euspen's 15th International Conference & Exhibition, Leuven, Belgium, June 2015
www.euspen.eu

Optimal design of a compliant parallel XY nano-positioning stage for high dynamic performance.

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Abstract
This paper presents a novel compliant parallel XY nano-positioning stage drive by 2 piezo-actuators(PZT). With the purpose to obtain high dynamic performance, the stage is designed using 2 bridge amplification mechanisms and 8-bar guiding flexures. Analytic modelling and optimal design of the stage were performed. To verify the results of the optimal design of the stage, FEM simulation are carried out. The designed stage has 118um in the X, Y directional motion and 278Hz in the first resonance frequency. The designed stage has also compact size of 150 X 150 X 30 mm^3.

Keywords: nano-positioning, Flexure, Piezoelectric actuator, parallel structure

1. Introduction

XY nano-positioning stages, based on flexure mechanisms and piezoelectric actuators, are widely used in industrial and scientific area such as an atomic force microscopes(AFM), micro-aligners and a semiconductor manufacturing. In many cases, nano-positioning stages require nano-scale resolution, high dynamic performance and compact size.

In previous research, various XY nano-positioning stages have been developed. Y. Li et al developed a compliant parallel XY micromotion stage with complete kinematic decoupling[1]. They developed high dynamic performance stage which has 720.52Hz in the first resonance frequency. However, this stage has small working range which is 19.2um X 8.8um. Y. Qin et al developed a decoupled XY stage [2]. This stage also has high first resonance frequency 665.4Hz. However, this stage also has small working range of 11.6um X 11.6um. PI developed a decoupled XY stage[3]. This stage has working range 100um X 100um and 255Hz first resonance frequency. But, proposed stage has higher performances(working range, dynamic characteristics) than PI stage using novel flexure structure.

In this paper, we propose a compliant parallel XY nano-positioning stage for high dynamic performance. And also proposed stage has 118um X 118um working ranges. Modeling and optimal design of proposed stage is carried out.

2. Design of stage

The main purpose of this proposed stage is the measurement equipment stage. Biological specimen stages are required large working range and compact size. Then, objectives of the proposed stage are compact size and long range motion with X, Y directional motion. Design goal of system are represented at Table 1.

Table 1 Design goal of stage

System size	150 X 150 X 30 mm3
Working range	> 120um (X, Y)
Runout(θz)	< 10urad

A proposed parallel XY nano-positioning stage is shown in figure 1. To achieve a high dynamic performance, we use parallel compliant mechanism which is consists of two bridge amplification mechanisms and two guiding structures.

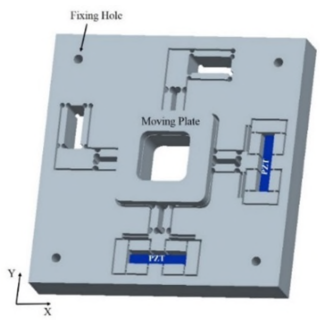


Figure 1. A proposed parallel XY nano-positioning stage.

Figure 2 represents bridge amplification mechanism which is composed of eight leaf-springs. Each of amplification mechanism has a piezoelectric actuator.

Figure 2. Bridge amplification mechanism

The proposed stage was designed and constructed from AL7075-T6 because the material offered a good machinability and a high ratio of yield stress to elastic modulus for good repeatability.

The moving plate is guided 8-bar structures which can prevent a rotational motion. And bridge amplification mechanism is guided two leaf springs to prevent a distortion of structure. Two-bodies on the opposite side of bridge amplification mechanisms guide moving plate to prevent a run-out motion(θ_z) and a rotational motion.

The size of the proposed stage is 150 X 150X 30mm^3. The aperture size of the proposed stage is 30 X 30 mm^2. Therefore, the proposed stage can generates X,Y axis motion.

3. Analytic model

An analytical model was developed to calculate the static motion such as open-loop maximum range, the dynamic resonance frequency of the flexure system without a piezoelectric actuator, and the stress based on a geometric model of the stage. Assuming a hinge as the 6-DOF spring element, the leaf-spring and circular hinge compliance equations derived as eq(1). This equation derived by Koseki[4].

$$\begin{bmatrix} \delta_x \\ \delta_y \\ \delta_z \\ \theta_x \\ \theta_y \\ \theta_z \end{bmatrix} = \begin{bmatrix} c_1 & 0 & 0 & 0 & c_3 & 0 \\ 0 & c_2 & 0 & -c_4 & 0 & 0 \\ 0 & 0 & c_5 & 0 & 0 & 0 \\ 0 & -c_4 & 0 & c_6 & 0 & 0 \\ c_3 & 0 & 0 & 0 & c_7 & 0 \\ 0 & 0 & 0 & 0 & 0 & c_8 \end{bmatrix} \begin{bmatrix} f_x \\ f_y \\ f_z \\ M_x \\ M_y \\ M_z \end{bmatrix} \quad (1)$$

Relationships between rigid body and springs(leaf-springs and circular hinges) are modeled mathematically using Ryu's multi-body dynamics[5]. After completing the total system analytic modeling, the optimal design is performed respectively.

