Proceedings of the 13th
KACC, October 1998

Displacement Characteristics of a Parallel Leaf Spring Mechanism with Large-Deflective Elastic Hinges for Optical Mount

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Abstract

In this paper, we examine the displacement characteristics of the parallel leaf spring mechanism with large-deflective elastic hinges, and the validity of this mechanism as a translational and rotational mechanism is confirmed with multi-input system. This study is focused on the linear driving force as an input force, which is applied to the large-deflective elastic mechanism, and the displacement characteristics are discussed with theoretically and experimentally. The motions of this mechanism due to large-deflective hinges are changed by the position of loading force regardless of a single driving force. The numbers of degree of freedom are increased with the hinges, and we can be used to a multiple driving force in order to obtain many types of output.

1. Introduction

The optical systems which are composed optical elements such as lens, mirrors, detectors, etc., need the alignment and the positioning of optical elements as shown in Fig.1. And the positioning of those is carried with translational and rotational motion. As an application of micro optical system, input light is intercepted or guided to other direction as translational and rotational movement of optical mount, and then is used optical switch, wave guide [1][2], etc. Also, in order to realization of a compact system, it is desirable that one optical mount can be moved translationally and rotationally. As an example of optical pickup system in Fig.1, optical mount with objective lens and flexible springs are moved to x and y direction. As a guide mechanism, optical mount needs multi-degree of freedom in field of optical system and others.

As an effective method to eliminate friction and the need to lubricate between the pairs in the link mechanism as in Fig.2(a), we have proposed to use large-deflective elastic hinges instead of those currently used, and constructed a parallel mechanism as a prototype example and analyzed its characteristics[3]. In addition, we have proposed the concepts of a pseudo-movable revolute pair and a pseudo-changeable link in the positioning mechanism with large-deflective elastic hinges which is the input link with the fixed pair as in Fig.2(b), and analyzed the displacement characteristics using the large

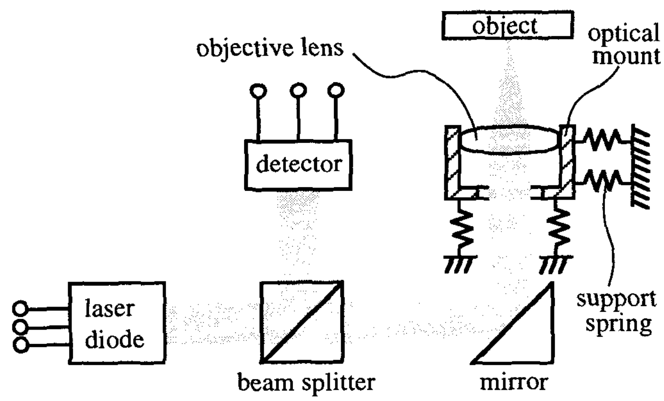
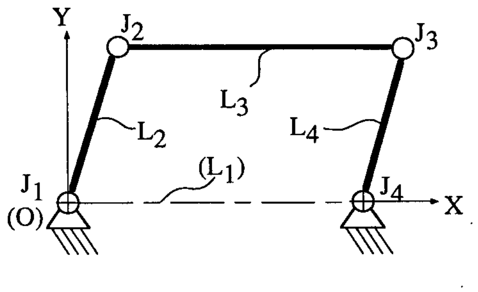
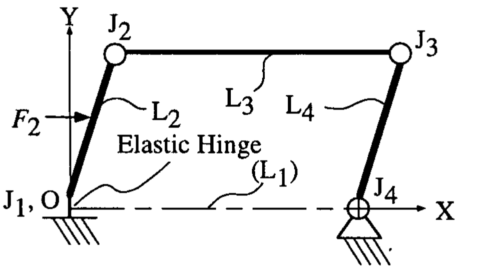


