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DESIGN AND ANALYSIS OF AUTO-BALANCER OF AN OPTICAL DISK DRIVE USING ROTATIONAL VIBRATION ABSORBERS

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Abstract

With the increased demands in performance, the optical disk drive needs to be high speed and more stable. A novel design of a vibration absorber used in reducing the vibration caused by the unbalance of an optical disk drive is proposed. When an optical disk is rotating, the vibration caused by the unbalance resulting from the non-homogeneous disk increases as the rotating speed increases. The proposed vibration absorber with a circular shape is installed beneath the optical disk and rotates with the optical disk. The objective is to use shape optimization technique to make the first radial natural frequency of the absorber coincides with the rotating speed of the optical disk in a specific rotational frequency range. Therefore the natural frequency corresponding to the radial mode of the vibration absorber becomes a speed-dependent. It can effectively suppress the vibration in the radial direction caused by the unbalance of the optical disk and acts as an auto-balancer when the optical disk drive operates in this frequency range. Numerical results from numerical simulations using finite element method prove the feasibility and show that the unbalance of the optical disk can be reduced effectively.
Keywords: Vibration absorber, unbalance, auto-balancer, optical disk, shape optimization

INTRODUCTION

Increasing the data storage capacity of an optical disk accompanied by the demands of high data access rates and high positioning accuracy of read/write heads has recently become a stringent necessity. With the increased demands in performance, the optical disk drive needs to be rotating with high speed and more stable.

The optical disk possesses a certain degree of imbalance due to manufacturing tolerance, and this imbalance causes sever radial vibration when the disk is rotating with a high speed. In order to suppress the excessive radial vibrations caused by the imbalance, the automatic balancer system (ABS) which consists of several free-moving ball-type masses running in specific circular races around the rotor is utilized. The ball-type masses in ABS tend to settle at positions to counter-balance the disk imbalance when the motor spindle is rotating. Owing to its simple mechanism, the ABS becomes a favorable device for reducing the unbalance in an optical disk drive system. Several implementations and novel designs were proposed and patented [1-3]. Several studies have been conducted, either experimentally or theoretically, in order to have a better understanding of the dynamic characteristics of the ABS [4-6]. However some issues, such as rolling friction of balancing balls [5] [6], effectiveness limitation caused by rotating speed [7], instability [4], etc. still need to be addressed when an optical disk drive system is equipped with an ABS. The ball mis-positioning caused by the rolling friction hampers the effectiveness of the ABS and sometimes even results in an excess vibration.

A novel design of a dynamic balancer is proposed to reduce the vibration caused by the imbalance of an optical disk drive and provides an alternative besides the ABS. The proposed dynamics balancer is in fact a circular structure and is installed below the optical disk. It will rotate with the disk, therefore its natural frequency varies with the rotating speed due to the change in-plane stress caused by the centrifugal force. The idea is to make the varying first radial natural frequency of this rotating circular structure to coincide with the rotating speed of the optical disk in a specific rotational frequency range. Under this circumstance, it will keep resonant and act as a vibration absorber in this specific frequency range. Theoretically it can effectively suppress the vibration in the radial direction of the rotor-disk system caused by the imbalance. Although the mechanism of a vibration absorber is different from that of ABS, it serves the same purpose of reducing the unbalance of the rotor-disk system. As compared to the ABS, it does not have the shortcoming, such as mis-positioning caused by the rolling friction and instability. And most of all, it has a much simpler mechanism and can be fabricated using an injection mold machine in a cost-effective manufacturing process.

DESIGN CONCEPT OF SPEED-DEPENDENT VIBRATION ABSORBER

The vibration absorber has been proved to be effective in reducing the machine vibration. Fig. 1 shows a typical dynamic response of a machine before and after the vibration absorber is implemented [8-9]. By simply attaching the vibration absorber to the machine and tuning its natural frequency to that of the machine, the original resonance of the machine disappears. However, two new natural frequencies where resonances may occur take place instead. When the new resonances occur, the absorber vibrates in phase and out of phase with respect to the machine, respectively. If the machine operates at a frequency between these two new natural frequencies, the machine vibration can be reduced dramatically. It is well known that the frequency range between the two natural frequencies depends on the mass of the vibration absorber as compared to that of the machine itself. Usually the machine operates at the original resonant frequency of the absorber and enjoys the great vibration reduction. A wide frequency range between the two natural frequencies is preferred in order to avoid situations where excitation frequency might drift around the nominal value. One may use a heavier absorber to increase the operating frequency range; however, the weight of vibration absorber is always limited in the design. If the natural frequency of a vibration absorber can be tuned in such a way that it varies along with the changes of excitation frequency, the vibration absorber may become more effective in a wide frequency range. It is also well-known that the excitation frequency for a typical imbalance for a rotary machine is the same as the rotating speed of the shaft. If the natural frequency of a vibration absorber is designed to be the same as the rotating speed of the disk shaft and can also adapt itself to follow the varying rotating speed, under this circumstance, the vibration absorber will have its best performance in countering the imbalance in a wide frequency range.

