dam area
- \(r_2\) — the outer radius
- \(\theta_1\) — the width of groove area
- \(\theta_2\) — the width of platform area

The steady-state nonlinear dimensionless control equation of spiral grooved thrust gas bearing is as follows:
The groove body of the spiral grooved thrust gas bearing is spiral structure, in which the calculation of film pressure is very complicated by finite difference method (FDM). In order to simplify the calculation, conformal transformation is needed for the governing equation and the solution domain. The working surface of the spiral groove thrust gas bearing is expanded into a plane and the conformal transformation method is used to transform the spiral groove into a fan-shaped element [8], as shown in Figure 3.

\[
\begin{align*}
\zeta &= r \\
\eta &= \theta - \frac{1}{\cot \beta} \ln \left( \frac{r}{r_0} \right)
\end{align*}
\]

\[\zeta = r, \quad \eta = \theta - \frac{1}{\cot \beta} \ln \left( \frac{r}{r_0} \right)\]

\[\left\{ \begin{array}{l}
\zeta = r \\
\eta = \theta - \frac{1}{\cot \beta} \ln \left( \frac{r}{r_0} \right)
\end{array} \right.\]

**Figure 3.** The conformal transformation

From the above two formulas, ‘r’ is a function of ‘ζ’ and ‘η’, and ‘Φ’ is only a function of ‘η’. In the new coordinate system, we derived the new Reynolds equation: the dimensionless Reynolds equation of gas film pressure of spiral grooved thrust gas bearing under steady state condition is:

\[
\begin{align*}
\frac{1}{R} \frac{\partial}{\partial R} \left( RH^3 \frac{\partial P^2}{\partial R} \right) + \frac{1}{R^2} \frac{\partial}{\partial \theta} \left( H^3 \frac{\partial P^2}{\partial \theta} \right) &= 2 \Lambda \frac{\partial (PH)}{\partial \theta} \\
\frac{\partial}{\partial \zeta} \left( \zeta H^3 \frac{\partial P^2}{\partial \zeta} \right) + \frac{1}{\zeta} \left( 1 + \frac{1}{\cot^2 \beta} \right) \frac{\partial}{\partial \eta} \left( H^3 \frac{\partial P^2}{\partial \eta} \right) - \frac{1}{\cot \beta} \left[ \frac{\partial}{\partial \zeta} \left( H^3 \frac{\partial P^2}{\partial \zeta} \right) + \frac{\partial}{\partial \eta} \left( H^3 \frac{\partial P^2}{\partial \zeta} \right) \right] + H^3 \frac{\partial P^2}{\partial \zeta^2} &= 2 \Lambda \zeta \frac{\partial (PH)}{\partial \eta}
\end{align*}
\]

The Boundary Condition:

1. The inlet and outlet of spiral grooved thrust gas bearing is environmental pressure:

\[P(\zeta = 0, \eta) = P\left( \zeta = \frac{r}{r_0}, \eta \right) = 1\]

2. The spiral grooved thrust gas bearing satisfies periodic boundary conditions:

\[P(\zeta, \eta) = P\left( \zeta, \eta + \frac{2\pi}{N} \right)\]

2.2 Static characteristic of spiral grooved thrust gas bearing

The distribution of gas film pressure of spiral grooved thrust gas bearing can be obtained by solving Reynolds equation (4):

The bearing loading capacity:
The film static stiffness:

\[ K = \frac{dF}{dh} \]  

(8)

3. Solution of Reynolds for spiral grooved thrust gas bearing

The gas film pressure between the rotor disc and the spiral grooved thrust gas bearing is solved by finite difference method (FDM) in this paper. The distribution of the gas film pressure— \( P_{i,j} \) is obtained by using the over relaxation iterative method (SOR), as shown in Figure 4. The flow chart of solving Reynolds equation of gas lubrication of the spiral grooved thrust gas bearing was shown in Figure 5.

| Table. 1 Physical properties of air and helium |
|----------------|----------------|
| Name            | Density( Kg / m³) | Viscosity( Pa·s ) |
| Air             | 1.225            | 1.79 \times 10^{-5} |
| He              | 0.163            | 1.99 \times 10^{-5} |

The spiral grooved calculation domain is expanded into a fan-shaped calculation domain, which can speed up the convergence speed and improve the accuracy \(^9\).

![Figure 4. The solution domain of fan-shaped](image)

![Figure 5. Flow chart of film pressure solution](image)

| Table. 2 Basic structural parameters of spiral grooved thrust gas bearing |
|---------------------------------|--------------|------|
| Structural parameters of bearing | Symbol / unit | Value  |
| Base circle radius of bearing    | \( r_0 \)(mm) | 8    |
| Radius of bearing dam area       | \( r_1 \)(mm) | 12.5 |
| Outer radius of bearing          | \( r_2 \)(mm) | 22   |
| Spiral angle                     | \( \alpha \)(°) | 14~24 |
| Groove length ratio              | \( \lambda = (r_2 - r_1)/(r_2 - r_0) \) | 0.45~0.75 |
| Groove width ratio               | \( \gamma = \theta / (\theta + \theta_l) \) | 0.4~0.8 |
| Minimum film thickness (Dam area) | \( h_0 \)(μm) | 10~30 |
| Depth of spiral groove           | \( h_f \)(μm) | 25~45 |
Groove depth ratio \( \delta = (h + h_0)/h_0 \) 3.5~5.5
Number of spiral grooves \( N \) 18
Environmental pressure \( P_a/(\text{MPa}) \) 0.1

4. The Calculation Results and Theoretical Analysis

4.1 Verification of calculation method

The finite difference method (FDM) is used to calculate the static characteristics of spiral grooved thrust gas bearing. The calculation results are compared with the results of finite element method (FEM) using FLUENT software in this paper.

