Dynamic Performances of the Herringbone-Grooved Gas Bearing

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Abstract: Aiming at studying the dynamic performances of herringbone-grooved gas bearings, a model is built to analysis the relationship between herringbone-grooved gas bearings parameters and its performances such as friction torque, stiffness, stamping, etc. In this model, the hydrodynamic effect of compressible fluid and the shunting effect of herringbone groove are taken into account, and the partial derivative method which based on local integral finite difference method is used to calculate the modal at the same time. The result shows that when the average gap and the length-diameter ratio are decreasing, the rotate speed and eccentricity ratio will be increasing and the hydrodynamic effect of the gas bearing will be enhanced. Increasing groove angle, decreasing width-ridge ratio or aggrandizing length-groove ratio can improve its load capacity. The parameters such as rotate speed and eccentricity ratio are the most important factors which affect the dynamic characteristics of herringbone-groove gas bearing greatly. Then the analysis method is verified through an experiment systematically. In the experiment, the moment balance method is used. The experiment result shows that the result of model is correct and reliable.

1. Introduction
Now, CNC machine tools are moving towards the direction of precision, high speed, intelligence, high efficiency and environmental protection. Electric spindle is the key part of CNC machine, which support the main machining power and handle the cutting tools directly. Therefore, high-speed spindle bearing is one of the key factors that affect the overall performance of CNC machine tools and other high-end equipment[1]. Gas bearing is a kind of non-contact bearing with gas as lubricant, which has many obvious features such as easy to rotate, clean, high rotary precision, long service life and so on.

It has been a long time about the study of gas bearings. In paper [2], the method for solving the transient stiffness coefficient of aerodynamic bearing was studied when the shaft neck was subjected to a constant radial velocity perturbation. Based on the small perturbation and linearizing PH methods, the equations of the static and dynamic characteristics of three-pad-tilting bearing were deduced[3]. Wang employed a hybrid numerical method combined the differential transformation method (DTM) and the finite difference method (FDM) to analyze the nonlinear dynamic behavior of a rigid rotor supported by a micro gas bearing system (MGBS)[4]. In 2017, Theisen had solved the problem to enhance the damping of gas bearing and improved its stability through robust control approaches[5].

For the study on characteristics of herringbone-groove bearing, Vohr and Chow proposed a narrow slot theory[6]. In Hris’s paper, considering eccentricity, the characteristics of the herringbone groove
bearing were studied by using the narrow slot theory, and the bearing stability and load capacity under the condition of neglecting the axial deflection and incompressible fluid lubrication were calculated[7]. Different types of spiral groove thrust bearing performance were introduced in paper and the differences with traditional thrust bearing are analyzed as well[8]. Murata proposed a two-dimensional analysis of oil film lubrication of herringbone groove[9]. Kawabata put forward a new type of herringbone-groove bearing which could turn positive inversion, and analyzed its load capacity under the condition of steady state by the theory of narrow slot[10]. Bonneau used the finite element method to calculate the characteristics of herringbone-groove gas bearing which have small slot number and compare with the characteristics of smooth surface bearing based on narrow slot theory[11]. Gad obtained the maximum radial force and radial stiffness by optimizing the geometric parameters of the bearing[12]. The influence of the geometric parameters and the lubrication gas on the performance of the herringbone-groove gas bearing was analysed[13].

In this paper, based on the theory of aerodynamics and taking the hydrodynamic effect of compressible fluid and the shunting effect of herringbone groove into account, the dynamic performances of herringbone-grooved gas bearing were studied.