It is well-known that the natural frequency of a rotating structure will change due to the centrifugal force [10]. Fig. 2 shows a rotating structure of which the fundamental frequency varies with respect to the rotating speed. The stress stiffening effect caused by the centrifugal force increases the structural bending rigidity and thus it has an influence in increasing the natural frequency of the rotating structure. On the contrary, the spin softening effect caused by the same centrifugal force tends to increase the structure length; therefore it decreases the natural frequency of the rotating structure. Inspired by the phenomenon of the varying natural frequency of a structure due to the rotational effect, a rotating speed-dependent vibration absorber is proposed to suppress the imbalance caused by the optical disk at varying rotating speed.

In order to fulfill the requirement of self-tuning the natural frequency at a varying rotating speed, a vibration absorber as shown in Fig. 3 is introduced. It has a simple circular geometry and is designed to be installed beneath the disk, which is also the same position as the traditional ABS. This vibration absorber consists of several ribs connected by a rim. When the vibration absorber is rotating with the shaft, each rib behaves like a rotating beam of which axial stiffness will increase due to the centrifugal force and thus its radial natural frequency can be altered. Note that the vibration absorber possesses different kinds of modes that depend on the excitation frequency. Among all the vibration modes, the radial mode that represents the vibration absorber vibrates in the radial direction is the only vibration motion to be utilized to counter the imbalance. Therefore the objective is to make the radial natural frequency of this absorber to vary with the rotating speed. An optimal algorithm based on size optimization is developed to find the optimal dimension of the rib to ensure that the natural frequency corresponding to the first radial mode of the vibration absorber is close to the rotating speed in a specific frequency range, i.e. 6000 RPM to 11000 RPM in this study.

SIZE OPTIMIZATION OF SPEED-DEPENDENT VIBRATION ABSORBER

Shape optimization involves the determination of parts of or all of the profiles of the absorber boundaries. There are two ways to describe the structural profile. One is to represent the structural shape by a set of mathematical functions; another is to describe the profiles by a set of control points. In order to reduce the number of design variables, the sizes of the absorber, h_1 the inner rim thickness, h_2 the outer rim thickness, R_1 the inside radius of outer rim, R_2 the radius of outer rim, w the rib width as shown in Fig. 3, are chosen to be the design variables. Now the shape optimization problem can be defined as follows:

Minimize

$$F_2(x_1, x_2, ..., x_j, .., x_n) = \sum_{i=1}^{m} (\omega_i - \Omega_i)^2 \qquad \text{for} \quad \Omega_1 \le \Omega_i \le \Omega_u \tag{1a}$$

subject to

$$L_{lj} \le x_j \le L_{uj}, \tag{1b}$$

where x_j is the design variable, n is the total number of the design variables, Ω_i is the shaft rotating speed in Hz, ω_i is the natural frequency corresponding to the first radial mode of the vibration absorber, m is the total number of the discretized frequencies in the rotating speed from Ω_l=100 Hz (6000 RPM) to Ω_u=170 Hz (10200 RPM), L_{lj} and L_{uj} represent the upper and lower bounds for the design variable x_j.

As a first step, a numerical algorithm, which incorporates commercial structural analysis code ANSYS, mathematical subroutines IMSL libraries and the user-supplied routines is proposed. The advantage is simply that it extends the capability and flexibility of ANSYS. ANSYS calculates the responses of a complicate structure such as stress analysis, dynamic response, etc., whereas IMSL is responsible for providing subroutines including optimization, transformations, matrix operations, etc. The user-supplied subroutine is required for calculating the objective function, interfacing between IMSL and ANSYS; and controlling the iteration. In other words, this algorithm provides a general and an automatic procedure in a general optimization framework. Among all the optimization methods, the sub-problem approximation method was chosen to perform the optimization procedure. It has the advantage that the derivatives of the objective function are not required, i.e. a zero-order method. One may choose other zero order method, such as the principal axis method to accomplish the same task [11, 12].

