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Design and Control of Flexure based XYθ_z Stage

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This article presents the design and control of an ultraprecision XYθ_z stage with nanometer accuracy. The stage has a plane mechanism and symmetric hexagonal structure which consists of a monolithic flexure hinge mechanism with three piezoelectric actuators and six flexures preserving the plane motion. The symmetric design reduces the effect of temperature gradient on the structure. Because the relationship between design variables and system parameters are quite complicated and there are some trade-offs among them, it is very difficult to set design variables manually and optimal design procedure is used. The objective of design is maximizing the 1st resonant frequency to improve the dynamic characteristics. The reason is that the stage must move with heavy load of about 20 kg. The higher resonant frequency also makes the stage stiffer and stronger against the dynamic force and moment. This paper describes the procedures of selecting parameters for the optimal design and a mathematical formulation for the optimization problem. The stage was designed to attain $\pm$10 um in the X- and Y-direction and $\pm$90arcsec in the yaw direction at the same time and have the 1st resonant frequencies of 455.5 Hz in X- and Y-direction and 275.3 Hz for yaw direction without load. The stage was fabricated according to the optimal design results and experimental results indicate that the design procedure is effective. A conventional PI control results are presented for ultraprecision motion.

Key Words : Precision Stage, Flexure, Piezoelectric Actuator, Nanotechnology, Optimal Design

1. Introduction

Nowadays, manufacturing technologies such as machine tool and semiconductor manufacturing processes and measuring technologies such as scanning probe microscopes are rapidly progressed and stimulate the development of precision positioning devices. The precision positioning devices require long travel range and high speed for large throughput of semiconductors. In order to satisfy these requirements, dual stages have been proposed and developed. (Shigeo et al., 1988; Yuichi et al., 1993 ; Lee and Kim, 1997 ; Kang et al., 2002) Dual stage have a coarse stage for large travel and a fine stage for high precision. In latter case, because frictions often account for the major portion of the precision motion errors there are many approaches avoiding them. (Lee and Baek, 2003 ; Kim and Trumper, 1998) and Piezoelectric actuator (PZT) and flexure guide mechanism are very common solution due to their simple structure. As flexure guide mechanism uses the elastic deformation at the connection between two rigid bodies to provide relative motion, thus eliminating sliding contact. It has many advantages for getting the precision motion : negligible backlash and stick-slip friction ; smooth and continuous

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displacement ; almost linear input-output displacement relation ; and inherently infinite resolution. (Smith and Chetwynd, 1992) Piezoeletric actuator uses inverse piezoelectircity that an electrical field applied to piezoelectric crystal leads to a deformation of the material. It has characteristics of unlimited resolution, large force generation, fast response, low power consumption and no wear. (PI catalog, 2001) There are so many researches about fine stage with PZT and Flexures. But the design objective of each is different, as a results, the structure and design strategy of stage must become different. Lee (Lee and Kim, 1997) proposed the XYθ_z fine stage as a part of dual stage, but his work was focused mainly on design and control of total dual stage system and mentioned only the structure and kinematics of the fine stage. Ryu (Ryu et al., 1997) proposed the XYθ_z stage as a wafer mask aligner and the design method of the flexure based fine stage. His work was aimed maximizing yaw motion, as a results, the structure composed of the double compound lever amplifier which uses lever amplifier twice was proposed and designed. While above two stages are parallel type, the stack type was proposed to achieve low crosstalk interference. (Chang et al., 1999) Besides, in order to extend the travel range, so many type of flexure structure were proposed. (Chang and Teague, 1998 ; Eiichi and Makoto, 1992) In this paper, the stage must carry a massive load of about 20 kg, therefore, the objective is maximizing bandwidth and stiffness in order to improve dynamic properties such as settling and rising time and reduce the positioning error due to dynamic force in out-of-plane direction. For this purpose, a novel mechanism utilizing the coupled hexagonal structure with three PZT and six leaf-spring-type flexures is presented.

