Dynamic Characteristics of a Hard Disk Drive Spindle System Due to Imperfect Shaft Roundness
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This paper proposes a modified Reynolds equation for the coupled journal and thrust fluid dynamic bearings (FDBs) to include variable film thickness due to imperfect roundness of a rotating shaft. A finite element method is used to solve the modified Reynolds equation to calculate the pressure. Reaction force, moment, and friction torque of FDBs are calculated by integrating the pressure and shear stress along the fluid film. The dynamic behavior of a hard disk drive (HDD) spindle system is investigated by solving the equations of motion with six degrees of freedom using the Runge–Kutta method. This research shows that the imperfect roundness of the shaft increases the nonlinearity of FDBs. Imperfect roundness of the shaft generates harmonics of the groove number ±1 in the bearing reaction force and the displacement of the HDD spindle system even in the case of stationary grooved FDBs.

Index Terms—Dynamic behavior, fluid dynamic bearings (FDBs), groove, Reynolds equation, roundness.

I. INTRODUCTION

FLUID dynamic bearings (FDBs) affect the dynamic performance of the spindle system of a computer hard disk drive (HDD). They are coupled bearings composed of a plain or grooved journal and thrust bearings, as shown in Fig. 1. They provide better dynamics and stability than ball bearings not only because fluid lubricant prohibits solid contact between the rotating and stationary parts but also because they provide a damping effect in addition to stiffness.

Many researchers have studied static characteristics such as pressure, load, and friction torque, as well as dynamic characteristics such as stiffness and damping coefficients. Zang and Hatch analyzed the coupled journal and thrust bearings by using a finite volume method [1]. Rahman and Leuthold calculated the static characteristics and dynamic coefficients with respect to translational motion of the coupled journal and thrust bearings by using a finite element method (FEM) [2]. Jang et al. investigated static characteristics by using the FEM and the Reynolds boundary condition [3]. They verified their method by comparing the simulated flying heights with the experimental ones at various speeds. Jang et al. calculated the nonlinear equations of motion of a HDD spindle system supported by the coupled journal and thrust bearing due to the effect of the bearing width and asymmetric groove [4]. Jang and Yoon calculated the locus of the rotor with rotating grooves with respect to translational motion and show that it generates a bearing reaction force with the harmonics of rotational speed [5].

Prior researchers assumed that FDBs are perfectly manufactured without any machining or assembling errors. However, the clearances of journal and thrust bearings in FDBs are within the range of manufacturing tolerance, so that it is very difficult to maintain high manufacturing precision. Fig. 2 shows the measurements of the roundness of the shaft used in a HDD. It shows that the shaft of a spindle system does not have perfect roundness. The roundness error of a shaft will change the clearance of the journal bearing. As a result, it will affect the reaction forces of journal bearings and the dynamic behavior of the spindle system. Fig. 3 shows the frequency spectrum of the roundness. It shows that imperfect roundness of a shaft contains several significant harmonics, so that these harmonics may affect the dynamic behavior of the HDD spindle system.

This paper proposes a method to determine dynamic behavior of a spindle system caused by the imperfect roundness of a shaft. The variable film thickness of a journal bearing is determined by considering the roundness of a shaft. The Reynolds equations are solved to calculate the pressure distribution by using the FEM, and reaction force and friction torque are obtained by integrating the pressure and shear stress along the fluid film, respectively. Then, the nonlinear equations of motion of a HDD spindle system derived in six degrees of freedom are solved by the Runge–Kutta method.
Fig. 3. Frequency spectrum of the measured roundness of a shaft.

(a) Translational displacement and (b) rotational displacement.

II. METHOD OF ANALYSIS

The Reynolds equations for journal and thrust bearings are given in cylindrical coordinates as follows:

\[
\begin{align*}
\frac{\partial}{\partial \theta} \left( \frac{h_J^3}{12\mu} \frac{\partial p}{\partial \theta} \right) + \frac{\partial}{\partial z} \left( \frac{h_J^3}{12\mu} \frac{\partial p}{\partial z} \right) &= \frac{\dot{h}_J}{2} \frac{\partial h_J}{\partial \theta} + \frac{\partial h_T}{\partial t} \\
\frac{\partial}{r \partial r} \left( r \frac{h_T^3}{12\mu} \frac{\partial p}{\partial r} \right) + \frac{\partial}{r \partial \theta} \left( \frac{h_T^3}{12\mu} \frac{\partial p}{\partial \theta} \right) &= \frac{\dot{h}_T}{2} \frac{\partial h_T}{r \partial \theta} + \frac{\partial h_T}{\partial t}
\end{align*}
\]

where \( R, \ p, \ h_J, \ h_T, \ \dot{h}_J, \ \dot{h}_T, \ \theta, \) and \( \mu \) are the radius of the journal, pressure, thickness of fluid lubricant of the journal and thrust bearings, rotating speed, and viscosity, respectively.

Fig. 4 shows the thickness change of fluid film of the journal bearing for translational and tilting motion. For translational motion of the journal bearing with eccentricity \( e_r \), as shown in Fig. 4(a), the film thickness of the journal bearing is calculated in the following equation:

\[
h_{ij} = c - e_r \cos(\theta - \phi) \tag{3}
\]

where \( c, \ \phi, \) and \( \theta \) are the journal bearing clearance, the attitude angle, and the angular coordinate measured from the fixed negative \( z \)-axis. For additional tilting motions in \( \theta_r \) and \( \theta_\varphi \), the eccentricity \( e \) and the angle change \( \varphi \) can be determined as follows:

\[
e = \sqrt{\left( e_r + (z - z_{mc})\sin \theta_\varphi \right)^2 + \left( (z - z_{mc})\sin \theta_r \right)^2} \tag{4}
\]

\[
\varphi = \tan^{-1} \left( \frac{(z - z_{mc})\sin \theta_r}{e_r + (z - z_{mc})\sin \theta_\varphi} \right) \tag{5}
\]

where \( z_{mc} \) is the height from a mass center of a HDD spindle system in \( z \)-direction.

