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Reaction Torque of Gas Foil Journal Bearing with Misalignment

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Abstract. In a rotor gas foil bearing system, zero misalignment can usually not be ensured for assembly errors or creep deformation of the base. The tilting attitude of the journal makes lubricant performance worse. To study the performance of a gas foil journal bearing with misalignment, effect of misalignment on gas film thickness is introduced into the lubrication model. The two typical misalignment attitudes of the journal were defined as radial tilting and tangential tilting. The results show that reaction torques increase with the tilting angle of the journal. A larger eccentricity leads to a larger reaction torque. This work provides a reference for prediction about the self-action of gas foil journal bearing for misalignment.

1. Introduction

Gas foil bearing have a broad application prospect in high speed rotating machinery for its compactness, endurance of foreign matter, improvement of damping. It is difficult to ensure zero misalignment in a rotor foil bearing system for assembly errors or creep deformation of the base. Despite of the ability to accommodate misalignments, the tilting attitudes of the journal change the gas film thickness distribution, making lubricant performance worse.

Many studies were about the performance of gas foil bearings with perfect alignment. Heshmat et al.[1, 2] made early applications of numerical method on gas foil bearing analysis. Based on different numerical methods, many factors were added into model to predict the performance of gas bearings. On structure modeling, uniform stiffness model was used in Heshmat's early work[1, 2]. Further, top foil can be treated as beam element[3-6], plate element[7, 8] or shell element[9, 10]. Friction effect and bearing sleeve was also taken into consideration [11-14]. Besides, considering real gas property, variations of gas physical parameters were introduced in. Peng[15] accounted for viscosity-temperature characteristic, and compared numerical results with experimental study. Park et al.[16] involved rarefaction gas coefficients into the lubrication model for foil thrust bearings. Conboy and Wright[21] and Kim[25] presented a foil thrust bearing model with S-CO2 gas lubrication.

Performances of gas foil journal bearings under misalignment operation conditions were studies experimentally. Howard[17] measured the temperature raise and vibration on a rotor foil bearing system test rig with misalignment. In his another work [18], vibration of this rotor system was analyzed by plotting the waterfall diagram under different angular misalignments, wearness of the tested bearing was observed. Feng [19] conducted a series of static and dynamic load tests to measure the structural stiffness and equivalent viscous damping of the prototype multi-cantilever foil bearing with misalignment, find a high level of misalignment can lead to larger static and dynamic bearing stiffness as well as to larger equivalent viscous damping. Gad[20] evaluated the effects of static and dynamic angular misalignments on the performance of a gas foil thrust bearing.

Most of studies about gas foil journal bearings with misalignment were based on experiments. In this paper, effect of misalignment on gas film thickness is introduced into the lubrication model. The two typical misalignment attitudes of the journal were defined as radial tilting and tangential tilting. The reaction torques caused by misalignment are calculated by solving the pressure distribution of gas film numerically. This work provides a reference for the prediction about the self-action of gas foil journal bearings for misalignment.

2. Theoretical model

As shown in Figure 1a), coordinate frame o-xyz is set in the centre of the bearing. θ is the circumferential coordinate. R is the radius of the rotor journal, and h is the film local thickness, c is the bearing clearance, L is the bearing length. A general attitude of the journal respect with the bearing is shown in Figure 1b), the translational motion of the journal is described according to the radial motion of the middle section ($z=0$) centre shown in Figure 1a).
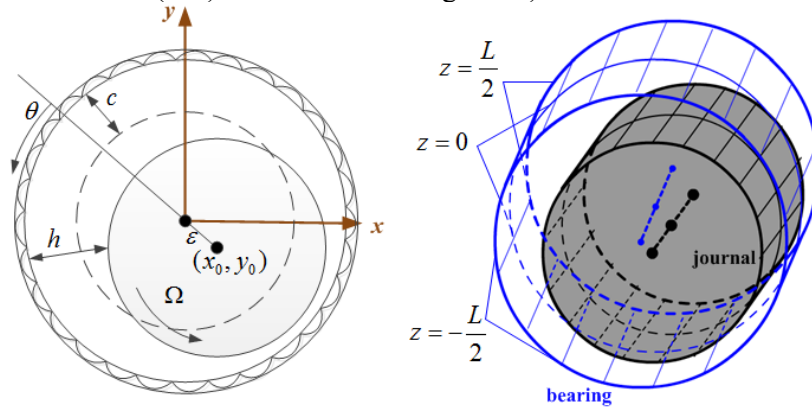
a) Geometry in the middle section ($z=0$)   b) Misalignment in a foil journal bearing
Figure 1 Configuration of gas foil journal bearing

