Influence of Floating Ring Seal Working Condition Parameters on the Dynamic Characteristics of Transient Gas Film

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Abstract—The high requirements for sealing performance in high-speed rotating machinery has led to the design of floating seal with annular spiral groove that offer the advantages of low leakage and extended stability. However, efforts to model the dynamic performance of these floating seal have suffered from the great complexity of the flow field. The present work addresses this issue by establishing a transient Reynolds formulation of a floating seal with annular spiral groove in a rotating coordinate system based on the small perturbation method. In addition, the influence of radial eccentricity and film thickness on the solution divergence and calculation accuracy is calculated. The dynamic stiffness and dynamic damping matrixes are built. Then the variation rules of the dynamic stiffness and damping coefficient of the gas film with structure and working conditions are investigated in detail. The results show that the floating ring seal is more suitable for the service conditions of small film thickness, low pressure, high speed and large eccentricity. Accordingly, the results obtained lay a theoretical foundation for evaluating real-world applications of floating ring seal.

1. Introduction

High-speed rotating machinery require ultra-high sealing performance[1], which led to the development of floating seal with annular spiral groove for applications such as aircraft engines and rocket turbine pumps[2-4]. As illustrated in Fig. 1, stable separation of the floating ring is supported by a gas film between the rotating and floating rings, and the floating capacity of this film is based on the hydrodynamic pressure effect generated by the spiral groove and the gas film wedge effect resulting from eccentricity.

Several studies have evaluated the steady-state sealing performances of floating seal from the perspective of seal structure based on numerical solutions of gas flow equations. For example, the National Aeronautics and Space Agency (NASA) developed the GCYLT computer program to analyze the gas film pressure and thickness of floating seal with different structures and dimensions[5]. The China Gas Turbine Research Institute optimized the design of floating seal [6]. Other studies have employed numerical analyses conducted at the microscale to investigate the lubricity of a groove floating seal[3], and the effects of the spiral groove angle and rotation speed on seal leakage[5]. These studies have demonstrated that the micro-gap between the rotating and floating rings and the characteristics of spiral groove profoundly influence gas flow through the seal. Other researchers have pursued the steady-state analysis of floating seal using computational methods. For example, Chochua et al.[7] developed a computational model for a honeycomb-stator gas floating seal. Mel’nik et al.[6] proposed a modified computational algorithm, and thereby analyzed the effects of changes in gas flow and inlet loss in the...
spiral groove of floating seal. More recently, the dynamic characteristics of floating seal have been investigated numerically. For example, Tokunaga et al.\[9\] solved the three-dimensional (3D) Reynolds equation for a floating seal using the finite difference method (FDM), and employed the solution to evaluate the hydrodynamic pressure distributions and leakage flow rate of the seal, as well as evaluate the Lomakin effect. Peng et al.\[10\] also applied the FDM to solve a coupled model of the clearance flow and elastic structure of floating seal, and analyzed the obtained principle stiffness and cross damping elements of the dynamic stiffness and damping matrices of the system. Banakh and Barmina\[11\] investigated the influence of bearing capacity and contact force in a dry friction micro-gap on oscillation in a rotor floating ring system. While these past studies have contributed greatly to modeling the dynamic characteristics of floating seal, the shear flow of the discontinuous spiral groove structure caused by variations in pressure and rotation further increased the complexity of the flow field. Therefore, numerical models for analyzing the dynamic performance of floating seal require further development.

The present work addresses this issue by establishing a transient Reynolds formulation and film thickness equation of an annular spiral groove floating seal in a rotating coordinate system based on the small perturbation method, high-precision eight-point difference method and Newton-Raphson iterative method. The proposed formulation accounts for variations in the gas film thickness arising from eccentricity between the rotating and floating rings, and the effect of the Rayleigh step on the gas film shape. The dynamic stiffness and damping coefficients of the gas film in the floating ring seal are evaluated in detail based on the principles of eddy energy.

2. Calculation model

2.1. Dynamic Reynolds differential equation

The floating ring seal model employed in the present study is illustrated in Fig. 2. Here, the model regards the sealed gas film as several springs and dampers.

![Fig.2 Physical model of an annular spiral groove floating seal employed in the analysis](image)
The dimensionless dynamic Reynolds equation of the floating ring seal model is given as

\[
\frac{\partial}{\partial \theta} \left( \frac{ph}{\partial \theta} \right) + \left( \frac{R}{L} \right)^2 \frac{\partial}{\partial z} \left( \frac{ph}{\partial z} \right) = \Lambda_x \frac{\partial}{\partial \theta} \left( \frac{ph}{\partial \theta} \right) + 2\gamma \Lambda_x \frac{\partial}{\partial t} \left( \frac{ph}{\partial t} \right)
\]

