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Design of a Rubber Mount for Vibration Reduction in a Slim Optical Disk Drive

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슬림형 광디스크 드라이브의 방진마운트 설계

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Abstract

With the increase of track density, high rotational speed and the compatibility for various media in optical disk drives, the effective design to vibration reduction is very important for robust operation. Especially when a slim optical disk drive for a notebook PC is excited by a mass-unbalanced disk, internal vibration and its transmission to external case bring about severer problem than that of conventional one. In this paper a design process of a rubber mount in a slim optical disk drive for vibration reduction is presented. The characteristics of rubbery materials – hyper-elastic and visco-elastic – are measured with standard specimens. The static stiffness of a rubber mount was calculated by FEM and the dynamic stiffness is predicted with the static stiffness and the impedance test data of the standard specimen. The transmissibility tests are performed for the purpose of verification of the design process.

Key Words : Rubber mount(고무마운트), Slim optical disk drive(슬림형 광디스크 드라이브), Finite element analysis(유한요소해석), Vibration reduction(진동저감)

1. Introduction

Recently many technological advances have been accomplished in the optical disk drives. The CD-ROM drives of 52× were already commercialized and they are operated with high rotation speed over 10,000rpm. One of technological challenges in the design of the high speed optical disk drives is how to reduce the vibration caused by rotation of a mass-unbalanced disk[1]. The vibration can be a disturbance to the tracking/focusing servo system. The disturbance in the high speed optical disk drives may cause severe failures in reading and writing

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process. The vibration is also transmitted to the external case and causes users to feel unpleasant vibration and acoustic noise during operation. An automatic ball balancer system and a dynamic vibration absorber are some effective solutions to the vibration induced problems[2~5]. For a slim optical disk drive, however, it is not easy to achieve reliability of their balancing functions because of the spatial limit of mechanical design. Actually the height limit of a slim optical disk drive for a notebook PC is less than 13mm. The only approach may be to maximize vibration reduction performance of a rubber mount.

Generally rubbery materials have high performance on the vibration absorber, easiness to handle and cost effectiveness. That's reason why the rubbery materials are widely used in most commercial optical disk drives as shock and vibration absorbers. But there is a problem on determining the stiffness of a rubber mount, to reduce the vibration of the pick-up base and the external case at the same time. As the stiffness of a rubber mount increases, the vibration transferred to the external case increases and the vibration of the pick-up base decreases. On the contrary, as the stiffness of a rubber mount decreases, the vibration of the external case decreases and the vibration of the pick-up base increases. Accordingly the stiffness of a rubber mount should be precisely selected to satisfy the allowable specification level of the vibrations.

Generally rubbery materials have some difficulties in modeling their static and dynamic behavior. Those are originated from the non-linearity in stress-strain relation and dependency on frequency, pre-strain and temperature. In this paper, a design process for a rubber mount in a slim optical disk drive through finite element analysis, considering in rubbery material's non-linearity and the dependency on frequency, is presented.

2. Determination of material properties

The vibration characteristics of a rubber mount can be described by its stiffness, which depends on such parameters as the hardness and the shape. And the stiffness determines the static deflection and the natural frequency of the system supported by the rubber mounts. To determine the stiffness of a rubber mount, the static behavior (hyper-elastic) and the dynamic characteristics(visco- elastic) are investigated.

2.1 Static behavior

The foundations of hyper-elastic theory and applications to the modeling for rubbery materials appear to be well established[6,7]. The basic assumption is that of a strain energy function U such that

$$S_{ij}=\frac{\partial U}{\partial \varepsilon_{ij}} \tag{1}$$

where S_{ij} are the second(symmetric) Piloa-Kirchhoff stress tensor and ε_{ij} are components of the Green-Lagrange strain tensor. For the rubbery materials that is assumed to be initially isotropic, the strain energy function is given by

$$U = U(I_1, I_2, I_3) \tag{2}$$

Here, strain invariants I_1, I_2, I_3 can be expressed as principal stretches λ_1, λ_2, λ_3:

$$\begin{aligned} I_1 &= \lambda_1^2+\lambda_2^2+\lambda_3^2 && \text{(a)} \\ I_2 &= \lambda_1^2\lambda_2^2+\lambda_2^2\lambda_3^2+\lambda_3^2\lambda_1^2 && \text{(b)} \\ I_3 &= \lambda_1^2\lambda_2^2\lambda_3^2 && \text{(c)} \end{aligned} \tag{3}$$

