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Stability analysis of hydrodynamic bearing with herringbone grooved sleeve

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ABSTRACT

This study presents a numerical method to investigate the stability analysis of a hydrodynamic journal micro-bearing. The governing dimensionless Reynolds equation is solved by using the finite difference method. The Green's theorem is utilized to deal with the discontinuity of the oil-film thickness, and the Gaussian elimination method is used to solve the simultaneous discrete equations. Optimal values for various design parameters are obtained to maximize the load capacity and to improve the stability characteristics. The accomplishments of this study will help the designers who deal with the grooved micro-bearing to select the design parameters to satisfy the required characteristics.

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1. Introduction

The herringbone grooved journal bearing (*HGJB*) is designed for micro-spindle in order to improve the lubrication characteristics and it gradually replaces the early design of plain journal bearing (*PJB*) used in the storage media, such as CD, VCD, and DVD. However the bearing characteristics and spindle stability are always worse than those of a *PJB* except the *HGJB* operates under low eccentricity conditions. This deterioration is caused by the saw-toothed pressure distribution generated by the grooves located on the internal surfaces of a *HGJB*.

The narrow groove theory (*NGT*) was used to analyze the *HGJB* by early researchers; it is assumed that the number of grooves in the bearing is infinite and the pressure distribution induced by the grooves is simplified as the mean value of the saw-toothed pressure distribution, which is higher than that in a *PJB*. Hirs [1] applied the *NGT* to obtain the pressure distribution, friction moment, and stability of a *HGJB*. Bootsma [2,3] investigated load capacity and stability of a *HGJB* for different shapes of spiral grooves including spherical and conical profiles by the *NGT*. They both compared the analysis results with experimental data.

Kang et al. [4] applied the finite difference method with the help of conservation of mass, Grümbel boundary conditions, and coordinate transformation to analyze the static and dynamic characteristics of a *HGJB*. The results showed that the stability of a *PJB* at low eccentricity can be improved by applying either rectangular or circular grooves. Jang and Yoon [5,6] analyzed the difference of static and dynamic characteristics between a grooved bearing with plain journal (*GBPJ*) and a plain bearing with grooved journal (*PBGJ*); the results revealed that the whirl vibration can be induced in *PBGJ* because the groove number multiplying the rotating speed can be an excitation frequency to disturb the journal leaving its equilibrium position. Lee and his colleagues [7–9] modified Elrod's method [10] and applied coordinate transformation to analyze three different groove designs in the investigation of the footprints in cavitation distributions within a *HGJB*. In addition, they investigated the differences between *HGJB* and *PJB* to conclude that *HGJB* is better than *PJB* on the operating stability and the region of cavitation distributions is also decreased. Rao and Sawicki [11] used the perturbation method and coordinate transformation to solve Elrod's cavitation equation to study the static load capacity, dynamic stiffness and damping coefficients, stability threshold, and critical whirl ratio to yield that *HGJB* is better than *PJB* at a low eccentricity ratio.

This study applies the finite difference method associated with Gaussian elimination method to solve the lubricant film Reynolds equation with the Grümbel boundary conditions. Except Kang et al., the aforementioned researchers all use other various numerical methods. The Gaussian elimination method is faster than an iterative method in solving a set of simultaneous equations. This study will also pay attention to the characteristics of the lubricant flow both in the regular and inverse directions. The aim of this study is to choose and optimize the design parameters of a *HGJB* to establish the design rules and analysis techniques for the miniature spindle motors with high rotating speed.

2. Finite difference method for solving perturbation of oil film

For a hydrodynamic *HGJB* as shown in Fig. 1, the herringbone grooves with the rectangular cross section having depth h_g and

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Nomenclature

$A_{ij,k}$	applied by eight point discrete grid method; the eight points are separated equally on the closed border $\Gamma_{ij,l}$ surrounding the chosen grid node G_{ij}, where $k=1$–8 and $l=1$–4
B_{ij},$\overline{B}_{ij}$	the normal and cross damping coefficients and their dimensionless forms $\overline{B}_{ij}=B_{ij}c\omega/P_aLD$ (i, j=x, y) in the Cartesian coordinate, and (i,$j=\varepsilon,\phi$) in the perturbation coordinate
c	radial clearance (m)
c_d	decaying exponent of perturbed whirl
D, R	diameter and radius of bearing (m)
e_o,e_p	static eccentricity and radial displacement of perturbed whirl (m)
F_f,$\overline{F}_f$	friction force due to shear stress within the film (N), and its dimensionless form, $\overline{F}_f=F_f/P_aLD$
F_x,F_y	force components due to lubricant film pressure in the x and y directions (N)
G_{ij}	grid node with the position coordinate (i, j)
h,$\overline{h}$	local film thickness (m), and its dimensionless form, $\overline{h}=h/c$
h_o,$\overline{h}_o$	static film thickness (m), and its dimensionless form, $\overline{h}_o=h_o/c$
h_g	groove depth (m)
I_c	the threshold of instability, $I_c=\overline{m}_c\lambda_c^2$
K_{ij},$\overline{K}_{ij}$	the normal and cross stiffness coefficients and their dimensionless forms $\overline{K}_{ij}=K_{ij}c/P_aLD$ (i, j=x, y) in the Cartesian coordinate, and (i,$j=\varepsilon,\phi$) in the perturbation coordinate
L	axial length of bearing (m)
l_g, l_r	groove and ridge width (m)
$\overline{m}$,$\overline{m}_c$	dimensionless mass parameter, $\overline{m}=mc\omega^2/2P_aLD$; and critical value for $\overline{m}$
N	number of herringbone grooves
O_b,O_j	bearing center, and journal (spindle) center
P,$\overline{P}$	the pressure distribution of lubricant film (N/m^2), $\overline{P}=P/P_a$
P_a	atmospheric pressure (10^5 N/m^2)
P_k^l,$\overline{P}_k^l$	lubricant film pressure distributions (N/m^2); $\overline{P}_k^l=P_k^l/P_a$; subscript k denoted for o (static), ε and ϕ (for perturbation); superscript l denoted for real part by R, and for imaginary part by I
Q_η,$\overline{Q}_\eta$	volumetric flow rate (m^3/s) of lubricant, and its dimensionless form $\overline{Q}_\eta=Q_\eta\left(P_ac^3/12\mu R\right)^{-1}$, $\eta=c,s$ for the circumferential flow and the side leakage
s	characteristic frequency ratio of the journal motion, $s=(c_d+i\omega_p)/\omega$
t	time (s)
W_k,$\overline{W}_k$	unidirectional load capacity (N), and its dimensionless form, $\overline{W}_k=W_k/P_aLD$, $k=x,y$ in the Cartesian coordinate, and $k=\varepsilon,\phi$ in the perturbation coordinate
$\overline{X}$,$\overline{Y}$	dimensionless amplitudes of whirl motion in x and y directions, $\overline{X}=X/c$, $\overline{Y}=Y/c$
x, y, z	Cartesian coordinates of lubricant film