(1)

where is circumferential coordinate (rad), \( R \) is the outer radius of the rotating ring (m), \( L \) is the axial length of the floating ring (m), \( \Lambda_x \) is the compressibility coefficient, and \( \gamma \) is the perturbation ratio. The dimensionless parameters employed in Equation (1) above are defined as

\[
z = \frac{z}{h}; \quad \bar{h} = \frac{h}{L} \quad \bar{p} = \frac{p}{p_0}; \quad \bar{\mu} = \frac{\mu}{\mu_0}; \quad \bar{v} = \frac{v}{\omega_t}
\]

(2)

where \( z \) is the axial coordinate (m), \( h \) is the gas film thickness (m), \( C \) is the average gas film thickness (m), which is the difference between the floating ring radius \( R_1 \) and the rotating ring radius \( R \) (i.e., \( C = R_1 - R \)), \( p \) is the gas film pressure (Pa), \( p_0 \) is the atmospheric pressure (Pa), \( \mu \) is the gas viscosity \((\text{N} \cdot \text{s} / \text{m}^2)\), \( \mu_0 \) is the initial gas viscosity \((\text{N} \cdot \text{s} / \text{m}^2)\), \( v \) is the eddy velocity (rad/s), and \( t \) is the time (s).

In addition, we define \( \Lambda_x = \frac{6a_0 \mu R^3}{p_0 C^3} \) and \( \gamma = \frac{v}{\omega} \), where \( \omega \) is the angular velocity of the shaft (rad/s).

Dimensionless pressure \( p \) and gas film thickness \( h \) under microscale perturbation can be given as follows.

\[
\begin{align*}
\bar{p} &= \bar{p}_0 + \bar{p}_x \Delta X + \bar{p}_y \Delta Y + \bar{p}_x' \Delta \theta + \bar{p}_y' \Delta \theta \\
\bar{h} &= \bar{h}_0 + \Delta x \sin \theta - \Delta y \cos \theta
\end{align*}
\]

(3)

2.2. Dynamic characteristic coefficients

The model system of springs and dampers illustrated in Fig. 2 includes four stiffness coefficients \( k_{xx}, k_{yy}, k_{xy}, \) and \( k_{yx}, \) and four damping coefficients \( c_{xx}, c_{yy}, c_{xy}, \) and \( c_{yx}, \) which are defined as follows.

\[
\begin{align*}
[k_{xx} & \quad k_{xy}] = -\frac{RLp_1}{C} \int_0^{2\pi} \int_0^h \bar{p}_x \cos \theta \bar{p}_x' \cos \theta \quad d\theta \quad d\bar{z} \\
[k_{xy} & \quad k_{yy}] = \frac{RLp_1}{C} \int_0^{2\pi} \int_0^h \bar{p}_y \cos \theta \bar{p}_y' \cos \theta \quad d\theta \quad d\bar{z}
\end{align*}
\]

(4)

\[
\begin{align*}
[c_{xx} & \quad c_{xy}] = -\frac{RLp_1}{Cv} \int_0^{2\pi} \int_0^h \bar{p}_x \sin \theta \bar{p}_x' \sin \theta \quad d\theta \quad d\bar{z} \\
[c_{xy} & \quad c_{yy}] = \frac{RLp_1}{Cv} \int_0^{2\pi} \int_0^h \bar{p}_y \sin \theta \bar{p}_y' \sin \theta \quad d\theta \quad d\bar{z}
\end{align*}
\]

(5)

In addition, the pressure boundary condition is given as follows.

\[
\begin{align*}
z &= 0, \quad p_x = p_y = p_0 = p_h \\
z &= L, \quad p_x = p_y = p_r = p_l
\end{align*}
\]

(6)

Here, \( p_l \) is the medium pressure on the low pressure side, \( p_h \) is the medium pressure on the high pressure side. Finally, the cyclic circumferential boundary condition is given as follows.

\[
\begin{align*}
p_x(\theta, z) = p_x(\theta + 2\pi, z) \\
p_y(\theta, z) = p_y(\theta + 2\pi, z)
\end{align*}
\]

(7)

3. Results and Discussions

3.1. Seal parameters

The gas property parameters of the analyzed the floating ring seal are listed in Table 1.
Table 1 Gas property parameters of the seal

| Symbol | Parameter                      | Value     | Symbol | Parameter                      | Value     |
|--------|--------------------------------|-----------|--------|--------------------------------|-----------|
| R      | Outer diameter of the          | 0.025 m   | T₀     | Environmental temperature     | 298 K     |
|        | rotating ring                  |           |        | Environmental pressure        | 101325 Pa |
| L      | Axial length of the            | 0.052 m   | p₁     | Viscosity                      | 1.8 × 10⁻⁵ Pa·s |
|        | floating ring                  |           |        |                                |           |
| C      | Average thickness of gas film  | 5 × 10⁻⁶ m| µ₀     | Density                        | 1.1452 kg/m³ |
|        | Eccentricity ratio             | 0.6       | ρ₀     |                                |           |

3.2. Model validation

We first validate the accuracy of the proposed calculation method using the results obtained by Cheng et al. [12]. The perturbation ratio γ is plotted with respect to the compressibility coefficient Λₓ in Fig. 3 for an eccentricity ratio of 0.2 and a width-to-diameter ratio of 0.5. The reference value of the perturbation ratio γ is close to 0.5 for all Λₓ, and the results obtained by the proposed method are consistent with those in reference Cheng et al. [12].

