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Static characteristics of a water-lubricated hydrostatic thrust bearing using a membrane restrictor

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ABSTRACT

It is well known that water does not pollute the environment as much as oil, but it has lower viscosity. We investigate the static characteristics of a water-lubricated hydrostatic thrust bearing using a membrane restrictor to achieve higher stiffness and lower power consumption at high speeds. The static characteristics of the proposed hydrostatic thrust bearing were studied numerically and experimentally for a nonrotating shaft. In the numerical calculations, the orifice effect was taken into account at the inlet of the viscous resistance area of the membrane restrictor. We found that the proposed water-lubricated hydrostatic thrust bearing could achieve a very high static stiffness by using the membrane restrictor, and the numerical and experimental results showed good agreement.

1. Introduction

Hydrostatic bearings have been successfully applied to ultra-precision machine tools, which often require higher stiffness to improve their accuracy of motion. However, when accuracy of the order of a micron or submicron is required, the bearing stiffness in conventional hydrostatic bearings may be insufficient. To improve their stiffness and accuracy, a passively controlled restrictor using a membrane has been proposed by researchers.

Mohsin [1] studied hydrostatic bearings using a membrane restrictor to improve their static and dynamic characteristics. Rowe [2,3] and co-workers published patents related to controlled restrictors for hydrostatic bearings. These controlled hydrostatic bearings could enhance the bearing stiffness but reduced the reliability of the machine tools. Accordingly, hydrostatic bearings with controlled restrictors have rarely been implemented in actual machine tools. However, in the 2000s, some machine tool suppliers applied hydrostatic bearings with controlled restrictors to commercially available machine tools to achieve higher bearing stiffness and to improve the accuracy of movement of a working table. Since then, hydrostatic bearings with controlled restrictors have attracted considerable attention as a means of improving the precision of machine tools. Several researchers studied hydrostatic bearings with membrane restrictors in the 2000s [4,5,7].

Singh et al. [4] and Vikas et al. [5] investigated the static characteristics of multirecess hybrid journal bearings using a membrane restrictor, taking into account the effects of the shapes of the recess and bearing. The membrane restrictor treated in [4,5] consisted of a circular membrane supported by a circular flat base, with narrow, annular gaps located at the upper and lower sides of the circular membrane. These gaps had viscous resistance. The upper and lower gaps on both sides of the membrane were connected to the opposite pockets of the hydrostatic journal bearing. The mass flow rates through the two gaps were controlled by the balance between the net water pressure force (the sum of the upper and lower forces exerted on the circular membrane) and the elastic deformation force of the membrane.

Ono et al. [6] and Kang et al. [7] treated hydrostatic thrust bearings with a membrane restrictor that had a single narrow, annular gap. Ono et al. proposed a new membrane restrictor that consisted of an annular gap and a membrane in the 1980s. The annular gap was controlled by the pocket pressure of a hydrostatic thrust bearing, which varied according to the shaft displacement. As a result, Ono et al. reported that the hydrostatic thrust bearing could achieve nearly infinite stiffness if one selected appropriate design parameters for the membrane. Kang et al. proposed a modified method for predicting the mass flow rate through the gap, and the gap was changed in the radial direction in an actual membrane restrictor. They assumed an equivalent uniform gap instead of the actual nonuniform gap in order to obtain the mass flow rate through the gap easily. From these reports, it is clear that the use of a membrane restrictor is very effective for achieving a high stiffness in hydrostatic bearings.

Recently, lower power consumption has become an important requirement for machine tools, and water-lubricated hydrostatic bearings have attracted attention because of water's low viscosity.

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Nomenclature

C_d	flow coefficient at the inlet of the membrane restrictor
D	bending strength of the membrane$=Et^3/12(1-\nu^2)$ [Pa/m^3]
E	Young's modulus of the membrane material used in this paper$=$193.0 [GPa]
f	deformation of the membrane [m]
F	dimensionless deformation of the membrane $(=4Df/r_{r2}^4 p_s)$
h_b	bearing clearance [m]
h_r	gap of the membrane restrictor [m]
h_{r0}	initial gap of the membrane restrictor [m]
H_r	dimensionless gap of the membrane restrictor $(=h_r/h_{r0})$
K	dimensionless elastic coefficient of the membrane $(=4Dh_{r0}/r_{r2}^4 p_s)$
p	pressure (gauge) [Pa]
p_a	ambient pressure (assumed zero) [Pa]
p_b	pressure in the bearing clearance [Pa]
p_i	pressure at the inlet of the gap of the membrane restrictor [Pa]
p_p	pressure in the bearing pocket [Pa]
p_r	pressure in the gap of the membrane restrictor [Pa]
p_s	supply pressure (gauge) [Pa]
P	dimensionless pressure$=p/p_s$
q	volume flow rate [l/min]
r	radial coordinate in the bearing clearance and gap [m]
r_{bo}	bearing outer radius [m]
r_{bi}	bearing inner radius [m]
r_{b1}	inner radius of the pocket [m]
r_{b2}	outer radius of the pocket [m]
r_{r0}	inner radius of the gap in the membrane restrictor [m]
r_{r1}	outer radius of the gap in the membrane restrictor [m]
r_{r2}	radius at the simple support position of the membrane [m]
R_{r0}, R_{r1}	dimensionless radii $(=r_{r0}/r_{r2},\ =r_{r1}/r_{r2})$
t	thickness of the membrane [m]
w	load capacity [N]
W	dimensionless load capacity $(=w/\pi r_{bo}^2 p_s)$
μ	viscosity of water [Pa s]
ν	Poisson's ratio$=$0.3
ρ	density of water [kg/m^3]
θ	circumferential coordinate in the bearing clearance

