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HDD 서스펜션의 모달 튜닝

김 동운* 박영필**

Modal Tuning of HDD suspension system

Young-Phil Park and Dong-Woohn Kim

ABSTRACT

The dynamic characteristics of a HDD suspension system are investigated by finite element analysis and experimental modal analysis. A finite element model of the suspension Type850 was developed for unloaded case. The calculated vibration modes were compared with measurements and agree well in shape and frequency except some local modes. Local thickness and Young's modulus of the finite element model are updated by modal tuning method to develop the precise FE model. A sensitivity matrix of the natural frequencies for some design variables was calculated using finite difference method. Most natural frequencies calculated by the tuned FE model coincide with the measurements and the errors between them are less than 2%.

1. Introduction

HDD contains high performance electro-mechanical components and actuator mechanism is one of major components. Suspension in Actuator mechanism connect the sliders to the VCM actuator. Magnetic read/write heads are mounted onto the sliders and high accuracy and high speed positioning of the sliders on the disks is established by the suspension.

To increase the recording density of HDD, both linear and track density must be increased. Since the linear actuator was replaced to therotary actuator for high speed small size disk drive, access acceleration occurs in lateral direction of suspension assembly. Therefore, the bandwidth of the positioning servo is limited by the resonant frequency of the suspension assembly in access direction.[1] Throughout the disk drive industry, It is accepted that in order to ensure adequate system performance, the frequency of the lowest structural resonance must be at least two octaves above the bandwidth of the servo system.[2,3]

The suspension is a very important component which required stringent design condition. The suspension assembly must be compliant enough to follow undulations in the disk and stiff enough to accurately and quickly position the slider. So, It must be very soft in three direction, i.e. vertical, pitch, and roll and stiff in the remaining direction, i.e., radial, tangential and yaw. [3]

This paper describes the dynamic characteristics of an unloaded Type 850 suspension assembly with pico-slider. Finite Element Model is constructed for numerical analysis and validated using an experimental modal analysis system consisting of an impact hammer and an

* Dep. of Mech. Engineering, Yonsei Univ.

** Professor, Dep. of Mech. Engineering, Yonsei Univ.

LDV. Local thickness and Young's modulus are selected as design variables to tune the modal parameter. The sensitivity matrix for modal frequency was calculated using finite difference method.

2. Suspension Description

The suspension assembly used in this study is shown in Fig.1. This suspension on which the pico slider is mounted is a component of Samsung HDD model SW0434A. It is manufactured by Hutchinson Technology Inc.

The main component of the suspension system is the load beam made of a rolled stainless steel. It is attached to the base plate by seven spotwelds. The base plate, also stainless steel, is mounted on the E-Block. The flexure is made of the same material as the load beam and fixed to the other side of the load beam by two spotwelds. The aluminum oxide-titanium carbide slider is glued to the center tab of the flexure. There is a hemispherical dimple at the end part of the load beam which accommodates the pitch and roll of the slider.

The suspension is analyzed in its unloaded state. In this condition, it is not deflected as if flying over the disk. Understanding of the behaviour of the unloaded suspension system provides a better frame of reference for understanding the dynamic behaviour of the loaded one. The compliance modes which disappeared in loaded case are evaluated in unloaded case analysis. [4]

3. Finite Element Analysis

3.1 FE Modeling

FE Analysis of the suspension system is performed using ANSYS , commercial FEA S/W package. 4 node shell elements(SHELL63) were used for the sheet metal parts, and 8 node brick elements(SOLID45) were used for the slider and base plate. The entire model consisted of 1725 elements(1593 SHELL63 and 132 SOLID45) and 2174 nodes. Connected nodes at the spotweld location are coupled with each other using node coupling. one pair of node for dimple point is coupled in the vertical direction to allow to rotate relative to each other. Material properties for the model are shown in Table 1.

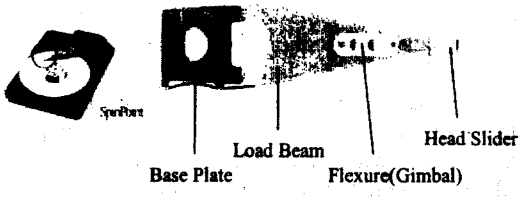

Fig. 1 Picture of the 850 type suspension assembly

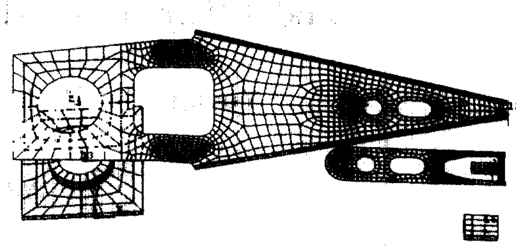

Fig.2 FE Model of Suspension system

Table 1 Material properties used in model

Component	Young's modulus [GPa]	Density [kg/m^3]	Poisson ratio	Thickness [mm]
Load Beam	193	7890	0.32	0.076
Flexure	193	7890	0.32	0.035
Base Plate	193	8020	0.32	
Slider	393	3125	0.23	