4. Optimal design

The design variables of the stage are the leaf spring length and thickness of amplification mechanism, the circular hinge radius and thickness of guiding structure, the gap between two leaf springs in amplification mechanism.

The objective of the optimization is to maximize the system bandwidth to improve the dynamic performance. It guarantees a fast response and robustness against dynamic disturbances. In other words, the first natural frequency of the stage is maximized as follows.

$$f(L_1, t_1, gap, L_2, t_2, L_3, L_4, r_1, t_3, r_2, t_4, w_6) = \min\left(\frac{1}{f_n^{\,2}}\right) \quad (2)$$

Also, there are constraints which consider maximum displacement, runout, fatigue stress of material and geometric constraints.

$$g_1 = 120\times10^{-6} - N\times(38\times10^{-6})\times\frac{K_{PZT}}{K_{PZT}+K_{flexure}} \le 0 \quad (3)$$

$$g_2 = \theta_{distort\_z} - (10\times10^{-6}\,rad) \le 0 \quad (4)$$

$$g_i = \sigma_{fatigue} - K_{stress}\left(\frac{Et_i}{2L_i}\theta_i + \frac{(F_{ext}/2)}{bt_i}\right) \le 0 \ldots (i = 3 \sim 22) \quad (5)$$

$$g_i = \sigma_{fatigue} - \frac{6K_{\theta_z}\theta_i}{b_i t_i^2} \le 0 \ldots (i = 23 \sim 46) \quad (6)$$

$$g_{47} = (85\times10^{-3}) - (L_4 + 4r_1 + 2t_3 + 2L_2) \ge 0 \quad (7)$$

To find the solutions, a sequential quadratic programming(SQP) method and MATLAB were used.

Table 2 Optimal design results

Design variable	Optimal design Results(mm)	Final Design value (mm)
L1	0.8998	0.9
t1	0.5000	0.5
Gap	2.0458	2
L2	30.000	30
t2	0.5000	0.5
L3	9.9769	10
L4	6.6836	7
r1	2.5000	2.5
t3	0.5000	0.5
r2	2.9911	3
t4	0.7587	0.8
w6	29.9964	30

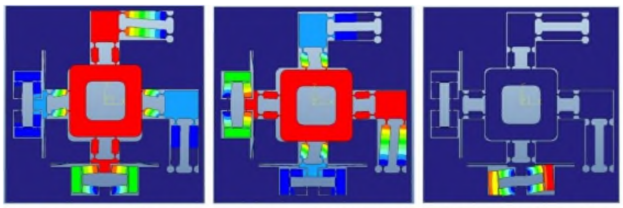
Table 2 shows results of optimal design. Final dimensions are determined considering minimum tolerance of manufacturing. To verify these analytical results, FEM simulation is performed according to final dimensions.

Table 3 Analytic results and FEM simulation results of the stage

	Analytic model	FEM simulation	Error
Working Range (X,Y)	123.3um	118.5um	4.05%
Runout (θz)	3.037urad	3.1urad	2.03%
Max Stress	142.8 Mpa	145.9 Mpa	2.12%
Mode 1 (y-axis)	267.67 Hz	278.845 Hz	4.01%
Mode 2 (x-axis)	267.74 Hz	279.098 Hz	4.07%
Mode 3 (θ_z-axis of bridge)	763.44 Hz	728.71 Hz	4.76%

Table 3 and Figure 3 shows analytic results and FEM simulation results. Both static and dynamic analytical results correspond with the FEM simulation result.

We use ProEngineering/Mechanica simulation tool to check the effectiveness of the analytic model.



Mode 1 (y-axis : 278.845Hz) Mode 2 (x-axis : 279.098Hz) Mode 3 (rotation z-axis of bridge : 728.71Hz)

Figure 3. Analytic results and FEM simulation results of the stage

5. Conclusion and future work

We proposed a compliant parallel XY nano-positioning stage which has 118um X, Y directional moving range and 278Hz in first resonance frequency. The analytic model and the optimal design of stage were carried out. The results from analytic model, we can achieve a high dynamic performance of the stage using optimal design.

Future research includes the fabrication of this stage with final dimension and experiments to verify the performance.

Acknowledgement

This research was supported by grant from the Wold-Class 300 Project R&D Program funded by the Ministry of Knowledge Economy, Republic of Korea

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