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Dynamic property and settling stability of ultraprecision positioning mechanism with sub-nanometre resolution driven by preloaded ball screw

Shigeo Fukada[1], Takehiko Shimura[1], Soichiro Koike[1]

[1]*Shinshu University, Japan*

sfukada@shinshu-u.ac.jp

Abstract

Dynamic property and settling stability of the mechanism driven by preloaded ball screw are discussed here in experiments using sub-nanometre-level motion. First, the relation between the rotational angle of the screw shaft and table displacement was clarified in the sub-nanometre range. Next, frequency responses from motor input to stage acceleration and from disturbance load on the stage to stage acceleration were both measured. Based on experiment results, a lumped parameter dynamic model with seven degrees of freedom was constructed. It was determined that a resonance frequency of 130 Hz is ruled by the axial stiffness of the flexible coupling between the nut and the stage. An oil film damper was applied to the mechanism to dampen the stage vibration. The residual vibration was attenuated to one third of RMS value, and ultra-fine positioning with resolution of 0.2 nm was achieved with this ball screw mechanism.

Keywords: Positioning mechanism, Ball screw, Ultraprecision positioning, Residual vibration

1. Introduction

Ball screws are most commonly used as positioning elements. Recently, the resolution of linear encoder has been improved to the pico-meter level, so a mechanism using the ball screw is expected to achieve positioning performance at sub-nanometre level resolution. In a previous report, an experimental positioning system was constructed using a linear encoder with resolution of 69 pm and aerostatic guide-way to show the quasi-static property of the positioning mechanism using ball screw without external axial load [1]. In addition, an axially loading device using an electromagnetic linear actuator was attached to the mechanism to clearly show microscopic behaviour under various loaded conditions [2]. In this report, the quasi-static relation between the rotational angle of the screw shaft and the table displacement is discussed experimentally using ultrafine motion with sub-nanometre range. And in order to clarify the limit of controlled settling stability of the mechanism, the dynamic property of the mechanism is evaluated by frequency response test to show a vibration model of the mechanism. Finally, to improve the positioning resolution, a simple method of oil film damper is applied to the mechanism, and the effect of the damper is then evaluated.

2. Experimental apparatus

Figure 1 shows the experimental positioning system schematically. A preloaded ball screw with a lead of 5 mm drives the stage, supported by an aerostatic guide-way with stroke of 100 mm. The nut is coupled to the stage by a special universal coupling device, which has four degrees of freedom as shown in the Figure, which eliminates excessive constraint between the nut and the stage. To apply axial load to the stage, an electromagnetic linear actuator using a voice coil motor (VCM) is set between the stage and the base. A linear encoder with resolution of 0.069 nm measures stage displacement, and an optical rotary encoder measures the rotational angle with a resolution of 1.75 μrad. To measure the microscopic rotational angle smaller than the encoder resolution, a special measurement device with resolution of 0.022 μrad was designed with capacitive probes as shown in Figure 1. This device was attached between the screw shaft and the rotary encoder. The positioning mechanism is set on a granite surface plate supported by a pneumatic isolation bench.