Fig.1 Optical mount in optical pickup system

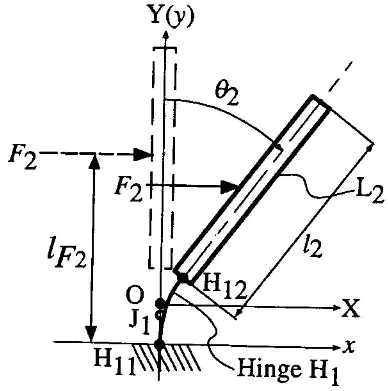


(a) Mechanism with four revolute pairs

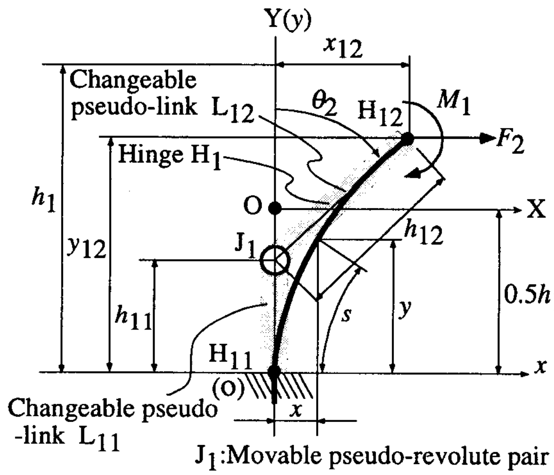


(b) Mechanism with a large-deflective elastic hinge

Fig.2 Parallel four-bar mechanism



(a) Schematic representation of hinge and link



(b) Model of movable peseudo-revolute pair

Fig.3 Concept of movable pseudo-revolute pair

deflection theory and confirmed the validity of the theory [4]. The tiny sizing of micromechanism with revolute pairs increases the friction force and relative clearance in comparison with the link length [5]. Micro-positioning mechanisms with circular notched flexure hinges have been developed to adjust the mirror [6] and are used to measure the line widths of photomasks [7]. However, in order to obtain a linear output displacement, all of those developments have used within the small deformed region. As a result, in most mechanisms with flexure hinges, the displacement magnifying mechanisms were used to obtain a large displacement. There are a few studies of large-deflective elastic hinges (LDEH) that obtained large displacement per hinge length. Therefore we permitted a LDEH to allow large displacement with one hinge, then replaced all revolute pairs with LDEHs in the mechanism as in Fig.2(a).

2. Beam Element of Large-Deflective Elastic Hinge

When we analyze the large deflection, the elliptical integral can solve the simple structure and load condition, however, it is difficult to solve the complex structure and multiple load conditions. The large deflection problem of hinges in the mechanism which appear in Fig.4 are necessary to establish the equilibrium equations after they have been deflected due to large deflection where it becomes a geometric nonlinear problem.

In the nonlinear large deflection problem, it is necessary to consider the deflected shape effect in conjunction with load increasing. In this case, the nonlinear term gets into relative equation of strain and displacement in the calculation of stiffness, and the stiffness of element k is defined by

$$k = k_E + k_G \tag{1}$$

where k_E is elastic stiffness of the element at the initial status and k_G is geometrical stiffness [8]. It is a simple method to compare with the method that needs the calculation of the strain energy. Nonlinear stiffness as Eq. (1) can be considered in a six-degree-of-freedom beam element model as shown in Fig.5, using the fixed coordinates system O-XY and internal axial force F_{ax}, transverse force F_{tr} and bending moment M are applied to the middle of both ends of the element, defined by

