2.5 Inch HDD Spindle Vibration with a Flexible Base Plate

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ABSTRACT

The purpose of this paper is to study the effects of the flexibility of HDD base plates on spindle vibration via theoretical predictions and experimental measurements. The flexibility of the base plate can significantly affect HDD spindle vibration. This is the most prominent feature in 2.5 inch HDD. Nevertheless, theoretical analysis of the spindle vibration often neglects the flexibility of the non-rotating part including spindle, base plate, and top cover. Our theoretical model developed in University of Washington can include the flexibilities of spindle and base plate. As a result, our theoretical prediction generally agrees well with our experimental measurements in vibration analysis. Moreover, because of its small form factor, industrial practice is to use flanged disks instead of regular disks in vibration testing of prototypes. Our experimental measurements indicate that flanged disks and regular disks have very different behavior when the frequency is above 1 KHz.

Key Words : HDD, Spindle Vibration, Boundary Conditions, Flexible Base

ABBREVIATIONS

HDD Hard Disk Drive
FDB Fluid-Dynamic Bearing
FEM Finite Element Method
NRP Non-Rotating Part

1. Introduction

2.5 inch HDDs are getting more attention in marketplace as laptop computers gain more popularities. The spindle vibration of 2.5 inch HDD is extremely complicated because spindle vibration is strongly affected by the flexible base and boundary conditions. In other words, when the base deforms, unbalanced (0,0) modes, unbalanced (0,1) modes, and base modes are coupled together. By the demand of HDD industry, the mathematical model developed in University of Washington (Tseng et al., 2003)[4] needs to be verified for 2.5 inch HDD. This paper has two subjects; One is to verify the theoretical model developed by Tseng (2003) for 2.5 inch HDD spindles which is significantly affected by base flexibility. The other is to investigate instrumentation of spindle vibration tests.

The unbalanced momentum from the disk deflection leads to spindle vibration. If the base plate is not rigid, spindle vibration must have a coupling with base motions because the spindle is mounted on the base plate. In general, the first frequency of the base mode in 3.5 inch HDD is much higher than disk (0,1) modes. Therefore, only higher unbalanced modes have a coupling with base modes. In 2.5 inch HDD the first frequency of the base mode is even lower than disk (0,1) modes. As shown in table 1, frequency of (0,1) disk mode for 3.5 inch HDD is 900 Hz; whereas, the first base mode is around 1600 Hz. In contrast, frequency of (0,1) disk modes for 2.5 inch HDD is 900 Hz, which is higher than the first base frequency (400 Hz). Therefore, unbalanced (0,1) and unbalanced (0,0) modes in 2.5 inch HDD have strong coupling with base modes. Figure 1 illustrates the coupling between unbalanced (0,1) modes and unbalanced (0,0) modes with base vibration modes.

Table 1 Frequencies of (0,1) disk and base mode

	3.5-inch	2.5-inch
(0,1) Disk Frequency	900 Hz	900 Hz
Base Frequency	1600 Hz	400 Hz
Coupling	Small	Large

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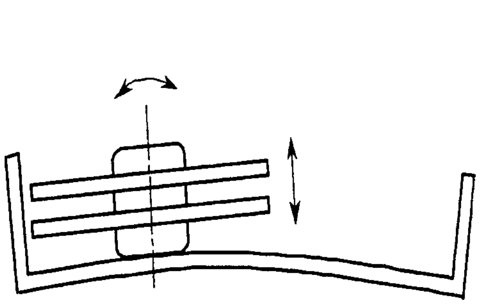

Fig. 1 Spindle vibration coupled with base plate

2. Mathematical Model Adopted

In the last 5 years, University of Washington has developed an accurate mathematical model and numerical simulations to predict response of an HDD disk/spindle system supported by a flexible non-rotating part (NRP) (Tseng et al., 2003)[4]. This flexible NRP can have arbitrary geometry, material properties, boundary conditions, joints, and damping treatments. In addition, the mathematical model can predict free vibration (e.g., natural frequencies, mode shapes, and damping) as well as forced response (e.g., frequency response functions and shock response).

Figure 2 illustrates the mathematical model for the simulation program. This mathematical model consists of three parts; a rotating part, a non-rotating part, and multiple bearings. The rotating part can be either a rotating-shaft or a rotating hub design carrying multiple disks. Rotating part and non-rotating part are connected through bearings. In order to use this mathematical model, FEM is first used to calculate frequencies and modeshapes of the rotating part and the non-rotating part, respectively. A Matlab software tool then uses the FEM results to compensate the gyroscopic effects of the rotating part to predict the response of the spindle vibration.