With incompressibility as a condition of constraint, I_3=1, U becomes a function of only two variables I_1 and I_2. Among several models that represent the strain energy function of rubbery materials, it is widely known that the Mooney-Rivlin model is well matched with experimental data up to 150% strain:

$$U = C_1(I_1-3) + C_2(I_2-3) \tag{4}$$

where C_1 and C_2 are material constants to be determined from experiment.

Uniaxial tension tests with UTM(Universial Testing Machine, Instrone) were performed to extract the material

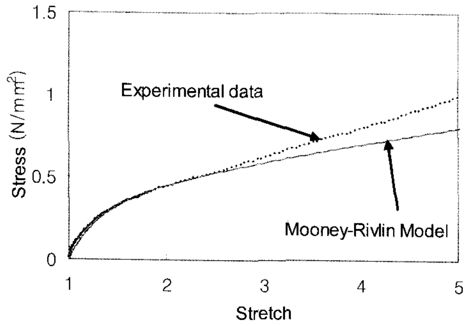


Fig. 1 Stress-stretch relationship from uni-axial test and curve fitting for Mooney-Rivlin coefficient

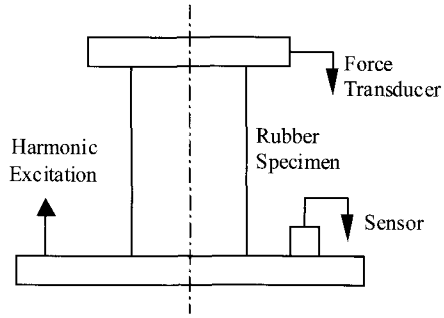


Fig. 2 Schematic of the impedance test for a cylinder block rubber specimen

constants of a butyl rubber. The test specimen was selected according to KS M6782(Korean Standards). Tension rate was 5mm/min and temperature was 23℃. Figure 1 shows the result of nominal stress-stretch relation of durometer(ASTM) hardness 30° specimen and a fitted curve using least square method. From the test, C_1 and C_2 were obtained: 4.91×10^4, $1.59\times10^4 N/m^2$.

2.2 Dynamic behavior

For the rubbery materials subjected to linear small dynamic deformations superimposed on nonlinear static strains, it has been suggested that the storage modulus can be factored into a function of frequency and a function of strain[8]. Let λ_s means the static stretch, then,

$$E_c(\omega,\lambda_s) = E_c(\lambda_s)\cdot G(\omega) \tag{5}$$

where $E_c(\omega,\lambda_s)$ and $E_c(\lambda_s)$ are the storage moduli in a dynamic and a static state under a certain strain. Therefore, $G(\omega)$ means the ratio between static and dynamic property. The storage modulus is obtained from the impedance test.

$$E_c(\omega,\lambda_s) = \frac{L}{A}\cdot\frac{1}{H(\omega)} \tag{6}$$

where, A and L are the cross sectional area and the height of a cylinder block specimen, and $H(\omega)$ is the compliance frequency response function. Figure 2 shows the impedance test.

The dynamic stiffness, therefore, can be derived as follows,

$$K(\omega)=K(0)\frac{E_c(\omega,\lambda_s)}{E_c(0,\lambda_s)} \tag{7}$$

where, $K(0)$ and $K(\omega)$ are the static and dynamic stiffness of the rubber mount and $E_c(0, \lambda_s)$ and $E_c(\omega, \lambda_s)$are the static and dynamic storage modulus of the standard specimen, respectively. In this paper, it is assumed that $E_c(\omega, \lambda_s)/E_c(0, \lambda_s)$ is not dependent upon the shape of a rubbery material and the dynamic characteristics are linear under the small dynamic amplitude. So, it is possible to predict the dynamic stiffness of a rubber mount which has an arbitrary shape from the test data of a standard specimen.

