CFD research on hydrodynamic gas bearings with different styles of grooves

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Abstract: The gas bearing is an appealing technology which is widely used in turbo-machine in large-scale cryogenic systems for its inherent characteristic of oil-free and high-speed capability. The hydrodynamic gas bearings are more effective on cryogenic system but have less load capacity and dynamic stability by comparing with externally pressurized gas bearings. Etching some grooves on shaft or bearing is proved to be effective on improving the static and dynamic performance by experiments. The performance parameters of hydrodynamic gas bearing, such as load capacity and stiffness, are dominated by the styles and the geometric parameters of groove. In this paper, the effect of groove styles and geometric parameters on the performance parameters of hydrodynamic gas bearings is presented. And a novel style of groove is designed which can effectively improve the static and dynamic performance. The results based on the calculation of CFD show that: geometric parameters of the groove do have some influence on static and dynamic performance of hydrodynamic gas bearings, and there is an optimal objective value for each parameter of groove which makes load capacity and stiffness maximum. Compared with the spiral-style and $\pi$-style groove, the novel style of groove presented in this paper has better static performance.

1. Introduction
Gas bearings use gas as lubricant. Compared with traditional oil and rolling bearings, gas bearings have the advantages of higher precision, less heat generation under high rotating speed and environmentally friendly. In addition, a gas bearing does not need a complex lubricant circulation system which allowing
the system structure simpler. As a result, they are widely used in various applications such as turbo-expanders, dentist drills and high-speed milling machines (i.e., small rotors operating under high speeds) [1]. However, due to the inherently low viscosity of gas, the gas bearing needs a quite small film thickness to guarantee their intended function, and thus the fabrication and installation are expensive and time-consuming. Furthermore, the low viscosity also results in the low load capacity and low stability.

A successful implementation of the gas bearing in large-scale cryogenic systems is the turbo-expander, a key-equipment which determines the safety and the relative Carnot efficiency of the cryogenic system. The bearings applied to turbo-expander include both self-acting and externally pressurized bearings. Self-acting bearings are more efficient compared with externally pressurized gas bearing due to no pressured gas consumption, but don’t have enough stability to reach the designed operating speed. Thus, the analysis of static and dynamic characteristics for the design of a gas bearing is necessary.

Over the past decades, many investigations have been made on the stability of gas journal bearings used in various fields. In 1961, A.A.Raimondi [2] showed the numerical solutions of Reynold’s equation pertaining to the finite journal bearings, and discussed the differences of numerical techniques between gas and liquid lubricated. In the same year, Castelli and Elrod [3] used small perturbation technique and obits of shaft to present a better understanding of complex fluid dynamic phenomenon of self-acting gas lubricating journal bearing. In 1972, B.C.Majumdar [4] studied the externally pressured gas bearing using a numerical method, and experimental results showed the validity of the method. In 1981, S.S.Wadhwa [5] studied the steady state performance of pocket-type orifice-compensated gas bearings by using the finite element and extending the incremental formulation. In 1992, Dimofte and Florin [6] adopted the alternating direction implicit method and Liebmann’s iterative solution to solve the compressible pressure-differential equation, and compared with the experimental data which showed a good accuracy and time-saving. Wang Chengchi used the finite difference method to solve the static and dynamic performance of self-active and aerodynamic gas journal bearing in 2001 [7] and 2005 [8] respectively and analyzed the nonlinear behavior by coupling the dynamic equations. In 2012, Wei Zhang and Louis Chiappetta [9] showed the modeling capability and foil deformation by building a 3 dimension, fully-coupled, fluid-structure-interaction model. Most researches used FDM or FEM methods which often reduced the three-dimensional flow to two-dimensional under some reasonable assumption. However, the static and dynamic performance of gas bearing is improved by etching some grooves on shaft or bearing. For these bearings, it is complicated to get the static and dynamic performance by using FEM or FDM.