The performance of flexure mechanism is heavily infleunced by variation of parameters such as flexure thickness and length. In the procedure of determining the design parameters, design objective must be considered. The method to select parameters and their numerical value usually devided into two, one is finite element analysis (Elmustafa and Lagally, 2001) and the other is mathematical model based analysis. (Ryu et al., 1997 ; Chang et al., 1999) The former is so accurate analysis method but time consuming. On the other hand, the latter is not so accurate and requires the mathematical formulation effort. However it doesn't require many time to get simulation results, therefore it is suitable to investigate the effects of variation of several parameters and apply systematic design method such as the optimal design to determine the parameters. We present the mathematical formulation of flexrue mechanism and conduct the optimal design in order to attain the design objective.

2. Conceptual Design of the Stage

In this section, we present the structure of XY θ_z stage in order to get higher resonant frequency and stiffness. A XYθ_z stage consists of three piezoelectric actuators and monolithic flexure mechanism. Two types of flexure mechanism are used. Hinge type flexure is used as a joint mechanism between an output body and the actuators and there is no displacement amplifier. Piezoelectric actuators with motion amplifiers have several advantages over standard piezoelectric actuators : compact size compared to stack actuators with equal displacement and reduced capacitance. But it is necessary that the stiffness of drive system decreases. If a motion amplifier behaves like ideal joint, the following relations apply.

$$k_{sys} = \frac{k_0}{r^2} \tag{1}$$

where k_{sys} is stiffness of the amplified system, k_0 is stiffness of the primary drive system (PZT) and r is amplification ratio. The stiffness of drive system restricts the stiffness of guide mechanism and overall system. Therefore we don't use motion amplifier. And leaf spring type flexure is used as a three-DOF joint mechanism improving the stiffness of stage and preserving plane motion. When only a few flexures are used, there is a limitation on increasing system stiffness due to maximum stress constraint of material. Therefore we use six flexures in parallel.

The overall structure is a symmetric plane mechanism of hexagonal type. Three actuators are arranged at intervals of 120 degrees with offset by which rotational moment is obtained and six flexures are placed at the center of each side. The symmetric design reduces the effect of temperature gradient on the structure. As all actuators are directly connected into moving part in order to improve the stage's dynamic characteristics by reducing moving mass, all actuators must work properly according to kinematics between actuator inputs (e.g. input voltages) and moving platform outputs (e.g. position) to get a desired motion. This relationship is modelled mathematically by Ryu (Ryu, 1997)'s algorithm. To reduce moving mass, part of outside is used as a moving part. Fig. 1 shows the proposed mechanism and design parameters.

The $XY\theta_z$ stage is used as a fine stage in dual stage system and compensates the errors and dynamics of coarse stage. Therefore the fine stage requires only a small travel range and fast dynamic characteristics. The specifications of fine stage is as follows ; First, in order to get the travel range specification, it is assumed that the total motion error of coarse stage due to assembly alignment error between coarse and fine stage, straightness error of coarse stage guide and so on is less than ±10 um in the translational axes (X and Y axes) and less than ±90 arcsec in the rotational axis (θ_z axis). These values were verified by actuating only coarse stage. From above assumption, we determined the travel range specifications so that the stage attains ±10 um in translational directions and ±90 arcsec in rotational direction at the same time. And we select 1000 V/60 um PZT (Pst 1000/25/60 piezomechanik) to offer the sufficient displacement and force. Second, the required resolution of fine stage is about 10 nm because nowadays manufacturing and measuring semiconductor industries and some other precision oriented technologies require reproducibility less than 20 nm. (Lee and Kim, 1997) Because the resolution of PZT stage is decided by only electrical noise level of all devices, e.g. sensor, digital to analog device, power supply and amplifier, the noise level must be less than 2 mV to actuate 1000 V/60 um PZT nearly around 12 nm resolution. Lastly, because the stage must carry approximately 20 kg load, it should have as fast settling and rising time as possible to increase the throughput. As the qualitative guide about time response specifications, the approximation for the second-order case with no zeros is commonly used.

$$t_r \cong \frac{1.8}{\omega_n} \tag{2}$$

$$t_s \approx \frac{4.6}{\zeta\omega_n} \tag{3}$$

where t_r is the rising time, t_s is the settling time, ζ is the damping ratio and ω_n is the undamped natural frequency. From above equations, we can know that the time response is improved by increasing the damping ratio and undamped natural frequency. But the damping ratio can be got from only control law and cannot be increased in design stage because the flexure inherently has very small damping. Therefore a design strategy is to increase the natural frequency. But it is limited by a maximum stress generated at the flexure mechanism and a maximum force that PZT actuator can generate, because while the stiffer and lighter mechanism is need to get higher nat-