3. Experimental Investigation

Impact hammer and LDV successfully measures the disk/spindle vibration of 2.5 inch HDD. Figure 3 illustrates the experimental setup. It consists of rotating-shaft design FDB spindle, two identical 2.5 inch aluminum disks, base casting, and fixture. The setup is placed on the isolation table to minimize measurement noise. The base casting is fixed to the fixtures using four screws at four corners. Tiny impact hammer excites the inner rim of top disk and LDV measures disk/spindle vibration at the inner rim of top disk. When impact hammer hits the inner rim of the disk, it excites the spindle modes more than disk modes. Also, when LDV focuses on the inner rim of the disk, it only picks up the spindle modes. According to the experimental results shown in Fig. 4 and Fig. 5, impact hammer and LDV are still good ways to measure vibration of 2.5 inch HDD. Dummy disks are commonly used to test the performance of HDD spindle motors. The top and the bottom disks have different sizes and masses. Also, one of the dummy disks has wider flange to facilitate the measurement of radial motion. Ideally, dummy disks should behave just like the real disks. In order to check how closely dummy disks represent the real disks, first experiment uses real disks then second experiment uses dummy disks. Figure 4 compares the responses of dummy disks and regular disks. We observed that balanced modes of dummy disks seem to be somewhat unbalanced and show some rocking motions of the spindle. This rocking motion is caused by the different mass moment of inertias of the top and the bottom disks. In addition, dummy disks do not simulate regular disks very well at higher frequencies as shown in Fig. 4 because of significant disk/base coupling phenomenon. Therfore, test setups for FDB spindles in 2.5 inch HDD using dummy disks are not adequate.

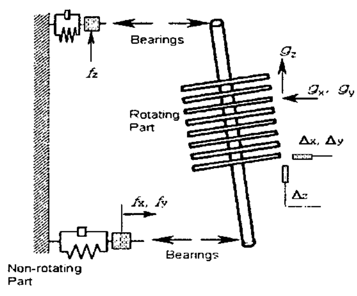

Fig. 2 Theoretical spindle model

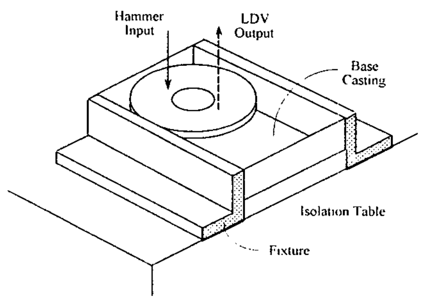

Fig. 3 Experimental setup for 2.5 inch HDD

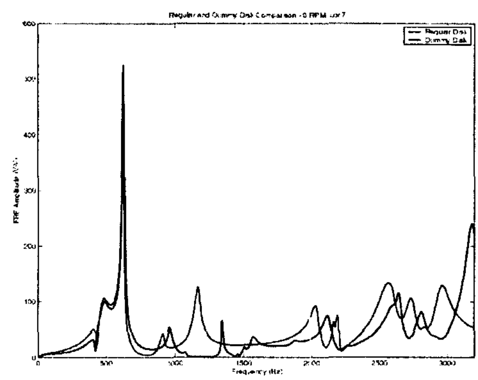

Fig. 4 Dummy disks vs. regular disks

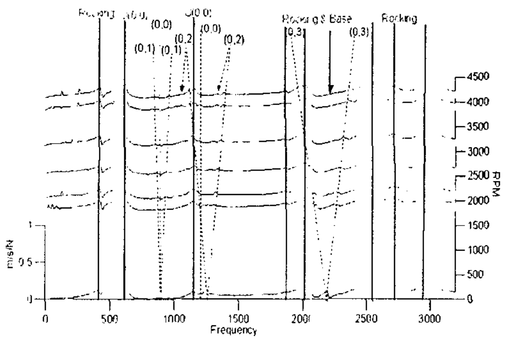

Fig. 5 Waterfall plot for 2.5 inch HDD

Figure 5 is a waterfall plot for 2.5 inch HDD with regular disks and all the modes are identified through STAR modal analysis. Solid line denotes unbalanced modes and dotted line denotes balanced disk modes. The first pair of rocking modes appear between 425Hz and 470 Hz. All the frequencies of unbalanced modes at 4200 RPM are summarized at the second column of Table 3.