3.2 Modeling Results

It is commonly known by suspension manufacturers that the geometry of the bend region can have substantial effects on certain modes. [4] FE Model in this study has a 9.2 ° bending angle and 25% bend ratio.

Fig. 3 shows the calculated natural frequencies and mode shapes of the suspension system. Large movement of the slider to the lateral direction appeared in 2nd torsion mode and sway mode. In sway mode, local deflection of the flexure in lateral direction occurred, also.

4. Experimental Modal Analysis

Fig. 4 shows a schematic representation of the modal testing system utilized to measure the suspension modal parameters. Measured modal parameters are used to validate and tune the finite element model developed in this study.

Experimental system consists mainly of a laser doppler vibrometer (LDV), impact hammer, dynamic signal analyzer and modal parameter

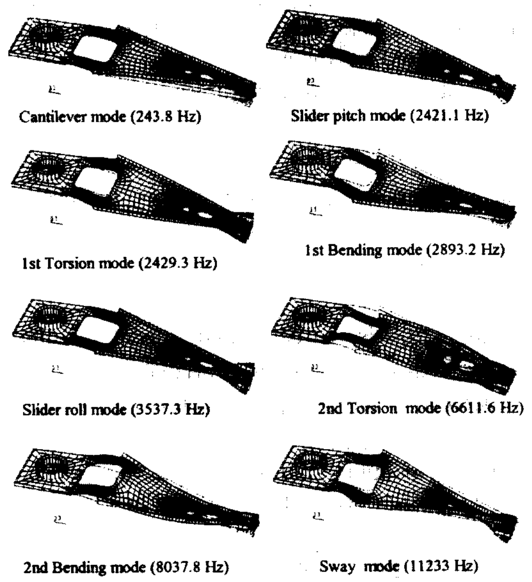

Fig 3. Calculated natural frequencies and mode shapes of the suspension system

estimation software. The base plate of the suspension assembly bonded to the base block by adhesive. Base block was rigid enough to have high natural frequencies of flexible modes

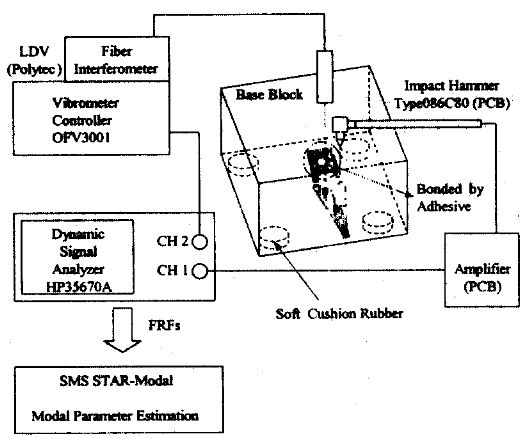

Fig. 4 Schematic of the experimental system

compare to suspension assembly. Similarly base block was heavy with soft cushion rubber to have lower frequency of rigid mode than the natural frequency of the cantilever mode of the suspension. In the preliminary measurements, The natural frequency of the rigid mode of the base block was 32.78 Hz. The natural frequency of the lowest flexible mode of the base block was 17 kHz. Therefore, the natural modes of base block assembly are decoupled with those of the suspension assembly, perfectly.

The impact hammer impacts the base plate near the bending region of the suspension and this is so called as the base excitation method because the base plate is mounted on the base block. A Polytec Laser Doppler Vibrometer was used to measure the response velocity of the suspension. Impact force signal and response velocity signal are processed in the dynamic signal analyzer to acquire the frequency response function. Tere are 15 response points to measure the frequency response function as shown in fig. 5. The modal parameter estimation software STAR-MODAL was applied in the geometric modeling of the suspension assembly and natural frequencies and mode shapes are estimated by polynomial curve fitting method. Fig. 6 shows a measured mode shapes and natural frequencies.

