STUDY ON THE DYNAMIC CHARACTERISTICS OF A WATER-LUBRICATED PUMP BEARING

(1st Report, Cylindrical Bearing Under High Ambient Pressure)

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Water (working fluid) lubricated bearings are used in various pump designs. This type of bearing has many characteristics different from usual oil lubricated bearings. This paper describes the following characteristics of the water lubricated bearings:

(1) Turbulent flow
(2) Full film due to high ambient pressure
(3) Large additional mass effect

Theoretical analysis by numerical calculation and model test using a special test system are performed for a cylindrical bearing. As a result, the characteristics of the water lubricated bearing are recognized and good agreement between theoretical and experimental results is confirmed.

Keywords: water lubricated bearing, dynamic characteristics of bearing

1. Preface

Generally, the research related to dynamic characteristics of bearing is mainly concerned with the oil film bearings exposed to the atmosphere[1]. Water lubricated bearings utilized in pumps, specifically in vertical pumps, are different in working conditions and characteristics from the conventional bearings. Therefore, the shaft systems must be designed by fully investigating the features of the water lubricated bearings used. Recently, on account of demand for improvement in reliability of a plant, highly reliable pumps are in demand and thus a study into pump bearings is needed to prevent the troubles related to bearings and vibration[2].

In this paper, on the following three points, the results of studies using a special test system and the latest technique of analyzing the dynamic characteristics of bearings are presented to provide a guideline for application of the results of research on the bearing.

dynamic characteristics.

(1) Effect of turbulent flow: Since water has low viscosity compared to oil, flow in a water lubricated bearing becomes turbulent at relatively low speed.
(2) Full film bearing: In case of a water lubricated bearing used in pump, the bearing ambient pressure may become higher than the atmospheric pressure. In this case, cavitation does not develop the same area as in the conventional bearings and thus a full circular film is formed.
(3) Added mass effect: Added mass effect which can be neglected in oil film bearings is significantly important for water lubricated bearings.

2. Research Object

Water lubricated bearings in the pump have various configurations, but in this study a cylindrical bearing with L/D of 1.0 is chosen as the object, as shown in Table 1.

Table 1. Subject of This Study
(Bearing with L/D=1, Cr=0.25mm)

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Nomenclature

Cr : Bearing radial clearance
D : Bearing diameter (R: radius)
G : Turbulent flow coefficient
h : Water film thickness
F : Excitation force
L : Bearing width
N : Rotating speed
P : Bearing internal pressure
U : Circumferential velocity
W : Axial velocity
S : Sommerfeld number
S': Sommerfeld correction number after turbulent flow compensated
Re : Reynolds number
W : Static load
ε : Eccentricity
(From static loading direction to rotating direction)
x : Circumferential direction
z : Axial direction
X,Y: Rectangular coordinate (Fig.4)
E : Nondimensional stiffness
(nondimensionalized used by W/C)
C : Nondimensional damping
(nondimensionalized used by W/C)
M : Nondimensionalized added mass effect

3. Theoretical Analysis

Numerical analysis used in this paper is based on Reynolds equations[4],[9] in which the influences of turbulent flow and inertia are considered by approximation. Thus, a steady state is expressed by the following equation.

\[
\frac{\partial}{\partial x} \left[ \frac{G}{\mu} \frac{\partial P}{\partial x} \right] + \frac{\partial}{\partial z} \left[ \frac{G}{\mu} \frac{\partial P}{\partial z} \right] = -U \frac{\partial h}{\partial x} + I
\]

Where, I is a compensating term by inertia force and is given by:

\[
I = \frac{\partial}{\partial x} \left( \frac{\rho G h^3}{\mu} l_x \right) + \frac{\partial}{\partial z} \left( \frac{\rho G h^3}{\mu} l_z \right)
\]

\[
l_x = \frac{\partial}{\partial x} (a U l h + \beta U^2 h - \gamma U_w h)
\]

\[
+ \frac{\partial}{\partial z} (a U h w - \frac{\gamma}{2} U_w h)
\]

\[
l_z = \frac{\partial}{\partial x} (a U h w - \frac{\gamma}{2} U_w h)
\]

\[
+ \frac{\partial}{\partial z} (a U h w)
\]

I of equation (2) is determined by repeating the calculation. As for the turbulent flow correction coefficients, the following values are used[3],[9].

\[
G_{x1} = \frac{1}{(12 + 0.0136 R_e^{0.44})}
\]

\[
G_{x2} = \frac{1}{(12 + 0.0043 R_e^{0.44})}
\]

\[
G_p = 4.7265/R_e^{0.4286}
\]

(provided that 1/12 is used if \( R_e < 12344 \)).

\[
R_e = \frac{C U}{\nu}
\]

\[
R_{xp} = \frac{k^2}{\mu \nu} \left[ \frac{\partial P}{\partial x} \right]^2 + \left[ \frac{\partial P}{\partial z} \right]^2
\]

Furthermore, the dynamic characteristics were determined by repeating the calculation in which small displacement and velocity were given, considering the unsteady term.

