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Research on Dynamic Pressure of Hydrostatic Thrust Bearing Under the Different Recess Depth and Rotating Velocity

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Abstract

In order to solve the loading capacity of the hydrostatic thrust bearing, a numerical simulation concerning dynamic pressure of multi-pad hydrostatic thrust bearing under the different recess depth and rotating velocity is been described. Three-dimensional dynamic pressure field of gap fluid between the rotational worktable and the base has been simulated by using the Computational Fluid Dynamics. This study theoretically researches the influence of recess depth and rotating velocity on dynamic pressure of the bearing according to the lubricating theory and Finite Volume Method, and the simulation results indicate that an improved characteristic will be affected by recess depth and rotating velocity easily. Through this method, the optimal loading capacity of such products can be achieved.

Keywords: *Recess depth, Dynamic pressure, Rotating velocity，Hydrostatic thrust bearing, Computational Fluid Dynamics (CFD)*

1. Introduction

Hydrostatic thrust bearings have been used in many industrial applications such as the machine tool industry, the forest product industry, the slippers of axial piston pumps and motors and the Halle optical telescope due to their favorable performance characteristics. These include high load-carrying capacity, zero wear of bearing surfaces, low friction at low or zero speeds, large fluid film stiffness and damping, reduced vibrations and good positional accuracy. Hydrostatic thrust bearing systems have been extensively investigated by many researchers during the last few decals and their research efforts have been focused on various aspects concerning this class of the bearings. The following reports are important investigations concerning hydrostatic thrust bearings this time. Recently Statish C. Sharma analyzed the capillary compensated four cavities hydrostatic journal bearing with different geometric shapes of recess. It was further reported that influence of recess shape on the performance of capillary compensated circular thrust pad hydrostatic bearing. Statish C. Sharma used the Finite Element Method to compute the performance characteristics of a circular thrust pad hydrostatic bearing with circular, rectangular, elliptical and annular cavities. They finally gave out conclusions that the value of the fluid film pressure, the value of the bearing flow, the value of the load carrying capacity, the value of stiffness coefficient of four different kinds of cavities shapes [1]. F. Shen revealed the effect of flow characteristics on the load capacity of oil cavity, the flow of round Rayleigh step cavity in

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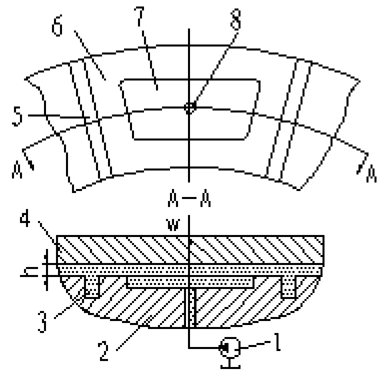
hydrostatic lift was divided into three regions. The ax symmetric jet impinging flow, radial source flow and film flow. The Computational Fluid Dynamics (CFD) method was used to build a calculation model and the influences of geometry parameters and the Reynolds number on the vortices were simulated. The results show that there are complex vortices in the flow field of the round Rayleigh step cavity, of which the pressure is higher than parallel disks flow [2]. Younes studied a circular thrust pad hydrostatic bearing with a central step added to a rotating shaft that rested in a deep recess from the point of view of optimal pumping power. Different bearing configurations were considered and it was reported that the addition of the stem enabled this bearing to support radial loads [3]. Christian studied the dynamic effects of hydrostatic bearings. The geometry of the hydrostatic bearing pockets and their restrictors are optimized by using time continuous pressure distribution at the bearing pocket, laminar flow behavior as well as constant velocity of the bearing. The dynamic effects of the flow at high velocities are not considered. The paper reflects the common design and calculation methods and shows their limitations in regard to the calculation of hydrostatic bearings at high velocities. It analyzes the results of complex dynamic flow simulations of hydrostatic bearings and presents a new design and optimization concept of hydrostatic bearings. This concept analyses the oil flow at high bearing velocities and it optimizes the bearing geometry, the restrictor geometry as well as the geometry of the main mechanical components [4]. L. Guo studied influence of geometric configuration on the characteristics of high speed hybrid bearings. Four hybrid bearings having different geometric configurations were analyses for their static and dynamic characteristics, including flow rate, load capacity, stiffness, and whirl frequency ratio. The four bearings included a square- recess bearing, a circular recess bearing, a triangular- recess bearing, and an angled orifice bearing. Comparisons of the results were made between bearings, the angled- orifice bearing has the most favorable overall performance and has good applications in high speed spindle [5]. Z. Q. Zhao described pressure of the hybrid bearings based on MATLAB software, the ladder hybrid bearings' static performance was solved with finite element method, oil film bearing pressure field was calculated, and researched the oil film pressure distribution when changing the depth of the ladder of the hybrid bearing [6]. C. Y. Zhang studied that influences of different factors on the flow field of center entrance oil cavity were studied. In static state, the number of vortex will reduce and the stability of the oil cavity will be enhanced with the increasing of lubricant viscosity. However, the increase of inlet speed, oil cavity's depth and inlet radius will lead to the vortex's effect increasing and reduce the stability of oil cavity. The increase of hydrostatic turntable's speed would directly lead to the vortexes appearing nearby the oil sealing edge and the vortexes near the inlet gradually being weakened until disappearing [7-18].This paper studied that influence of recess depth on the dynamic effect of a constant flow hydrostatic thrust bearing.