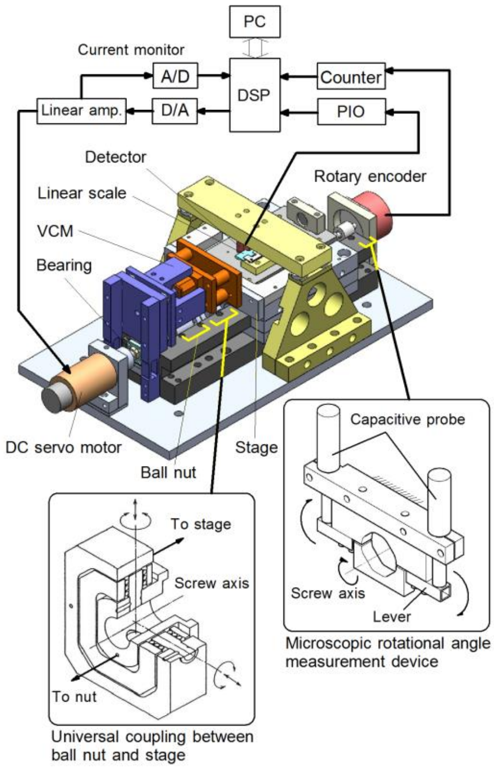


Figure 1 Experimental apparatus

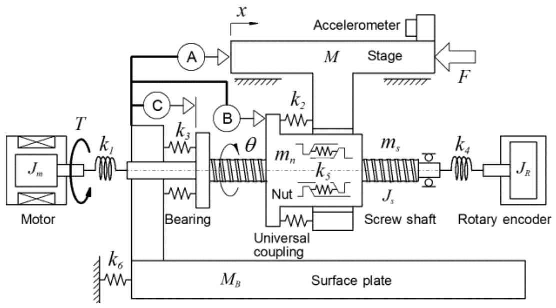


Figure 2 Schematic structure of mechanism

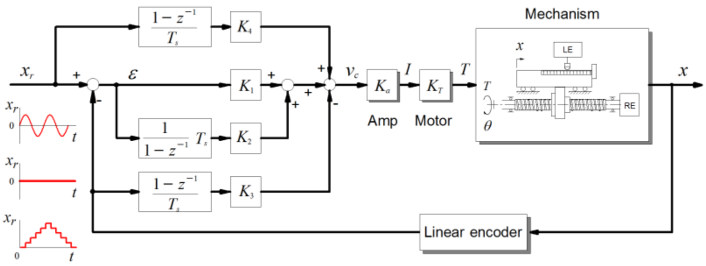


Figure 3 Control system

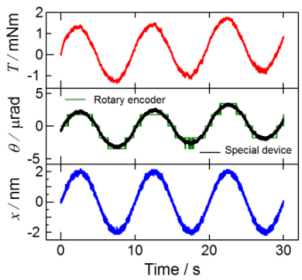
Figure 2 shows the structure of the mechanism. To measure stiffness in the following chapter, three displacement sensors - A, B and C - are used. To measure frequency response, a servo-type accelerometer is set on the stage. This mechanism has two input ports of torque or force: One is a DC servo motor of 130 W rotating the screw shaft of the ball screw with motor and screw shaft connected by a spring-type flexible coupling; the other is a VCM with force to current ratio of 50 N/A applying axial load to the stage.

Figure 3 shows the control system used in the following experiments. A three-degrees-of-freedom control system with PI-D feedback controller and feedforward compensators with derivative operation was realized by using a digital signal processor with sampling rate of 8 kHz.

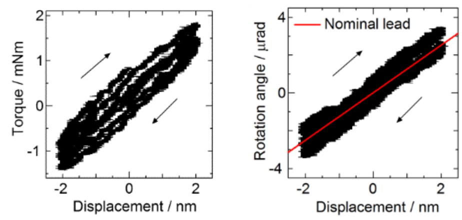
3. Property of mechanism

3.1. Quasi static behaviour

First, the quasi-static microscopic behaviour of the mechanism was measured in sub-nanometre range using small sinusoidal position commands x_r in Figure 3. Figure 4 shows an experimental result when the amplitude of stage displacement is 2 nm and period of sinusoid is 12 s. Figure (a) shows time domain response with rotation angle of the screw shaft θ, motor torque T and stage displacement x. The motor torque was determined by multiplying motor current by torque constant of the motor. The rotation angle is shown in values obtained by both the rotary encoder and the angle measurement device as shown in Figure 1. The rotational angle and displacement follow the torque change without any time lag, so the elastic property can be simply shown [3]. Figure 4 (b) and (c) show the relation among the measured variables in (a). In this sub-nanometre range, the relation between θ and x follows the nominal lead of the screw quasi-statically, while the relation between T and x shows nearly linear characteristics, though it also shows slight hysteresis or drift. If the amplitude of x_r is increased, the hysteresis is enlarged as shown in Figure 5, where the amplitude is 600 nm. And in this range the relation between θ and x has completely linear relation. If the amplitude is enlarged over micrometres, the torque is saturated with a certain value determined by macroscopic rolling friction of the ball screw and the bearing. However, it is important that, within the microscopic range, the stage motion is ruled by the elastic property in T- x relation and the linear relation between x and θ according to the lead of screw.