In this paper, three-dimensional models with grooves are built, and finite volume method (FVM) based on the software Ansys is used to calculate the three-dimensional flow field. The influence of the geometrical parameters and styles of grooves on the static and dynamic performance of self-acting gas bearing are studied, and this will provide a three-dimensional method for the design.

| Nomenclatures:                                      |
|----------------------------------|------------------|
| $D$                               | Diameter of bearings |
| $d$                               | Diameter of shaft  |
2. Calculation Modeling and Method

Figure 1 shows the structure of hydrodynamic gas bearing system. There is a gas clearance between bearing and shaft. As bearing and shaft are not coaxial, a force will be generated which can support the shaft through high enough rotation. To improve the effect of hydrodynamic, some grooves are etched on the shaft. Two types of the groove are designed which are shown in Figure 2. For type 1 (shown in Figure 2 (a)), the style can be named as the spiral-style groove, while for type 2 (shown in Figure 2 (b)), the style is called \( \pi \)-style groove. The basic structure of a bearing is as follows: length of bearing: \( L = 20 \text{mm} \), diameter of bearing: \( D = 17 \text{mm} \), average thickness of clearance: \( C = 10 \text{um} \). And the geometry structure parameters of grooves are listed in Table 1.

Ansys is commercial software which is widely used in the flow field calculation. In this study, some assumptions should be specified before the three-dimension flow field calculation:

(a) Because the ability of bearing materials (Copper) to conduct away heat is much greater than the heat generating capacity of gas film (low viscosity), the gas lubricating films are very nearly isothermal. Thus, we assume that the flow is isothermal.

(b) The gas in the film can be regard as ideal gas, so the density is only the function of pressure and temperature.

(c) The side flow (flow of gas in and out of the side of the bearing) is neglected.

(d) We assumed that the gas viscosity has nothing to do with the pressure, and the temperature of gas virtually is constant, thus, the gas viscosity is constant, too.

The three-dimension flow field calculation is based on the solution of Navier-Stokes equations coupled with mass conservation equation. Due to the symmetric configuration of bearing, only half of length in axial direction is modeled. In the calculation of gas bearing without grooves, the laminar calculation method was usually used. But, for the bearing with grooves, the flow field is more complex. Thus, the \( k - \varepsilon \) calculation method which is more precise to calculate the flow field and pressure distribution is chosen to capture the load capacity under various conditions. The Simplic iteration method is chosen to improve the stability of calculation. The residual error is set as \( 10^{-4} \) to capture the
precise calculation results. The boundary conditions for gas bearing with grooves in this paper are as follow:

(a) On one side of bearing, the pressure inlet boundary condition was set as the ambient pressure: \( P_a = 1 \text{atm} \). The temperature of gas was 300K.

(b) The rotor speed was designed as 220000rpm, thus, the boundary of journal was set as moving wall, and the type of motion was rotating.

(c) On the other side of model for calculation, the symmetry boundary condition was set.

(d) Helium was chosen as the lubricant, the density of helium changes with the pressure, which obeys the characteristic of ideal gas.

![Figure 1. Schematic diagram of hydrodynamic gas bearing](image1)

![Figure 2. Structure diagram of grooves in hydrodynamic gas bearings](image2)

**Table 1. Parameters of Grooves on the Bearing**

| Dimensionless length of groove | Dimensionless width of groove | Dimensionless depth of groove | Spiral angle | Number of groove |
|-------------------------------|-------------------------------|-------------------------------|--------------|-----------------|
| \( \lambda = L_{n1} / L \)   | \( \zeta = W_s / (W_b + W_g) \) | \( \kappa = C_s / C \)        | \( \beta_i \) | 11              |

**3. Results and Discussion**

The three-dimension pressure distribution is calculated by the finite volume method (FVM). The total number of nodes for half physical model is about 3 million. The length of element along the clearance is between 1 micrometer and 2 micrometers. Figure 3 shows the pressure distribution along the spiral-style grooves and \( \pi \)-styles grooves. Apparently, the gas is compressed both in spiral grooves and the spiral section of \( \pi \)-style grooves. But there is no compression effect in the straight groove. Compared with
the spiral-style groove, the pressure in the middle of the bearing with π grooves is higher. Thus, the hydrodynamic effective of π -style groove is better than spiral-style.