Greek symbols

θ,$\overline{y}$,$\overline{z}$	dimensionless coordinates, $\theta=x/R$, $\overline{y}=y/c$, $\overline{z}=z/(L/2)$
$\tilde{\theta}$	relative angle measured from the line connecting equilibrium positions of journal center and bearing center, $\tilde{\theta}=\theta-\phi_0$
Γ_{ij},$\Gamma_{ij,k}$	the virtual closed border and its four segments surrounding the grid node G_{ij}, $k=1$–4
Λ	bearing parameter, $\Lambda=6\mu\omega/P_a(c/R)^2$
Ω_{ij}	the computational mesh of the node G_{ij} surrounded by $\Gamma_{ij,k}$
α	groove width ratio, $\alpha=l_g/(l_r+l_g)$
β	groove angle
γ	groove depth ratio, $\gamma=h_g/c$
δ_k	criterion values of convergence, $k=1$–3
ε_0,ε_p	eccentricity ratios, $\varepsilon_0=e_0/c$ for static equilibrium, and $\varepsilon_p=e_p/c$ for perturbation
ϕ_o,ϕ_p	static attitude angle, and angular displacement of perturbed whirl
λ,λ_c	whirl frequency ratio, $\lambda=\omega_p/\omega$; and the critical one, $\lambda_c=\omega_p/\omega_c$
μ	dynamic viscosity of lubricant (N s/m^2)
μ_f	the friction coefficient, $\mu_f=\overline{F}_f/\overline{W}$
τ	dimensionless time, $\tau=\omega$t
ω	rotating speed of spindle (rad/s)
ω_p	whirl frequency of perturbed motion (rad/s)

width l_g are arranged with bilateral symmetry along the z-axis. The groove angle β is positive when the direction of circumferential flow Q_c is the same as shown in this figure, and β is negative when the direction of Q_c is inverse. The Reynolds number for this *HGJB* at the maximum rotational speed is

$$\mathrm{Re}=\frac{\rho VD}{\mu}=\frac{886\times0.002\times7500\times0.004}{0.036}=1476.67,$$

which is less than the upper limit of the value: Re $=2300$ for laminar flow. With the assumptions of incompressible and laminar flow for lubricant, which is fed into the clearance between the plain journal and the grooved sleeve, and using the following dimensionless parameters: $\overline{h}=h/c$, $\overline{P}=P/P_a$, $\theta=x/R$, $\overline{z}=z/L/2$, and $\tau=\omega t$, the governing dimensionless Reynolds equation can be expressed in the θz plane as

$$\frac{\partial}{\partial\theta}\left(\overline{h}^3\frac{\partial\overline{P}}{\partial\theta}\right)+\left(\frac{D}{L}\right)^2\frac{\partial}{\partial\overline{z}}\left(\overline{h}^3\frac{\partial\overline{P}}{\partial\overline{z}}\right)=\Lambda\left(\frac{\partial\overline{h}}{\partial\theta}+2\lambda\frac{\partial\overline{h}}{\partial\tau}\right) \quad (1)$$

where $\Lambda=6\mu\omega/P_a(c/R)^2$ is the bearing parameter, $\lambda=\omega_p/\omega$ is the whirl frequency ratio, ω_p is the journal whirl frequency, ω is the journal rotating speed, and L and D are the length and diameter of the bearing, respectively.

Because of the bilateral symmetry of the herringbone grooves, the calculating domain of Eq. (1), which is only the lower half of the lubricant film as shown in Fig. 1(b), can be numerically solved to obtain its load capacity, side leakage, total friction force, and dynamic characteristics. As the results are multiplied by two the exact complete characteristics of this film are obtained. It is divided into $m\times n$ grids as shown in Fig. 2, with m nodes along the lateral axis and n nodes along the vertical axis for numerical computations. The node spans are $\Delta\theta=2\pi/m$ in the θ-axis and $\Delta\overline{z}=1/n$ in the z-axis. Three representative grids A, B and C, located on the interior, the symmetric axis and the periphery of the calculating domain respectively, and the details of their grid structure are shown in Fig. 3, where ◉ refers to a grid node and • refers to the eight points of neighborhood in the eight-point discrete grid [12]. As shown in Fig. 3, Ω_{ij} contains node G_{ij} that is

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