INERTIA EFFECT AND THE TURBULENCE ON THE LUBRICATION PERFORMANCE OF THE HIGH SPEED WATER-LUBRICATED THRUST BEARING

TRACK OR CATEGORY
Fluid Bearing

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INTRODUCTION
Water-lubricated bearings are more and more popular because of the harmlessness to the environment. But the viscosity of water is much smaller than the oil’s so the film thickness in lubrication of water is smaller than oil. A question is come up with that what is the difference between water lubrication and common oil lubrication besides the physical characteristics. Researchers found several problems worth attention with the promotion of water lubrication research. Zhang Xiuli thought the main concern is the low temperature rising [1] and the possibility of cavitation [2]. Zhang studied a misaligned hydrodynamic water-lubricated plain journal bearing considering the differences between the physical properties and cavitation. Saeid Dousti [3] and Wang Lei [4] treated the inertia effect was a noticeable in water lubrication. Saeid focused on the low viscosity of water and proneness to turbulence and fluid inertia effects in fixed geometry journal bearings. Wang simulated the dynamic characteristics of the hydrostatic thrust bearings considering the inertia effect with the bulk flow theory and the influence of centrifugal force at different speed.

In general lubrication problems, oil has a big enough viscosity so the magnitude of inertia force is tiny, comparing to the viscous force. Researchers usually ignore the influence of inertia force to simplify the mathematical model, the classical Reynolds Equation. For water lubrication, the viscosity is much smaller than oil and the magnitude of viscous force decreases a level, which has similar size to the inertia force. Meanwhile, viscous force is proportional to the linear velocity, while inertia force is proportional to the linear velocity square. The dimensions are closer with a high speed, which put forward the necessity to study the inertia effect in high speed water-lubricated thrust bearings.

In this study, a generalized Reynolds equation in the cylindrical coordinate system is conducted and solved, which takes the full inertia effect and the regional flow state into account. The solutions are provided for the effect of the inertia of the high speed thrust bearings lubricated by water. The lubrication model is established by simplifying the Navier-Stokes equation and the assuming of the profile of the velocity not affected by the inertia. The inertia effects on the lubrication performance of the water-lubricated thrust bearings is simulated under the different thickness, load, and rotating speed conditions.

METHOD
The geometry of the tilting-pad thrust bearings
Figure 1 shows a simplified geometry of a tilting-pad thrust bearing. With the tilting pad, the bearing can adapt to various operation conditions. It’s the wedge-shaped fluid film between collar and shoe that generate the carrying capacity. In order to simplify the model complexity, the model is simplified to a one-sixth model, a sector slider. Furthermore, assume that it’s a line-pivoted bearing and there is only a degree of freedom of rotation around the axis $\theta=\theta_p$. 
A cylindrical coordinate is used in thrust bearing to express the model conveniently, in which \( r, \theta, z \) are the coordinate axis. The collar rotates round about axis with a rotate speed \( \omega \). \( B \) is the breadth, and \( L \) is the length. \( r_1 \) is inner radii and \( r_2 \) is outer radii. \( \theta_0 \) is the angle of the shoe. The expression of the film thickness is
\[
H = h_{\text{min}} + \Delta h \frac{r \theta}{L}
\]

\( \text{(1)} \)

**THE CONTROL EQUATIONS**

The pressure distribution in liquid film between collar and shoes is described by the Navier-Stokes equation. The equation taking the inertia effect into account in the cylindrical coordinate system is given by
\[
\begin{align*}
\rho \left( v_r \frac{\partial v_r}{\partial r} + \frac{v_\theta}{r} \frac{\partial v_\theta}{\partial \theta} + \frac{v_z}{z} \frac{\partial v_z}{\partial z} - \frac{v_\theta^2}{r} \right) &= \frac{\partial p}{\partial r} + \frac{\partial}{\partial z} \left( \mu \frac{\partial v_r}{\partial z} \right) \\
\rho \left( v_\theta \frac{\partial v_\theta}{\partial r} + \frac{v_\theta}{r} \frac{\partial v_\theta}{\partial \theta} + \frac{v_z}{z} \frac{\partial v_z}{\partial z} + \frac{v_\theta v_\theta}{r} \right) &= \frac{1}{r} \frac{\partial p}{\partial \theta} + \frac{\partial}{\partial z} \left( \mu \frac{\partial v_\theta}{\partial z} \right) \\
0 &= \frac{\partial p}{\partial z} + \frac{\partial \tau_{vy}}{\partial z}
\end{align*}
\]

\( \text{(2)} \)

in which the left terms stand for the inertia force in N-S equation, including the centrifugal term and convective terms. Convective terms (the first terms) is the inertia force of speed changing, while centrifugal term (the 4th term) is the inertia force of direction changing in the circular motion.