NUMERICAL RESULTS AND DISCUSSIONS

Assume that the vibration absorber is made of rubber with a Young's modulus E=0.012×10^9 Pa and density ρ = 960 kg/m^3. The dimension of the vibration absorber before size optimization is listed in Table 1. The first four mode shapes are illustrated in Fig. 4. The first mode is easily identified as a torsional mode with four ribs bending in phase circumferentially. The second mode is a radial mode where two ribs exhibit in phase and in plane bending; and the other two ribs show longitudinal vibration. In this mode the rim of the absorber vibrates in the radial direction, so the absorber response is similar to that of a disk caused by the imbalance. The third mode is also a radial mode whose natural frequency is a little higher than the second mode. Two adjacent ribs vibrate transversely but out of phase, therefore the rim of the absorber moves in the direction of 45 degree relative to rib axial direction. Notice that only the second mode which exhibits vibration in the radial direction will be used to counter the imbalance caused by the rotating disk. Therefore the objective of the speed-dependent vibration absorber is to vary the natural frequency corresponding to this radial mode in accordance with the disk rotational speed in a specific rotational speed range. Under this circumstance, the vibration absorber will keep resonant at varying disk rotating speed and theoretically acts as an auto-balancer to counteract the imbalance caused by the disk.

By utilizing the algorithm which incorporates the ANSYS and the IMSL introduced in the previous section, the vibration absorber was optimized with in-equality constraints as shown in Table 2. The final dimension of the vibration absorber after optimization is listed in Table 3. The natural frequency corresponding to the second mode at rotating speeds of 0 RPM, 6000 RPM, 8040 RPM, 9000 RPM and 10200 RPM of the vibration absorber after size optimization is listed in Table 4; and the corresponding mode shape are shown in Fig. 5. It shows that the second mode keeps the same mode shape where two ribs exhibit in phase and in plane bending and the other two ribs show longitudinal vibration during the disk run-up from 6000 RPM to 10020 RPM. The natural frequency corresponding to this mode during the disk run-up is shown in Fig. 6. In this figure, notice that the natural frequency of the second mode of the speed-dependent absorber varies in accordance with the disk rotational speed in the range from 6000 RPM to 10020 RPM with a discrepancy less than 4 Hz. This result numerically validates that the radial natural frequency of the vibration absorber can be altered to approximately match with the disk rotating speed. The question left is what the performance is if the vibration absorber is installed in the optical drive and will be addressed in the following section.

PERFORMANCES OF SPEED-DEPENDENT VIBRATION ABSORBER ACTING AS AUTO-BALANCER

To evaluate the performance in vibration reduction of the proposed vibration absorber used as auto-balancer on an optical disk drive, several numerical examples are presented.

As shown in Fig. 7, a finite element model according to an optical disk drive is created using the commercial package ANSYS. Owing to the complexity of a real optical disk drive, this finite element model is simplified and consists of only the key components which are a spindle, a disk, a turntable, a sleeve, a washer and a clamp. In addition, an imbalance of 3g is assumed to be located on the disk. The material properties for each component are listed in Table 5. The disk is fixed at its inner circle by the clamp on the turntable but the disk is free to rotate with the turntable and the clamp without slip due to the washer. Similarly, the displacement at the bottom of the spindle is assumed to be fixed both in the axial and radial directions but it are free to rotate in the axial direction. This finite element model was meshed using a total 22856 solid 45 elements and the frequency response of the disk caused by the imbalance can be determined straightforward. The non-dimensional radial displacement responses $(MX)/(me)$ at the outer rim of the disk with and without the speed-dependent vibration absorber are shown in Fig. 8 for the rotating speed from 0 RPM to 10200 RPM, where M is the total mass of the disk with the imbalance, m is the imbalance mass, e is the mass center of the imbalance and X is the radial displacement. It shows that the radial vibration is obviously reduced with the absorber being installed as expected, especially at 4000RPM to 7000RPM where three vibration peaks are suppressed.

CONCLUSIONS

A novel design of a vibration absorber used in reducing the disk vibration caused by the imbalance of an optical disk drive is proposed. It provides an alternative in countering balance of the disk besides the traditional automatic balancer system. The proposed vibration absorber is installed at the same position as the traditional ABS and rotates with the shaft. When the rotor of the optical driver is rotating, the natural frequency of the proposed vibration absorber corresponding to its first radial mode varies due to the inertia force. The objective is to design a vibration absorber whose radial natural frequency not only varies with the rotating speed but also coincides with the varying the rotating shaft. The design task is accomplished using shape optimization technique. Results from numerical simulations show that it can effectively suppress the imbalance of the optical drive system when the drive operates in a specific frequency range. Moreover, as compared to the traditional ABS, it has a much simpler mechanism and can be fabricated using an injection mold machine in a cost-effective manufacturing process.

