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3축 나노 스테이지 동특성 해석 및 개선
Analysis and Improvement of Dynamics Characteristic of 3-axis Nano Stage

김충*·이강녕*·이동주*·이문구**·최형길**· 이석원**·박노철*·박영필***

Choong Kim* Kang-Nyung Lee* Dong-Ju Lee* No-Cheol Park* Young-Pil Park***
Moon Gu Lee Hyoung Gil Choi**Suk-Won Lee****

Key Words : Nano Stage, Design of experiments, Modal test, Finite Element Analysis, Flexible mode

ABSTRACT

The precision positioning system requires robust design to obtain enough bandwidth. Therefore, The sub-resonance occurred by the disaccord of force center and mass center should be oppressed. And it is necessary to move the flexible mode to a higher frequency. In this paper, the 3-axis nano stage was proposed and dynamic characteristics was improved by design of experiments (DOE).

1. Introduction

Precision positioning devices have been widely used in various areas, precision machines, surface scanners, semiconductor manufacturing machines, and optical devices. The quality for a precision positioning device is evaluated by precision of manufacturing and inspection technology [1]. In the present industry, it is required that the precision positioning system maintains the resolution of a nanometer scale and the long stroke of hundreds of nm [2]. Specially, in precision stages that use to semiconductor and liquid crystal display (LCD) processes, the moving precision with a nanometer scale and fast response characteristics are required. In order to achieve these targets, the effective driving mechanism and the control method are needed. And, the precision positioning system should have a robust design to obtain enough bandwidth.

Previous research has explored the following topics. The new structure of voice coil motor (VCM) for a fine actuator was suggested by the optimal design of the stage to obtain the maximum force [2]. A novel electromagnetically driven elastic stage for precision instrumentation purposes was presented [3]. And the piezoelectric actuator has been used for precision positioning from micrometer down to nanometer scale. It was designed with a low-stiffness spring element to achieve a high-accuracy and large-displacement characteristic [4], [5]. A design for a small linear optical pickup actuator, with a size similar to a PC II card, to provide seeking and fine tracking motions in an optical disk drive was proposed using FE analysis and DOE procedure [6].

In this paper, we proposed the 3-axis nano stage using VCM for a fine actuator and the method obtaining enough bandwidth for precision positioning device through a modification of our structure. In details, the actuator was modeled and was analyzed using finite element (FE) analysis tool. This FE model was tuned by experiments through modal test. And it was optimized by applying design of experiments (DOE) technique. Finally, it was checked that the first flexible mode was moved to a higher frequency than before.

2. Overall Structure

The initial model is designed with satisfying the following specification as shown in Table. 1. The target velocity profile of the actuator is shown in Fig. 1. The acceleration of the actuator can be expressed by

$$a_{\max} = \frac{4 \cdot s}{\tau^2} = \frac{(\vec{B}_{eff} \times \vec{I}) \cdot l_{eff} \cdot n}{m} \qquad (1)$$

where, s is the driving stroke, τ is the access time, B_{eff} is the effective magnetic field strength in the effective area, I is the input current, l_{eff} is the effective length of each turn of the coils, n is the total number of turns of the coils, and m is the mass of the moving part of the actuator. Using this equation, the maximum acceleration of the actuator is 5 m/s^2.

We propose the VCM actuator design that is shown in Fig. 2. When the current is applied to coil 1 and coil 3, it is driven in the x direction. The current is applied to the 2 and 4 coils for y direction driving. Finally, it is driven in θ_x direction when the current is applied to all the coils. The moving part is kept at the $50\mu m$ height from the surface by air bearing.

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* 연세대학교 정보저장기기연구센터
E-mail : jumulri@yonsei.ac.kr
Tel : (02) 2123-4677, Fax : (02) 365-8460

** 삼성전자 메카트로닉스 센터

*** 연세대학교 기계공학과

Table. 1 Specifications

	Stroke	Resolution	Load Capacity	Maximum Accelerometer
Value	*5mm* x *5mm*	±50 *nm*	*40 kg*	*5* m/s^2