2.1. Hydrodynamic Lubrication Equation

Under laminar flow assumption, Reynolds equation in cylindrical frame is given below.

$$12\frac{\partial(\rho h)}{\partial t}+6\Omega\frac{\partial(\rho h)}{\partial \theta}=\frac{1}{R^2}\frac{\partial}{\partial \theta}\left(\frac{\rho h^3}{\mu}\frac{\partial p}{\partial \theta}\right)+\frac{\partial}{\partial z}\left(\frac{\rho h^3}{\mu}\frac{\partial p}{\partial z}\right) \tag{1}$$

Where, h is the gas film thickness, c is bearing clearance, $u(\theta,z)$ is the deflection of the foil. Gas film thickness $h=h_0+u$ consists of the geometry thickness h_0 and foil deflection u.

Neglecting the transient item, the non-dimensional form of the Eqn.(1) is

$$\frac{\partial}{\partial \theta}\left(\frac{\bar{\rho}\bar{h}^3}{\bar{\mu}}\frac{\partial \bar{p}}{\partial \theta}\right)+\left(\frac{2R}{L}\right)^2\frac{\partial}{\partial \bar{z}}\left(\frac{\bar{\rho}\bar{h}^3}{\bar{\mu}}\frac{\partial \bar{p}}{\partial \bar{z}}\right)=\Lambda\frac{\partial(\bar{\rho}\bar{h})}{\partial \theta} \tag{2}$$

The non-dimensional variables are listed below.

$$\bar{\rho}=\frac{\rho}{\rho_0}, \quad \bar{h}=\frac{h}{c}=\frac{h_0}{c}+\bar{u}(\theta,\bar{z}), \quad \bar{u}(\theta,\bar{z})=\frac{u(\theta,z)}{c}$$
$$\varepsilon=\frac{e}{c}, \quad \bar{\mu}=\frac{\mu}{\mu_0}, \quad \bar{p}=\frac{p}{p_a}, \quad \bar{z}=\frac{2z}{L}, \quad \bar{t}=t\cdot\Omega, \quad \Lambda=\frac{6\mu_0\Omega}{p_a}\left(\frac{R}{c}\right)^2 \tag{3}$$

Where ρ_0,μ_0,p_0 are the density, viscosity, pressure of the ambient gas, respectively. L is the axial length of the bearing. ε is the eccentricity ratio. Λ is the non-dimensional bearing number.

2.2. Structural model

A uniform stiffness model of the foil structure based on Heshmat's[1] is adopted to acquire the deflection of the foil structure, as shown in Figure 2. Assume that the stiffness of the foil is taken to be

uniformly distributed and constant throughout the bearing surface. The overall foil structure stiffness is mainly contributed by bump foil stiffness.
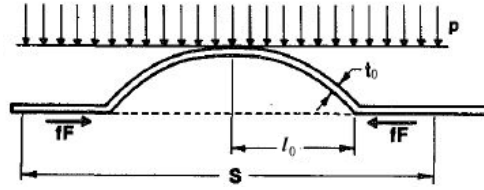

Figure 2 Configuration of bump foil[2]

Under the assumptions above, the relationship between the point pressure and deflection is given by

$$\bar{p} - 1 = \frac{1}{\bar{\alpha}}\bar{u} + \bar{C}_f \frac{\partial \bar{u}}{\partial \bar{t}} \tag{4}$$

Where, $\bar{\alpha}, \bar{C}_f$ are the non-dimensional compliance and dumping coefficients of structure, written as[1]

$$\bar{\alpha} = \frac{1}{k_b}\frac{p_a s}{c}, \quad \bar{C}_f = \frac{C_f}{p_a/(c\Omega)} \tag{5}$$

Where s denotes spacing between adjacent bumps for a given radial section, k_b is the stiffness of the bump foil, $k_b = \left[0.5E_b/(1 - v_b^2)\right] \cdot (t_0/l_0)^3$ with E_b, v_b are the modulus, posisson ratio of the bump foil material, t_b, l_0 are the thickness and length of half bump sector.