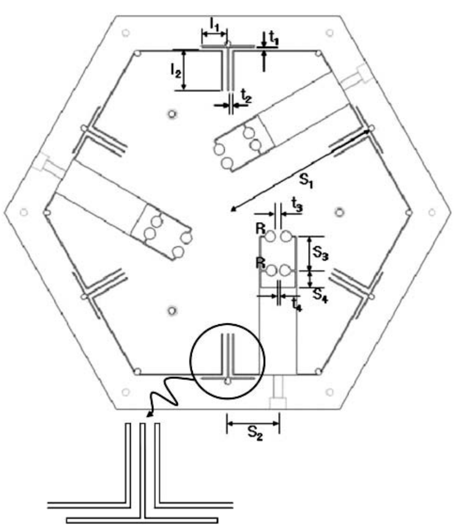


Fig. 1 Drawing of the proposed X-Y-Theta stage and design parameters

ural frequency it generates higher stress at the flexure in order to get a same travel and requires a higher force actuator. Besides, there are some trade-offs which will be solved by optimal design procedure.

3. Modeling of the Stage

Since the relationship between design variables and system parameters are complicated, it is very difficult to set design variables manually. Therefore optimal design is used. In order to perform the optimization, static and dynamic control model of the stage must be made. Flexure mechanism is modelled as rigid bodies connected through translational/rotational springs. In the proposed stage, total ten moving bodies is connected by several springs. (Fig. 2) The Lagrange's equation of the system can be written as :

$$\frac{d}{dt}\left[\frac{\partial(T-V)}{\partial \dot{q}^i}\right]-\frac{\partial(T-V)}{\partial \dot{q}^i}=Q^i, \quad i=1, 2, \cdots, N_b \tag{4}$$

Where T and V denote the kinetic energy and potential energy of the system, Q^i is the force vector, N_b is the number of moving bodies and q^i is the displacement vector of the origin of the body i or $\mathbf{q}^i=[x^i\ y^i\ z^i\ \theta_x^i\ \theta_x^i\ \theta_x^i]^T$.

$$T=\frac{1}{2}\dot{\mathbf{x}}^T\mathbf{M}\dot{\mathbf{x}}, \quad V=\frac{1}{2}\mathbf{x}^T\mathbf{K}\mathbf{x} \tag{5}$$

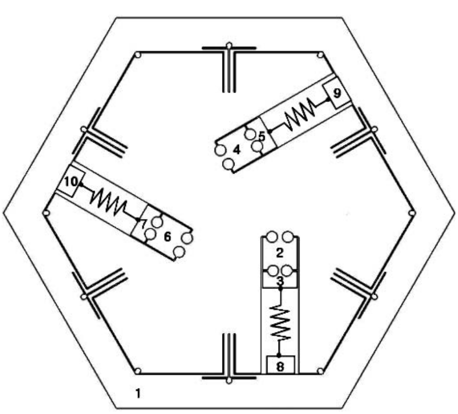


Fig. 2 Moving bodies connected by springs

where M and K represent mass and stiffness matrix and $\mathbf{x}$ is the system displacement vector and defined as follow.

$$\mathbf{x}^T=[\mathbf{q}^1\ \mathbf{q}^{2^T}\ \cdots\ \mathbf{q}^{N_b{}^T}]^T \tag{6}$$

In order to derive $\mathbf{K}$ matrix, the spring rate of hinge and leaf spring (Paros and Weisbord, 1965 ; Timoshenko and Goodier, 1970) and the coordinate transformations (Ryu, 1997) are used. As a result, following equation of motion is derived.

$$\mathbf{M}\ddot{\mathbf{x}}+\mathbf{K}\mathbf{x}=\mathbf{F} \tag{7}$$

Because there are 10 moving bodies and each body has six degree of freedoms, the size of $\mathbf{M}$ matrix and $\mathbf{K}$ matrix is $(6\times10)\times(6\times10)$ and the size of $\mathbf{x}$ vector is $(6\times10)\times1$. And the resonant frequencies are obtained by solving the following eigenvalue equation.

