1 INTRODUCTION

To increase memory capacities of file memory devices, such as hard disk drives and optical disk drives, improvement of recording density is necessary. Both reduction of recording track width and higher head positioning accuracy are important to improve recording density. Ball bearings are generally used in spindles of disk drives at present, but they have been known that ball bearings generate high frequency vibration. Therefore, improvement of head positioning accuracy in high frequency region is difficult when ball bearings are used. Several improvements of ball bearing have been done [1], but they have limits to suppress ball bearing vibrations. Then, another type of spindles using hydrodynamic fluid bearings having high-rotational accuracy and small vibration characteristics have been developed as next generation spindles. Especially, a ferro fluid bearing spindle has advantage to prevent leakage and dispersion of lubricating oil using a ferro seal. In this study, we measured damping characteristics and frequency characteristics of ferro fluid bearing spindles using a high-frequency vibration base driven by a piezo actuator. Using the high-frequency vibration base, high frequency excitation was added to ferro fluid bearing spindles mounted on the vibration base, and we proved that ferro fluid bearing spindles have effective damping. The damping became effective for decreasing non repeatable run out in higher spindle rotational speed.

2 EXPERIMENTAL SETUP

Figure 1 shows the experimental setup. A spindle has been mounted on a newly developed vibration base. High frequency excitation has been enabled using a piezo actuator (Tokin NLA-10x10x18). Acceleration sensors (Fuji ceramics B21S-N704) have been installed to the vibration base. Also displacement sensors (MTI ASP-1-ILA) have been attached to a spindle for measurement of spindle run out. Outputs of sensors have been measured by an oscilloscope and an FFT analyzer.
3 EXPERIMENTAL RESULT

3.1 Measurement of damping effect on spindle

We applied step waveform signal to the PZT to vibrate the vibration base, and have measured step response of the vibration base by an FFT analyser using acceleration sensor under following conditions.

1: Vibration base
2: Vibration base + mass (equal to spindle)
3: Vibration base + a ferro fluid bearing spindle (0 - 15,000 rpm)
4: Vibration base + a ball bearing spindle (0 - 15,000 rpm)

In Figure 3, from the comparison with the cases of vibration base only and the vibration base + mass, the resonance point is moved from 2.1 kHz to 1.8 kHz. In Figure 4, the resonance point change is similar to Figure 3, but the peak value near resonance points of "FFB 0rpm" case is smaller than that of "+mass" case. It shows that ferro fluid bearing in spindle works as a dumper in this case.

In Figure 4, the peak value near 1800 Hz becomes smaller according to the rise of spindle rotational speed. It means that damping effect becomes larger with spindle rotational speed. This can be explained as the following. When the spindle rotated, the shaft rotates near the center of the bearing, and the clearance is almost equally (about 1.5 µm in this case) retained over whole circumferential clearance. Then, an oil film in the bearing clearance seems to demonstrate effective damping.

On the other hand, there is no bearing stiffness when the spindle does not rotate, and the shaft and the bearings seem to have partially contact each other. As the result, the damping effect seems to be larger when the spindle rotates. In addition, measurements when the spindle rotates at 6,000 rpm and at 15,000 rpm were done. By those measurement, a tendency in which damping effect increment was depend on the rise of revolution up to 15,000 rpm was observed.

Next, we compare damping factor of ferro fluid bearing spindle and that of a ball bearing spindle (Figure 6). Though damping effect of the ball bearing spindle is exist (Figure 5), this effect is less than that of the ferro fluid bearing spindle. And, in case of the ball bearing spindle, many peak values appear between 700 Hz to first resonance point and 2,100 Hz to High frequency region. This shows that the ball bearing has much more resonant points than that of the ferro fluid bearing spindle. These many resonant points remain as NRRO (Non-Repeatable Run out) in a high frequency range.

In the near future, when frequency band for track positioning control is expanded, these resonance point will become a problem as NRRO in the high frequency range.

We think that ferro fluid bearings have advantage to ball bearing on this point.
3.2 Measurements of Spindle Run out

We applied sin waveform signal to the PZT to vibrate the vibration base, and have measured spindle run out of a ferro fluid bearing spindle using displacement sensor and acceleration sensor under following conditions.

**Applied SIN waveform:**
- frequency 200 Hz, 280 Hz
- wave amplitude 1.5 to 21 V
- Spindle rotational speed 6000 rpm

We used a lock-in amplifier to increase S/N ratio and succeeded to operate nano meter order measurements. Figure 7 shows spindle run out of X-axis direction (it is parallel to PZT excitation) component, and Figure 8 shows spindle run out of Y-axis direction (it is normal to PZT excitation) component. In this experiments, the vibration base was excited to only X-axis direction, spindle run out was appeared in X-axis direction and Y-axis direction. But run out of Y-axis direction was smaller than that of X-axis direction. This reduction seems to be caused by that experimental ferro fluid bearing is herringbone-grooved bearing.

From the results, it is showed that the ratio acceleration to displacement (i.e. stiffness of spindle shaft) is almost constant with same rotation speed.

![Figure 7: Spindle Run out (X-axis)](image)

![Figure 8: Spindle Run out (Y-axis)](image)

![Figure 9: Simulation model](image)

![Figure 10: Simulation result sample](image)
4 FREQUENCY RESPONSE SIMULATION

Figure 9 shows a simple model of the damping system [2] in this experiment. Ground is vibration base, \( m_1 \) is mass of spindle base, and \( m_2 \) is mass of rotation parts of spindle. \( k_1 \) is sum of suspension and piezo actuator stiffness, \( k_2 \) is bearing stiffness, \( c_1 \) is damping factor of suspension and piezo actuator, and \( c_2 \) is damping factor of bearing. \( m_1 \) and \( m_2 \) are known, \( k_1 \) can be calculated from resonance frequency, and \( k_2 \) can be calculated by spindle run out measurements. We use equation (1) for simulation.

\[
X = \frac{-m_1w^2 + jc_2w + k^2}{(-m_2w^2 + jCw + K)(-m_2w^2 + jc_2w + k_2) - (jc_2w + k_2)^2} F
\]

Here, \( X \) is vibration amplitude of \( m_1 \) by external force \( F \), \( w \) is frequency of sin waveform signal, \( K \) is the sum of \( k_1 \) and \( k_2 \), and \( C \) is sum of \( c_1 \) and \( c_2 \). We made simulation by mathematic application (MathCAD). Figure 10 shows a sample of simulation results. The simulation makes good agreement with experimental results. We are going to consider to adapt this simple mode for the experimental results, and we are going to develop a reasonable model for the realistic vibration system.

5 CONCLUSIONS

Ferro fluid bearing spindles and ball bearing spindles were installed on vibration base, and step response for the excitation by piezo actuator was measured, each characteristics in frequency domain were measured. As the result, it is proven that magnetic fluid bearings and ball bearings have damping effects. And it was proven that magnetic fluid bearings are superior to ball bearings in damping effect.

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7 REFERENCES