In the 1970s, Hother-Lushington [8] reported that the use of water as lubricant was unfamiliar because of its comparatively low viscosity and its corrosive action on ferrous materials. However, when Hother-Lushington published his paper, there was an increasing tendency to use process fluids for lubricating bearings in all types of plant and water was one of the most widely used "process fluids". Hother-Lushington discussed the advantages and reliability of a water lubricated bearing. In the 1990s, environmental pollution became a very important issue, and several reports on water-lubricated hydrostatic bearings for machine tools were published. Slocum et al. [9] proposed a water-lubricated hydrostatic thrust bearing with a self-controlled ceramic restrictor for grinding machines. Yoshimoto et al. [10,11] investigated the static and dynamic characteristics of water-lubricated conical bearings with spiral grooves for a high-speed spindle, to improve its load capacity and bearing stiffness. Liu et al. [12] and Durazo-Cardenas et al. [13] investigated water-lubricated hydrostatic journal bearings with a porous restrictor, to reduce the power consumption at high speeds compared with oil-lubricated hydrostatic journal bearings.

It is clear from the above reports that a water-lubricated hydrostatic thrust bearing with a membrane restrictor is a very suitable candidate for achieving very high stiffness, reduced power consumption at high speeds, and minimal pollution of the environment. The objectives of this paper were, therefore, to clarify the static characteristics of a water-lubricated hydrostatic thrust bearing with a membrane restrictor, numerically and experimentally, and to confirm the usefulness of the proposed hydrostatic thrust bearing in achieving the requirements indicated above.

2. The proposed bearing structure using a membrane restrictor

Fig. 1 shows the geometrical configuration of the proposed water-lubricated annular hydrostatic thrust bearing with a membrane restrictor. As shown in Fig. 1(a), a circular membrane with thickness t is mounted on a circular flat base with an initial gap h_{r0} of 50 μm and is simply supported at the outer diameter of $2r_{r2}$. The gap beneath the membrane acts as a viscous resistance area, and one can control the mass flow rate through the gap by means of the force balance between the water pressure and the elastic stiffness of the membrane. Water flowing out from the membrane restrictor enters the four pockets of the hydrostatic thrust bearing through a circumferential feed groove, as shown in Fig. 1(b). The operating principle of this restrictor is as follows. When the load imposed on the hydrostatic thrust bearing increases and the bearing clearance h_b decreases, the pocket pressure p_p increases because of the reduced flow rate through the bearing clearance. When p_p increases, the gap of the membrane restrictor becomes wider, and the flow rate of water entering the pocket increases, and finally the bearing clearance increases. By this operating principle, it is possible to reduce the changes in the bearing clearance caused by varying imposed loads, compared with the bearing clearance changes in conventional hydrostatic bearings.

3. Numerical calculation method

3.1. Governing equations of the pressure distribution

To solve for the pressure distributions in the bearing clearance and the gap of the membrane restrictor, the following Reynolds equations in polar coordinates with no rotation are adopted.

For the bearing clearance, we have:

$$\frac{1}{r}\frac{\partial}{\partial r}\left(rh_b^3\frac{\partial p_b}{\partial r}\right)+\frac{\partial}{r\partial\theta}\left(h_b^3\frac{\partial p_b}{r\partial\theta}\right)=0. \tag{1}$$

For the gap of the membrane restrictor, we have:

$$\frac{\partial}{\partial r}\left(rh_r^3\frac{\partial p_r}{\partial r}\right)=0. \tag{2}$$

These Reynolds equations were numerically solved using the finite difference method. In addition, although the widths of grooves between the four thrust pads were constant in the radial direction, as seen in Fig. 1(b), in the calculations the angular coordinate corresponding to the edge of the land region was represented by the value at the radial center position, as shown in Fig. 1(b).

The dimensionless load capacity W was given as follows by integrating the pressure on the entire thrust pad region:

$$W=w/(\pi r_{bo}^2 p_s)=4\int_{\theta_1}^{\theta_2}\int_{Rbi}^{1}PRdRd\theta, \tag{3}$$

where $\theta_1=\theta_{b0}$, $\theta_2=\pi/2-\theta_{b0}$.