Design and Optimization of a Novel Compact and Long-range 3-DOF Flexure Stage

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Abstract

A novel compact and long-range 3-DOF flexure stage using three piezo-actuators is designed and optimized. The stage is the combination mechanism which composed of serial type X-stage and parallel type Y, θ_Z -stage. Stage consists of bridge-type amplifying mechanism for amplification of PZT deformation, double compound guide mechanisms for performing only desired motion, and circular hinge mechanism which permit rotational motion in Y, θ_Z -stage. To set the design variables of stage, optimal design is carried out. To verify the results of optimal design, FEM simulation is used. The stage is designed to simultaneously attain $800\mu m$ in X and Y directions and $\pm 5mrad$ in θ_Z direction. 1st natural frequency of the X-stage is 45Hz and the Y, -stage is 89Hz. Size of the stage is $200 \times 200\ mm^2$.

1 Introduction

Recently, flexure stages that provide ultra-precision, multi degree of freedom, long range are required for nano-positioning. In this article, a new concept design of 3-DOF flexure stage which is driven by three piezo-actuators with compact size and large working space are proposed and optimized. The stage consists of three amplification and guide mechanisms. Finite element analysis is used to verify the theoretical model and optimization.

2 Design and Optimization of The Flexure Stage

2.1 Conceptual design

Objectives of the proposed stage are compact size and long range motion with 3-DOF ($XY\theta_Z$) motion. System basic specifications are represented at Table 1.

Table1: Stage Basic Specifiaction

System size	200 X 200 mm^2
Maximum displacement	> 800 um (X, Y)
Maximum rotation angle	> ± 5 mrad (θ_Z)

To actuate the 3-DOF ($XY\theta_Z$), the stage requires three or more PZT actuators. It is more cost effective to use fewer actuators. Therefore, only three PZT actuators are used for driving in-plane motion. Combination of serial and parallel mechanism is proposed to reduce size of the stage system.

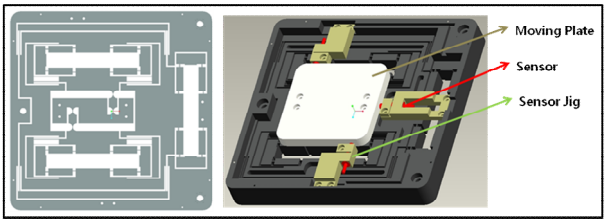


Figure 1: Proposed $XY\theta_Z$ Stage

A serial mechanism is used for X-directional motion with one PZT actuator, and a parallel mechanisms are used for Y, θ_Z -directional motions with two PZT actuators. Proposed stage can be achieved large working space with very compact size, and the positional errors produced by X-directional motion of serial mechanism can be compensated by Y, θ_Z -directional motions of parallel mechanisms.

2.2 Design of Amplification and Guide Mechanism

As mentioned above PZT actuators have very small deformation range. For that reason, amplification mechanisms are required for long-range motion.

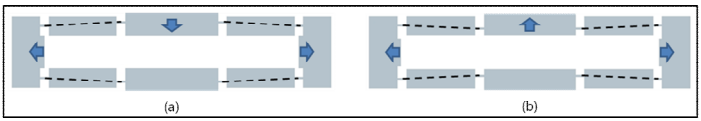


Figure 2: Amplification mechanism

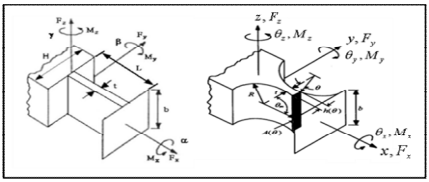
Figure 2 represents two kinds of amplification mechanisms which are composed of eight leaf-springs. For high amplification ratio of bridge type amplification mechanism, leaf-spring is the most effective way. Mechanism (a) is used for amplification of X-directional motion. Both (a) and (b) mechanisms are used for

amplification of Y, θ_Z -directional motions. Each of amplification mechanisms has a PZT actuator.

Guide mechanisms are also required to carry out desired motion and to eliminate parasitic motions. The stage adopts one double compound mechanism for x-direction and two double compounds mechanisms for y-directional motion. Four circular hinges are adopted to permit rotational motion of moving body.