4. Test Apparatus and Testing Methods

For experiments, a special test system is used as shown in Figures 1 through 3. This test system consists of a relatively rigid shaft supported by tilting pad bearings, a floating casing and a test casing, and test bearings. A pair of test bearings are fixed in the casing and the inlet and outlet pressures can be set arbitrarily. City water is used for lubrication, the temperature of water can also be controlled.

Figure 1. Appearance of Bearing Test Apparatus

Figure 2. Bearing Test Apparatus
where inertia coefficients $M_{XX}$, $M_{XY}$, $M_{YX}$ and $M_{YY}$, which are usually neglected for oil film bearings, have an added mass effect on the bearing journal if the bearing metal is fixed, and they are significantly important for a water lubricated bearing since they may affect the stability of the bearing.

Various methods can be used to determine the dynamic coefficients on a bearing test stand. In this paper, a method which determines the dynamic coefficients by the relationship between the sinusoidal excitation forces, which are applied by hydraulic actuators in $x$ and $y$ directions, and the displacement of the lubrication film.

That is, equation (6) can be written as equation (8) by using the following complex expression:

\[
\begin{align*}
Z_{xx} &= K_{xx} - M_{xx} \omega^2 + iC_{xx} \omega \\
Z_{ty} &= K_{ty} - M_{ty} \omega^2 + iC_{ty} \omega \\
Z_{yx} &= K_{yx} - M_{yx} \omega^2 + iC_{yx} \omega \\
Z_{yy} &= K_{yy} - M_{yy} \omega^2 + iC_{yy} \omega
\end{align*}
\]

\[
\begin{bmatrix}
Z_{xx} & Z_{ty} & X^r & F_x \\
Z_{yx} & Z_{yy} & Y^r & F_y
\end{bmatrix}
= \begin{bmatrix}
F_x \\
F_y
\end{bmatrix}
\]

To determine the unknown $Z_{xx}$, $Z_{ty}$, $Z_{yx}$ and $Z_{yy}$ from equation (8), several relationships of equation (9) are required. Therefore, the following two methods were applied.

(a) A method which determines the required numbers of relationships by applying the excitation forces with phase difference of $90^\circ$ in $x$ and $y$ directions (forward circular excitation and backward circular excitation) and by testing twice under the same conditions.

\[
\begin{align*}
\frac{Z_{xx}}{Z_{yy}} &= [X] \begin{bmatrix}
F_{xx} \\
F_{yy}
\end{bmatrix} \\
\frac{Z_{yx}}{Z_{ty}} &= [Y] \begin{bmatrix}
F_{yx} \\
F_{ty}
\end{bmatrix}
\end{align*}
\]

(b) A method which determines the required numbers of relationships at one time by applying the excitation forces of two different frequencies $\omega_1$ and $\omega_2$ in $x$ and $y$ directions simultaneously. Thus, equation (12) can be obtained, same as for equations (9) - (11).

\[
\begin{bmatrix}
X_{w1} & Y_{w1} & 0 & 0 & Z_{xx} & F_{w1x} \\
0 & 0 & X_{w1} & Y_{w1} & Z_{ty} & F_{w1y} \\
X_{w2} & Y_{w2} & 0 & 0 & Z_{yx} & F_{w2x} \\
0 & 0 & X_{w2} & Y_{w2} & Z_{yy} & F_{w2y}
\end{bmatrix}
\]

\[
\begin{bmatrix}
F_{w1x} \\
F_{w1y} \\
F_{w2x} \\
F_{w2y}
\end{bmatrix}
\]
The twelve coefficients can be determined by solving equation (11) or (12) for several different frequencies and by curve fitting the obtained $x_{zy} \ldots$.

5. Static Characteristics

In this section, the experimental results and analytical results with regard to the static characteristics are compared and discussed.

5.1 Shaft center loci

Figure 5 shows the static loci of the shaft center. It is clear from Figure 5 that the conventional bearings exposed in atmosphere have an eccentricity in the loading direction in highly eccentric region, but in this test, the bearing has the eccentricity in a direction perpendicular to the loading direction and it has the characteristics of a wedge oil film bearing. A tendency that the experimental values exceed 90° in the angle of eccentricity may be caused by the influence of inertia.