2. The journal governing equation

2.1 The basic form of governing equation

The dynamic characteristics of gas film reflect the change of gas film force as journal in deviating from the static equilibrium position. Therefore, the Reynolds equation in unsteady condition must be set as the basis of calculation and analysis[14]. The partial differential derivation and dimensionless form of Equation (1) is[15][16]:

\[ \frac{\partial}{\partial \lambda} \left( R \frac{\partial}{\partial \lambda} \left( H \frac{\partial P}{\partial \lambda} \right) \right) = 2\Delta \frac{\partial(PH)}{\partial \phi} + 2\sigma P(\varepsilon \cos \varphi + \varepsilon \theta \sin \varphi) \]

where, \( \varphi \) is the circumferential angle coordinate \( (\varphi = \pi R) \), \( x \) is the bearing rotating direction coordinate. \( R \) is bearing radius. \( \lambda \) is dimensionless axial coordinate \( (\lambda = z / L) \). \( z \) is axial coordinate. \( \mu \) is the dynamic viscosity of the gas. \( U \) is the linear velocity of the bearing. \( L \) is bearing length. \( P \) is dimensionless pressure \( (P = p / p_a) \). \( p \) is pressure. \( p_a \) is ambient pressure. \( H \) is dimensionless gas film thickness \( (H = h / c) \). \( h \) is gas film thickness. \( c \) is bearing clearance. \( \varepsilon \) is eccentricity ratio \( = e / c \). \( e \) is eccentricity of journal. \( \Delta \) is compressibility number \( (\Delta = 6 \mu \omega^2 R^3 / (p_a c^5)) \). \( \sigma \) is extrusion coefficient \( (\sigma = 12 \mu R^2 / p_a c^5) \); \( \mu \) is viscosity of fluid, \( \omega \) is rotating speed, \( (e', \theta') \) is the axis of instantaneous position, \( (e'', \theta'') = \dot{\theta} / \omega = d\theta / (\omega dt) \).

2.2 The expression of bearing stiffness and damping coefficients

The relationship between the changes of the force and the perturbations is nonlinear[17]. When the perturbation is small; the relationship can be linearized. The four stiffness coefficients and four damping coefficients are called the eight dynamic characteristics of aerodynamic bearing. The gas film force is consisted of pressure distribution, so the dynamic characteristics of gas film directly depend on the perturbation pressure. Therefore, the dimensionless expressions of stiffness and damping coefficients can be obtained.

\[ K_w = \int_{-1}^{1} \int_{-1}^{1} P_e \left( \cos \phi \over \sin \phi \right) d\phi d\lambda, \quad K_{w'} = \int_{-1}^{1} \int_{-1}^{1} P_e \left( \cos \phi \over \sin \phi \right) d\phi d\lambda \]

\[ C_w = \int_{-1}^{1} \int_{-1}^{1} P_e \left( \cos \phi \over \sin \phi \right) d\phi d\lambda, \quad C_{w'} = \int_{-1}^{1} \int_{-1}^{1} P_e \left( \cos \phi \over \sin \phi \right) d\phi d\lambda \]

Where

\[ P_e = \frac{\partial P}{\partial e}, \quad P_{e'} = \frac{\partial P}{\partial e'}, \quad P_{e''} = \frac{\partial P}{\partial \theta'}, \quad P_{e'''} = \frac{\partial P}{\partial \theta''} \]
3. Solution of pressure perturbation

3.1 Establishment of perturbation equations

The crux of calculating the eight characteristics is to solve the four perturbation pressure \( P_\varepsilon, P_\theta, P_\varepsilon', P_\theta' \), the results can be get by solving the partial derivative of governing equations. Four perturbation equations are:

\[
\text{Re}(y(P^2_\varepsilon)) = -6\Delta \cos \theta \frac{\partial (PH)}{\partial \phi} + 3H \left[ \frac{\partial H}{\partial \phi} \cos \theta - H \sin \phi \right] \frac{\partial P^2_\varepsilon}{\partial \phi} + 3H \left( \frac{R}{L} \right)^2 \left[ \frac{\partial H}{\partial \lambda} \cos \phi \right] \frac{\partial P^2_\varepsilon}{\partial \lambda} + 2\sigma \left[ P_\varepsilon \left( \varepsilon \cos \phi + \varepsilon \theta' \sin \phi \right) + P_\theta' \sin \phi \right]
\]