The equation (1) cannot be solved without the continuity equation of the incompressible liquid
\[
\frac{\partial v_r}{\partial r} + \frac{v_\theta}{r} \frac{\partial v_\theta}{\partial \theta} + \frac{\partial v_z}{\partial z} = 0
\]

\( \text{(3)} \)

An assumption which was put forward by Constantinescu [5] that the inertia effect does not affect the profile of the pressure-flow velocity, except the average values \( v_{rp}, v_{\theta p} \) is used.

Integrated across the film thickness, and simplified, the extended Reynolds Equation is given by:
\[
\begin{align*}
\frac{\partial}{\partial r} \left( rh^3 \frac{\partial p}{\partial r} \right) + \frac{\partial}{\partial \theta} \left( h^3 \frac{\partial p}{\partial \theta} \right) &= \frac{\omega r \partial h}{2} + I_c
\end{align*}
\]

\( \text{(4)} \)

**Regional flow state**

Every grid in computed region has a different Reynolds number. The flow turns to disorder if Reynolds number is bigger than threshold value. If \( \text{Re}<1000, \) the flow is laminar flow. If \( \text{Re} \geq 1500, \) there is turbulence flow while a transition region of \( 1000 \leq \text{Re} < 1500 \) can be calculated by cubic polynomial interpolation. The flow state is related with the coefficient of the formula. A most widely accepted method taking turbulence into account is adding the coefficients to the Reynolds equation based on the local Reynolds number. We use the Ng-Pan turbulence model here:
\[ \alpha = \frac{6}{5}, \beta = \frac{1}{3}, \gamma = 1, k_r = 12, k_\varphi = 12 \]

Laminar:

\[ \alpha = 1, \beta = 0.25 + \frac{0.885}{Re^{0.367}}, \gamma = 1, k_r = 12 + 0.0036 Re^{0.06}, k_\varphi = 12 + 0.00113 Re^{0.9} \]

Turbulence:

**Boundary conditions**

The boundary conditions of Reynolds equation for the tilting-pad thrust bearing are given as below:

\[ p_{\text{in}} = 0 \]

**Solution procedure**

A finite difference scheme is implemented to solve the equations. The generalize Reynolds equation is solved by successive over relaxation iterative method (SOR). And the optimal relaxation factor is gotten by the numerical calculation formula.

Integration of the pressure on the bearing surface gives the opening force \( F_z \) and the friction force \( F \):

\[ W = \iint p r d\theta d\varphi \]

\[ F = \iint \tau r d\theta d\varphi = \iint \left( \frac{nU}{h} + \frac{h}{2} \right) r d\theta d\varphi \]

**RESULTS AND DISCUSSION**

**Validation of computational results**

Hashimoto [6] carried out theoretical results and experimental load capacities with film thickness ratio. Figure 4 shows the dimensionless load force versus the thickness ratio. The radius ratio is equal to 0.5 and Re equals 5000. Pad extent angle is 60°. The simulate results agrees well. Figure 5 shows the experimental results done by Hashimoto. The experiment agree with the full inertia model well.

![Figure 2](image_url)

**Study of a tilting-pad bearing**

The parameters of the tilting-pad thrust bearing are listed in Table 1.
Table 1. Parameters of tilting-pad thrust bearing lubricated by water

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outer Diameter $D_o$(mm)</td>
<td>900</td>
</tr>
<tr>
<td>Inner Diameter $D_i$(mm)</td>
<td>400</td>
</tr>
<tr>
<td>Pad sector angle ($^\circ$)</td>
<td>50</td>
</tr>
<tr>
<td>Circumferential Partial Coefficient $a_0$</td>
<td>0.543</td>
</tr>
<tr>
<td>Radial Partial Coefficient $a_r$</td>
<td>0.5</td>
</tr>
<tr>
<td>Density $\rho$(kg/m$^3$)</td>
<td>998</td>
</tr>
<tr>
<td>Viscosity $\mu$(Pa.s)</td>
<td>0.001</td>
</tr>
<tr>
<td>Bump height $h_c$/μm</td>
<td>25</td>
</tr>
</tbody>
</table>

Figure 3(a) shows the inertia effects in the tilting-pad bearing which is central pivoted. For insuring the needful load capacity, a circular arc bump was pre-processed on the friction surfaces of shoes. The load capacity increases when take inertia effect into account. Both the centrifugal force and the convective term weaken the load capacity, which is different with the numerical results in the plane surface shoe. The difference between inertia less load capacity and full inertia results can reach 17% when rotate speed equals 1800 r/min, which is cannot be neglected. Figure 3(b)(c) shows that the pressure center with full inertia approaches the outer diameter. The deviation increase with the crescent rotate speed.

Figure 3. Numerical results for tilting-pad bearings (a) load capacity, (b) pressure center, (c) deviation of pressure center.

ACKNOWLEDGMENTS
The work described in this paper was supported by National Basic Research Program of China (973) (2015CB057303).

REFERENCES

KEYWORDS
Inertia Effect, Water-Lubricated, Tilting-Pad Bearing