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Vibration Analysis and Damage Detection of the Bearing on Hard Disk Drive Tester

Jiraphon Srisertpol, and Kittisak Photiseang

Abstract**-----**This article is to study, conduct an experiment, and analyze the vibration of the Gemini tester, which is an apparatus widely used in various industries. The vibrations normally cause errors and damages on parts. In case of Hard Disk Drive (HDD) industry, the vibration at the Pocket Slot also presents the poor performance. This study had analyzed the property of a Bearing by using Modal Analysis. The natural frequency of the Bearing was examined by the Measurement Technique. The experiment results demonstrated a useful technique for the vibration control of the Bearing in the manufacturing process.

Keywords**—** Vibration Analysis, Bearing, Parameter Estimation

I. Introduction

COMPUTER is considered a useful tool nowadays since its high speed computing with accuracy, precision, and its sharp display. One of the main key inside - components of this powerful tool is the Hard Disk Drive (HDD). Therefore, the HDD industry has been developed for its better efficiency and reliability in order to satisfy the customers. The test process of HDD is an important step to claim for the reliability of products. The test duration of a HDD varies to its capacity. The vibration generated by a Bearing is a key factor to examine HDD's error and damage as shown in Figure 1. And when replace bearing with the new one then Test Slot become to normal. The degeneration of bearing can't be observed as normal shown in Figure 2. This study aimed to analyse the vibration of a Test Slot affecting the mechanical properties of HDD and to classify the bearing that were suitable for the test process to control the vibrating effect of HDD. [1].

In 1984, Clarence W. and Sam S. [2] had analysed the natural frequency of vibration in parts of a shaker by a method of Experimental Modal Analysis (EMA). The results were displayed in both time and frequency domains.

Jiraphon Srisertpol and corps [3, 4] had analyzed the vibration of transfer module that effects to Head Gimbal Assembly(HGA) base on operating Walter D. Pilkey and Sergev V.Purtsezov [5] to present finding of the appropriate variables to get a shock isolator to control the mechanical properties.

Jiraphon Srisertpol is with Department of Mechanical Engineering, Suranaree University of Technology, Nakornratchasima, Thailand. (corresponding author's e-mail: jiraphon@sut.ac.th).

Kittisak Photiseang, Department of Mechatronics Engineering, Suranaree University of Technology, Nakornratchasima 30000 Thailand, and Seagate Technology, 90 Moo 15 Tambol Sungnoen Amphur Sungnoen Nakornratchasima 30170 (e-mail: kittisak.photiseang@seagate.com)

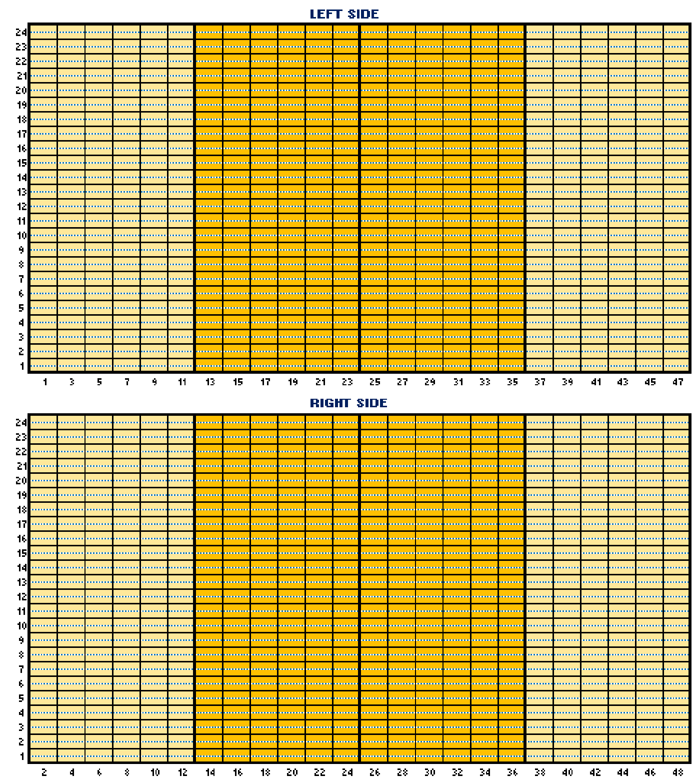

Fig. 1 Test Slot error location

Fig. 2 Bearing appearance

II. Procedure

The model of a shaker used to test for the natural frequency of the HDD vibrations was depicted in Figure 3. The shaker was tested with the method of Base Excitation which its amplitude was constant, $Y(t)=28.284\sin\omega t$, and it was shook under various frequencies to examine for the natural frequency.