(5)

\[
\text{Re}(y(P^2_\theta)) = -6\Delta \sin \theta \frac{\partial (PH)}{\partial \phi} + 3He \left[ \frac{\partial H}{\partial \phi} \sin \theta - H \cos \phi \right] \frac{\partial P^2_\theta}{\partial \phi} + 3H \left( \frac{R}{L} \right)^2 \left[ \frac{\partial H}{\partial \lambda} \sin \phi \right] \frac{\partial P^2_\theta}{\partial \lambda} + 2\sigma \left[ P_\varepsilon \left( \varepsilon \cos \phi + \varepsilon \theta' \sin \phi \right) + P_\theta \cos \phi \right]
\]

(6)

\[
\text{Re}(y(P^2_\varepsilon')) = 2\Delta \frac{\partial}{\partial \phi} \left( HP_\varepsilon \right) + 2\sigma \left[ P_\varepsilon \left( \varepsilon \cos \phi + \varepsilon \theta' \sin \phi \right) + P \cos \phi \right]
\]

(7)

\[
\text{Re}(y(P^2_\theta')) = 2\Delta \frac{\partial}{\partial \phi} \left( HeP_\theta \right) + 2\sigma \left[ \varepsilon P_\theta \left( \varepsilon \cos \phi + \varepsilon \theta' \sin \phi \right) + P \varepsilon \cos \phi \right]
\]

(8)

3.2 Solution of perturbation equations by partial derivative method

The crux is to solve the four perturbation pressure \( P_\varepsilon, P_\theta, P_\varepsilon', P_\theta' \), the results can be get by solving the partial derivative of governing equations. Four perturbation equations are:

\[
\text{Re}(y(P^2_\varepsilon)) = -6\Delta \cos \theta \frac{\partial (PH)}{\partial \phi} + 3H \left[ \frac{\partial H}{\partial \phi} \cos \theta - H \sin \phi \right] \frac{\partial P^2_\varepsilon}{\partial \phi} + 3H \left( \frac{R}{L} \right)^2 \left[ \frac{\partial H}{\partial \lambda} \cos \phi \right] \frac{\partial P^2_\varepsilon}{\partial \lambda} + 2\sigma \left[ P_\varepsilon \left( \varepsilon \cos \phi + \varepsilon \theta' \sin \phi \right) + P_\theta' \sin \phi \right]
\]

(5)

\[
\text{Re}(y(P^2_\theta)) = -6\Delta \sin \theta \frac{\partial (PH)}{\partial \phi} + 3He \left[ \frac{\partial H}{\partial \phi} \sin \theta - H \cos \phi \right] \frac{\partial P^2_\theta}{\partial \phi} + 3H \left( \frac{R}{L} \right)^2 \left[ \frac{\partial H}{\partial \lambda} \sin \phi \right] \frac{\partial P^2_\theta}{\partial \lambda} + 2\sigma \left[ P_\varepsilon \left( \varepsilon \cos \phi + \varepsilon \theta' \sin \phi \right) + P_\theta \cos \phi \right]
\]

(6)

\[
\text{Re}(y(P^2_\varepsilon')) = 2\Delta \frac{\partial}{\partial \phi} \left( HP_\varepsilon \right) + 2\sigma \left[ P_\varepsilon \left( \varepsilon \cos \phi + \varepsilon \theta' \sin \phi \right) + P \cos \phi \right]
\]

(7)

\[
\text{Re}(y(P^2_\theta')) = 2\Delta \frac{\partial}{\partial \phi} \left( HeP_\theta \right) + 2\sigma \left[ \varepsilon P_\theta \left( \varepsilon \cos \phi + \varepsilon \theta' \sin \phi \right) + P \varepsilon \cos \phi \right]
\]

(8)

The corresponding partial derivative:

\[
\frac{\partial P^2_\varepsilon}{\partial \phi} = \frac{1}{\sin \beta} \left[ \frac{\partial P^2_\varepsilon}{\partial \gamma} - \frac{\partial P^2_\varepsilon}{\partial \xi} \cos \beta \right]
\]

(9)

Where, \( \phi, \lambda \) are coordinates in the rectangular coordinate system; \( x, y \) are coordinates in the skew coordinate system; \( \beta \) is the angle of herringbone groove.