2.3 Modeling and Optimization

X-stage and Y, θ_Z -stage are decoupled. Therefore, modeling and optimization of stages are carried out respectively. Assuming a hinge as a 6-DOF spring elements, the leaf-spring and circular hinge compliance equations are as shown in figure 3. The compliance matrix consists of eight unknown equations. The compliance matrices of leaf-spring and circular hinge can be derived respectively from beam theory.[2]



$$C^h = \begin{bmatrix} \frac{\Delta x}{F_x} & 0 & 0 & 0 & 0 & 0 \\ 0 & \frac{\Delta y}{F_y} & 0 & 0 & 0 & \frac{\Delta y}{M_z} \\ 0 & 0 & \frac{\Delta z}{F_z} & 0 & \frac{\Delta z}{M_y} & 0 \\ 0 & 0 & 0 & \frac{\Delta \alpha}{M_x} & 0 & 0 \\ 0 & 0 & \frac{\Delta \beta}{F_z} & 0 & \frac{\Delta \beta}{M_y} & 0 \\ 0 & \frac{\Delta \gamma}{F_y} & 0 & 0 & 0 & \frac{\Delta \gamma}{M_z} \end{bmatrix}$$

Figure 3: Coordinates of a leaf-spring and a circular hinge and a compliance matrix

Relationships between rigid body and springs are modeled mathematically using Ryu's algorithm.[1] After completing the total system analytic modeling, the optimal design is performed respectively.

The design variables of the X-stage are the leaf spring length and thickness of the amplification mechanism and the guide mechanism (l_1, t_1, l_2, t_2), and the gap related to amplification ratio between two leaf springs in the amplification mechanism (gap_1).

The design variables of the Y, θ_Z -stage are the leaf spring length and thickness of the amplification mechanism and the guide mechanism (l_1, t_1, l_2, t_2), the gap between two leaf springs in the amplification mechanism (gap_1), the thickness and radius of circular hinge (R, t_3) and another gap between center of moving body and the circular hinge (gap_2). The objective of the optimization is to maximize the system bandwidth (1st natural frequency) to improve the dynamic characteristics. It guarantees a fast response and robustness against dynamic disturbances.

$$Minimize\ \ f\left(l_1,t_1,l_2,t_2,gap_1\right)=\frac{1}{\omega_{n_1}^2},\quad Minimize\ \ f\left(l_1,t_1,l_2,t_2,gap_1,R,t_3,gap_2\right)=\frac{1}{\omega_{n_1}^2}$$

Also, there are constraints which consider maximum displacement (880um, $\pm$5.5 mrad – 10% margin) and maximum stress at each spring. To find the solution, a sequential quadratic programming (SQP) method and MATLAB were used. Final dimensions are determined considering minimum tolerance of manufacturing.

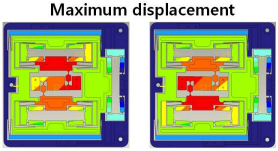
Table2: Results of optimal design

Design variables (mm)		l_1	t_1	l_2	t_2	gap_1	R	t_3	gap_2
X- stage	Optimal design	3	0.924	25	0.4	1.410			
	Final dimension	3	0.92	25	0.4	1.4			
Y,Θ_z-stage	Optimal design	3	0.714	25	0.4	1.049	4	0.4	20.009
	Final dimension	3	0.72	25	0.4	1.05	4	0.4	20

3 FEM Simulation

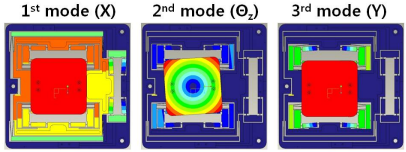
To verify these analytical results, FEM simulation is performed according to final dimensions. The stage performance is fully satisfied.

Maximum displacement



Static	MATLAB	FEM
Max. disp. (X)(um)	884.4	890.1
Max. disp. (Y) (um)	871.4	850.6
Max. rot. (Θ_z) (mrad)	5.5	5.45

1st mode (X) 2nd mode (Θ_z) 3rd mode (Y)



Mode	1st	2nd	3rd
Frequency (Hz)	45.9	69.2	89.1

Figure 4: Analytical results and FEM results of the stage

4 Conclusion

In this study, a novel 3-DOF flexure stage which has large working space with very compact size is proposed and optimized. Future work includes the fabrication of this stage with final dimension and experiments with the manufactured stage.

References:

[1] Ryu, J. W. and Gweon, D. G., M.KS, “Optimal design of a flexure hinge based wafer stage,” The Journal of Precision Engineering 21, pp. 18-28, 1997.

[2] Koseki, Y., et al., “Kinematic Analysis of Translational 3-DOF Micro Parallel Mechanism Using Matrix Method,” Advanced Robotics, 2002.