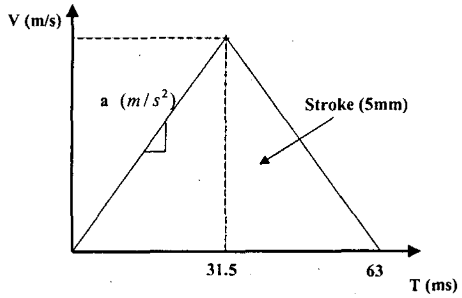


Fig. 1 Velocity profile

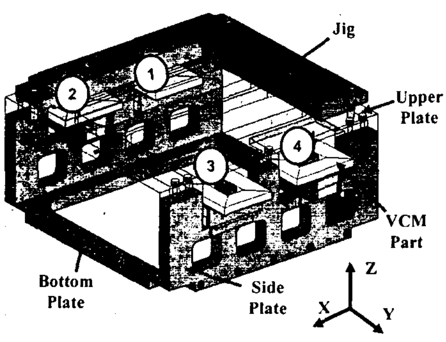


Fig. 2 Structure of 3-axis nano stage

3. Modal Test and Tuning

Initial model is fabricated as shown in Fig. 3. In order to check whether it has sufficient bandwidth or not, modal test is performed. Fig. 3 shows the experimental set-up. The modal test is accomplished by the following procedure. We measured the response displacements at 74 points using the 3-axis accelerometer after exciting the moving part in x, y and z directions. The measured data are saved by the pulse fast fourier transform (FFT) system and the resonance frequencies and flexible modes are extracted by ME scope. Fig. 4 is the first, the second and the third flexible modes. As shown in this figure, it can be checked that the first flexible mode is very low. And so, the sufficient bandwidth can not be secured. Therefore, structural improvements are needed. In order to improve structural parts, FE model is tuned with experimental results. And, we select each bolting area as tuning parameters to reflect the real model. The density and the stiffness are not considered as a tuning parameter because they are hardly changed between simulations and experiments. Through this procedure, tuned FE model of 3-axis stage is made. As shown in Table. 2, it can be found that tuning parameters are good choices because the error is within 10%.

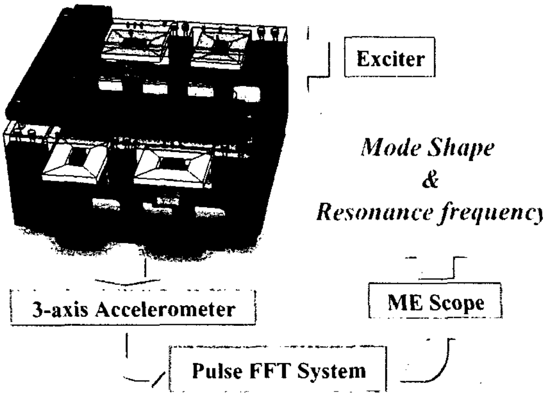
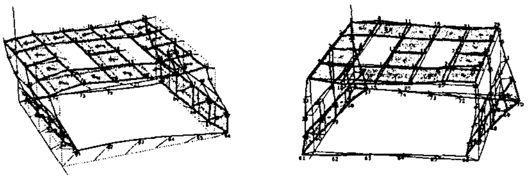
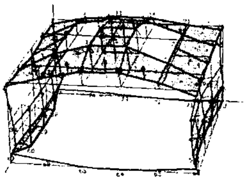


Fig. 3 Experimental set-up for modal test

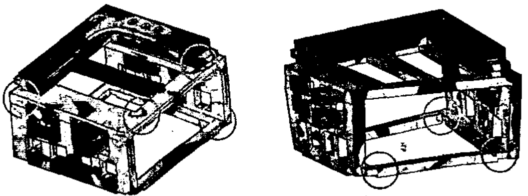


(a) 1st mode (196Hz)   (b) 2nd mode (326Hz)

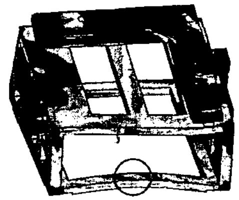


(c) 3rd mode (385Hz)

Fig. 4 Flexible modes for modal test



(a) 1st mode (190.10Hz)   (b) 2nd mode (383.6Hz)



(c) 3rd mode (471.3Hz)

Fig. 5 Stress distribution for initial model

Table. 2 Errors between tuned model and experiment results

	1^{st}	2^{nd}	3^{rd}	Total mass
Tuned model	184.36	356.02	375.33	22.85
Experiment	196	326	385	22.78
Error	6.31	8.43	2.58	0.3

4. Optimization

To improve dynamic characteristics of our nano stage, we maintain the initial structure and target to move the first flexible mode to a high frequency by the modification of moving parts through DOE procedure. Constraints are as followings. Mass and force centers must be accord within 1mm to remove the sub-resonance mode and the total weight of moving part must be less than 30 kg.

The following DOE procedure was used to optimize the dimensions of each part for improvement of dynamic characteristics based on the Taguchi approach. At first, we observe the stress distribution of three flexible modes, as shown in Fig. 5. Through the stress distribution, parts that are closely connected with the strain energy distribution of a considered mode should be reinforced while parts that are not should be removed. Mass balancers and reinforces are added at these structural parts.

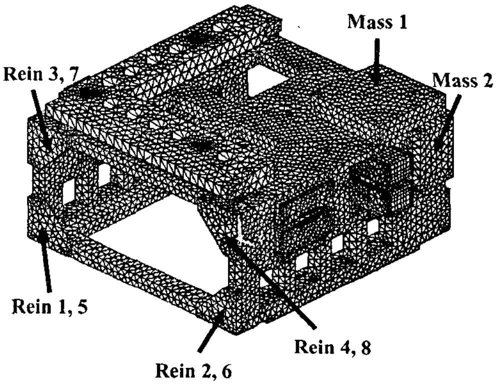
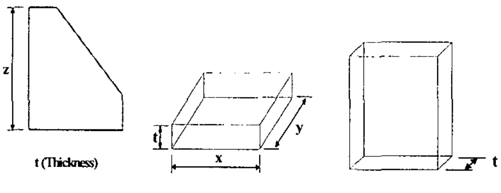