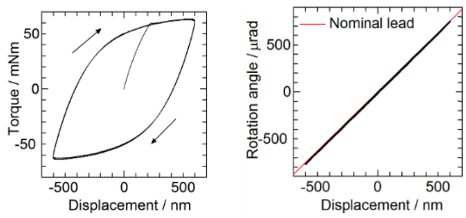


(a) Time domain response



(b) $T-x$ relation (c) $\theta-x$ relation

Figure 4 Microscopic behavior of mechanism



(a) $T-x$ relation (b) x-θ relation

Figure 5 Nonlinear elastic behavior in case of amplitude of 600 nm

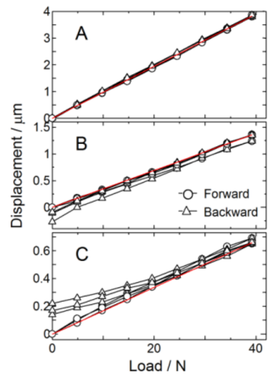


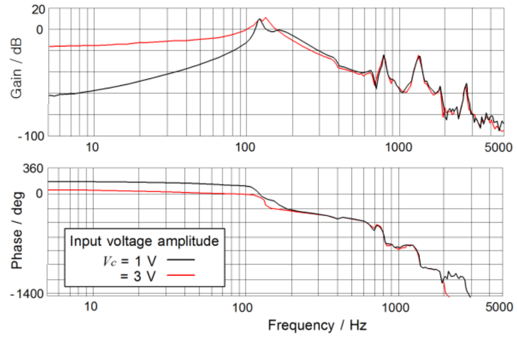
Figure 6 Axial stiffness

3.2. Static Axial stiffness

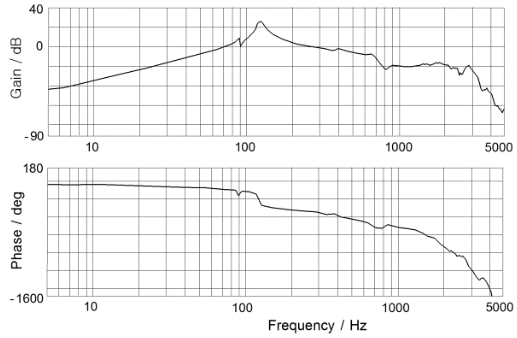
Figure 6 shows experimental results of axial displacement measured by displacement sensors A, B and C in Figure 2 when axial load was applied to the stage statically. The gradient of the linear line represents the stiffness of each part: A-B is the stiffness k_2 of the coupling between the stage and the nut, B-C is the stiffness k_5 between the screw shaft and the nut, and C is the stiffness k_3 of the thrust bearing supporting the screw shaft.

3.3. Dynamic property

Next, frequency responses of the mechanism were measured. Figure 7 (a) shows the response from motor input to stage acceleration, and (b) shows the response from disturbance load

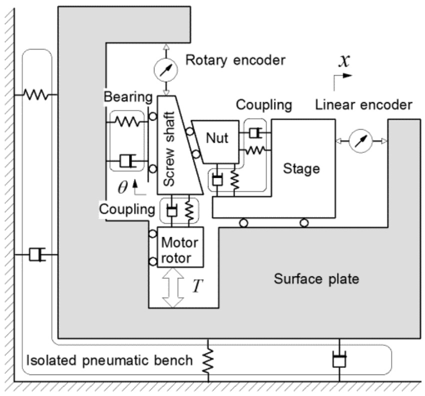


(a) Response from motor input to stage acceleration

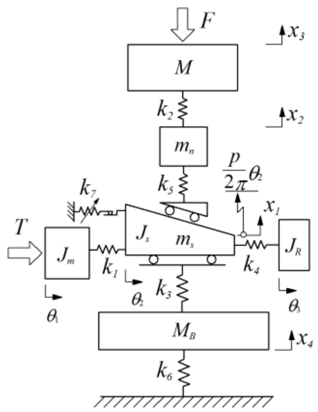


(b) Response from disturbance load to stage acceleration

Figure 7 Frequency response of the mechanism



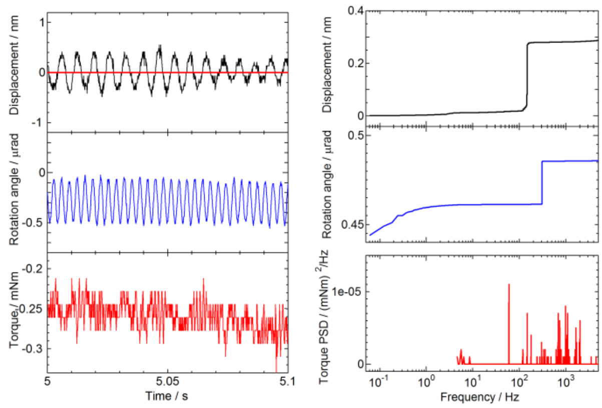
(a) Structure



(b) Model for natural frequency analysis

Figure 8 Vibrating structure

on the stage to stage acceleration. (a) includes two results with different input voltage amplitudes , 1 V and 3 V, of the amplifier driving the motor. When the amplitude is above 3 V, the motor rotates macroscopically; when the amplitude is smaller than 1 V, the motion is within the range of microscopic behaviour ruled by nonlinear elastic property. Therefore, both curves show different gradients in low frequency range under 100 Hz. Both (a) and (b) clearly show 1st resonance at 130 Hz.