2.3. Gas film thickness
Gas film thickness consists of the geometry clearance and foil deflection, the film thickness in the middle section (z=0) is given by

$$h = c + e\cos\theta + u(\theta, z = 0) \tag{6}$$

For the journal is tilting, the attitude of the axial line can by define by the tilting angle θ_x and θ_y as shown in Figure 3. The positions of the journal center in different axial sections can be described by the axial distance and tilting angles.

$$(x, y, z) = (x_0, y_0, 0) + (-z\tan\theta_y, z\tan\theta_x, z) \tag{7}$$

Where (x_0, y_0) is the position of the journal center in the middle section (z=0) described in Figure 1a). Thus, the film thickness is written as

$$h = c - y\cos\theta + x\cos\theta - z\tan\theta_y, z\tan\theta_x + u(\theta, z) \tag{8}$$
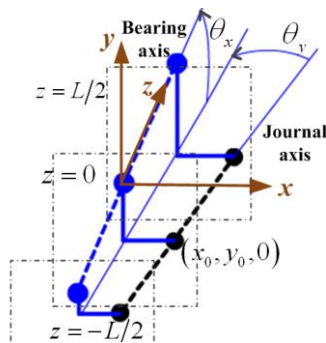

Figure 3 Eccentricities in different axial section of a tilting journal
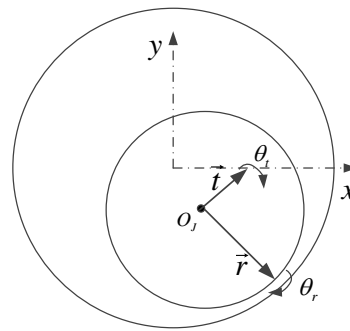
Figure 4 Tilting attitude of journal with two typical rotational axes.

2.4. Boundary conditions
Because the top foil deviates from the bump film under negative pressure, the pressure should be larger than the ambient pressure in the downstream area following the minimum film thickness position. Thus, Reynolds boundary condition is applied as given in Eqn.(7), where the position θ^* at zero pressure point is figured out during the numerical process.

$$\left.\bar{p}\right|_{z=\pm\frac{L}{2}}=1,\quad \left.\bar{p}\right|_{\theta=0}=\left.\bar{p}\right|_{\theta=2\pi},\quad \left.\frac{\partial\overline{p}}{\partial\theta}\right|_{\theta=\theta^{*}}=0 \tag{9}$$

2.5. Bearing Force

In post process, non-dimensional load capacity and torque evaluated by integral on the pressure distribution in the calculating field[2].

$$\bar{F}_{x}=-\int_{-1}^{1}\int_{0}^{2\pi}\left(\bar{p}-1\right)\cos\theta d\theta d\bar{z},\bar{F}_{y}=-\int_{-1}^{1}\int_{0}^{2\pi}\left(\bar{p}-1\right)\sin\theta d\theta d\bar{z},\bar{W}=\sqrt{\bar{F}_{x}^{2}+\bar{F}_{y}^{2}},\bar{T}=\int_{-1}^{1}\int_{0}^{2\pi}\left(\frac{\overline{h}}{2}\frac{\partial\overline{p}}{\partial\theta}+\frac{\Lambda}{6\overline{h}}\right)d\theta d\bar{z} \tag{10}$$

Reaction torque caused by misalignment is

$$\bar{M}_{x}=-\int_{0}^{2}\bar{z}\int_{0}^{2\pi}\left(\bar{p}-1\right)\cos\theta\cdot d\theta d\bar{z},\bar{M}_{y}=-\int_{0}^{2}\bar{z}\int_{0}^{2\pi}\left(\bar{p}-1\right)\sin\theta\cdot d\theta d\bar{z},\quad \bar{M}=\sqrt{\bar{M}_{x}^{2}+\bar{M}_{y}^{2}} \tag{11}$$

Recovering dimensional form of the load capacity and equivalent torque are shown below.