Table 1 – Dimensions of Absorber before optimization (Unit: mm)

R_1	R_2	r_1	r_2	w	h_1	h_2	r_3	L
13.5	14	1.5	2	0.1	3	0.3	2	28

Table 2 – Constraints of Absorber during optimization procedure (Unit: mm)

	R_1	r_2	r_3	w	h_1	h_2	R_2
Low bound	13	1.7	0.5	0.1	0.3	0.3	13.1
Upper bound	15.4	6	3.5	0.4	3.8	3.8	15.5

Table 3 – Dimensions of Absorber after optimization (Unit: mm)

R_1	R_2	r_1	r_2	w	h_1	h_2	r_3
15.18	15.28	1.5	4.9	0.14	3.8	0.52	1.6

Table 4 – Natural frequencies and mode shapes of vibration absorber at varying rotational speed

Mode number	0 RPM (0Hz)	6000 RPM (100Hz)	7020 RPM (117Hz)	Mode type
1	15.396	66.397	76.265	Torsional (θ)
2	23.797	106.62	121.65	Radial (r)
3	25.633	111.47	127.50	Radial (r)
Mode number	8040 RPM (134Hz)	9000 RPM (150Hz)	10020 RPM (167Hz)	Mode type
1	86.144	95.444	105.32	Torsional (θ)
2	136.48	150.24	164.64	Radial (r)
3	143.45	158.38	174.14	Radial (r)

Table 5 – Dimensions and material properties of a simplified disk drive

Components	Material	Young's modulus (Pa)	Density (kg/m^3)
Disk	Plastic	2.16×10^9	1200
Spindle	Steel	1.9×10^{11}	7850
Sleeve	Aluminum	72×10^9	2800
Washer	Rubber	4×10^6	1300
Turntable	Aluminum	7.2×10^{10}	2800
Clamp	Plastic	1.4×10^9	960
Imbalance	Plastic	2.16×10^9	1200

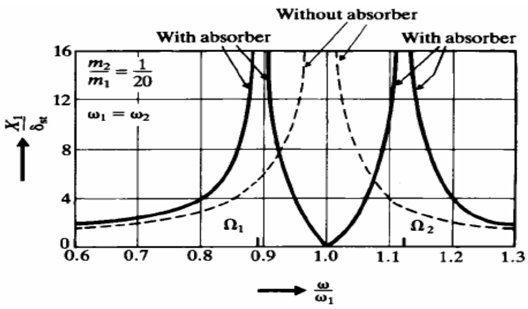


Figure 1 – The dynamic response of the machine with/without the vibration absorber [8]

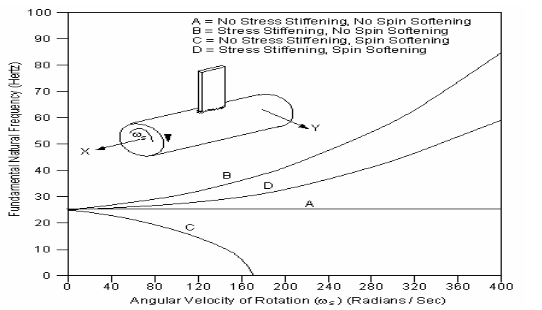
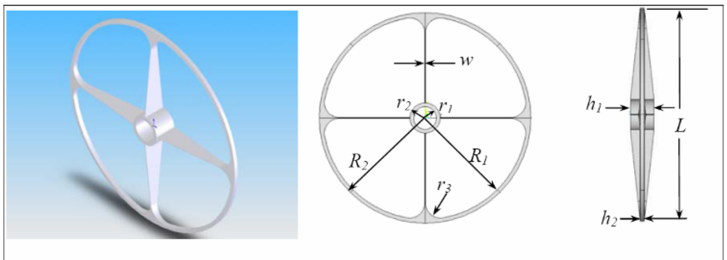
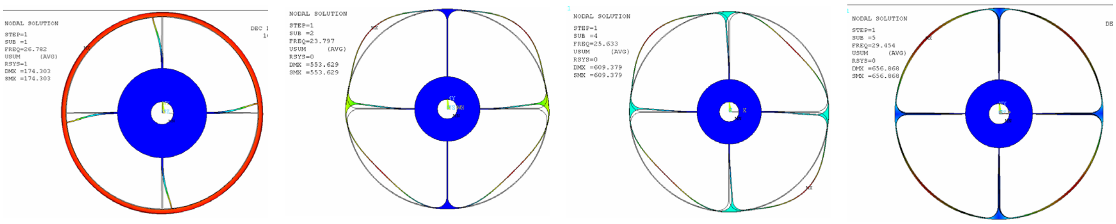