4. Verification of Mathematical Model for 2.5 inch Spindle Vibration

The mathematical model uses natural frequencies and modeshapes of the hub and the stationary part as input [3]. Therefore, the numerical accuracy of the mathematical model largely depends on the accuracy of the FEM analyses that are used to obtain the natural frequencies and modeshapes of the hub and the stationary part. For 2.5 inch HDD, FEM of base plate is very important because all the spindle vibration is coupled with the base. In addition, modeling of base plate is extremely difficult due to the complicated geometry. As shown in Table 2 and Fig. 6, all the modes in FEM and STAR modal analysis agree very well in frequencies and modeshapes. Enough number of modes from non-rotating part is a critical issue in obtaining accurate simulations. However, we can not include too many modes because of capacity of PCs. This issue leads to convergence tests in order to identify proper number of FEM modes for non-rotating part. Figure 7 shows frequency convergence of unbalanced (0,1) modes respect to number of modes for non-rotating part. Unbalanced (0,1) modes in simulation converge after 40 modes of non-rotating part.

Fig. 6 First five modes in FEM and in experiments for the base plate

Convergence tests for rotating parts are also performed and results for unbalanced (0,1) modes are shown in Fig. 8. Frequencies of rotating part converge after 10 modes. The reason of fast convergence is that the hub of 2.5 inch HDD is very stiff. 2.5 inch HDD has a steel hub (E=210 GPa) and hub flexibility effects are not significant. In contrast, 3.5 inch HDD has an aluminum hub (E=69 GPa) and hub deformation affects spindle rocking vibration significantly.

Table 2 Verification of the base plate in FEM

FEM (Hz)	Exp. (Hz)	Error (%)
676	618	9.38
1498	1620	7.51
2116	2120	0.20
2204	2284	3.52
3094	3268	5.32
3499	3536	1.06
4281	4368	2.00
4668	4464	4.57

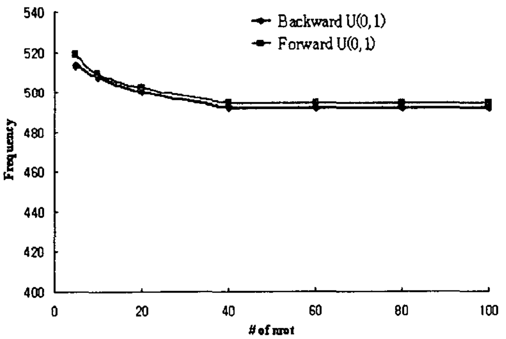

Fig. 7 Frequency convergences for non-rotating part

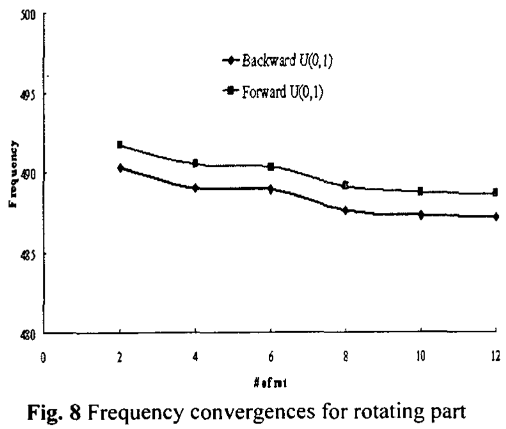

Fig. 8 Frequency convergences for rotating part

Table 3 Comparison of simulations with experiments

	Theory (Hz)	Exp. (Hz)	Error
1^{st}	472.2	425	10%
U(0,1)	612.5	-	-
2^{nd} (0,1)	2797.1	-	-
	2889.8	-	-
U(0,0)	663.5	608	8.8%
U(0,0)	1174.5	1160	0.5%
Base	1998.6	1860	6.9%
	2089.7	2020	3.3%
	2164.3	-	-

The software is able to simulate both free response and forced response of disk/spindle system. The simulation results need to be verified through experimental measurements. The free response provides the frequency and modeshape of each mode. In order to verify the free response, the resonance peaks of the measured FRF must be identified using existing methods including finite element method (FEM) and STAR modal analysis software. Figure 9 compares the frequencies in simulation with experimental measurements from 0 to 4200 RPM. Dotted line is theoretical prediction in frequency respect to disk spin speed and solid line is experimental measurements. Table 3 is frequency comparisons at 4200 RPM. The errors between the simulations and the experiments are 0.5% to 10%. Overall prediction in frequency and modeshape matchs well with experimental measurements.