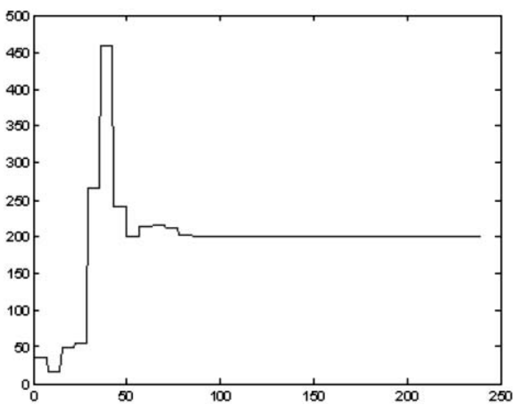
$$|\mathbf{K}-\omega^2\mathbf{M}|=0 \tag{8}$$

4. Optimal Design

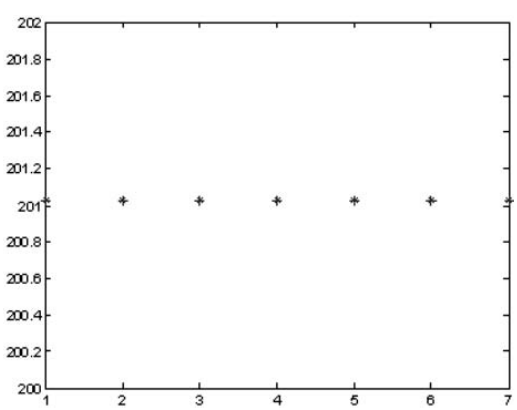
The performance of flexure mechanism is heavily infleunced by selection of parameters such as flexure thickness and length. In the procedure of determining the design parameters, design objective must be considered. As mestioned previously, the objective is maximizing the system bandwidth to improve the dynamic characteristics. It guarantees fast response and robustness against several dynamic disturbances. Therefore following cost function is used.

$$\text{Minimize}\left(\frac{1}{\omega_x}\right)^2+\left(\frac{1}{\omega_{\theta_z}}\right)^2$$

Design parameters are presented in Fig. 1, but some parameters less effective to stage performance are determined as far as constraints are not violated by designer's subjectivity, and others are used as design variables. Following constraints is used ; moving range (X- and Y-axis must be more than ±10 um and θ_z-axis must be more than ±90 arcsec at the same time), stress at flexure (the maximum stress at flexure system must be less than material yield stress divided by stress safety factor), degree of plane motion (rates between out-of-plane stiffness and in-plane stiffness must be more than ten), overall dimension (the volume of stage must be less than 400 mm × 400 mm × 35 mm) and interference condition for fabrication and assembly (there is no interference among components). Above constraints except degree of plane motion are general things for flexure system design. But the degree of plane motion defined by rates between out-of-plane stiffness and in-plane stiffness isn't common and doesn't be considered in other works. Because the flexure system is moved by compliance, when the force doesn't be operated exactly, there are out-of-plane moments. In order to reduce these effects, the degree of plane motion must be increased. In addition, because a heavy load occurs dynamic moments in fast operations, this constraint becomes more important.



(a)



(b)

Fig. 3 Cost function convergence and convergence value with 7 initial points

For the solution, a sequential quadratic programming (SQP) method and MATLAB have been used. Fig. 3(a) shows the cost function convergence. Fig. 3(b) verifies that the optimal design process has a global minimum to shows it has identical converged cost function value with different 7 initial points by Baysian Stopping Rules (Boender and Rinnooy Kan, 1987).

Design optimization results are presented in Tables 1～3. The stage was fabricated according to the optimal design results as prescribed in Fig. 4.