A period is selected from the fluid domain of spiral groove gas bearing as shown in the Figure 6. According to the actual situation, the corresponding boundary conditions are set. The structured hexahedral mesh is divided by ICEM software, and the grid quality is shown in Figure 7.

The grid is imported into FLUENT fluid calculation software. The inlet and outlet pressures are ambient pressures, and the platform area is set as periodic boundary conditions. When the spiral groove is on the bearing, only the thrust disc rotates at high speed, and the speed of the thrust disc is 60000 rpm.

After iterative calculation, each parameter reaches the convergence state. The gas film pressure distribution of spiral groove thrust gas bearing is shown in Figure 8. A negative pressure area is formed at the entrance of the spiral groove. Under the action of viscosity, the lubricating gas brought in by the thrust plate flows along the spiral groove, and the lubricating gas is compressed in the dam area to form a high pressure area.

The Rotating Speed (10^4 rpm)  | Helium (Matlab-FDM) | Helium (Fluent-FEM) |
|-----------------|-------------------|-------------------|
| Capacity (N)     |                   |                   |

4.2 Comparison of calculation results with different methods

Figure 9. Comparison of calculation results with different methods
As shown in Figure 9, the maximum calculation error is less than 10%. The results of the two methods are in good agreement, which verifies the correctness of the MATLAB program. The Figure 10 and Figure 11 show the film pressure distribution of the helium lubricated spiral grooved thrust gas bearing. The Figure 10 is a two-dimensional view from the top, and the Figure 11 is a three-dimensional axonometric drawing.

The Bernoulli Equation:

\[ P_1 + \frac{1}{2} \rho v_1^2 + \rho gh_1 = P_2 + \frac{1}{2} \rho v_2^2 + \rho gh_2 \]  

(9)

Combined with the physical structure of spiral groove thrust gas bearing, the distribution mechanism of gas film pressure is analyzed: At the entrance of the spiral groove, the pressure can be converted into velocity energy, and the inlet pressure becomes low, which forms a pumping effect on the gas outside the groove from the Bernoulli Equation; most of the viscous gas flows along the spiral groove into the groove, and is compressed in the dam area to form a high-pressure gas film area, producing bearing loading capacity; the bearing gas flows to the environment passing through the dam area finally.

4.2 Influence of different parameters on static characteristics

The spiral groove on the end face of the bearing, it’s the thrust disc of the rotor which brings viscous gas into the spiral groove under the action of gas viscosity as shown in Figure 12.

As shown in Figure 13, different spiral angle has different influence on axial loading capacity of the spiral grooved thrust gas bearing. When the spiral grooved thrust bearing is lubricated by air and the optimum spiral angle is 18 degrees, it produces the best bearing loading capacity \(^{14}\). However, the optimum spiral angle which can produce maximum loading capacity is 16 degrees under the condition of helium as lubricant.
Due to the spiral angle is small, the spiral groove is close to the annular groove just as the left picture in Figure 14, the circumferential step effect is very weak, and the bearing loading capacity is low [8]; when the spiral angle is large, the structure of the bearing gradually tends to the radial stepped bearing [9], and the pumping effect of the gas bearing is gradually weakened, which makes the bearing capacity gradually reduce. Therefore, there must be an optimal spiral angle, which makes the spiral grooved thrust bearing have the best pumping effect and circumferential step effect at the same time to produce the best bearing loading capacity.

When the spiral angle is larger, the structure of spiral groove thrust gas bearing is closer to radial stepped bearing as shown in Figure 14. Helium with low density and molecular weight is easier to cross through the platform area and enter the next groove area, resulting in the reduction of the air volume of the bearing that can be compressed. Therefore, compared with the air lubricated spiral groove thrust gas bearing, the spiral angle is need to be designed smaller when helium is used as lubricating gas.

In the process of studying the influence of groove length ratio on bearing loading capacity of spiral grooved thrust gas bearing, the groove length ratio has the same influence trend on bearing loading capacity whether the lubricant is air or helium.

The groove length ratio represents the ratio of the length of the spiral groove to the radius of the whole thrust disc, and also reflects the sealing effect of the dam area on bearing gas [15]. When groove length is very small, the combined effect of pumping effect and circumferential step effect is weak, resulting in small loading capacity; when groove length ratio is large, sealing effect of dam area is small with enormous leakage, so the bearing loading capacity is also small. As shown in Figure 15, the optimal groove length ratio is about 0.68-0.7.
The groove width ratio represents the ratio of the width of the spiral groove to the whole thrust disc surface. As shown in the figure 16, the influence of the groove width ratio on the bearing loading capacity showed a trend of increasing and then decreasing. When the groove width ratio is 0.5-0.6, the spiral grooved thrust bearing produce the best bearing loading capacity to the thrust plate.