Figure 10 compares FRFs at 4200 RPM. Unbalanced (0,0) modes have a good match in amplitude and frequency. However, unbalance (0,1) modes do not appear in the simulation. The source of the mismatch on

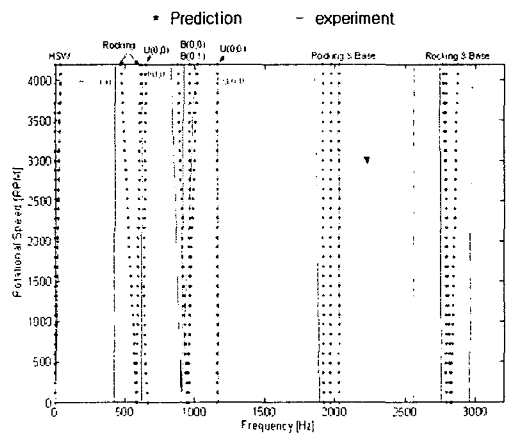

Fig. 9 Campbell Diagram

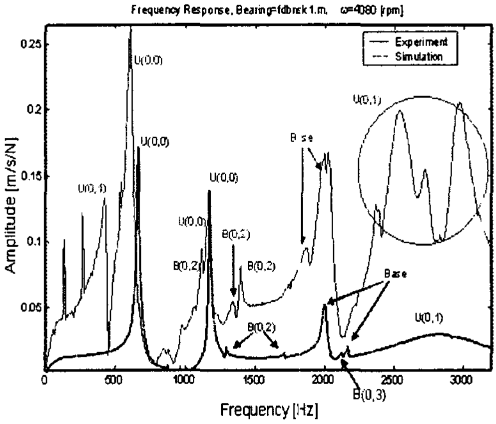

Fig. 10 FRF comparison

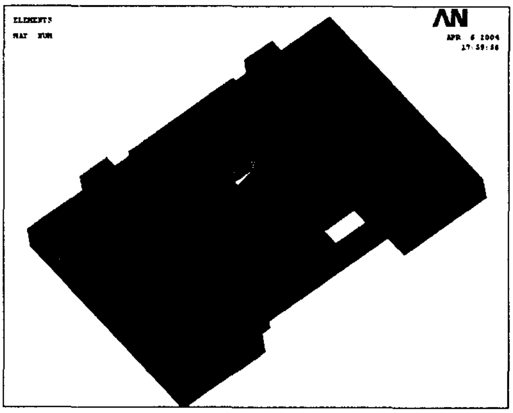

Fig. 11 Modification of boundary condition

the amplitude is three uncertainties; experimental errors, wrong bearing coefficients, and FEM of the base plate. Experimental errors may come from the lack of spindle motor samples. HDD companies use their own simulations to calculate the bearing stiffness and damping coefficients. However, it is extremely difficult to calculate accurate bearing coefficients. FEM of base plate simplifies the geometry because the shape of base plate is extremely complicated. The amplitude in FRF simulation has a room to improve in the resonance amplitude of unbalanced (0,1).

5. Boundary conditions of base plate

The effects of non-rotating part are very significant in 2.5 inch HDD FDB spindle performance. Non-rotating part is also very sensitive to the boundary conditions. Therefore, different boundary conditions of non-rotating part lead to different spindle vibration. To study boundary conditions of non-rotating part FEM simulates three modified boundary conditions of non-rotating part. First boundary condition is perfectly fixed in FEM at the four corners of base plate, and this boundary condition is the base line. Second boundary condition includes glue layer between the sleeve and the base plate because HDD manufactures apply epoxy and press fit the sleeve into the sleeve holder on the base plate. Last boundary condition includes fixture. Four screws at four corners of the base plate bolt down to fixtures as shown in Fig. 11. In summary, the first boundary condition is that four scew nodes are perfectly fixed in the space, and the second boundary condition includes glue effect between motor sleeve and sleeve holder, and the last boundary condition includes the effect of the flexibility of fixtures.