(a) Time domain (b) Frequency domain

Figure 9 Vibration under controlled settling condition

And a slight resonance is confirmed around 400 Hz, while various resonances are shown at 800 Hz, 1300 Hz, 1900 Hz and 2800 Hz in (a). These results show that the mechanism has a vibrating system with multiple degrees of freedom

4. Vibration model

Figure 8 (a) shows the schematic structure of the mechanism, consisting of several inertial elements connected by springs and dampers. To determine the natural frequency of the vibration system, variables and parameters are defined as shown in Figure 8 (b). Spring constants of k_1 - k_6 corresponds to parameters in Figure 2. And k_7 represents the nonlinear elastic spring constant approximately determined in Figure 4. Since this vibrating system has seven degrees of freedom, the natural vibration is described by following equation of motion [4].

$$\mathbf{M}\ddot{\mathbf{X}} + \mathbf{K}\mathbf{X} = \mathbf{0} \tag{1}$$

Vector $\mathbf{X}$ and matrix $\mathbf{M}$ and $\mathbf{K}$ are defined as follows.

$$\mathbf{X} = [\theta_1\ \theta_2\ \theta_3\ x_1\ x_2\ x_3\ x_4]^{\mathrm{T}}$$

$$\mathbf{M} = \mathrm{diag}[J_m\ J_s\ J_R\ m_s\ m_n\ M\ M_B]$$

$$\mathbf{K} = \begin{bmatrix} k_1 & -k_1 & 0 & 0 & 0 & 0 & 0 \\ -k_1 & k_1 + k_4 + \left(\frac{p}{2\pi}\right)^2 k_5 + k_7 & -k_4 & \frac{p}{2\pi}k_5 & -\frac{p}{2\pi}k_5 & 0 & 0 \\ 0 & -k_4 & k_4 & 0 & 0 & 0 & 0 \\ 0 & \frac{p}{2\pi}k_5 & 0 & k_3 + k_5 & -k_5 & 0 & -k_3 \\ 0 & -\frac{p}{2\pi}k_5 & 0 & -k_5 & k_2 + k_5 & -k_2 & 0 \\ 0 & 0 & 0 & 0 & -k_2 & k_2 & 0 \\ 0 & 0 & 0 & -k_3 & 0 & 0 & k_3 + k_6 \end{bmatrix},$$

where J_m, J_s , J_R, m_s, m_n, M and M_B in matrix $\mathbf{M}$ are inertial moments or masses of elements as shown in Figure 2 and Figure 8 (b). p in matrix $\mathbf{K}$ is lead of screw, and k_i are spring constants. The values of k_2, k_3 and k_5 were determined in Figure 6, and k_1, k_4 and k_6 were known from maker's manuals.

The natural frequency is obtained by eigenvalue of matrix $\mathbf{KM}^{-1}$. The calculated results are seven natural frequencies: 1.1, 129, 400, 954, 1526, 1982 and 3981 Hz. These calculated values correspond approximately to the experimental results of Figure 7(a) except for the slowest frequency of 1.1 Hz.

5. Controlled property

5.1. Settling stability

The vibration discussed in the previous section represents open-loop characteristics of the mechanism without feedback control. Next, the vibration was observed under settling condition with feedback control. Figure 9 shows the experimental results on variation of each variable when the position command x_r in Figure 3 was held at 0. (a) shows time

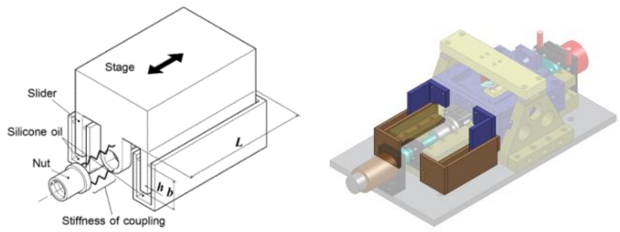
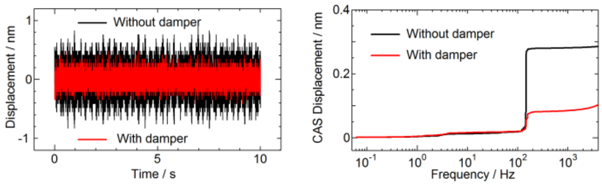


Figure 10 Parameters of oil film damper



(a) Time domain (b) Frequency domain (CAS)

Figure 11 Effect of oil film damper

domain profile of the variation, while (b) shows frequency domain characteristics of each variable in the cumulative amplitude spectrum (CAS) or power spectrum density (PSD) [5]. The mean position of the stage is kept at 0 by the feedback control. However, the stage displacement shows residual vibration of 130 Hz with amplitude of 0.5 nm, and the rotational angle of the screw shaft shows vibration of 300 Hz. The vibrations of the stage displacement and the rotation of the screw shaft are respectively governed by different natural frequencies. In quasi-static condition, the relation between the stage displacement and the rotation of the screw shaft follows the linear relation determined by nominal lead of the screw as shown in Figure 4 (c), but in dynamic settling conditions, each vibrates with each its own frequency.