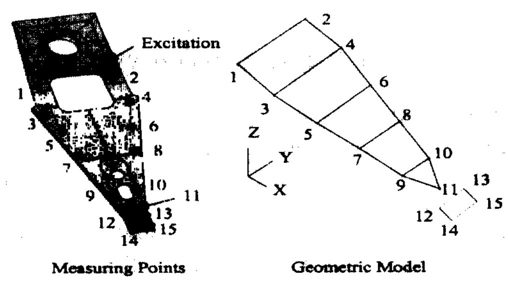

Fig. 5 Geometric model of the suspension assembly in modal testing

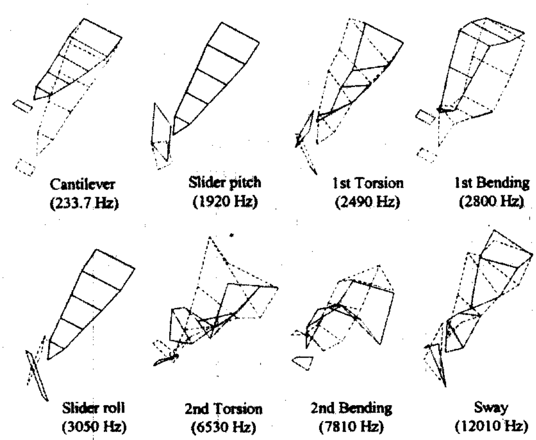

Fig. 6 Measured natural frequencies and mode shapes of the suspension assembly

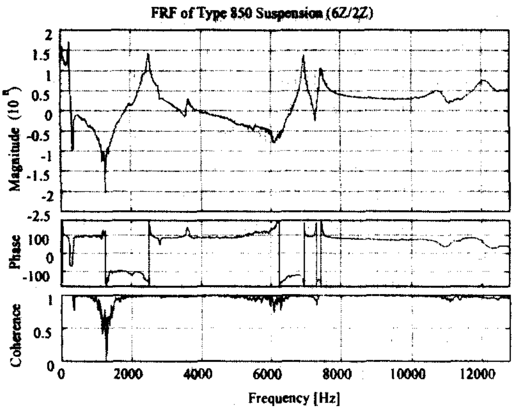

Fig 7. Experimental frequency response function of the suspension

Effective frequency range was secured from 0 Hz to 12 kHz by using the steel tip impact hammer. As a result, good coherence was acquired as shown in fig 7.

5. Tuning of the suspension system

Calculated natural frequencies are compared with the experimental results in table 2, the difference being shown as percentages of the measured results. The maximum deviation was only 7 %, except the results related to the local flexure modes, slider pitch and roll mode. The error of the slider pitch mode was about 26%. This large error originated in the inaccurate modeling of the flexure. Small halls of the flexure are omitted and the dimensions are measured by rough scale. Therefore the difference errors of the slider pitch and roll mode could be reduced by accurate and precise modeling of the flexure.

Table. 2 measured and calculated natural frequencies of the suspension system

Mode No.	Description	Measured Frequency[Hz]	Calculated Frequency[Hz]	Difference error [%]
1	Canti-lever	233.67	243.8	-4.34
2	Slider pitch	1920	2421.1	-26.10
3	1st Torsion	2490	2429.3	2.44
4	1st Bending	2800	2893.2	-3.33
5	Slider roll	3050	3537.3	-15.98
6	2nd Torsion	6530	6611.6	-1.25
7	2nd Bending	7810	8037.8	-2.92
8	Sway	12010	11233	6.47

To develop a more accurate FE model, rough model must be updated using measured modal parameters. The algorithm of FE model update is represented as follows. [5]

1) Construct the modal parameter vector θ to be compared.

$$\theta = [\theta_1, \theta_2, \cdots, \theta_S]^T \tag{1}$$

Measured modal parameter vector θ_E becomes an target value. The error vector $\Delta\theta$ is represented as

$$\Delta\theta = \theta_E - \theta_C \tag{2}$$

where, θ_C is a calculated modal parameter vector with current design variables.