Using partial derivative finite differential method discretize the four perturbation, and the differential forms of perturbation equations are deduced.

\[
\text{Re}(y(P^2_\varepsilon)) = -6\Delta \cos \theta \frac{\partial (PH)}{\partial \phi} + 3H \left[ \frac{\partial H}{\partial \phi} \cos \theta - H \sin \phi \right] \frac{\partial P^2_\varepsilon}{\partial \phi} + 3H \left( \frac{R}{L} \right)^2 \left[ \frac{\partial H}{\partial \lambda} \cos \phi \right] \frac{\partial P^2_\varepsilon}{\partial \lambda} + 2\sigma \left[ P_\varepsilon \left( \varepsilon \cos \phi + \varepsilon \theta' \sin \phi \right) + P_\theta' \sin \phi \right]
\]

(5)

\[
\text{Re}(y(P^2_\theta)) = -6\Delta \sin \theta \frac{\partial (PH)}{\partial \phi} + 3He \left[ \frac{\partial H}{\partial \phi} \sin \theta - H \cos \phi \right] \frac{\partial P^2_\theta}{\partial \phi} + 3H \left( \frac{R}{L} \right)^2 \left[ \frac{\partial H}{\partial \lambda} \sin \phi \right] \frac{\partial P^2_\theta}{\partial \lambda} + 2\sigma \left[ P_\varepsilon \left( \varepsilon \cos \phi + \varepsilon \theta' \sin \phi \right) + P_\theta \cos \phi \right]
\]

(6)

\[
\text{Re}(y(P^2_\varepsilon')) = 2\Delta \frac{\partial}{\partial \phi} \left( HP_\varepsilon \right) + 2\sigma \left[ P_\varepsilon \left( \varepsilon \cos \phi + \varepsilon \theta' \sin \phi \right) + P \cos \phi \right]
\]

(7)

\[
\text{Re}(y(P^2_\theta')) = 2\Delta \frac{\partial}{\partial \phi} \left( HeP_\theta \right) + 2\sigma \left[ \varepsilon P_\theta \left( \varepsilon \cos \phi + \varepsilon \theta' \sin \phi \right) + P \varepsilon \cos \phi \right]
\]

(8)

Then the following equation can be got:

\[
(P_{ij}) = \frac{A_{ij} (P_a)_{ij} + B_{ij} (P_a)_{i+1,j} + C_{ij} (P_a)_{i,j+1} + D_{ij} (P_a)_{i+1,j+1} + (F_1)_{ij}}{E_{ij}}
\]

(12)
Where, $A_{ij}, B_{ij}, C_{ij}, D_{ij}, E_{ij}, (F_n)_{ij}$ are coefficient related with $H_{ij}, P_{i-1,j}, P_{i+1,j}, P_{i,j-1}, P_{i,j+1}$; $n$ indicates $\varepsilon, \varepsilon', \theta, \theta'$, etc.

4. Calculation results and analysis

4.1 The Comparison of Dynamic Characteristics between smooth-surface gas Bearing and herringbone-groove gas bearing

Under the same condition, there is difference of the dynamic coefficients between two type gas bearings. Figure 2 to figure 5 show that under the same condition, with the change of eccentricity, the change rate of dynamic characteristic coefficient of smooth-surface gas bearing is larger, than herringbone-groove gas bearing. Therefore, we can get that the herringbone groove can make the gas film smoother, but the shunting effect of herringbone groove will decrease the load capacity.