$$W=\bar{W}\cdot p_{a}R\frac{L}{2},\quad T=\bar{T}\cdot p_{a}cR\frac{L}{2},\quad M=\bar{M}\cdot p_{a}cR\frac{L^{2}}{4} \tag{12}$$

3. Results

To analyze the gas film and the performance of the foil journal bearing with misalignment, two kinds of typical tilting attitude of are defined according to the rotational axis of the journal as shown in Figure 4: a) Radial tilting, rotation around a radial axis along eccentric line, shown in Figure 5b); b) Tangential tilting, journal centers in all sections are located on the eccentric line, shown in Figure 5a).
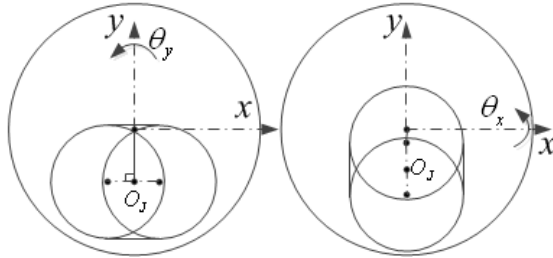

Figure 5 Typical attitudes with misalignment: a) Radial tilting; b) Tangential tilting

Table 1 Data used in the simulation	
Foil bearing parameters	
Bearing diameter (2*R*)	38.1×10^{-3}m
Bearing length (*L*)	38.1×10^{-3}m
Diametral clearance (2*C*)	63.6×10^{-6}m
Rotational speed(Ω)	30000r/min
Pressure(p_a)	101325Pa
Viscosity(μ)	1.932×10^{-5}Pas
Density(ρ_0)	1.29kg/m^3

The foil bearing configuration and lubrication parameters used in this simulation are listed in Table 1. To simplify the input for the simulation, the eccentricity is set only in vertical direction with(x_0, y_0)=(0,e_y) shown in Figure 5.

Figure 6 presents the film thickness distribution and pressure distribution of a typical misalignment with radial tilting journal corresponding to Figure 5a). From Figure 6a), the minimum film thickness positions are distributed along the tilting journal axial direction. The gas film achieves thinnest near two bearing edges. Figure 6b) shows the pressure distribution of the misalignment with radial tilting. Different from the symmetrical pressure distribution with perfect alignment, two pressure peaks emerge along the tilting direction. It is deduced that the two peaks are not at the same height due to the existence of eccentricity.

Figure 7 presents the film thickness distribution and pressure distribution of another typical misalignment with tangential tilting journal corresponding to Figure 5b). From Figure 7a), the film thickness shows a tilting surface along the tilting journal axial direction. The gas film achieves thinnest near an edge of the bearing. Figure 7b) shows the pressure distribution of the misalignment with tangential tilting. Different form the symmetrical pressure distribution with perfect alignment, the pressure peak moves close to an edge of the bearing.

The reaction torque caused by the misalignment varies with the tilting angle. The resultant reaction torque variations are given in Figure 8. It is shown that resultant reaction torques increase with the tilting angles. Compared with tangential tilting, radial tilting usually leads to a larger reaction

torque. Small tilting angle leads to a very thin gas film under large eccentricities, the curves with $e_x = 0.6$ and $e_x = 0.8$ contain only several data points in this simulation.
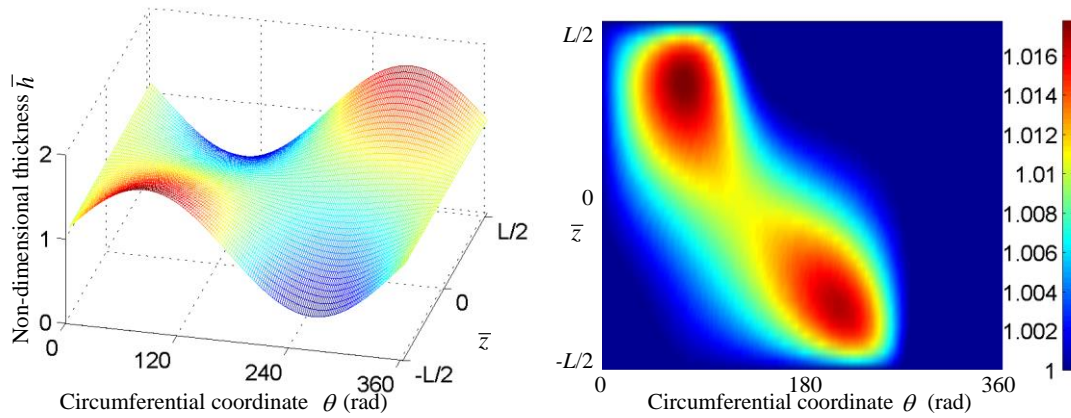