2) Determine the design variables $\zeta_i (i = 1, 2, \cdots, m)$ for structural modification. Thickness, Young's modulus and dimension are applicable to the design variables. Modification vector $\Delta\zeta$ is defined as

$$\Delta\zeta = [\Delta\zeta_1, \Delta\zeta_2, \cdots, \delta\zeta_m]^T \tag{3}$$

3) Calculate a sensitivity matrix using finite difference method. The first order sensitivity matrix of modal parameter for design variable is

$$Z = \begin{bmatrix} \frac{\partial\theta_1}{\partial\zeta_1}, & \frac{\partial\theta_1}{\partial\zeta_2}, & \cdots, & \frac{\partial\theta_1}{\partial\zeta_m} \\ \frac{\partial\theta_2}{\partial\zeta_1}, & \frac{\partial\theta_2}{\partial\zeta_2}, & \cdots, & \frac{\partial\theta_2}{\partial\zeta_m} \\ \cdots & \cdots & \cdots & \cdots \\ \frac{\partial\theta_s}{\partial\zeta_1}, & \frac{\partial\theta_s}{\partial\zeta_2}, & \cdots, & \frac{\partial\theta_s}{\partial\zeta_m} \end{bmatrix} \tag{4}$$

Therefore, variation of the modal parameter due to the structural modification is represented as

$$\Delta\theta = Z\Delta\zeta \tag{5}$$

4) Modal tuning is performed using the equation (5) reversely. Error vector between the experiment and FE analysis was known for initial design parameter vector.

$$\Delta\zeta = Z^{-1}\Delta\theta \quad , \quad s = m \tag{6}$$

Generally, number of modal parameter s does not equal to the number of the design variables m. If s is less than m, least square method is used to solve the equation (6). If s is greater than m, pseudo inverse method is applied as in equation (7).

$$\Delta\zeta = (Z^T Z)^{-1} Z^T \Delta\theta \quad , \quad s > m \tag{7}$$

5) Update the design parameter vector and iterate FE analysis until the convergence criteria is satisfied.

$$\zeta^{i+1} = \zeta^i + \Delta\zeta \tag{8}$$

Design variables of the suspension assembly in this study are thickness and elastic modulus

Table 3. Sensitivity matrix of the natural frequencies for the design variables

Mode \ DV	t1	t2	t3	t4	t5	t6	t7	E
Canti-Lever	-1.632E+02	-3.213E+02	-5.902E+01	2.534E+03	2.012E+03	2.605E+02	-6.921E+02	6.472E-07
Slider Pitch	3.947E+01	1.092E+05	1.902E+03	3.421E+02	1.053E+02	0.000E+00	0.000E+00	6.435E-06
1st Torsion	4.079E+03	4.164E+03	1.279E+03	1.593E+04	7.592E+03	-2.763E+02	-1.553E+03	6.332E-06
2nd Bending	3.684E+02	4.262E+02	-1.967E+02	1.426E+04	6.921E+03	5.026E+03	1.355E+03	7.663E-06
Slider Roll	1.974E+02	1.139E+05	4.574E+04	9.474E+02	7.237E+02	-6.053E+02	-7.895E+01	9.399E-06
2nd Torsion	7.618E+03	-3.672E+03	-3.279E+02	2.279E+04	1.362E+04	1.359E+04	2.763E+02	1.726E-05
3rd Bending	-7.763E+02	-2.754E+03	-9.836E+01	1.380E+04	1.078E+04	1.589E+04	2.086E+04	2.133E-05

Table 4. Natural frequencies of suspension system in modal tuning iteration

Description	Measured	0		1		2		3		4		5	
Canti-lever	233.67	243.8	-4.34	239.85	-2.64	234.82	-0.49	234.69	-0.44	235.46	-0.77	236.11	-1.04
Slider pitch	1920	2421.1	-26.10	2263.3	-17.88	2102.5	-9.51	1950.2	-1.57	1919.5	0.03	1914.1	0.31
1st Torsion	2490	2429.3	2.44	2440.4	1.99	2443.3	1.88	2436.3	2.16	2440.9	1.97	2444.7	1.82
1st Bending	2800	2893.2	-3.33	2851.2	-1.83	2805.2	-0.19	2806.6	-0.24	2819.9	-0.71	2826.9	-0.96
Slider roll	3050	3537.3	-15.98	3386.3	-11.03	3226.4	-5.78	3068.8	-0.62	3038.8	0.37	3033.9	0.53
2nd Torsion	6530	6611.6	-1.25	6549.9	-0.30	6496.7	0.51	6519.6	0.16	6543.9	-0.21	6557.9	-0.43
2nd Bending	7810	8037.8	-2.92	7939	-1.65	7785.6	0.31	7823	-0.17	7870.8	-0.78	7898.8	-1.14
Sway	12010	11233	6.47	11165	7.04	11088	7.68	11141	7.24	11192	6.81	11230	6.49
SER [%]	0		62.81		79.05		44.51		17.49		15.71		17.90

(SER ; Sum of Error Rate = $\Sigma \Delta f/f \times 100$ [%])