Table 1 Optimal design results : design variables

Design variables	S_1	S_2	S_3	S_4	b	R
Values (mm)	144.5	45.0	30.0	15.0	35.0	10.0
Design variables	l_1	l_2	t_1	t_2	t_3	t_4
Values (mm)	21.8	37.8	2.6	3.9	4.0	2.0

Table 2 Characteristics of stage according to optimal design results

Resonant freq.	Without load			With load		
	ω_x	ω_y	$\omega_{\theta z}$	ω_x	ω_y	$\omega_{\theta z}$
Values (Hz)	455.1	455.1	275.1	160.8	160.8	78.4

Table 3 Characteristics of stage according to optimal design results

Stiffness	K_x	K_y	K_z	$K_{\theta x}$	$K_{\theta y}$	$K_{\theta z}$
	Unit (N/um)			Unit (Nm/urad)		
Values	25.7	25.7	261.3	26.1	26.1	0.26

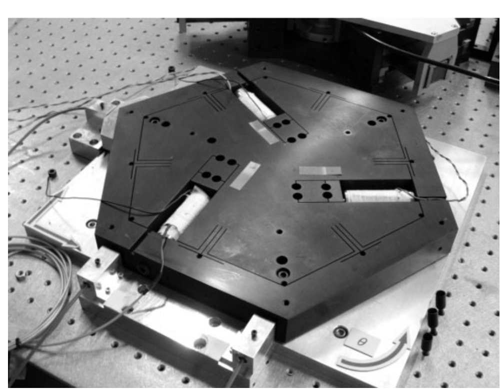


Fig. 4 Fabricated X-Y-Theta Stage

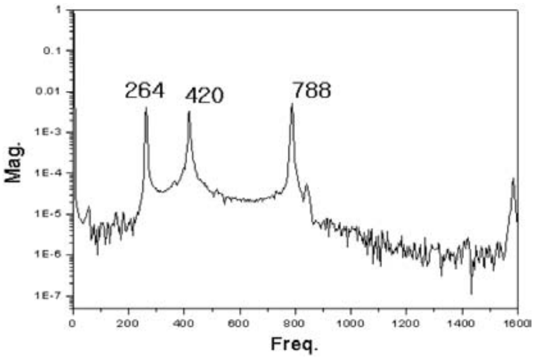


Fig. 5 Impulse response function

To measure impulse response function, impact hammer, capacitance gages (probe 2805 with the sensitivity of ±50 um/±10 V, ADE corp.) and dynamic signal analyser (Hewlett Packard 35670 A) are used. Fig. 5 shows the impulse response in x-direction measured at an offset point from the mass center. The first peak frequency (266 Hz) is the resonant frequency of yaw-direction and the second peak frequency (420 Hz) is that of x- and y-direction. The third peak frequency (788 Hz) is the resonant frequency of moving part itself. Measured natural frequencies are matched with designed natural frequencies by less than 5% error.

5. Control of the Stage

Fig. 6 shows the experiment setup of control. The controller is a DSP controller (dSPACE corp.), the amplifier is SVR 1000-3 (Piezomechanik corp.) and the position sensors are three capacitance gauges (probe 2805 with the sensitivity of ±50 um/±10 V, ADE corp.). One capacitance gauge measures x-direction position and two capacitance gauge measure y-direction and theta (yaw) direction.

The proposed stage is parallel mechanism in which three piezo actuators are coupled. Therefore, the kinematic relationship between input voltage of each actuator and motion of moving part must be examined. Since three sensors are also coupled, the same operation between sensor output and motion of moving part must be conducted. Because the proposed stage is linear sys-

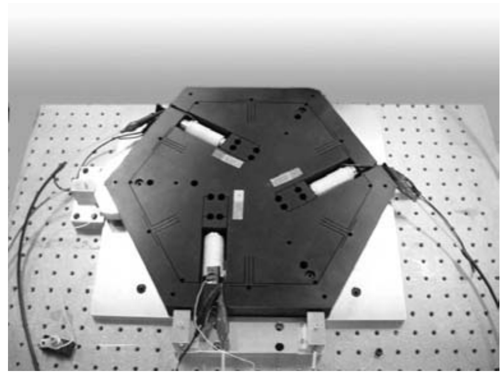


Fig. 6 Experimental setup

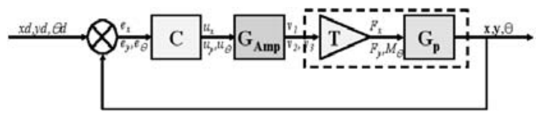


Fig. 7 Control block diagram

tem, above relationship in equation (7) is represented by following equations.

$$F = TV_{in} \tag{9}$$

$$X = T^{*}V_{out} \tag{10}$$

In above equations, F is the force vector by PZT actuator, **X** is the displacement vector of moving part expressed by rectangular coordinate, **T** and **T*** is the transpose matrix containing the kinametics of actuators and sensors, $\mathbf{V}_{in}$ is actuator input voltage vector and $\mathbf{V}_{out}$ is sensor output voltage vector. In addition, the $XY\theta_z$ stage is treated as three independent SISO systems not coupled MIMO system because there is so small interference among three actuators. Fig. 7 shows a control block diagram for conventional PI control and Figs. 8~10 show the control results.