![Graph 1](image1.png)

**Figure 3. Static pressure changes along the groove**

![Graph 2](image2.png)

**Figure 4. Load capacity and stiffness versus spiral angle β₁**

Figure 4 presents the effect of spiral angle on load capacity and stiffness of two styles of groove. Six angle values from 25° to 50° are chosen for calculation. The results show the spiral angle has an important influence on the effect of bumped in and compression. It is inferred that the load capacity and stiffness of the π -style groove are always larger than the spiral-style groove. Both two types of grooves have the optimum spiral angle which makes the load capacity maximum. The tendencies of the load capacity to spiral angle are quite similar for both two styles groove. When the spiral angle is below 25°, the load capacity increases with the increase of spiral angle, however, when the spiral angle is larger than 30°, the load capacity declines. Thus, the optimum spiral degree is in the range of (25°, 30°). The static stiffness for both two styles of groove reached the maximum when the spiral angle is in the range of (25°, 35°).

Figure 5 shows the influence of the length of spiral grooves and spiral section of π -style grooves on load capacity. In this paper, the lengths of spiral section of groove which are chosen to calculate are from 3mm to 7mm. The results show that: the load capacity reach maximum when the length of groove is about 7mm in the gas bearings with spiral-style of grooves, while the optimum length of spiral section groove for load capacity is about 6mm in the bearings with π -style grooves. The stiffness for both styles of grooves are not sensitive to the length of groove, and have the same tendency and optimum length compared with load capacity.

Figure 6 shows the influence of the width in the circumferential direction of two styles of grooves on load capacity and stiffness. The width of groove influenced the area of compression channel and the hydrodynamic effects in circumferential direction. When the non-dimensional width of groove tends to zero, it means no grooves etched on shaft. On the contrary, when non-dimensional width of groove tends to 1, it also means no grooves on the shaft but the average of gas clearance is $C + H_g$. Thus, there is an optimum width of groove which makes load capacity maximum. From the results of CFD calculation, the optimum non-dimensional width of groove is 0.45. The change of stiffness for both two styles of groove versus width of groove is quite small. The optimum non-dimensional width of groove for load capacity is also suitable for stiffness.
The tendency of stiffness for \( \pi \) grooves increased first and then declined, the optimum value is 14\( \mu m \). For spiral grooves, the tendency increased first and then almost constant, the optimum depth is 16\( \mu m \).

In order to improve the hydrodynamic effect of grooves, a novel style of grooves is designed which is shown in Figure 8. Compared with \( \pi \) grooves, the novel style of groove changes the straight groove to spiral groove. So the gas can still be compressed in the second part of spiral grooves. Before the optimizing calculation of spiral angle \( \beta_2 \), we make the spiral angle \( \beta_1 \) is constant, and equal to 25°. For the parameter \( \beta_2 \), eight values were chosen for calculation. The results are shown in Figure 9. When \( \beta_2 \) is equal to 25°, the same with \( \beta_1 \), the herringbone-groove appears. The load capacity increases first and then declines with the increase of the second spiral angle. The optimum value of \( \beta_2 \) is 50°.

Figure 10 shows the load capacity versus eccentricity ratio for three styles of grooves. 11 values are...
chosen from 0.2 to 0.7. The tendencies of the curves for three types of grooves are almost the same which means the stiffness is almost equal. The novel style of groove has better load capacity compared with the other two styles of grooves.

![Graph 1](image1.png)  ![Graph 2](image2.png)

**Figure 9. Load capacity versus spiral angle $\beta_2$**  
**Figure 10. Load capacity versus eccentricity ratio $\varepsilon$**

### 4. Conclusions

This paper investigated the structure parameters of grooves with different styles for gas bearing used in turbo-expander. The purpose of this paper is to optimum the structure of grooves and introduced a new style of grooves which has better performances than traditional styles. From the results, we can get some conclusions and summarized as bellow:

1. The bearings with $\pi$ grooves have better steady static performance than bearings with spiral groove due to the hydrodynamic effects of the straight grooves.
2. Spiral angle in both two styles of grooves has influence on compression effects. when the value is in the range $(25^\circ, 30^\circ)$, the load capacity and stiffness reached the maximum.
3. The load capacity and stiffness reaches the maximum when the length of spiral section for $\pi$ style groove is $6mm$, and for spiral-style groove the optimum length is $7mm$; the optimum width of grooves for both two styles of groove is $0.45$, which makes the load capacity and stiffness maximum.
4. The optimum depth of grooves is $18\mu m$. This value is suitable for both two styles of groove. The tendency of stiffness is similar for both two styles of grooves, but the change of value is small.
5. The novel style of groove has better static performance than both styles of grooves for the better compression effect in the groove.

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