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Research on static performance of hydrodynamically lubricated thrust slider bearing based on periodic harmonic

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ABSTRACT

In this paper, a universal methodology for investigating the static performance of hydrodynamically lubricated thrust slider bearing based on periodic harmonic is put forward. Because the lubricant film thickness reflects directly the bearing topology structure, the dimensionless lubricant film thickness in the circumference direction is expressed by harmonic functions. Next, each dimensionless pressure distribution related to each lubricant film thickness expanded term is researched. Finally, each dimensionless load capacity related to each pressure distribution is calculated. The results show the inherent relevance among lubricant film thickness expanded term and pressure field and load capacity.

1. Introduction

Hydrodynamic bearings occupy an important place in the field of aerospace, civil turbomachinery and so on. The lubrication conditions of a hydrodynamic bearing directly affect the performances of machines. With the rapid development of modern industry, the advanced rotating machinery requires bearings to have higher overall performances.

Though there are many kinds of circular and non-circular hydrodynamic bearings used in the industry for a long time, the current bearing design methods still depend on experience. Especially, there is little research about non-circular hydrodynamic bearing design theories. The conception of and research into wave bearing provided some inspiration in studying and building up the non-circular hydrodynamic bearing design theories. Dimofte [1,2] introduced the wave bearing and researched bearing performance characteristics. Ene et al. [3] studied theoretically and experimentally the influence of wave amplitude and supply pressure on the dynamic behavior of a hydrodynamic three-wave journal bearing. Ene et al. [4] experimentally investigated the attenuation of the gear mesh noise/vibration by fluid film wave bearings relative to rolling element bearings. Shamoto et al. [5] proposed a new non-contact fluid bearing which utilizes traveling waves and whose bearing force or clearance can be controlled electrically. These studies show that compared with the traditional fluid film bearings, wave bearings have many advantages. In fact, various circular and non-circular hydrodynamic bearings may have some common laws characterized by the periodic harmonics of their topology structures. So those bearings can be analyzed and processed uniformly and on this basis, the universal bearing design rules can be proposed. Li et al. [6] firstly built up the corresponding relation based on periodic harmonic between the components of the oil film thickness and pressure distribution of multi-lobe bearings. His study had a potential value in revealing the common laws about topology structure of multi-lobe bearing and in the design and analysis of the non-circular bearings.

Hydrodynamically lubricated thrust slider bearings are common non-circular bearings and often used in rotating machinery to support thrust loads and to axially locate rotors in operation. Much current research on the thrust slider bearings have focused on bearing lubrication problem and topological shape optimization problem. Oladeinde et al. [7] studied the effect of power law fluid on the static performance of a fixed inclined slider bearing by using the finite element method. Liu et al. [8] derived a set of analytical solutions to pressure, load capacity, flow rate and torque loss for hydrodynamic lubrication problems of the fan-shaped thrust step bearing and studied the effects of inner radius, step height and step location on pressure distribution and load capacity. Papadopoulos et al. [9] presented a computational study of performance indices in a dimpled parallel thrust bearing, considering both isothermal and nonisothermal flow. Fesanghary et al.

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Nomenclature

r	coordinate in the radial direction
R	dimensionless coordinate in the radial direction
θ	coordinate in the circumferential direction
h	lubricant film thickness of a hydrodynamically lubricated thrust slider bearing
H	dimensionless lubricant film thickness of a hydrodynamically lubricated thrust slider bearing
H_k	$(k+1)$th term of the expansions of H
h_{in}	inlet lubricant film thickness
h_{out}	outlet lubricant film thickness
A_k,B_k	Fourier coefficients
p	lubricant film pressure
P	dimensionless lubricant film pressure
P_k	$(k+1)$th term of the expansions of P
MP_k	maximum dimensionless pressure value of P_k
SP_k	sum of the first k pressure components
DP_k	difference between P and SP_k
W	dimensionless bearing load capacity
W_k	dimensionless bearing load capacity related to H_k
r_1	inner radius
r_2	outer radius
θ_t	end position coordinate of the tapered area
θ_p	pad angle
N	number of pads
μ	dynamic viscosity
ω	rotational speed