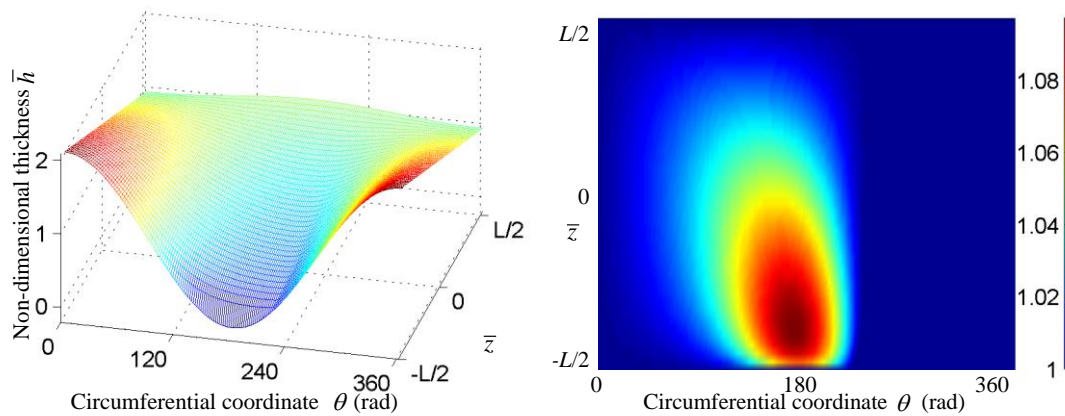Figure 6 Non-dimensional thickness and pressure distribution ($e_y = -0.1, \theta_y = 1 \times 10^{-3}$ rad)

Figure 7 Non-dimensional thickness and pressure distribution ($e_y = -0.5, \theta_x = 9 \times 10^{-4}$ rad)
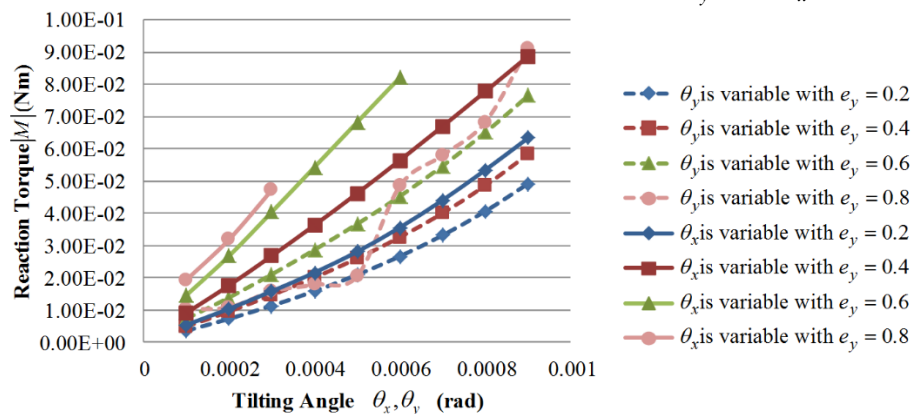

Figure 8 Reaction torque M v.s. tilting angles with different vertical eccentricities

4. Conclusions

Misalignment in a foil journal bearing system makes lubricant performance worse. In this paper, effect of misalignment on gas film thickness is introduced into the lubrication model. Main conclusions are summarized as follow.

a)   The two typical misalignment attitudes of the journal were defined as radial tilting and tangential tilting.

b) Two pressure peaks may emerge in some radial tilting attitudes conditions. For tangential tilting attitude, the unique pressure peak moves toward the edge of the bearing.

c) The reaction torques caused by misalignment increase with the tilting angle of the journal. A larger eccentricity leads to a larger reaction torque.

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