OPTIMIZATION OF GROOVE DIMENSIONS IN SPIRAL-GROOVED JOURNAL BEARINGS FOR SPINDLES OF PRECISION EQUIPMENTS

Tomoko HIRAYAMA¹, Naomi YAMAGUCHI², Shingo SAKAI³, Noriaki HISHIDA⁴, and Hiroshi YABE⁵

¹Doshisha University, Kyoto, JAPAN
²Ryukoku University, Shiga, JAPAN
³Osaka-Electro-Communication University, Osaka, JAPAN
⁴Spindle Device Laboratory, Kyoto, JAPAN

ABSTRACT

Spiral-grooved bearings have lately attracted considerable attentions as better substitutions for current ball bearings, especially in motor spindles of precision equipments such as hard disk drive systems. Their high stiffnesses and high damping abilities can realize good run-out characteristics even for such small-sized equipments.

Regarding the groove dimensions of the spiral-grooved journal bearings, it seems that we usually design them according blindly to a traditional standard aiming for maximizing the direct bearing stiffness. It is also the fact that the bearings designed according to this standard can realize comparatively good bearing performances. On the other hand, there have been considerably few discussions on optimum designs of the groove dimensions specialized for a reduction of operational run-out, which is a major characteristic for applications to precision equipments. The optimization of groove dimensions of the spiral-grooved journal bearings is, then, discussed in this paper. The subjects for optimizations are: 1) reduction of rotor displacement against a static load, 2) reduction of amplitude of the operating non-repeatable run-out (NRRO) due to an unbalanced rotor mass, and 3) reduction of amplitude of the operating non-repeatable run-out (NRRO) against an external impulse load.

KINEMATICAL MODEL FOR THEORETICAL APPROACH

A schematic diagram of the spiral-grooved journal bearings dealt in this study is shown in figure 1. It mainly consists of a stationary shaft and a rotating sleeve. Assumptions in theoretical approaches are: 1) The shapes of the shaft and the sleeve are geometrically perfect. The slenderness ratio, \(l/d\), is assumed to be 1.0 for the numerical calculations.

2) The displacement of the sleeve is only in parallel scheme against the shaft without tilting.

3) The bearing clearance is filled with an incompressible fluid, but free from occurrences of cavitation.

4) Linearization treatment for the bearing characteristics, stiffnesses and damping coefficients, can applied for any movement of the sleeve rotor.

EQUATION OF MOTION AND EVALUATION INDEXES

The dimensionless equation of motion for the rotor is:

\[
\frac{1}{36} M_\infty^2 \ddot{Z} + \frac{K Z}{\Omega} + \frac{B}{\Omega} \dot{Z} = \tau
\]

where \(Z\), \(K\), and \(B\) are displacement of rotor, stiffness, and damping coefficient, respectively, in dimensionless forms. They are complex variables constituted as:

\[
Z = X + iY
\]

\[
K = K_{XX} + iK_{XY}
\]

\[
B = B_{XX} + iB_{XY}
\]

Definitions for normalizations are:

\[
Z = \frac{z_0}{\epsilon_{yz}}
\]

\[
K_y = \frac{c_y}{\Omega^2 (p_{yd}) \cdot b_y}
\]

\[
B_y = \frac{c_{y,y}}{(p_{yd}) \cdot b_y}
\]

where \(p_{yd}\) is atmospheric pressure, and \(\nu\) is the frequency of revolution of the rotor.

\(\tau\), appearing in the equation (1), is external force term, which should be appropriately given according with the subject in consideration, as:

For the evaluation of static displacement of the rotor, we consider an external static load \(f_0\), and

\[
\tau = f_0 = \frac{f_0}{(p_{yd})}
\]

For the evaluation of amplitude of RRO, we consider a centrifugal force due to an unbalanced mass of the rotor, and

\[
\tau = \frac{M_r \cdot R_r^2 \cdot \nu^2}{\rho r c_z}
\]

where \(M_x\) is dimensionless unbalanced mass [see equation (10)] and \(R_r\) is dimensionless radius defined by \(r_0 \epsilon_{yz}\).

For the evaluation of amplitude of NRRO, we consider an external impulse load \(g_0\) applied to the rotor, and

\[
\tau = g_0 \delta (r) = g_0 (p_{yd}) \cdot \delta (r)
\]

where \(\delta (r)\) is Dirac delta function.