$$\begin{Bmatrix} F_{ax1} \\ F_{tr1} \\ M_1 \\ F_{ax2} \\ F_{tr2} \\ M_2 \end{Bmatrix}_i = \left[\frac{EI}{L^3} \begin{bmatrix} \frac{AL^2}{I} & 0 & 0 & \frac{AL^2}{I} & 0 & 0 \\ 0 & 12 & 6L & 0 & -12 & 6L \\ 0 & 6L & 4L^2 & 0 & -6L & 2L^2 \\ -\frac{AL^2}{I} & 0 & 0 & \frac{AL^2}{I} & 0 & 0 \\ 0 & -12 & -6L & 0 & 12 & -6L^2 \\ 0 & 6L & 2L^2 & 0 & -6L^2 & 4L^2 \end{bmatrix} + \frac{F_{ax2}}{L} \begin{bmatrix} 0 & 0 & 0 & \frac{AL^2}{I} & 0 & 0 \\ 0 & \frac{6}{5} & \frac{L}{10} & 0 & -\frac{6}{5}L & \frac{L}{10} \\ 0 & \frac{L}{10} & \frac{2}{15}L^2 & 0 & -6L & -\frac{1}{30}L^2 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & -12 & -\frac{L}{10} & 0 & \frac{6}{5} & -\frac{L}{10} \\ 0 & 6L & -\frac{1}{30}L^2 & 0 & -\frac{L}{10} & 4L^2 \end{bmatrix} \right]_i \begin{Bmatrix} u_1 \\ v_1 \\ \theta_1 \\ u_2 \\ v_2 \\ \theta_2 \end{Bmatrix}_i \quad (2)$$

where v, u and θ are transverse, axial and angular deformation of the element due to loads, respectively. Each beam element is treated as previously fixed one [9] and then node N_1 becomes hinged, so nodal displacement of node N_1 (v_1, u_1 and θ_1) is given

$$u_1 = v_1 = \theta_1 = 0 \quad (3)$$

Substituting Eq. (3) into (2) yields, the next equation is obtained.

$$\begin{Bmatrix} F_{ax2} \\ F_{tr2} \\ M_2 \end{Bmatrix}_i = \left[\frac{EI}{L} \begin{bmatrix} AL^2/I & 0 & 0 \\ 0 & 12 & -6L \\ 0 & -6L & 4L^2 \end{bmatrix} + \frac{F_{ax}}{L} \begin{bmatrix} 0 & 0 & 0 \\ 0 & 6/5 & -L/10 \\ 0 & -L/10 & 2L^2/15 \end{bmatrix} \right]_i \begin{Bmatrix} u_2 \\ v_2 \\ \theta_2 \end{Bmatrix}_i \quad (4)$$

And simplified Eq. (4) substituting with (1) is represented

$$\begin{Bmatrix} F_{ax2} \\ F_{tr2} \\ M_2 \end{Bmatrix}_i = \left[[K_E] + [K_G] \right]_i \begin{Bmatrix} u_2 \\ v_2 \\ \theta_2 \end{Bmatrix}_i \quad (5)$$

where $[K_E]_i$ and $[K_G]_i$ are the elastic stiffness matrix and the geometrical stiffness of element, respectively. In this paper, each element is treated as a small-deflection beam segment and its axial deformation (u) is negligible, and the transverse and angular deformation (v) of the element is sequentially calculated [10].