6. Improvement of stability

The vibration of the stage displacement with 130 Hz is mainly determined by the axial stiffness k_2 of the universal coupling between the nut and the stage, and the stage mass M, which is a single-degree-of-freedom vibration system. It is difficult to eliminate residual vibration by operating a motor torque. One method to improve the vibration is to increase the stiffness k_2. However, it is not advisable because it needs stiffer joints resulting in too heavy device for considering effect expected. Therefore, a simple passive method was attempted by using an oil film damper to improve the stability of the stage vibration. Figure 10 shows the concept and structure of the damper: Sliders were attached to both sides of the stage at symmetrical positions relative to the screw axis, and the sliders were sunk into grooves filled with silicone oil of 1×10^5 cSt. The gap h between the slider and the groove is 0.2 mm, and the width b and length L of the slider are 20 mm and 75 mm respectively. Damping ratio ζ obtained by this damper is about 0.08. Figure 11 shows the effect of the damper: Residual vibration was attenuated to one third of RMS value.

To determine the resolution of the positioning system, a multi-step position command was input to the control system of Figure 3. Figure 12 shows the experimental result for step width of 0.2 nm. Ultra-fine positioning with sub-nanometre level resolution of 0.2 nm was achieved with this ball screw mechanism. The realization of such an ultra-fine resolution with a relatively simple control system is owing to the simple elastic and linear property of the mechanism within microscopic range as shown in Figure 4. If the motion range is increased, the effect of hysteresis becomes significant as shown in Figure 5 (a). However, also in those conditions, the final settling of positioning is ruled by the microscopic elastic behaviour, and the elastic property is expected to contribute to the fine positioning with a feedback control.

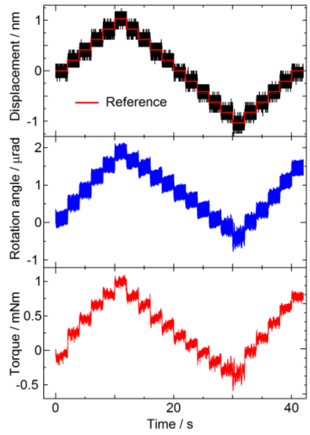


Figure 12 Result of 0.2 nm step positioning

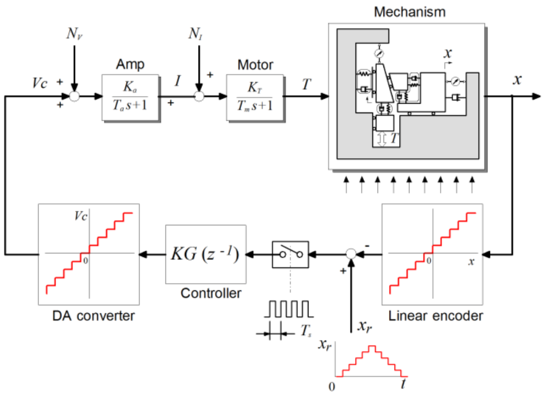


Figure 13 Signal flow and transformation of variables

It was determined that it was possible to attenuate residual vibration with the passive oil film damper. So the cause of the residual vibration is considered. Figure 13 shows the schematic process of signal flow and transformation of variables. The mechanism acts as a vibrating system with multiple degrees of freedom as shown in Figure 8, and various sources are exciting the vibration mechanically from floor or environmental air conditioning. Electrical noise in analogous electromagnetic and electronic devices disturbs the stability of the system. In addition, the sampling and quantization of the signal in digital processing system disturbs the stability in the sub-nanometre region. In future work, a more effective solution for these sources of disturbance should be considered.

7. Conclusion

Property of the positioning mechanism driven by preloaded ball screw was examined in sub-nanometre range. In addition to quasi-static behaviour and static axial stiffness, dynamic property was evaluated by frequency response test. It was determined that the mechanism acts as a vibrating system with multiple degrees of freedom, and the residual vibration in controlled settling condition is governed by the vibration system. To attenuate the residual vibration, a simple passive damper was applied, and ultra-fine positioning with resolution of 0.2 nm was realised with the ball screw mechanism.

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