When the groove width ratio is small, it means that the groove area is relatively narrow, and the gas flow rate pumped into the groove is very small, and there is not enough bearing gas to produce large bearing capacity; when the groove width ratio is large, it means that the platform area is relatively narrow, and the resistance of bearing gas flowing from the spiral groove area to the next spiral groove area becomes smaller, which leads to the large leakage of bearing gas as shown in Figure 17, The results show that the bearing gas which can be compressed in the dam area will be reduced and the bearing loading capacity will be smaller.

As shown in Figure 18, when air is the lubricant, the optimum groove depth is 35μm, and the gas film thickness ratio is $\delta = 4.5$, it is consistent with the classical 4.0 design value; when helium is used as lubricant, the best film depth ratio and bearing loading capacity are less than those of air.
Figure 18. Influence of depth of spiral groove on loading capacity

For the same lubricant, the thrust disc of the rotor rotates at a high speed, and under the action of viscosity, the gas is brought into the groove and compressed in the dam area; with the increase of groove depth, the shear force gradually decreases, the gas velocity decreases, and the average pressure at the inlet increases. The pressure difference with the atmospheric environment (the pumping pressure) as shown in the figure 20, decreases continuously. Although the inlet area increases, the total volume flow rate increases first and then decreases.

For two different lubricant, air and helium, the viscosity of helium is slightly higher than that of the air, and the carrying capacity of thrust disc to the groove is basically the same. However, the air density is 7.5 times that of helium. According to Bernoulli equation, the greater the gas density is, the smaller the pressure at this point. Therefore, the pumping capacity of air is greater than that of helium. Under the same spiral groove structure, the volume flow rate of air is larger than that of helium, and the bearing loading capacity is also greater than that of helium.

Figure 20. Comparison of pumping effect and mass flow rate

It can be seen from Figure 21 that the gas film thickness in the dam area is a very important factor affecting the bearing loading capacity. With the increase of the gas film thickness in the dam area, the bearing loading capacity characteristics decrease sharply. It cause that the leakage of bearing gas is greatly increased, and the step effect is weakened.

With the decrease of gas film thickness in dam area, the slope of bearing loading capacity curve increases gradually, and the static stiffness of bearing gradually increases as shown in Figure 22.
Figure 21. Influence of gas film thickness of dam area on loading capacity

Figure 22. Influence of gas film thickness of dam area on static stiffness

4.3 The static characteristics of the spiral groove on the thrust disc and on the bearing

In this paper, we found that the spiral groove on the end face of the bearing, it’s the thrust disc of the rotor which brings viscous gas into the spiral groove, and the lubricating gas is compressed to form a gas film pressure; but there is a big difference when the spiral groove was designed on the thrust disc as shown in figure 23.

The spiral groove is on the thrust disc of the rotor. The spiral groove and the rotor thrust disc have the same speed. The lubricating gas entering the spiral groove is no longer carried by the viscous effect, but has a relative speed with the spiral groove. In this case, the volume flow rate of the lubricating gas is higher than that of the spiral groove on the bearing.

As shown in figure 24-26, we found that the bearing loading capacity has been improved when the spiral groove was designed on the thrust disc. Under the same structural parameter condition, this kind of design increase the bearing loading capacity by 6 percent on average, when the spiral groove thrust gas bearing lubricated by helium.
To sum up, we calculated the influence of different structural parameters and design styles on the static characteristics of spiral groove thrust gas bearing. According to the design requirements of the bearing loading capacity of helium turbo expander, we can select different and appropriate structural parameters on both sides of the thrust disc of the rotor to meet the design conditions.

5. Conclusion

1. The spiral grooved thrust gas bearing lubricated by helium has the best bearing loading capacity with 16 degree spiral angle. Due to the smaller density and molecular weight of helium, the smaller spiral angle makes it easy for helium to drain into the dam area for compression and to reduce the leakage in the platform area.

2. The optimum ratio of groove width is the result of the joint action of inlet area and leakage of the platform area. The optimal ratio of groove length is only the result of leakage in dam area.

3. For the same kind of lubricating gas, the groove depth increases, the inlet area increases. But the shear force decreases, which makes the average inlet pressure decrease. Therefore, the optimal groove depth ratio is the result of the joint action of the inlet area and the average inlet pressure.

4. The viscosity only affects the gas carrying capacity of the rotor thrust disc, while the density affects the velocity and pressure of the lubricating gas at the inlet, resulting in different pump gas pressure and volume flow rate. Therefore, compared with the viscosity, the density of helium is the more important reason for the smaller bearing loading capacity.

5. Spiral groove on the bearing, under the action of gas viscosity, the thrust disc will bring lubricating gas into the spiral groove, which is a kind of passive compression; when the spiral groove is on the rotor thrust disc, this is a kind of active compression, and the bearing loading capacity increases by 6% on average with the lubricating gas flow rate increases.

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