Then, the film thickness of the journal bearing due to translational and tilting motions can be written as follows:

\[
h_{ij} = c - e \cos \left\{ \theta - (\phi - \varphi) \right\} \tag{6}
\]
Similarly, the film thickness of the thrust bearing is determined by using the flying height and total axial clearance of the thrust bearing. The film thickness of thrust bearing is also derived by considering the translational and tilting motions.

The film thickness due to the imperfect roundness of a shaft can be written by using the Fourier series

$$h_{R} = \sum_{n=1}^{m} A_n \cos(n\omega t - \phi_n)$$

where $\omega$, $A_n$, and $\phi_n$ are the rotating speed of a shaft and the amplitude and phase of the harmonics, respectively.

The final film thickness of the journal bearing can be calculated by the superposition of clearances due to translational and tilting motion of a spindle system and imperfect roundness of a shaft in the following equation:

$$h_{fj} = h_{j} - h_{R}.$$  

Once the thickness of the fluid film is determined, the nonlinear equations of motion of a rigid HDD spindle system can be derived in six degrees of freedom with the cylindrical coordinates, i.e., three translational motion in $r$, $\phi$, and $z$ and three rotational motions in $\theta_r$, $\theta_\phi$, and $\theta_z$ [4].

For the given initial conditions of the FDBs, the pressure distribution of FDBs is determined by solving the Reynolds equations with FEM. Reaction force, moment, and friction torque of FDBs are calculated by integrating the pressure and shear stress along the fluid film. Substituting these values into nonlinear equations of motion for a spindle system allows the calculation of position and angle by using the Runge–Kutta method. This numerical analysis is repeated until the calculated locus of a rotating part converges.

III. RESULT AND DISCUSSION

Fig. 5 shows the finite element model of FDBs. The journal and the thrust bearings have eight herringbone grooves and 20 spiral grooves, respectively. The analysis model is discretized by 2380 elements with four-node isoparametric elements. The spindle system rotates at 5400 rpm, and the mass unbalance of the spindle system for this analysis is assumed to be 10.52 g·0.023 mm, which is the maximum possible value due to the difference between the inner radius of two disks and the outer radius of hub.

The measured roundness of a shaft is included to analyze the effect of imperfect roundness. Figs. 6 and 7 show the radial reaction force and the radial displacement of a shaft with perfect and imperfect roundness. The groove is inscribed in the stationary sleeve in this model. They show that the imperfect roundness of a shaft generates periodic variation in the radial reaction force and radial displacement corresponding to groove number. Fig. 8 shows the frequency spectra, where 1X and 2X are the first and
second harmonics (90 and 180 Hz). It shows that the radial reaction force and radial displacement have the dominant eighth harmonic when there is imperfect roundness of the shaft. Prior researchers have shown that the groove effect is observed only for the rotating groove [5]. However, this research shows that reaction force and displacement can have frequency components corresponding to the groove number even for stationary groove in the sleeve if imperfect shaft roundness exists in FDBs. It also can be explained that the radial reaction force changes whenever the surface of a rotating shaft with imperfect roundness meets the stationary groove.

The radial displacement can be rewritten by using the Fourier series so the displacement in the \( x \)-direction can be represented as follows:

\[
r(t) = \sum_{k=1}^{n} r_k \sin(k \omega t - \phi_k)
\]

\[
x(t) = \cos \omega t \sum_{k=1}^{n} r_k \sin(k \omega t - \phi_k) \]

\[
\frac{1}{2} \sum_{k=1}^{n} r_k \left[ \sin \{(k - 1) \omega t - \phi_k\} + \sin \{(k + 1) \omega t - \phi_k\} \right].
\]

The above equations show that the \( k \)th harmonic of radial displacement is transformed to the \( k \pm 1 \)th harmonic of \( x \)-directional displacement. Fig. 9 shows the frequency spectrum of the \( x \)-directional displacement of a shaft. It shows that the eighth harmonic due to groove effect is modulated to the seventh and ninth harmonics.

This research investigates the reaction force and the displacement of a rotor due to the effect of static forces such as the weight of the spindle system. It assumes that the HDD is vertically positioned with respect to the ground, so that the weight of the spindle system (0.184 N) is applied to the mass center of the spindle system. Table I shows the \( x \)-directional displacement in three cases: perfect roundness of a shaft, perfect roundness of a shaft with static force, and imperfect roundness of a shaft with static force. It shows that the static force and imperfect roundness increase the amplitudes of harmonics of rotational speed due to the increased nonlinearity of FDBs. The amplitude of the second harmonic is much larger than other harmonics because the second harmonic of the roundness is the largest of all harmonics in this model.

### IV. Conclusion

This research proposes a modified Reynolds equation to investigate the coupled journal and thrust FDBs with the imperfect roundness of a rotating shaft. It shows that the imperfect roundness of the shaft increases the nonlinearity of FDBs. It shows that radial reaction force and radial displacement of the HDD spindle system can have a frequency component corresponding to the groove number even for stationary groove in the sleeve if imperfect roundness of a shaft exists in FDBs. Their frequency component corresponding to groove number is modulated to the harmonics of the groove number \( \pm 1 \) in \( x \) and \( y \) directions. It also shows that static force increases the nonlinearity of FDBs. This research may be effectively applied to design robust FDBs in order to increase the dynamic performance and, eventually, the capacity of HDDs.

### REFERENCES


