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Improvement on load performance of externally pressurized gas journal bearings by opening pressure-equalizing grooves

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ABSTRACT

To improve the load performance of externally pressurized gas journal bearings (EPGJBs), the influences of structural parameters of pressure-equalizing grooves (PEGs) such as the length, depth, number and location on the load capacity and stiffness are studied systematically. It is found that the depths of the circumferential PEGs have great impacts on the load performance. Compared with opening the circumferential PEGs, opening the axial PEGs is more helpful to improve the load capacity, even opening only one or two axial PEGs. The layout of the axial PEGs has great influence on the load performance. A group of experiments were performed to verify the simulation results, it is found that the experimental results were in agreement with the simulation results.

1. Introduction

Externally pressurized gas bearings (EPGBs) are widely used in precision and high speed systems due to their unique and absolute advantages of low friction loss, high accuracy and long life. However, these advantages cannot always be fully realized due to low load capacity and stiffness. Many methods such as increasing gas supply pressure, adopting active compensation, and increasing the number of orifices have been adopted by researchers to increase the load capacity and stiffness of EPGBs. Sun et al. [1] studied the supersonic flow field of aerostatic thrust bearings with high gas supply pressure for promoting load capacity. However, Talukder and Stowell [2] found that the aerostatic bearings are prone to pneumatic hammer vibration with the increase of gas supply pressure. Mizumoto et al. [3] developed an active control restrictor consisting of a piezoelectric actuator, the orifice area of which can be changed actively according to displacement of the journal supported by EPGBs. The gas film pressure of EPGBs can be changed to improve load capacity due to the variable orifice area. Aguirre et al. [4] designed the deformable bearing surfaces which can be driven by piezoelectric actuators to change the gas film clearance for high stiffness and accuracy. However, the complexity of active compensation limits the application of EPGBs. Chen et al. [5] indicated that the load capability and stiffness can be improved by increasing the number of orifices. However, the improvement effects of load capacity are no longer significant when the number of orifices exceeds a certain value.

Li and Ding [6] studied the influence of the chamber's volume at the outlet of orifices on the load capacity. They found that the load capacity can be improved by increasing the chamber's diameter, but the chamber's depth has little influence on load capacity. When the diameter and depth of the chamber are too big relative to the orifice diameter, the supersonic gas flow will appear as a factor of instability. Chen and He [7] studied the effects of the recess shape on performance of the gas-lubricated bearing and found that the rectangular recess can provide larger load capacity than the spherical recess. Belforte et al. [8] evaluated the discharge coefficients of orifice-type restrictor for aerostatic bearings by experiments. Renn and Hsiao [9] modified the critical pressure ratio of orifice-type restrictor based on the results of experiments and CFD simulations. Fourka and Bonis [10] compared the difference between orifice and porous feeding. They found that the number of feedholes should be between 8 and 12 and porous feeding can improve load capacity. But the manufacturing of gas bearings with porous feeding is difficult. Pande and Somasundaram [11] investigated the influence of manufacturing errors on EPGJBs and indicated that manufacturing and assembly errors can decrease load capacity. Kazimierski and Trojnarski [12]

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Nomenclature

c	radial clearance of bearings (m)
D	diameter of bearings (m)
d	orifice diameter (m)
e,ε	eccentricity (m), $\varepsilon = e/c$
$\mathbf{F}$	array of gas film pressure square to solve
$h, \overline{h}$	gas film thickness (m), $\overline{h} = h/c$
h_{gc}	depth of circumferential PEGs
h_{ga}	depth of axial PEGs
$\mathbf{K}$	integrated stiffness matrix of equation set for solving gas film pressure
K_w	stiffness of bearing (N/m)
k_{ij}	element of ith row and jth column of matrix $\mathbf{K}$
L	axial length of bearing (m)
l	axial length of PEGs (m)
$\dot{m}_r$	gas mass flow of rth orifice determined by gas pressure at the outlet of orifice $\overline{p}_r$ (kg/s)
N_i	shape function of triangular element, $i=i,j,m$
$p, \overline{p}$	gas film pressure of bearings (Pa), $\overline{p} = p/p_a$
p_0	gas supply pressure (Pa)
$p_a, \overline{p}_a$	ambient pressure (Pa), $\overline{p}_a = 1$
$p_r, \overline{p}_r$	gas film pressure at the outlet of rth orifice (Pa), $\overline{p}_r = p_r/p_a$
$\overline{Q}$	flow factor, $\overline{Q} = 24\mu\dot{m}_r/(c^3 p_a \rho_a)$
$\mathbf{T}$	constant array of equation set for solving gas film pressure
W	load capacity of bearings (N)
$z, \overline{z}$	axial coordinates of bearing (m), $\overline{z} = z/(D/2)$
δ_r	delta function, $\delta_r = \begin{cases} 1, & \text{at orifice} \\ 0, & \text{at non-orifice} \end{cases}$
θ	angle of circumferential PEGs (deg)
μ	dynamic viscosity of gas lubricant (N s m^2)
ρ	density of gas lubricant (kg/m^3)
ρ_a	density of ambient gas (kg/m^3)
φ	angular coordinates of bearing
Ω	computational domain of bearing for pressure distribution
Ω_i	an element including node i, $i=i, j$
Ω^e	computational domain of an element