Impedance tests with dynamic materials testing machine(Instron Model 8590) were performed to extract dynamic characteristics(storage modulus and loss factor) of rubbery materials. The cylindrical specimen of 15mm diameter and 5mm height was selected. Figure 3 shows the dynamic characteristics of durometer hardness 30° specimen. Both the storage modulus and the loss factor show the frequency dependency obviously.

3. Design procedure

3.1 Design requirements

In this paper, design requirements are given as follows: the natural frequency of the system supported by the

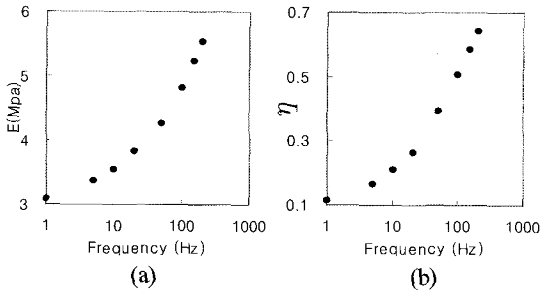


Fig. 3 Dynamic characteristics of a cylindrical rubber specimen of durometer hardness of 30°: (a) storage modulus; (b) loss factor

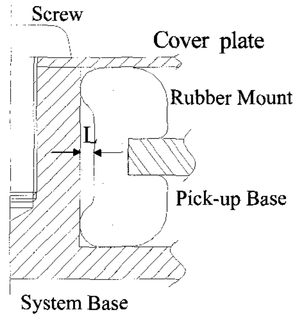


Fig. 4 Configuration of rubber mount

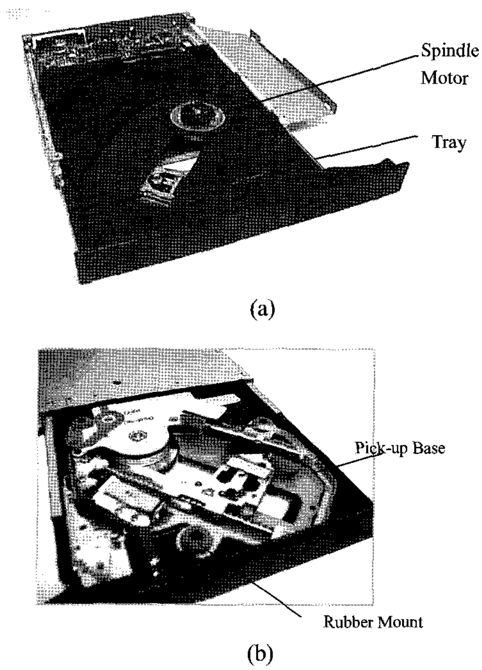


Fig. 5 Slim optical disk drive: (a) top view; (b) bottom view

rubber mounts is required over 200Hz and the maximum static deflection by the weight of the system supported by the rubber should be less than 0.020mm. Since the mass supported by a rubber mount is about 30g, from the natural frequency point of view, the required stiffness becomes over 47,400N/m. In this case, static deflection by system weight is 0.006mm. From the impedance test previously mentioned, the ratio of $E_c(\omega, \lambda_s)/E_c(0, \lambda_s)$ around 200Hz was obtained to about 2. This means that the dynamic stiffness at 200Hz extends to 2 times higher than the static stiffness. The required static stiffness of a rubber mount, therefore, becomes over 23,700N/m and the static deflection is 0.012mm which is smaller than the requirement.

3.2 Finite element analysis of rubber mount

A shape of the rubber mount was selected as shown in Fig. 4 with the consideration of assembly configuration and spatial limitation. Figure 5 also shows a typical assembly configuration, which is widely used in practice. The pick-up base, on which various important parts such as pick-up, spindle motor, feed motor, etc., are assembled, is supported by the rubber mounts. Rubber mounts are screwed on the system base with cover plate. Fig. 5 shows a slim optical disk drive.