![Figure 5. Static Shaft Center Loci](image)

5.2 Circumferential pressure distribution

Figure 6 shows an example of the circumferential pressure distribution on the shaft center line. The horizontal scale presents the circumference in degrees. In the figure, the continuous line shows the pressure distribution and the dashed line illustrates the clearance distribution, and both are measured continuously from the rotating part. Under this testing condition, it is clear that since the entire region is under positive pressure due to a sufficiently high ambient pressure, no cavitation develops and ideal Sommerfeld conditions i.e., full cylindrical bearing conditions are fulfilled.

5.3 Load carrying capacity curve

Corresponding to the shaft center loci in Figure 5, the relationship between the bearing static load and journal eccentricity is non-dimensionalized and plotted in Figure 7. The horizontal scale in the figure shows Sommerfeld number.

$$S = \frac{\mu N R (c/P LD)}{W}$$

![Figure 7. Static Load Carrying Capacity Curve (Sommerfeld Number used)](image)

In Figure 7, the continuous line shows the calculated results obtained by applying the fore-mentioned turbulent flow theory. It can be seen that the theoretical and experimental values are in good agreement.

As shown in Figure 7, the non-dimensional characteristics change as the rotating speed varies, even for the same bearing. This is due to a turbulent flow. Figure 8 presents the non-dimensional characteristics replotted by using Sommerfeld correction number for turbulent flow[3].

$$S' = S_0 = S(1 + 0.0013 R^2)$$

where

$$R_s = \frac{\pi D N C_r}{\nu}$$

It is reconfirmed from Figure 8 that the bearing used in turbulent flow can be approximated as the bearings in laminar flow by using the turbulent flow correction Sommerfeld number in equation (14)[3].

Furthermore, it can be expected that in the experiment of a water lubricated bearing, since the temperature distribution is smaller than that of oil lubricated bearing, more accurate results will be obtained.
6. Dynamic Characteristics

The measured and calculated values of stiffness and damping coefficients are non-dimensionalized and plotted against the eccentricity in Figure 8, 9 and 10. Figure 9 was obtained by the forward and backward circular excitation methods and Figure 10 by a complex frequency excitation method. It is clear from these figures that the measured and calculated results are in good agreement. Thus the two methods can be regarded as reasonable testing methods. Moreover, the calculated stiffness and damping coefficients as plotted against the eccentricity can be represented by a single line since the influence of the rotation Reynolds number on the coefficients is extremely small.

Figure 11 plots the experimental values of inertia coefficients which are non-dimensionalized by added mass approximation (16) (i.e., Friz equation [8] for a non-rotating concentric double cylinder.)

\[ M_0 = \frac{R}{C_r} \frac{\rho R^4 l}{1+12(R/L)^2} \]  \hspace{1cm} (16)

Figure 10. Dynamic Characteristics by Complex Frequency Excitation Method

Figure 11. Measurements of Inertia Coefficients

The cross-coupled terms of inertia coefficients are negligibly small and thus are not plotted.

From these results, it is confirmed that a somewhat smaller than the Friz approximation but significant amount of added mass effect exists. For example, in this model, the measured added mass for the bearing journal of 480N (49kgf) amounts to 1960x2940N (200-300kgf) which can be not neglected. Furthermore, sufficient attention must be paid to high speed and high frequency regions since the influence of inertia coefficients is proportional to the square of the vibration frequency and it is significantly large.

7. Conclusions

Analysis and experiment of static and dynamic characteristics of a full bearing as a typical example of water lubricated bearings have been conducted with the following conclusions. These conclusions
must be taken into account properly when performing the design and vibration analysis of a rotating machine in which bearings of this type are used.

(1) Generally, the dynamic characteristics of a bearing lubrication film are difficult to measure and, in many cases, they do not in satisfactorily agree with the theoretical results. However, by clarifying the boundary conditions of the bearing and by conducting experiments with high accuracy, it can be shown that the dynamic characteristics can be obtained with high accuracy.

(2) As for bearings operating under high ambient pressure, full circular-lubrication film is formed. Consequently, the static load carrying capacity increases, simplifying the dynamic characteristics to be expressed by $K_{xy}$, $K_{yx}$, $C_{xx}$ and $C_{yy}$.

(3) In a water lubricated bearing, the flow becomes turbulent at a relatively low circumferential velocity. However, the turbulent flow can be approximated as a laminar flow by using the turbulent flow correction coefficients.

(4) For a water lubricated bearing, inertia coefficients (i.e., added mass effect) are dominant and can roughly be approximated by using a simple Fritz equation.

References