Fig. 6 Mass balancers and reinforces



(a) Reinforce (b) Mass 1 (c) Mass 2

Fig. 7 Design parameters of mass balancers and reinforces

Table. 3 Design variables and level

	Name	Level 1	Level 2	Level 3
A	mass2_t	0.01	0.02	
B	rein1_t	0.005	0.01	0.015
C	rein1_z3	0.05	0.07	0.09
D	rein2_t	0.005	0.01	0.015
E	rein2_z3	0.05	0.07	0.09
F	rein5_t	0.005	0.01	0.015
G	rein5_z3	0.05	0.07	0.09
H	rein6_t	0.005	0.01	0.015
I	rein6_z3	0.05	0.07	0.09
J	rein3_t	0.005	0.01	0.015
K	rein3_z1	0.103	0.123	0.143
L	rein4_t	0.005	0.01	0.015
M	rein4_z1	0.103	0.123	0.143
N	rein7_t	0.005	0.01	0.015
O	rein7_z1	0.103	0.123	0.143
P	rein8_t	0.005	0.01	0.015
Q	rein8_z1	0.103	0.123	0.143
R	mass1_t	0.025	0.03	0.035
S	mass1_x2	0.06	0.07	0.08
T	mass1_y2	0.06	0.07	0.08

As shown in Fig. 6 and 7, the mass balancers are used as design parameters for the accord of mass and force centers and reinforces are selected as design variables to move the first flexible mode to a high frequency. The selected twenty design parameters are shown in Table. 3. Objective function is the maximum of the first flexible mode frequency. The mean sensitivity for the selected parameters can be obtained from the DOE procedure using orthogonal arrays. From an analysis of the mean (ANOM), we can obtain the sensitivity of each parameter as shown in Fig. 8. Among twenty design variables, A, G, I, N, P and R are used to optimize our structure because it is proved that they have higher sensitivity under these conditions.

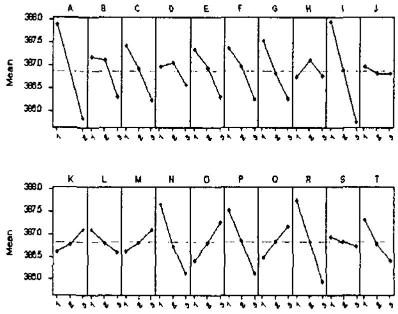


Fig. 8 Analysis of mean

The remaining design variables are selected to have a high level of sensitivity s in our design. Through the same procedure, the new orthogonal array is constructed by the selected design variables as shown in Table. 4. The sensitivity of each parameter from ANOM is obtained as shown in Fig. 9. Based on them, various cases can be made. Among them, we select the optimal case that satisfies objective function and constraint conditions like Fig. 10. Finally, the first flexible mode is improved about 47 %. And besides, the total weight of moving part is 29.4 kg and the deviation of mass and force centers is 0.934mm.

Table. 4 Design variables and level

	Name	Level 1	Level 2	Level 3
A	mass2_t	0.005	0.01	0.015
G	rein5_z3	0.04	0.05	0.06
I	rein6_z3	0.04	0.05	0.06
N	rein7_t	0.003	0.005	0.007
P	rein8_t	0.003	0.005	0.007
R	mass1_t	0.028	0.03	0.032

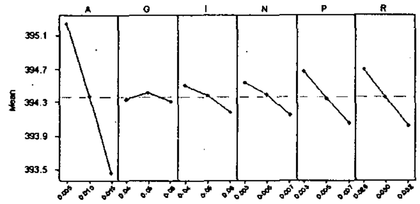
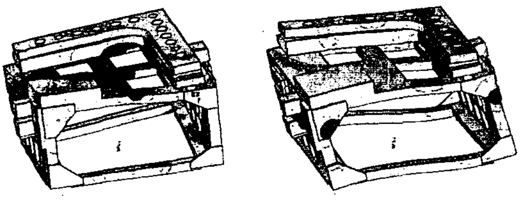
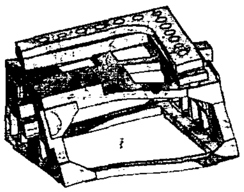


Fig. 9 Analysis of mean



(a) 1st mode (393.7Hz) (b) 2nd mode (410.5Hz)



(c) 3nd mode(550.5Hz)

Fig. 10 Flexible modes for optimized model

5. Conclusions

3-axis nano stage was proposed. This actuator was fabricated and modal test was performed. In order to make precise FE model, it was tuned with experimental results. Through stress distribution of flexible modes that affect the control bandwidth, design parameters were extracted to move the first flexible mode frequency to higher frequency area. Using DOE procedure, sensitivities of each design variable were checked. Based on them, a final model that satisfied the objective function and constraint conditions was proposed. Finally, the first flexible mode frequency increased about 47% than the initial model's. Therefore, we obtained wider bandwidth than before for nano precision positioning device after improvement of the overall structure.

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