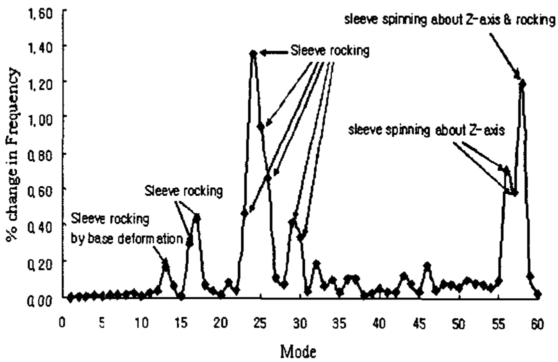

Fig. 12 Effects of Epoxy layer in NRP frequency

Ten micron of Epoxy layer (E=1GPa) is added between sleeve and sleeve holder. Figure 12 shows the % changes of frequencies in FEM for the non-rotating part. Epoxy layer changes the non-rotating part frequencies less than 1.5%. Therefore, it doesn't affect FRF results. When fixture is added to the base plate, fixture reduces higher mode frequencies of non-rotating part up to 27% as shown in Fig. 13. Before adding fixture, the errors in frequencies of FEM and experiments are 6.9% and 3.3% for the base modes as shown in Table 3. After adding fixture, the errors in frequency are improved to 2.8% and 2.4%. Therefore, accurate boundary conditions in FEM improve the frequency accuracy in simulations as shown in Table 4.

6. Conclusions

1. Hammer and LDV measurements remain reliable for 2.5-in HDD spindles.

2. Experimental measurements indicate a lot of coupling between disk (0,1) and (0,0) modes with base vibration.
3. Dummy disks do not simulate regular disks in spindle vibration because of the different mass moment of inertias of the top and the bottom disks.
4. Theoretical predictions on resonance frequencies are accurate. However, predictions on resonance magnitude have room for improvement.
5. Accurate boundary conditions in FEM for NRP improve base mode frequencies in FRF predictions.

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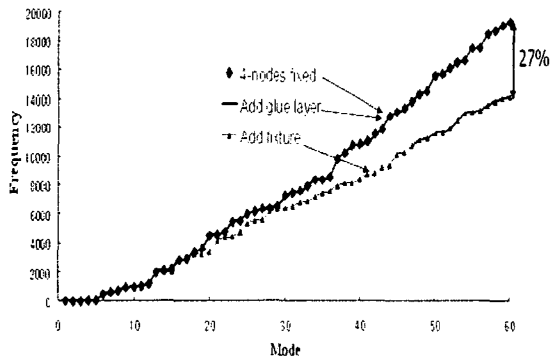

Fig. 13 Effects of fixture in NRP frequency

Table 4 Comparison of simulations with experiments after adding fixture

	Theory (Hz)	Exp. (Hz)	Error
1^{st} U(0,1)	472.2	425	10%
	612.5	-	-
2^{nd} (0,1)	2758.9	-	-
	2848.7	-	-
U(0,0)	**649.2**	608	**6.3%** (8.8%)
U(0,0)	**1161.9**	1160	**0.2%** (0.5%)
Base	**1912.6**	1860	**2.8%** (6.9%)
	1972.8	2020	**2.4%** (3.3%)
	2035.5	-	-

Reference

[1] T. Jintanawan and I. Y. Shen, 2000, "Free vibration of a rotating disk pack and spindle motor system with rotating shaft design", J. Inform. Storage Processing Syst., vol.2, pp.129–139.

[2] T. Jintanawa, I. Y. Shen, and K. Tanaka, 2001, "Vibration analysis of fluid bearing spindles with rotating-shaft design", IEEE Trans. Magn., vol.37, pp.799–805.

[3] J. Y. Shen, C. W. Tseng, and I. Y. Shen, "Vibration of rotating disk/spindle systems with flexible housing/stator assemblies", J. Sound Vibration, submitted for publication.

[4] Tseng CW, Shen JY, Shen IY, 2003, "Vibration of rotating-shaft HDD spindle motors with fexible stationary parts," IEEE Trans Magn., Vol.39, No.2, pp.794.799

[5] B. Heo and I. Y. Shen, 2005, "A Parametric Study on Rocking Vibration of Hard Disk Drive Spindle Motors with Fluid-Dynamic Bearings and Rotating-Shaft Design", Microsystem Technologies: Micro- & Nanosystems and Information Storage and Processing Systems.

[6] Baekho Heo, 2004, "Vibration Reduction of Rotating Disk/Spindle Systems via Structural Design", Ph.D. dissertation, Univ. of Washington.