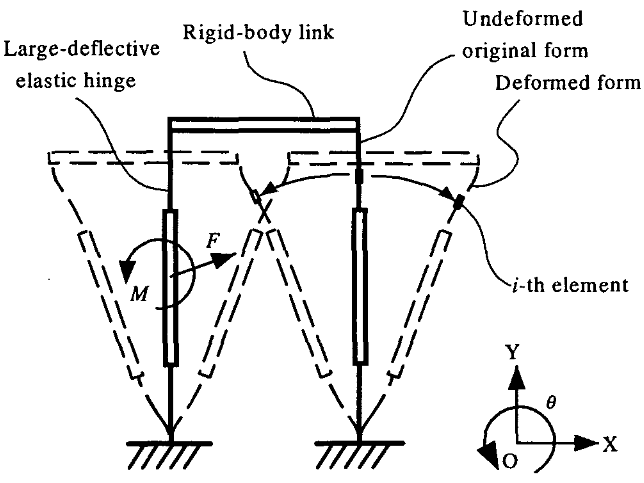


Fig.4 Parallel leaf spring mechanism with large-deflective elastic hinges

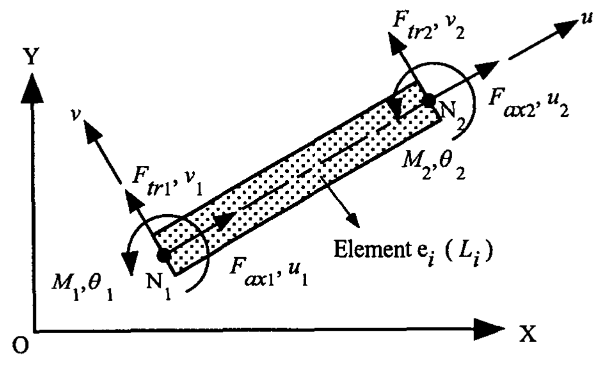


Fig.5 Six degree-of-freedom beam element in fixed coordinates system

3. Experimental and Calculated Results

The mechanism whose width is 15 *mm* and height is 25 *mm* with four large-deflective elastic hinges is represented in Fig.6(a), and the hinge length is made up of three times the link length (5-mm). The mechanism, which is a monolithic type, has links and hinges made by the same material, is made by super elastic alloy (SEA), so that it can avoid microslip [11]. Likewise, its thickness is 30 *μm* as clamped, and it is possible largely to deflect without microslip. The shape of the mechanism is formed by fixing in the assembled parts, and treated at 450℃.

The mechanism is modeled on divided beam elements, and then the mechanism's deformation is obtained with proposed method by authors which is called self-equilibrium convergence method(SECM)[12]. The variation of N_{41} on coupler as applied X-direction force at N_{21} is shown in Fig.7. In this case, the coupler is moved to 13 *mm* in X-direction and 5 mm in opposite Y-direction with 1gf applied. And then, the variation of N_{41} on coupler as applied X-direction force at N_{22} is shown in Fig.8. The coupler is moved to 2.77 *mm* in X-direction and 0.175 *mm* in opposite Y-direction with 1gf applied. We consider the calculated results with SECM well agree with measured data.

The concept of rotational motion in multi-input mechanism is represented in Fig.9. As the number of input becomes to four, it is possible that the coupler move to X, Y direction and rotation. In case of 4-bar link mechanism with revolute pairs, it has only one input because that mechanism's degree of freedom is one. However, in case of 4-bar link mechanism with large-deflective elastic hinges, it has input more than

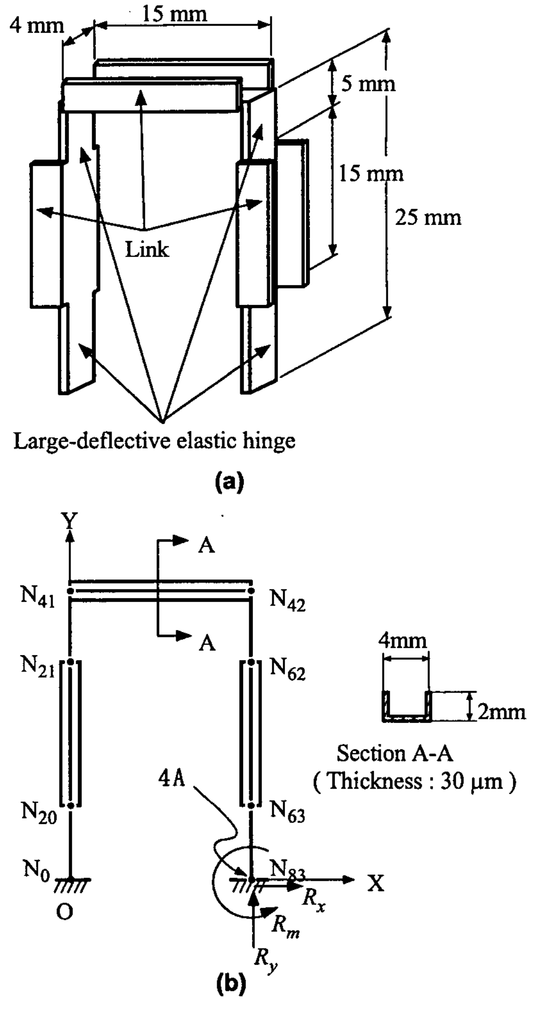


Fig.6 Dimension of parallel leaf spring mechanism (a) and modeling with beam element (b)