2. Working Principle of Hydrostatic Thrust Bearing

Liquid hydrostatic thrust bearing working principle is that lubricating oil which is compulsively injected into oil cavity forms bearing capacity of hydrostatic bearing through throttling action of the gap between resistive oil edges and the rotary table, lifts bearing spindle, and bears external loads. The working principle of hydrostatic bearing with quantitative oil supply is shown as Figure 1. Lubricating oil enters into oil cavity from pump along inlet and flows out along the radial shallow recess and resistive oil edges of external ring as shown in Figure 1 [19, 20].



1—pump; 2—guideway; 3—gap fluid; 4—rotary table; 5—oil groove; 6—resistive oil edges; 7—oil cavity; 8—oil inlet; W—external load; h—oil film thickness

Figure 1. Sketch map of principle of work of quantitative supply static thrust bearing

3. Mathematical Model

Flow problem of fluid between the rotation worktable and base belongs to a laminar flow problem, and its mobility must meet mass conservation equation, momentum conservation equation and energy conservation equation [21, 22].

3.1 Mass conservation equation

$$\frac{\partial \rho}{\partial t}+\nabla\cdot(\rho U)=0 . \tag{1}$$

Where ρ is density; t is time; U is speed vector.

3.2 Momentum conservation equation

$$\frac{\partial(\rho U)}{\partial t}+\nabla\bullet(\rho U\otimes U)=-\nabla p+\nabla\bullet\tau+S_M . \tag{2}$$

Where S_M is momentum source; τ is pressure tensor.

3.3 Energy conservation equation

$$\frac{\partial(\rho h_{tot})}{\partial t}-\frac{\partial \mathrm{p}}{\partial \mathrm{t}}+\nabla\cdot(\rho U h_{tot})=\nabla\cdot(\lambda\nabla T)+\nabla\cdot(U\cdot\tau)+U\cdot S_M+S_E . \tag{3}$$

Where h_{tot} is total enthalpy; T is temperature; λ is fluid transfer heat coefficient; S_E is heat source term.

3.4 Reynolds equations

The working principle of Rayleigh step bearing is shown as Figure 2.