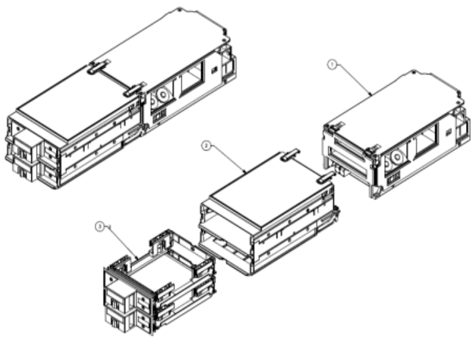

Fig. 3 Structure of the Test Slot

To finding of the isolator properties. The response of the Isolator was analyzed by the methods of Parameter Estimation for the spring stiffness and the damping coefficient comparing with measurement technic.

In the Figure 4, the experiment of the Test Slot with Base Excitation method was given constant amplitude and was observed by the Accelerometer Sensor.

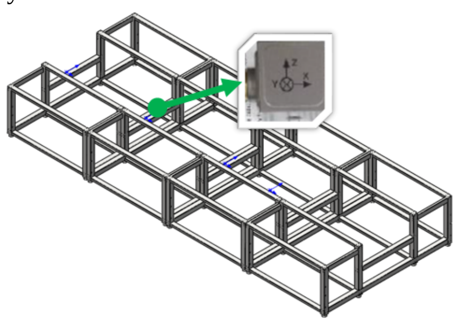

Fig. 4 Experimental Setup

In figure 5, The mathematical model and the Free Body Diagram of the simulating experiment were depicted as following,

$$
\begin{aligned}
m_1\ddot{x}_1 + c\dot{x}_1 + (k_1+k_2)x_1 - cx_2 - k_2x_2 &= k_1 y \\
m_2\ddot{x}_2 + c\dot{x}_2 + k_2x_2 - c\dot{x}_1 - k_2x_1 &= 0
\end{aligned}
\tag{1}
$$

$$
\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix}\begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{bmatrix} + \begin{bmatrix} c & -c \\ -c & c \end{bmatrix}\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} + \begin{bmatrix} k_1+k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix}\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} k_1 \\ 0 \end{bmatrix} y
\tag{2}
$$

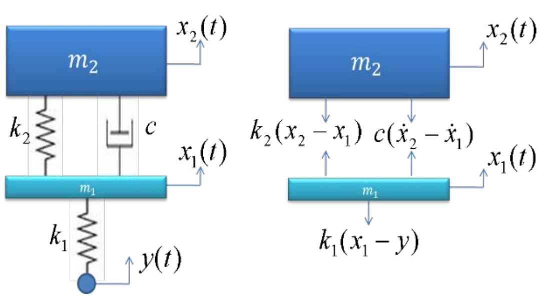

Fig. 5 Free Body Diagram of Simulating Experiment

The experimental of the model has an initial value for the variable as following.

- Mass value m_1 = 85 kg
- Mass value m_2 = 50 kg

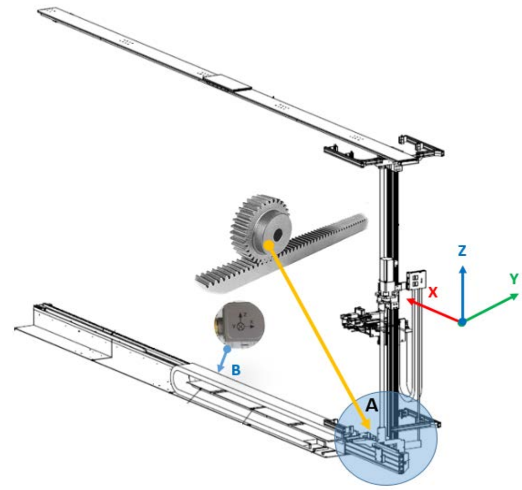

Fig. 6 Setup Position Bearing

This article has an experimental setup three proper and improper bearing property by separates in 3 case as following.