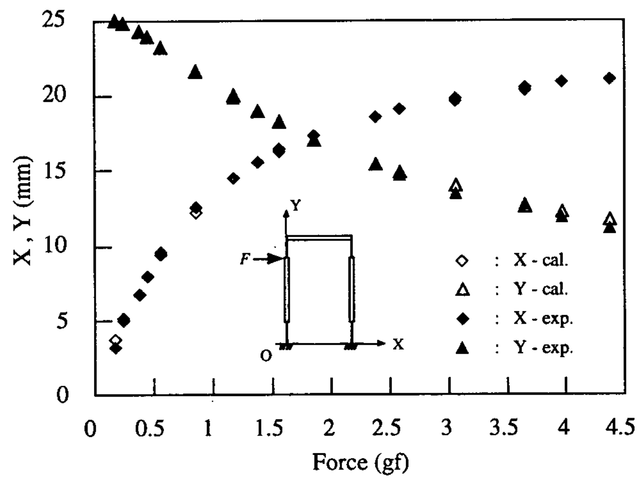


Fig.7 Displacement of X and Y direction with *F* applied N_{21}

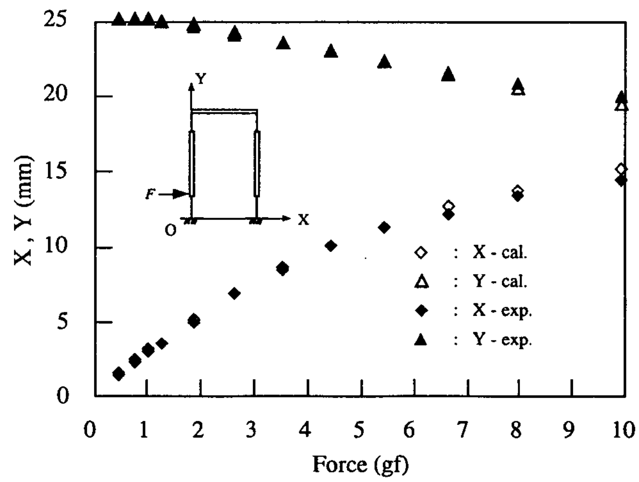


Fig.8 Displacement of X and Y direction with *F* applied N_{20}

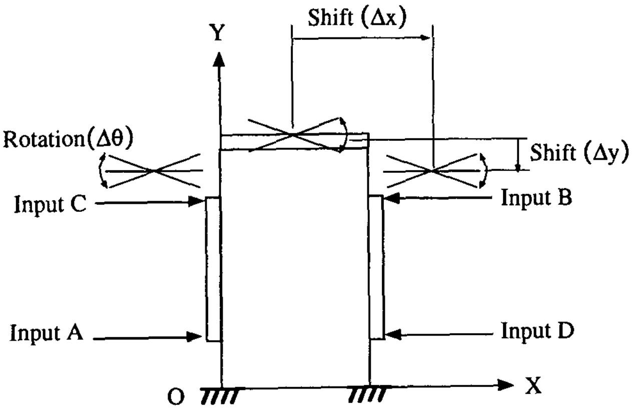


Fig.9 Concept of rotational motion of coupler in the mechanism with multi-input system