Control results show the stage can follow 10 nm step and 0.05 arcsec step. But a PI controller has the high-gain characteristics to achieve ultra-precision and high speed, so it is not robust to modeling error and external disturbances. Since the developed $XY\theta_z$ stage is a coupled mechanism, errors in the inverse kinematics also act as disturbances. Therefore, we will compensate these disturbances and present the results in near future.

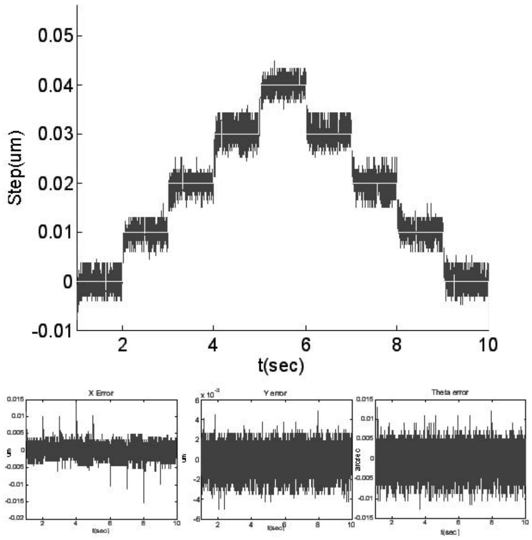


Fig. 8 X-axis 10nm step response and errors

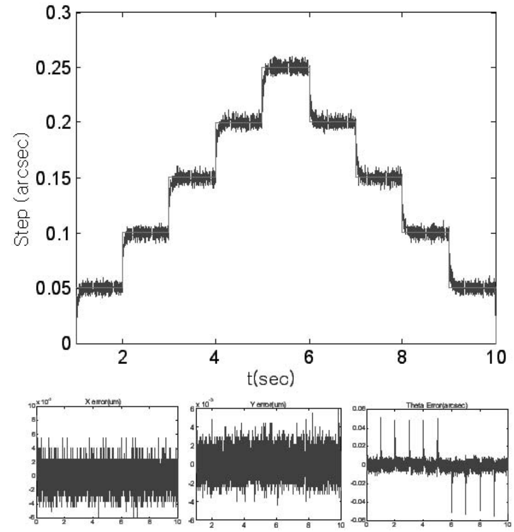


Fig. 9 Theta-axis 0.05 arcsec step response and errors

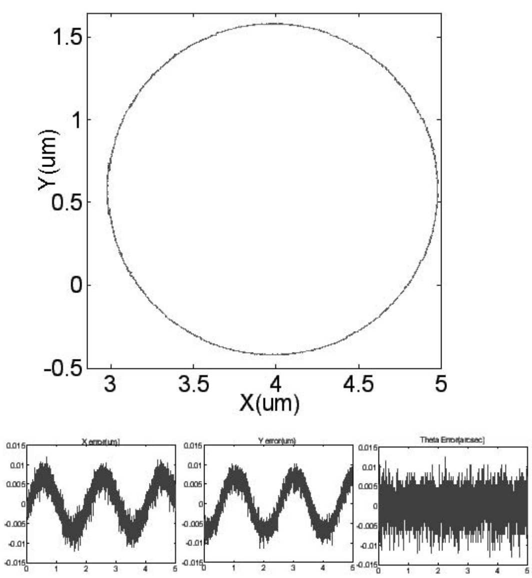


Fig. 10 2um circle control and errors

6. Conclusion

A novel mechanism of precision $XY\theta_z$ fine stage was proposed and designed. Using optimal design procedure, the stage is designed and fabricated and evaluated. Experimental results agree with design results in the deviation of less than 10%. We get 10 nm translation resolution and 0.05 arcsec rotational resolution using PI controller.

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