After the shape of rubber mount is determined, detailed design process, that is, determination of an appropriate hardness and fitting dimensions for a rubber mount is performed using finite element analysis. In this research, the durometer hardness 30° is selected and a dimensional variable L (shown in Fig. 4) as an important design parameter is selected. Two values of the variable L are considered, 0.4 and 0.6mm.

Figure 6 shows the finite element model of the rubber mount. Based on the symmetry of shape, loading and boundary conditions, an axisymmetric finite element model is used with HYPER 56, 4 node element, which is able to describe hyper-elastic behavior in the ANSYS com-

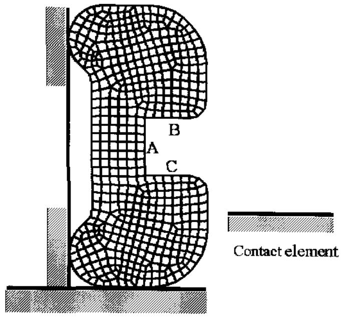
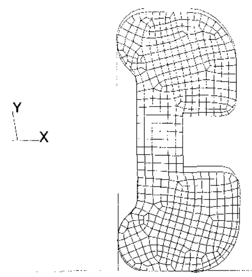


Fig. 6 Axisymmetric finite element model of the rubber mount

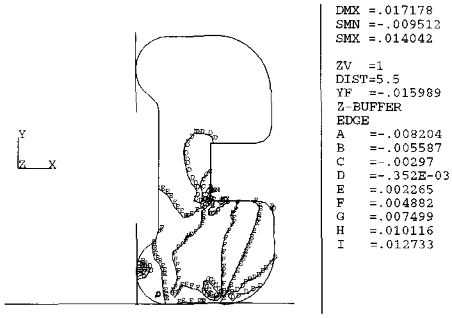
puter program[9]. The contact regions between the rubber mount and the pick-up base is considered as follows; (1) region A deformed by weights is considered to be fixed in tracking(radial) direction and free in focusing(axial) direction; (2) region B and C which supports the pick-up base assumed to be rigid is considered to be coupled in axial direction and free in radial direction. To numerically simulate these conditions, first, contact elements are used at the bottom side of rubber mount and at the boss surface of system base, second, the weight loaded to a rubber mount is 30g, which is used for the loading condition on region C.

The static deflections for two values of L(shown in Fig. 4), 0.4mm and 0.6mm were calculated as 0.0089mm and 0.0161mm, respectively, which means that the static stiffness are 33,000 and 18,300N/m respectively. Considering that $E(\omega)/E(0)$ is about 2, the dynamic stiffness are found to be 66,100 and 36,500N/m respectively. So, the expected natural frequencies of the system become 236Hz and 176Hz respectively. Accordingly, the value of 0.4mm for the dimensional variable L was selected.

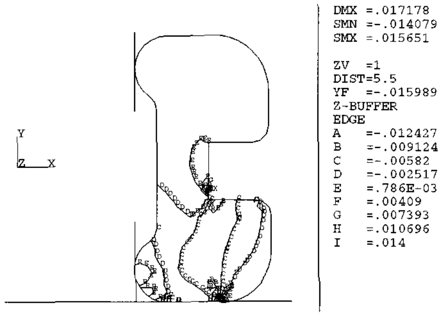
Figures 7(a) ~ 7(d) show the deformation shape and the strain distributions. The maximum tension and compression are respectively 1.4% and 0.95% in x direction and 1.5% and 1.4% in y direction. In the case of shear deformation, the maximum value is about 3.4% and the deformation resulted from the system weight in the below part becomes very large. Generally a rubbery material is weak to the shear deformation, so the information for the shear deformation is useful in the design of a rubber



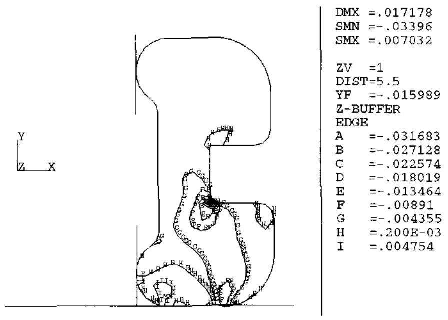
(a)