[10] presented an optimization method based on the sequential quadratic programming for hydrodynamic film shape that provides the greatest load capacity in sectorial-shape thrust bearings. Gohar et al. [11] presented the design method to define the optimum geometry of a 12 pad tapered-land thrust bearing, taking into account varying viscosity and hot oil carry over. Zhang et al. [12] proposed a reference to determine the water-lubricated step thrust bearing dimensions and a formula to check the minimum water film thickness. Zhu et al. [13] optimized the design parameters of herringbone, step-pocket, taper-pocket and taper-flat thrust bearings in terms of both maximum axial stiffness and maximum ratio of axial stiffness to friction torque and calculated bearing performance for both cases. Song et al. [14] analyzed a tapered-land thrust bearing with no oil grooves by solving the traditional Reynolds equation and 3D Navier–Stokes equation with a cavitation model, and the predicted results were examined against the experimental results. Karadere [15] studied the effects of the pad and runner deformations on the performance of fixed inclined thrust bearings. Almqvist et al. [16] developed a THD model of hydrodynamic thrust bearings, and the theoretical results were compared against the experimental results. Yuan et al. [17] developed a three-dimensional TEHD model for analyzing the effects of the rotational speed, the oil feed temperature and the surface profile of the pad on pivoted pad thrust bearing performance and discussed the optimization of the pad surface profile. Wu et al. [18] studied the relationship between dimensionless load capacity and outlet film thickness and the relationship between dimensionless load capacity and slope coefficient. Dadouche et al. [19] studied experimentally and theoretically the influence of the oil supply temperature, the bearing load, and the rotational speed on a fixed geometry thrust bearing performance characteristics such as temperature, oil flow, power loss and minimum film thickness.

Though there are many research results in the field of hydrodynamically lubricated thrust slider bearing, very few studies make comparisons among those bearings having various profiles, in order to make a unified explanation for the differences of bearing performances. In this paper, a new methodology is presented to evaluate the static performance of a fixed inclined thrust slider bearing and a tapered-land thrust slider bearing based on periodic harmonic. The goal of the research is to study the influence mechanism of periodic harmonics to static performance of the hydrodynamically lubricated thrust slider bearing and so reveal the inherent relevance between bearing static performance and bearing topology structure.

2. Mathematical model

2.1 Fourier series of lubricant film thickness

The configurations of a fixed inclined and a tapered-land thrust slider bearing are shown in Fig. 1. A fixed inclined thrust slider bearing has a fixed inclined upper surface while a tapered-land thrust slider bearing has a tapered bottom. There are some deep grooves to separate the entire bearing surface into several independent partial bearings so that only positive pressure generates on the bearing surface.

In the study, the lubricant film thickness is assumed to be constant along the radial direction, that is, the tilt of the thrust disc is disregarded. As a result, the lubricant film thickness is a function of θ.

For different configurations, the expressions for the lubricant film thickness between the pad and thrust disc are as follows:

for a fixed inclined thrust slider bearing,

$$h = h_{\text{out}} + h_s - h_s\frac{\theta}{\theta_p} \quad 0 \le \theta \le \theta_p \tag{1}$$

for a tapered-land thrust slider bearing,

$$h = \begin{cases} h_{\text{out}} + h_s - h_s\frac{\theta}{\theta_t} 0 \le \theta \le \theta_t \\ h_{\text{out}} \theta_t \le \theta \le \theta_p \end{cases} \tag{2}$$

where $h_s = h_{\text{in}} - h_{\text{out}}$. Introducing the following dimensionless quantity $H = h/h_s$, Eqs. (1) and (2) can be written as shown in Eqs. (3) and (4).

$$H = 1 + \frac{h_{\text{out}}}{h_s} - \frac{\theta}{\theta_p} \quad 0 \le \theta \le \theta_p \tag{3}$$

$$H = \begin{cases} 1 + \frac{h_{\text{out}}}{h_s} - \frac{\theta}{\theta_t} 0 \le \theta \le \theta_t \\ \frac{h_{\text{out}}}{h_s} \theta_t \le \theta \le \theta_p \end{cases} \tag{4}$$

Obviously, H whose domain is [0, θ_p] is an aperiodic function. It can be expanded to Fourier series as Eq. (5) on its well-defined interval.

$$H = \sum_{k=0}^{\infty} H_k = \frac{A_0}{2} + \sum_{k=1}^{\infty} \left[A_k \cos \frac{2k\pi\theta}{\theta_p} + B_k \sin \frac{2k\pi\theta}{\theta_p} \right] \tag{5}$$

for a fixed inclined thrust slider bearing,

$$A_0 = 1 + \frac{2h_{\text{out}}}{h_s} \tag{6}$$

$$A_k = 0 \quad (k = 1, 2, 3, \cdots) \tag{7}$$

$$B_k = \frac{1}{k\pi} \quad (k = 1, 2, 3, \cdots) \tag{8}$$