- Case I: Setup all positions with proper bearing property.
- Case II: Setup an almost improper bearing property in positions A.
- Case III: Setup improper bearing property in positions A.

III. Tool And Equipment

The Dynamic Signal Analyser series 35670A of Agilent Technologies shown in Figure 7 was employed to measure the vibrations.

Fig. 7 Dynamic Signal Analyser

The Accelerometer Sensor series 8395A of Kistler with the ability of measurement 3 axis along x, y and z axes was employed for the accelerate measurement depicted in Figure 8, and the Shaker series M1200W of Spectral Dynamic was depicted in Figure 9, respectively.

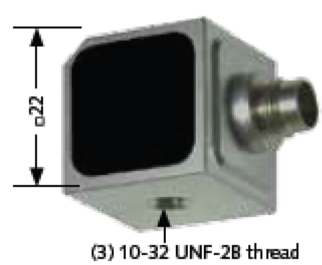

Fig. 8 The Accelerometer Sensor series 8395A

Fig. 9 The Vibration and amplifier for measurement

IV. Result And Discussion

From result and vibration analysis of bearing of Test slot that determine amplitude of base constant $y(t) = 28.28 \sin \omega t$ and various frequency in range 0 – 2400 Hz. The experiment will be finding stiffness (k1) by base excitation method (single degree of freedom), value is 7.3x106 N/m. And apply k1 to estimate stiffness (k2) and damped (c) in 2DOF (Experimental setup). The result display in fig 10, Xm is response of measurement and Xs is response of simulation. Error between Xm and Xs is small.

Case I: The result show that in fig 11, T1, T2 and T3 is good bearing, time trial 1-3. Frequency acting in range 201–400 Hz and 1001–1200 Hz that acceleration in (g) and display in table 1.

Case II: The result show that in fig 12, T1, T2 and T3 is almost improper bearing, time trial 1-3. Frequency acting in range 201– 400 Hz and 1001–1200 Hz that acceleration in (g) and display in table 2.

Case III: The result show that in fig 13, T1, T2 and T3 is improper bearing, time trial 1-3. Frequency acting in range 201– 400 Hz and 1001–1200 Hz that acceleration in (g) and display in table III.

TABLE I
PROPER BEARING, ACCELERATION IN (G)

Bearing classify	Frequency(Hz)	T1	T2	T3
G1 (8 MCs)	201 - 400	0.027	0.029	0.031
	1001 - 1200	0.016	0.018	0.022

TABLE 2
ALMOST IMPROPER BEARING, ACCELERATION IN (G)

Bearing classify	Frequency(Hz)	T1	T2	T3
G2 (8 MCs)	201 - 400	0.041	0.045	0.046
	1001 - 1200	0.064	0.065	0.065

TABLE III
IMPROPER BEARING, ACCELERATION IN (G)

Bearing classify	Frequency(Hz)	T1	T2	T3
G3 (8 MCs)	201 - 400	0.072	0.080	0.080
	1001 - 1200	0.103	0.111	0.106