![Fig. 3 The comparison between the calculational result and literature](image)

3.3. Effect of rotation speed on the dynamic characteristic coefficients

From Fig. 4(a) the principal stiffness coefficients of the spiral spiral groove initially increased with increasing nr and then decreased until becoming negative for nr values greater than 38000 r/min, indicating that sealing system instability might occur from that point onward. As shown in Fig. 4(b), the cross stiffness coefficients exhibited substantial symmetrical variation, and the differences between the values were particularly large for nr values ranging between 10000 r/min and 20000 r/min and between 40000 r/min and 50000 r/min. Thus, positive work was applied to eddy motion in these intervals, resulting in a negative effect on sealing system stability. The results shown in Fig. 4(c) indicate that the principal damping coefficients underwent complex variation with increasing nr. In contrast, the difference between cross damping values initially increases with nr, and then decreases gradually until the nr value is greater than 38000 r/min and finally tends to zero, as shown in the shaded area in Fig. 4(d).
3.4. Effect of pressure on the dynamic characteristic coefficients  
As shown in Fig. 5(a), both principal stiffness coefficients initially increased with increasing ph to 0.42 MPa, and then decreased until finally obtaining a constant value. For the subsequently decreasing principal stiffness coefficients, according to Fig. 5(b), the difference between the two cross stiffness values began increasing with increasing ph above about 0.34 MPa, and this increasing difference for ph values greater than 0.42 MPa resulted in positive work being applied to eddy motion, which increased system instability. It can be seen from Fig. 5(c) that cxx decreases with the increase of ph, but increases gradually when ph > 0.3 MPa, and tends to be stable when ph > 0.5 MPa. On the contrary, cyy decreases with increasing ph, but increases rapidly when ph exceeds 0.28 MPa, and cyy shows a parabolic distribution until ph reaches 0.42 MPa. However, the cross damping coefficients shown in Fig. 5(d) varied more simply and symmetrically, where the values of cxy and cyx increasingly diverged with increasing ph above 0.42 MPa, and eventually maintained a fairly stable difference.

Fig. 4 Dynamic characteristic coefficients with the rotation speed

Fig. 5 Dynamic characteristic coefficients with pressure
3.5. Effect of eccentricity ratio on the dynamic characteristic coefficients

It can be seen from Fig. 6(a) that with the increase of ε, the main stiffness coefficient increased at first and then decreased. When the value of ε was from 0.64 to 0.78, the main stiffness coefficient become negative. Therefore, the ring spiral groove floating seal held stable when ε was between 0.3 and 0.64 and remains stable when ε > 0.78. The two cross stiffness values were given in Fig. 6(b). Both of them were initially symmetrical, but then began to conform to the same increasing trend with increasing ε greater than 0.72, and the difference between them decreased progressively. As shown by the shaded region in Fig. 6(c), the sum of the principal damping values gradually increased for ε values between 0.64 and 0.72. As shown in Fig. 6(d), the damping coefficients fluctuated around zero, but the difference between the two cross damping values sharply increased for ε = 0.7. According to the observed influences of ε on the principal stiffness, cross stiffness, and principal damping coefficients that the floating ring seal would be more stable with a large eccentricity ratio (ε > 0.7).

![Fig. 6 Dynamic characteristic coefficients with the eccentricity ratio](image)

4. Conclusion

This study established a transient Reynolds formulation and film thickness equation for the floating ring seal in a rotating coordinate system based on the small perturbation method. The dynamic stiffness and damping matrices are thereby obtained. The dynamic characteristic coefficients were obtained under working conditions, and the results and analyses provided the following main conclusions.

1) The circular and symmetrical geometry of the floating ring seal ensured that the cross stiffness coefficients (k_{xy} and k_{yx}) and cross damping coefficients (c_{xy} and c_{yx}) of the gas film were symmetrical and conformed to the requirements for sealing structures. Moreover, the xx and yy components of the principal stiffness coefficients (k_{xx} and k_{yy}) and principal damping coefficients (c_{xx} and c_{yy}) conformed to equivalent trends.

2) The optimal sealing system became unstable for rotation speeds greater than 38000 rev/min. With respect to pressure, the divergent trends observed for the cross stiffness coefficients and the decreasing trends observed for the principal stiffness coefficients led to a decreasing stability of the sealing system with increasing pressure greater than 0.42 MPa. Accordingly, the increase in pressure had a noticeable influence on the pressure flow and shear flow of the seal, resulting in eddy divergence. Large eccentricity was observed to enhance the wedge effect, which resisted non-uniformities in the gas film pressure and thickness distribution, and was beneficial to eddy convergence.
Acknowledgments
This work was financially supported by the National Natural Science Foundation of China (51905480) and the Science and Technology Innovation 2025 Major Projects, China (2020Z112 and 2018B10006)

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