Table 5. Design variables in modal tuning iteration

DV	0	1	2	3	4	5
t1	7.600E-03	8.228E-03	8.902E-03	9.120E-03	9.120E-03	9.120E-03
t2	3.050E-03	2.906E-03	2.759E-03	2.616E-03	2.587E-03	2.582E-03
t3	3.050E-03	3.068E-03	3.084E-03	3.102E-03	3.111E-03	3.114E-03
t4	7.600E-03	7.538E-03	7.493E-03	7.445E-03	7.494E-03	7.510E-03
t5	7.600E-03	7.654E-03	7.692E-03	7.893E-03	8.002E-03	8.096E-03
t6	7.600E-03	6.840E-03	6.156E-03	6.080E-03	6.080E-03	6.080E-03
t7	7.600E-03	7.739E-03	7.865E-03	8.212E-03	8.492E-03	8.661E-03
E	1.930E+08	1.933E+08	1.935E+08	1.938E+08	1.939E+08	1.939E+08

(young's modulus). Load beam divided into 5 parts having own thickness as a design parameter. Flexure divided into 2 parts, also. The elastic modulus of the load beam is selected a design parameter. Another material properties and dimensions are fixed as a constant during the tuning process. Fig 8 shows the divided parts of the load beam and flexure.

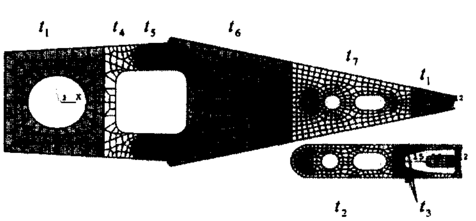

Fig 8. Thickness of load beam and flexure

Therefore the design parameter vector is defined as

$$\zeta = [t_1, t_2, t_3, t_4, t_5, t_6, t_7, E]^T \qquad (9)$$

Modal parameter vector consists of seven natural frequencies except sway mode. The error rate of the sway mode isn't large but calculated natural frequency is less than the measured one. This is different from another modes. Calculated natural frequencies are greater than the measured ones in another vibration modes. If the natural frequency of the sway mode is included in modal parameter vector, smooth convergence does not occurred.

Table 3 lists the sensitivity matrix of the natural frequencies for 8 design variables. Table 4 lists the result of the modal tuning for each iteration. Iteration stops when the sum of error rate does not decreased. In each iteration, Modification vector $\Delta\zeta$ was calculated by equation (7) and decrease it's magnitude if it exceeds a given limitation. In this study, $\Delta\zeta$ was limited to 20% of ζ. If $\Delta\zeta$ is greater than 20% of ζ^0(inital value), $\Delta\zeta$ is scaled according to the rule given by

$$\Delta\zeta_i^* = 0.2 \cdot \frac{\Delta\zeta_i}{\max|\Delta\zeta/\zeta|} \qquad (10)$$

where, $\Delta\zeta_i^*$ is an ith element of the scaled modification vector.

Design variable does not changed after its cumulated variation reaches 20% of its initial value.

Table 5 lists the design variables in each iteration. Variation of some design variables stopped after several iteration because of 20% limitation. Convergence process was affected by this limit rule. As shown in table 4, natural frequencies stop to converge after 4th iteration and start to diverge. Therefore, tuned result is a modal parameter with least SER(Sum of Error Rate). In this study, after 4th iteration, best result of the modal tuning is acquired. The errors of the slider pitch and roll mode decreased rapidly with reduction of thickness (t2) of flexure. The thickness t6 is decreased to the 20% limit lower bound value. From this result, It is known that the thickness t6 of the load beam has a powerful role in tuning process.

6. Conclusion

This paper analyzes the dynamic properties of an unloaded type850 suspension assembly using finite element analysis and experimental modal analysis. Good agreement between calculation and measurements was achieved except the results related to the local flexure modes, slider pitch and roll mode. Large error of the slider mode originated in the inaccurate modeling of the flexure. Large movement of the slider to the lateral direction appeared in 2nd torsion mode and sway mode. In sway mode, local deflection of the flexure in lateral direction occurred, also. To develop a more accurate FE model, Modal tuning method with sensitivity matrix was introduced. Rough model was updated using measured natural frequencies and the difference between calculation and measurements were decreased less than 2% except a sway mode. The error rate of the sway mode isn't large but calculated natural frequency is less than the measured one. This trend is different from another modes. If the natural frequency of the sway mode is included in modal parameter vector, smooth convergence does not occurred. Initial values of calculated natural frequencies must be close to measurements as possible to prevent the divergence.

7. Acknowledgement

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