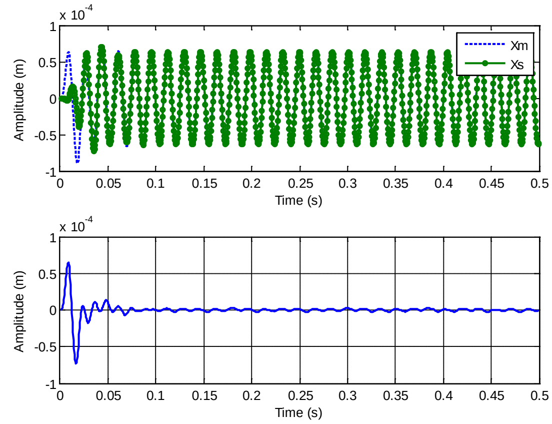

Fig. 10 Response between Measurement and Simulation

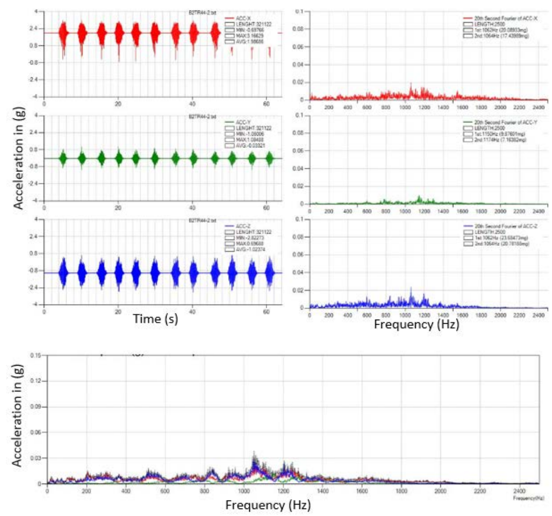

Fig. 11 Displacement transmissibility and Frequency of proper bearing

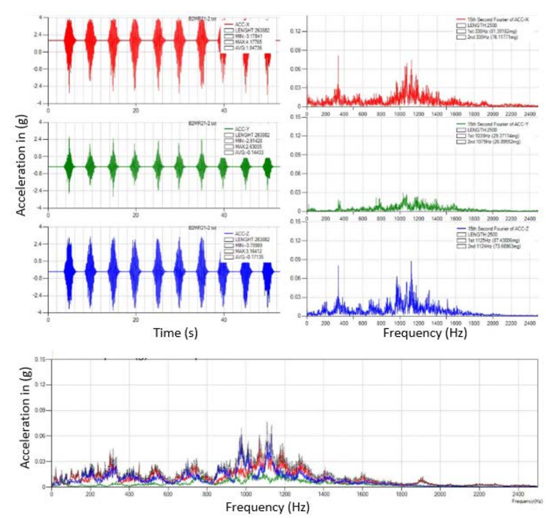

Fig. 12 Displacement transmissibility and Frequency of almost improper bearing

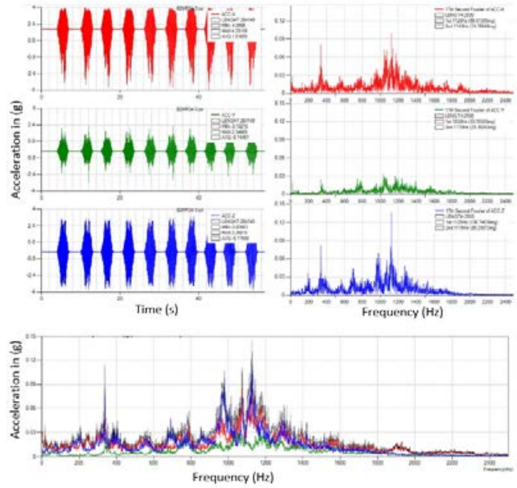

Fig. 13 Displacement transmissibility and Frequency of improper bearing

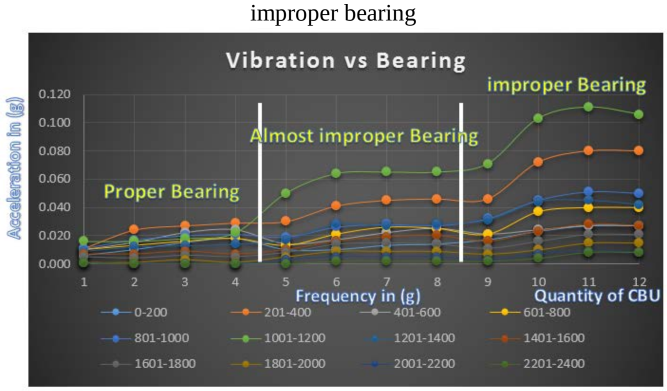

Fig. 14 Displacement transmissibility and Frequency of 3 bearing comparison

The mechanical property of proper bearing had 2 frequency range are 201-400 and 1001-1200 Hz, the vibration has classify almost improper bearing on acceleration in (g) at value 45 and 65 that specification useful for replacement bearing as shown in Table II.

If we will analyze vibration contribute to HDD in test process. We need to know force that generate from HDD test process, for analyzed and design to detect improper bearing to reduce vibration or increase Tester performance.

V. Conclusions

This paper was to study the change of mechanical properties in the vibration bearing of a Test Slot in order to classify the appropriate bearing for the Hard Disk Drive test process. The method of Base Excitation and the measurement were employed to analyze for the natural frequency. The investigation could decrease the damage from the vibrations effect.

Acknowledgment

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