Definitions of the other dimensionless parameters are enumerated as follows:

\[
M = \rho_r h \frac{\nu^2}{(p_{yd})} (c_r / f_{r0})^3 : \text{Dimensionless rotor mass}
\]

\[
M_b = \rho_r h \frac{\nu^2}{(p_{yd})} (c_r / f_{r0})^3 \cdot m_b: \text{Dimensionless bearing mass}
\]

\[
\lambda = \frac{6 \mu \omega / p_{yd} \cdot (r_0 / c_z)^2}{: \text{Bearing number}}
\]

\[
\sigma = \frac{12 \mu \nu / p_{yd} \cdot (r_0 / c_z)^2}{: \text{squeeze number}}
\]

\[
\Omega = \nu / \omega = \sigma / 2 \zeta: \text{Whirl ratio}
\]

The assumptions 1) and 3) mentioned above lead to the following expressions, which are independent on \(\lambda\) or \(\sigma\), for the bearing characteristics \(K\) and \(B\):

Fig. 1 Kinematical model of a bearing for theoretical approach
Fig. 6 Optimum design for minimizing $\Sigma$

Minimizing $\Sigma$

\[
\Sigma = \sum_{\lambda} G(\tau - \phi)
\]

Minimizing $\alpha$

\[
\alpha = \frac{\varepsilon_0 E_{\lambda}}{\lambda}
\]

Minimizing $\beta$

\[
\beta = \frac{\varepsilon_0 E_{\lambda}}{\lambda}
\]

Minimizing $\delta$

\[
\delta = \frac{\varepsilon_0 E_{\lambda}}{\lambda}
\]

Optimum groove designs on evaluation indexes

Designing parameters for groove configuration in the study are: groove width ratio $\alpha$, groove angle $\beta$, and dimensionless groove depth $\delta$ (normalized by $c_t$). The optimum design values are comprehensively searched by computational calculations for each subject expressed with the evaluation indexes. The subjects considered are maximization of $K_0$ (the currently recommended design criterion), minimizations of $E_0$, $A_0$, $F_{\max}$, and $\Sigma$ within the limits of the stable operation. A safety factor $F_s$ for rotor mass is now introduced for a margin of stable operation. The optimum groove design values in the following are; then, obtained under the condition that $M_{s,\lambda} \geq F_s M_{\lambda}$.

In this study, the bearing systems are supposed to be “smooth member rotating” state. $F_s$ is set to be 2.0. It is also noted that the currently recommended groove design (for maximization of $K_0$) is

$$K_0 = \frac{K_{XX}}{\lambda}$$

$$K_1 = -\frac{K_{XY}}{\lambda}$$

$$B_0 = \frac{B_{XX}}{\sigma}$$

$$B_1 = -\frac{B_{XY}}{\sigma}$$

Evaluation indexes in the followings are formulated in terms of these quantities.

(i) Evaluation index for the static displacement

When the load acting on the sleeve is static, then $dZ/d\tau = 0$ follows. Thus, the evaluation index $E_0$, for the static displacement of the rotor is obtained from the equations (1) and (8) as follows:

$$E_0 = E_\lambda \frac{A_0}{G_0} = \frac{1}{\lambda} \sqrt{\frac{1}{3} K_0^2 + K_1^2}$$

where $E_\lambda$ is dimensionless static displacement.

(ii) Evaluation index for the amplitude of RRO due to unbalanced mass

Substituting an expression $Z = A \exp(i(\tau - \phi))$ for rotor behavior into the equations (1) and (8); we obtain evaluation index $A_0$ for the amplitude of RRO as:

$$A_0 = \frac{1}{\lambda} \sqrt{\frac{1}{3} M_{s,\lambda}^2 K_0^2 + 2B_0 K_1^2 + (2B_0 K_1^2)}$$

(iii) Evaluation index for the amplitude of NRRO due to impulse load

When an impulse load is applied on the rotor, the center of the sleeve may move against the shaft with an orbit such as figure 2(a), for an example. Figure 2(b) shows a change of the absolute displacement against dimensionless time $\tau$ drawn from the orbit. Two kinds of evaluation indexes for the rotor behavior are then proposed; one is the maximum displacement $\varepsilon_{\max}$, and the other is the integrated area $\Sigma$ swept by the displacement. They are presented in the dimensionless forms, respectively, as follows:

$$\varepsilon_{\max} = \frac{\max \varepsilon_0 E_{\lambda}}{\lambda}$$

$$\Sigma = \frac{\varepsilon_0 E_{\lambda}}{\lambda}$$

Fig. 2 Example of an orbit of the sleeve against an impulse load.

In addition, we should never forget to examine whether the bearing operation is stable or unstable against ‘half-frequency whirl’, which is inherent in the hydrodynamic bearings. A characteristic value is formulated in the term of critical mass of the rotor, $M_{s,\lambda}$. The condition for critical state is obtained by applying the Routh-Hurwitz criterion to the equation (1) as follows:

$$M_{s,\lambda} = 144 \left( \frac{K_0 B_0}{K_1^2} + B_1 B_2 / K_1 \right)$$

CONCLUSIONS

The evaluation indexes for the operation of spiral-grooved journal bearings concerning static displacement of the rotor and amplitudes of RRO and NRRO are proposed. It is shown that the appropriate groove designs realizing the minimization of the proposed evaluation indexes can improve the bearing performances several times as good as the currently recommended design.

REFERENCES