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Design and Test of Hydrostatic Built-in Grinding Spindle with Orifice Restrictors

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Abstract. The simulation of orifice-compensated hydrostatic bearing was introduced by solving Reynolds equation with finite difference method to evaluate the recess pressure, load capacity, and the flow rate. By the results of the simulation, we designed a built-in grinding spindle, containing two hydrostatic thrust bearings and two hydrostatic journal bearings, which were all orifice-compensated. Experimental measurements would be focused on the stiffness of spindle shaft.

Keywords: Hydrostatic bearing, Orifice restrictor, Spindle.

INTRODUCTION

Hydrostatic bearings have been frequently used in high-precision machine tools nowadays to maintain desirable performance for fine surface roughness and durability. It is because of several advantages, such as low straightness ripple, good damping characteristic, low friction and wear. The working principle of multi-recess hydrostatic bearings is based on the mechanism of fluid film lubrication. The fluid film is supplied from an external pressure source to create the net load on the bearing. The bearings system forms the fluid film as soon as the pump is turned on.

One of the characteristics of hydrostatic system is the stiffness of the bearing and it is generally true to design a hydrostatic bearing with a stiffness as higher as possible, such that the variation of the deflection due to load variation is limited. Various approaches may be adopted for the improvement of hydrostatic bearings stiffness. One of the approaches is to reduce the nominal bearing clearance. When a fixed laminar-flow restrictor is used, the stiffness of the bearing is inversely proportional to its nominal clearance. However, higher machining accuracy might be needed and the thermal effects of viscosity friction will increase [1]. Another approach of using the orifice restrictor instead of capillary restrictor may increase the performance of the bearings. Previous studies have revealed that the stiffness of orifice-compensated bearings is higher than capillary-compensated counterparts [2] [3]. Furthermore concerning the compact characteristic of orifice restrictor, it is of interest to design a grinding spindle with the orifice restrictor.

Fluid Film Lubrication Model

To obtain the pressure distribution of the oil film in the hydrostatic journal bearing, the generalized 2-dimensional Reynolds equation was used (neglecting the time dependent terms).

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial y}\right)=0 \tag{1}$$

Equation (1) is written in the finite difference form and solved iteratively with following assumptions: [2] [3].

1. The dimensions of the fluid model is in the thin film regime: the thickness of the fluid film is small, compared to its size in the other directions.
2. No pressure gradient across the fluid film.
3. Laminar flow.
4. Nonslip boundary condition: The inertia effect is small and can be neglected.

5. The ambient pressure is assumed to be zero.
6. The fluid is Newtonian fluid.

The solution satisfying above conditions yields the pressure distribution of the oil film, which leads to the determination of load capacity, stiffness, and the flow rate of the bearing (see Fig.1). The load capacity is expressed as

$$W = \int_{-L/2}^{L/2} \int_{0}^{\pi D} p_{x,y} \cos\theta dxdy \tag{2}$$

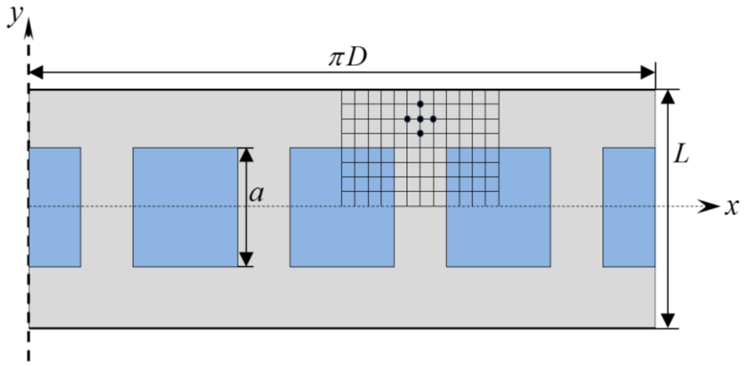

FIGURE 1. Developed view of journal bearing

The stiffness can be estimated through the load capacity over a small change of eccentricity. Design parameters in this paper are listed in Table 1. The stiffness performances versus eccentric with different initial restriction ratios are plotted in Fig.2.

TABLE 1. Design parameters of the bearing

Parameters	Range
L/D ratio	**1.1**
Recess numbers N	**4**
Bearing Clearance	**25μm**
a/L ratio	**0.2**
Concentric pressure ratio β_0	**0.5, 0.55, 0.6**
Max eccentricity ratio ε	**0.2**
Restrictor type	**Orifice**

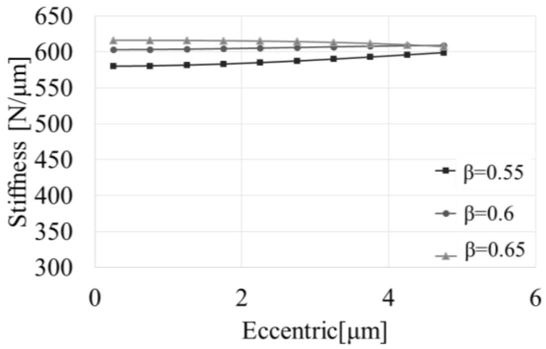

FIGURE 2. Stiffness versus eccentric

Generally, stiffness of bearing is proportional to the pressure of recess. As the eccentric increasing in Fig.2, the relationship of stiffness is no longer keeps in the same level. The bearing with higher pressure ratio may not

perform as well from initial status to large eccentric condition.
Hence, we are constructing an experimental platform for the spindle to verify the simulation of oil film. The bearing arrangement of experimental spindle is plotted in Fig.3. In order to confirm the performance of the spindle equipped with hydrostatic bearings, a real grinding test will be conducted.

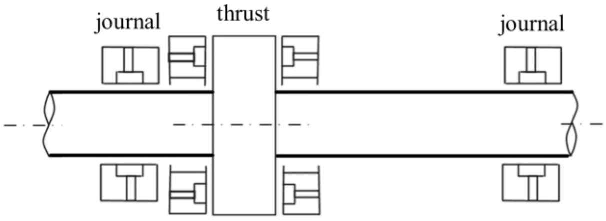

FIGURE 3. Bearings arrangement of spindle.

Experimental Setup

Fig.4 shows the facilities used to test the performance of this spindle. The spindle was mounted within the bracket which was vertically sat on the optical table. The pressure source was kept in 5 MPa. Ideally, the recess pressure was 2.75 MPa. Because our setting was not equipped with the frequency converter, we conduct the run-out test by rotating the shaft with hands. The run-out was less 1μm measured by the dial test indicator. The stiffness test of the shaft under rotation would be conducted after complete the setup of experiment. Although the dynamic effect of bearing under rotation was not concerned in the simulation yet, but it can still reveal the static performances of the bearing system.

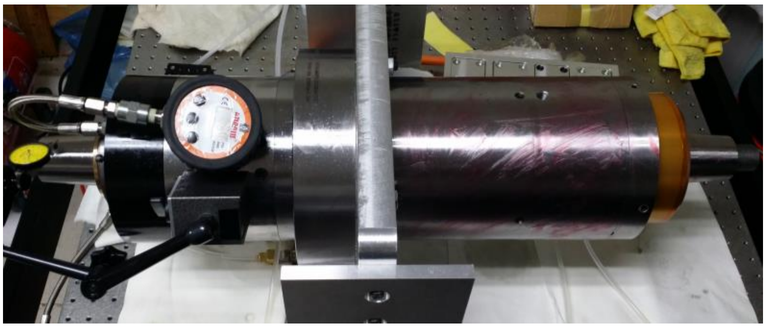

FIGURE 4. Experimental setup.

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