compared the difference between pocketed and annular orifice, and indicated that the bearing stiffness with pocketed orifice is bigger than that with annular orifice. Chen et al. [13] analyzed the stiffness of bearing designs with various geometric parameters, including ratio of length to diameter, orifice diameter, clearance and supply pressure for a high-speed aerostatic spindle. Brzeski and Kazimierski [14] developed a kind of EPGJB with complex structure, which can compensate for shaft displacement by the use of a float bushing to realize high stiffness.

Opening pressure-equalizing grooves (PEGs) on the bearing surface of EPGBs can inhibit the pressure decay of gas film away from orifices so as to improve the load capability and stiffness of EPGBs; some studies have been conducted in connection with this. Chen et al. [15] applied PEGs to arc-shaped gas aerostatic guideway and studied the influence of width and depth of PEGs on load capacity; however, the study did not take account of the influence of layout and number of PEGs. Zhang and Fang [16] proposed a flexible PEG technology, in which the bearing surface can be deformed by gas film pressure to form PEGs for enhancing the load capability and stiffness. Flexible PEG technology requires a complicated manufacturing process. Du et al. [17] studied the pneumatic hammer vibration of aerostatic thrust bearings with circumferential PEG and deduced the stability criteria. Chen et al. [18] indicated that the grooved aerostatic thrust bearing has higher load capacity and greater stiffness than that without grooves by the resistance network method, and increasing the groove width and depth can enhance load capacity. Belforte et al. [19] studied the aerostatic thrust bearing with the circumferential groove of the different depths. They indicated that the stiffness of the thrust bearing increases with increase of both supply pressure and groove depth, and orifice diameter at maximum stiffness decreases with the increase of groove depth. Their research results also show that the thrust bearing appears stable with a groove depth less than 20 μm, and the air hammer appears with groove depth of 20 μm at the supply pressure more than 0.76 MPa. Kogure et al. [20] through studying of the dynamic characteristics of T-shaped groove bearing, indicated that the surface-restriction compensated gas bearing with T-shaped grooves has the advantages of good damping performance and manufacturing ease as compared with the I-shaped groove bearing. Belforte et al. [21] found that the groove presence increase the load capacity and stiffness especially at the low clearance by analyzing a aerostatic pad bearing with circumferential groove, and also pointed out that the groove can avoid the effect of negative stiffness that may be experienced in pads without groove for very low clearance. However, research on PEGs has mainly focused on externally pressurized gas thrust bearings and guideways, while little attention has been paid to EPGJBs.

In EPGJBs, there are many structure options for PEGs. They can be opened either in circumferential or axial direction; the number of the PEGs can be more or less; the position, length and depth of the PEGs can be varied. And for EPGJBs, whatever kind of PEG is opened, the pressure distribution in whole bearing will be affected because the gas film is connected in the circumferential direction. However, rarely has a study systematically examined the influence of various PEGs on the load capacity of EPGJBs. Aiming at EPGJBs with single-row and double-row pocketed orifices, this study systematically evaluated the influences of the number, position, length and depth of PEGs on the load capacity and stiffness by using finite element numerical simulation, and verified the results with a series of experiments. The findings provide helpful suggestions for designing the EPGJBs with high load capacity and stiffness.

2. Mathematical model

The dimensionless Reynolds equation of EPJGBs can be described in the following form [22]

$$\frac{\partial}{\partial \varphi}\left(\overline{h}^3 \frac{\partial \overline{p}^2}{\partial \varphi}\right) + \frac{\partial}{\partial \overline{z}}\left(\overline{h}^3 \frac{\partial \overline{p}^2}{\partial \overline{z}}\right) + \overline{Q}\delta_r = 0 \tag{1}$$

where $\overline{p}$ is the dimensionless gas film pressure, $\overline{h}$ is the dimensionless gas film thickness, φ is the angular coordinates, $\overline{z}$ is the dimensionless axial coordinates, $\overline{Q}$ is the flow factor, and δ_r is the Dirac function.

The boundary conditions are as follows:

(1) Gas film pressure at the both open ends of the bearing are equal to ambient pressure $\overline{p}_a$, i.e., $\overline{p}(\varphi, \pm L/D) = \overline{p}_a$.
(2) Gas film pressure at the outlet of orifice $\overline{p} = \overline{p}_r$, and $\overline{p}_r$ determines the gas mass flow rate of rth orifice $\dot{m}_r$.
(3) Gas film pressure $\overline{p}$ is a periodic function of φ, i.e., $\overline{p}(\varphi, \overline{z}) = \overline{p}(2\pi + \varphi, \overline{z})$.