(b)



(c)



(d)

Fig. 7 Analysis results: (a) deformation; strain distribution of (b) axial direction, (c) radial direction

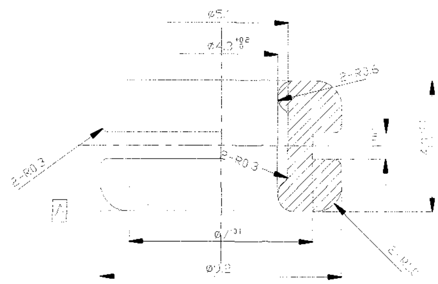


Fig. 8 Final design of the rubber mount

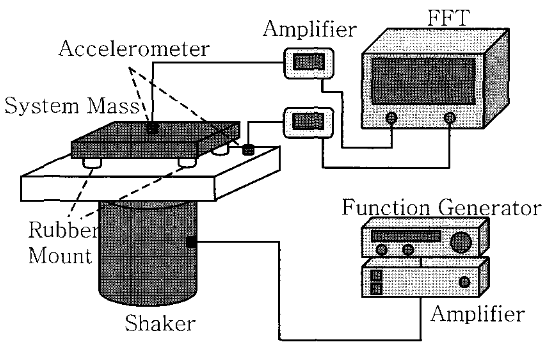


Fig. 9 Experimental setup for transmissibility measurement

mount.

4. Evaluation

4.1 Transmissibility tests

Based on the above research, a rubber mount is fabricated. Figure 8 shows the shape and detailed dimensions of the designed rubber mount.

Transmissibility tests are performed to evaluate the vibration characteristics of the designed rubber mounts. The experimental setup for the frequency response function is presented in Fig. 9. The system mass equivalent to the inertial property of all parts inside the pick-up base is excited by a shaker(LDS V4.50) according to the sweep signal generated by a function generator(LDS DSC4-CE) and the input and output signals picked up by two accelerometer(B&K 4374). These two signals are analyzed by a FFT analyzer(B&K 2816).

Figure 10 shows the transmissibility of a rubber mount. The measured natural frequency was 262Hz, which shows about 11% difference in compared with the predicted natural frequency, 236Hz. This difference is not large and is acceptable in practice. Table 1 shows the measured natural frequencies for 4 samples of the manufactured rubber mounts. Good agreement is found with the predicted value, which shows that the presented design process is valuable, that is, the dynamic stiffness of a rubber mount and the natural frequency of a system can be predicted if the impedance test data of standard rubbery specimen are in hand.

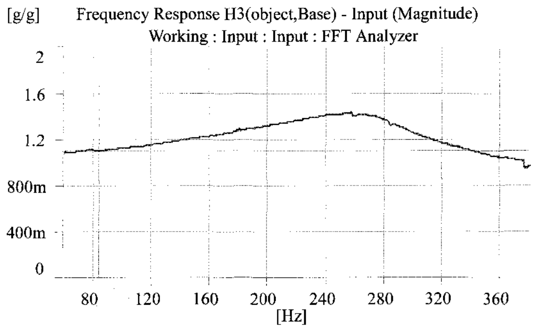


Fig. 10 Transmissibility of a rubber mount

Table 1 Natural frequencies of the rubber mounts [unit: Hz]

No	1	2	3	4	Average
Natural Frequency	255.5	263.5	265.0	264.0	262.0

5. Conclusion

The following conclusions can be drawn from the study.

(1) A design process for a rubber mount in a slim optical disk drive, considering the non-linearity in stress-strain relation and the dependency on frequency of the rubbery material, is presented, which can predict the dynamic stiffness of a rubber mount and the natural frequency of the system supported by the rubber mount.

(2) To validate the analysis, rubber mounts are fabricated and transmissibility experiments are performed. Good

agreement between experiments and simulations is found for the natural frequency. This suggests that the current analysis can be applicable for designing a new rubber mount.

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