Figure 2 – The effect of stress stiffening and spin softening on the fundamental frequency of a rotating structure [8]



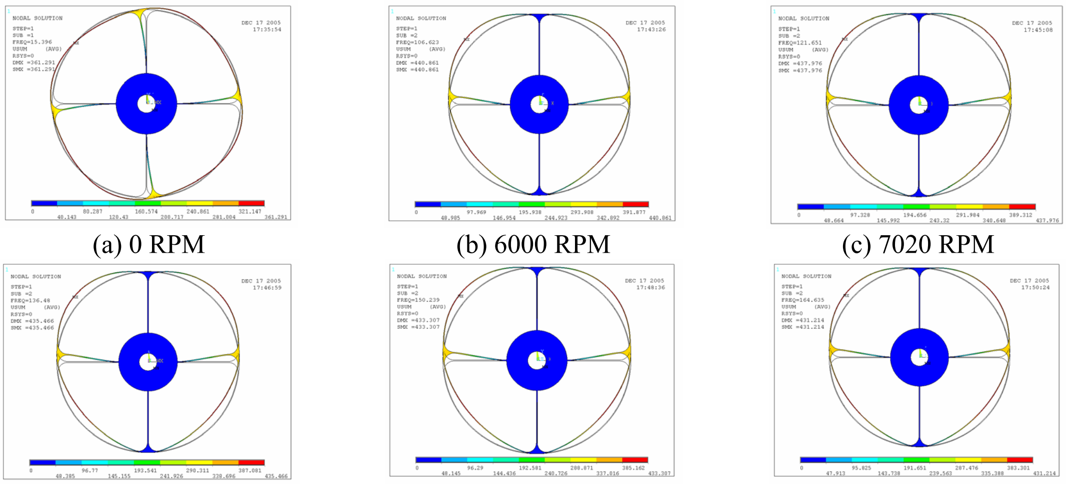
(a) 3-D view (b) Front view (c) Side view

Figure 3 – The proposed vibration absorber



(a) The 1st mode (b) The second mode (c) The third mode (d) The forth mode

Figure 4 – The first four mode shapes of the vibration absorber at 0 RPM before optimization



(a) 0 RPM (b) 6000 RPM (c) 7020 RPM

(d) 8040 RPM (e) 9000 RPM (f) 10020 RPM

Figure 5 – The natural frequency corresponding to the radial mode of vibration absorber at varying rotational speeds after optimization

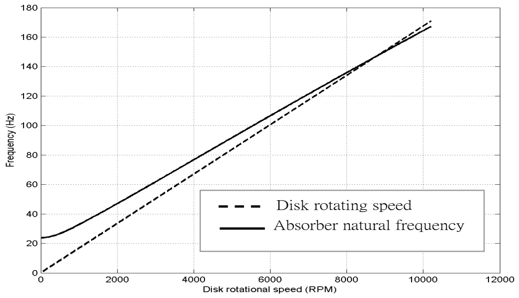


Figure 6 – The natural frequency corresponding to the 2nd mode of the vibration absorber at varying rotating speeds

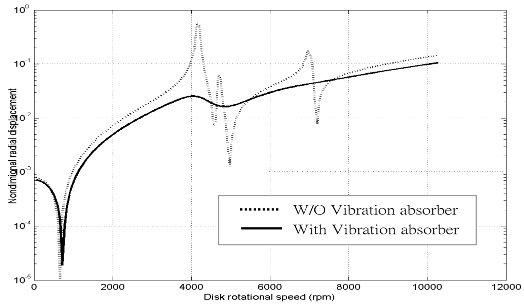


Figure 8 – Comparisons of Vibration displacement before and after vibration absorber being installed

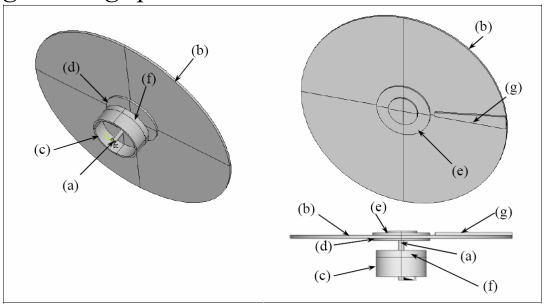


Figure 7 – Rotor-disk model; (a) spindle, (b) disk, (c) sleeve (d) turntable, (e) clamp, (f) washer, (g) imbalance

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