![Figure 2. The relationship between principal stiffness and eccentricity ratio](image1)

![Figure 3. The relationship between cross stiffness and eccentricity ratio](image2)

![Figure 4. The relationship between principal damping and eccentricity ratio](image3)

![Figure 5. The relationship between cross damping and eccentricity ratio](image4)

4.2 The effects of dimension parameter of bearing on dynamic characteristic coefficients

The figure 6 and figure 7 shows the variation of stiffness and damping coefficients as the bearing clearance changes, we can get that as the bearing clearance increases, the values of overall stiffness and overall damping coefficients are in the downward trend, and then keep stable. Therefore, the gas film pressure will increase as the bearing clearance decrease, and the load capacity will decrease as well.
Figure 6. The relationship between stiffness coefficients and bearing clearance

Figure 7. The relationship between damping coefficients and bearing clearance

Figure 8. The relationship between stiffness coefficients and ratio of length to diameter

Figure 9. The relationship between damping coefficients and ratio of length to diameter

The figure 8 and figure 9 show the variation of stiffness and damping coefficients as the ratio of length to diameter changes. As the ratio changes from 0.5 to 1.0, the values of overall stiffness coefficients decreases, while the values of overall damping coefficients increase. With the upward trend continuing, the dynamic characteristic coefficients remain unchanged.

4.3 The effects of working condition of bearing on dynamic characteristic coefficients

The working condition is important for bearing, the effect of bearing working condition can be obtained by analyzing the variations of the dynamic characteristic coefficients.

The figure 10 and figure 11 shows the variation of stiffness and damping coefficients as the eccentricity ratio changes, with the eccentricity ratio increasing, the values of overall stiffness and overall damping coefficients all have the upward trend. Hence, the increase of the eccentricity ratio can improve the load capacity, and then the dynamic characteristics increase correspondingly.

Figure 10. The relationship between stiffness coefficients and eccentricity ratio

Figure 11. The relationship between damping coefficients and eccentricity ratio
Figure 12. The relationship between stiffness coefficients and rotating speed

Figure 13. The relationship between damping coefficients and rotating speed

The figure 12 and figure 13 shows the variation of stiffness and damping coefficients as the rotating speed changes. As the rotating speed increase, the stiffness coefficients $K_{xy}$, $K_{yy}$ and coefficient $C_{yy}$ keep the downward trend, while the other stiffness coefficients and other damping coefficients increase. As the rotating speed increase, the hydrodynamic effect becomes more obvious. The load capacity and the dynamic characteristics will increase as well finally.

5. Experimental verification

According to the above moment balance method, the rotor of the tested bearing is directly fixed on the rotating shaft of the output end of the electric spindle, the herringbone grooves are machined on the rotor surface. The bearing is equipped on the rotor driven by electric spindle directly, so the bearing is in a rotational trend. However, due to the limitation of the string the bearing cannot rotate, the force is transmitted to the sensor though the string. Thus, the friction torque between the bearing and rotor can be measured. As shown in figure 14, there are four kinds of bearing structures. Figure 15 is friction torque test device.

Figure 14. Herringbone-groove gas bearing

Figure 15. Friction torque test device
As presented in figure 16, when the bearing is in low speed, the friction torque decreases with the increasing of rotating speed. When the bearing is floating and stable, the friction torque has an increase trend. As to the reason of difference between the two curves, it probably is that the bearing does not float and the dry friction between the bearing and the rotor makes the measured value larger than the theoretical value when the speed is low. In addition, the differences between the experimental value and the theoretical value may be caused by the machining accuracy, the calculation error caused by the numerical method and the accuracy and sensitivity of the tension sensor.

![Comparison between experimental value and the theoretical value of the friction torque](image)

**Figure 16.** Comparison between experimental value and the theoretical value of the friction torque

### 6. Conclusions

In this paper, the dynamic characteristics of herringbone-groove gas bearings are analyzed. Through the analysis and experiments, the following conclusions can be obtained:

1. Theoretical analysis results indicate that smooth-surface bearing will be out of stability under the condition of light load and high speed. Herringbone-groove gas bearings not only improve this defect and its pumping effect, but also increase the hydrodynamic effect.

2. With the increase of eccentricity and rotational speed, the change of bearing dynamic characteristics is obvious. So changing these parameters properly can get better stiffness and damping coefficients.

3. The groove depth has a significant influence on the stiffness and damping coefficients. Because the groove depth is large, the average air gap becomes large, and the air film pressure is affected. So the deeper the groove depth, the radial stiffness and damping coefficient of the bearing decreases gradually.

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