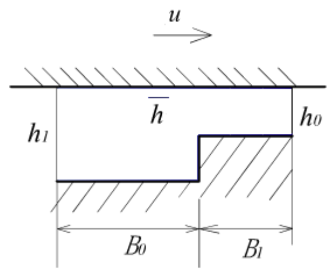


Figure 2. Working principle of Rayleigh step bearing

In the area B_0, Reynolds equation is

$$\frac{dp_1}{dx} = 6\eta u \frac{h - h_m}{h^3} . \tag{4}$$

In the area B_1, Reynolds equation is

$$\frac{dp_2}{dx} = 6\eta u \frac{h - h_m}{h^3} . \tag{5}$$

3.5 Dynamic pressure equation

In the area B_0, dynamic pressure equation is

$$p_1 = \frac{6\eta u B_1 \left(h_1 - h_0\right)}{B_0 h_0^3 + B_1 h_1^3} x . \tag{6}$$

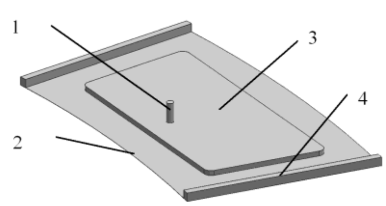
In the area B_1, dynamic pressure equation is

$$p_2 = \frac{6\eta u B_0 \left(h_1 - h_0\right)}{B_0 h_0^3 + B_1 h_1^3} x . \tag{7}$$

Where η is viscosity of lubricating oil; h_0, h_1 and h_m are thickness of oil film; B_0 and B_1 are length of step; p is oil film pressure.

4. Oil recess model

The hydrostatic thrust bearing having sector recess is shown in Figure 3.



1-Inlet; 2-Oil Cavity; 3-Throttle surface; 4-Return Oil groove

Figure 3. Clearance fluid modeling of sector recess

5. Grid of Oil Film

The total number of the grid of the gap oil film is 265376, and the quality distribution below 0.55 is 0, grid number between 0.55~0.9 is 4446, it accounts for the total 1.675%, 0.9~1.0 is 260930, it accounts for the total 98.325%. The grid and the grid quality of the gap oil film are shown as Figure 4.

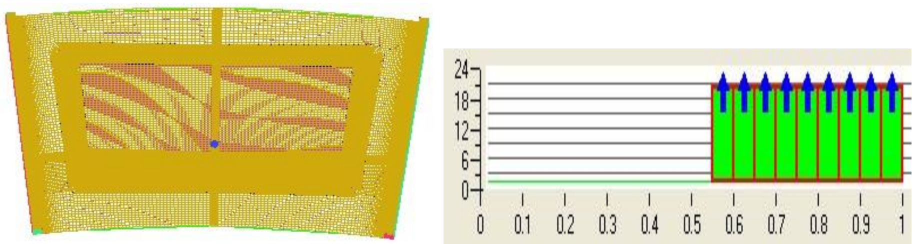


Figure 4. The grid and the grid quality of the gap oil film

6. Boundary Conditions

In the hydrostatic bearing interior flow field, the fluid regards as the incompressible fluid; There is no slipping between the lubricating oil and the solid; In the process of the worktable rotating, the lubricant film regards as the adiabatic; The lubricating oil selects No.46 hydraulic fluid, sets the material properties according to No.46 hydraulic fluid, the viscosity edits the viscosity-temperature function to replace the constant value; The inlet choice the flow capacity entrance, the inlet temperature assumed as 311k, the outlet selects the pressure export; The upper surface of the oil film is the rotating surface, the velocity is the fixed angular speed.

7. Results and Discussions

In order to study the recess depth impacting on the dynamic pressure, when the inlet flow capacity is 100 kg/s, rotation velocity are 6rpm and 20rpm, and choose viscosity of lubricating oil η equals 0.0288 Pa·s, density ρ equals 900 kg/m^3, the recess depth assign respectively 0.25mm, 0.5mm, 1mm, 2mm, 4mm, 6mm, 8mm, 10mm, 12mm, 14mm and 16mm, the three-dimensional dynamic pressure fields are obtained in CFX to be as Figure 5, Figure 6, Figure 7, Figure 8.

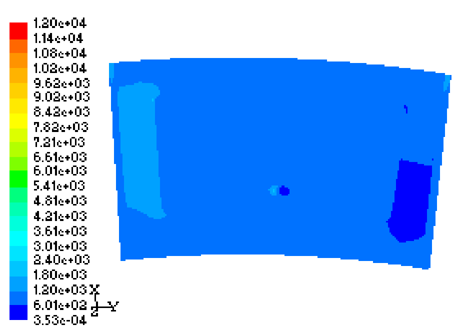


Figure 5. Dynamic pressure field of 0.5mm depth at 6r/min

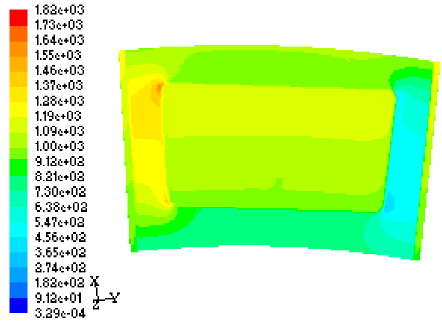


Figure 6. Dynamic pressure field of 8mm depth at 6r/min

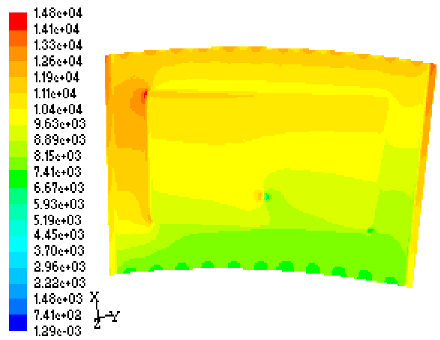


Figure 7. Dynamic pressure field of 0.5mm depth at 20r/min

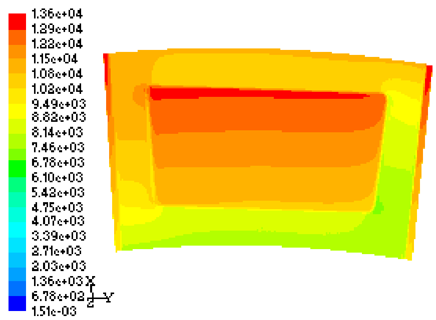


Figure 8. Dynamic pressure field of 8mm depth at 20r/min

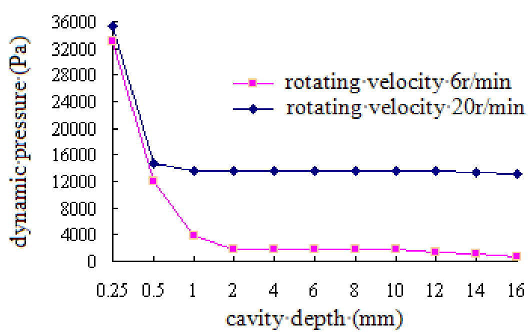


Figure 9. Relationship curve of oil cavity depth and dynamic pressure of oil film

Figure 9 shows the variation of dynamic pressure as a recess depth at rotating velocity of 6rpm and 20rpm. It is found that as the rotating velocity increases, the dynamic pressure increases. For the same value of rotating velocity, it is noted that the dynamic pressure is decreased with increasing the recess depth.

8. Conclusions

Based on the lubricating theory and Finite Volume Method, a simulation of three-dimensional dynamic pressure field of multi-pad hydrostatic thrust bearing having sector recess under the different recess depth and rotating velocity is been fulfilled by using general fluid computation software CFX. Conclusions are as follows:

1) Simulation research of the three-dimensional dynamic pressure of multi-pad hydrostatic thrust bearing under the different recess depth and rotating velocity had been achieved through Computational Fluid Dynamics and the Finite Volume Method.

2) The simulation results qualitatively agreed well with the theoretical calculation values. The results showed that oil recess dynamic pressure is affected by rotating velocity and recess depth.

3) Under the same value of rotating velocity, it was noted that the dynamic pressure was decreased with increasing of the recess depth.

4) It had found that as the rotating velocity increases, the dynamic pressure increases.

5) The safety of a multi-pad hydrostatic thrust bearing could be forecasted through this method, and laid a foundation for deformation of hydrostatic thrust bearing.

Acknowledgements

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