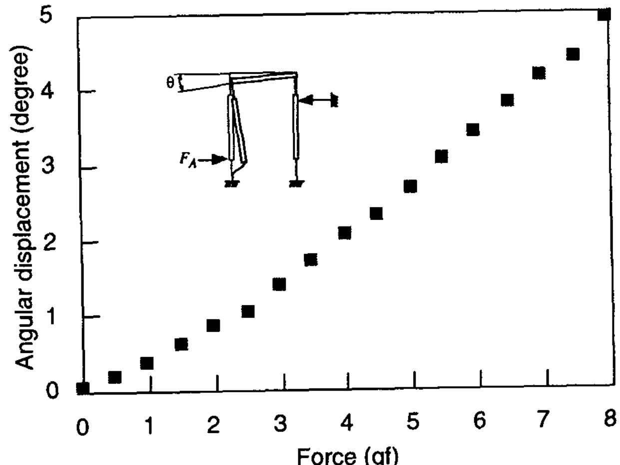


Fig.10 Rotational displacement of coupler when one input Is varied and the other input is constrained

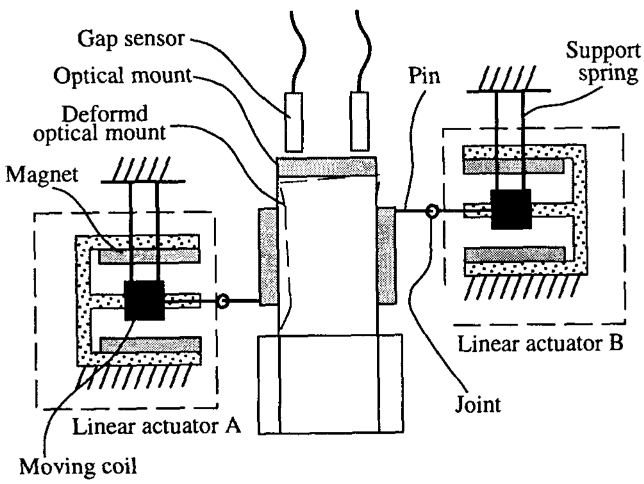


Fig.11 Rotational mechanism with two linear actuators

one, because that mechanism's degree of freedom is increase with the hinges. When the X directional force F_A is applied to N_{20} and N_{62} is constrained at the same time, the rotational displacement of coupler is shown in Fig.11. The coupler is rotated in counter-clockwise as increasing to F_A, and angular displacement is changed to almost linearly. From this result, we can consider that the coupler is rotated in clockwise when the X directional force F_A is applied to N_{63} and N_{21} is constrained at the same time. In order to exam the rotational motion of coupler with two input force, we made the two linear actuator system, one is applied to position of input A and the other is applied to input B respectively. The pin that is connected moving coil is possible to rotating in order to following the varying input position as deformation of mechanism and the up-down position of coupler is measured by two gap sensors.

In order to exam the rotational motion of the coupler which is shifted to X direction, two input forces are applied to input A and input B, respectively. As increasing the current I_A, to hold the coupler at 1 *mm* shifted in X direction with tuning the current I_B, and its angular displacement is shown in Fig.12. Where the horizontal and vertical axes represent the current of coil A and angular displacement of coupler, respectively, and the relatively minimum current is zero within the measured range. The angular displacement varies continuously as increasing the input current I_A, however, it varies with a square of the current. Also, as increasing the current I_A, to hold the coupler at 2 *mm* shifted in x direction with tuning the current I_B, and its angular displacement is shown in Fig.13. It varies with a square of the current as same with Fig.12.

4. Conclusions

As an effective method to eliminate friction and the need to lubricate between the pairs in the link mechanisms, revolute pairs are replaced with large-deflective elastic hinges. We propose the self-equilibrium convergence method (SECM) to solve the large deflection of hinges in the mechanism, and the calculated results with SECM well agree with measured data (Fig.7 and 8).

When the X directional force is applied to input A position and input B is constrained at the same time, the coupler is rotated in counter-clockwise as increasing to force, and the angular displacement is changed to almost linearly. And then, the angular displacement of coupler which is shifted to X direction, two input forces are applied to input A and input B, respectively, varies continuously as increasing the input current I_A. The proposed mechanisms with large-deflective elastic hinges are looking forward to using as a guide mechanism with multi-degree of freedom in field of optical system and others.

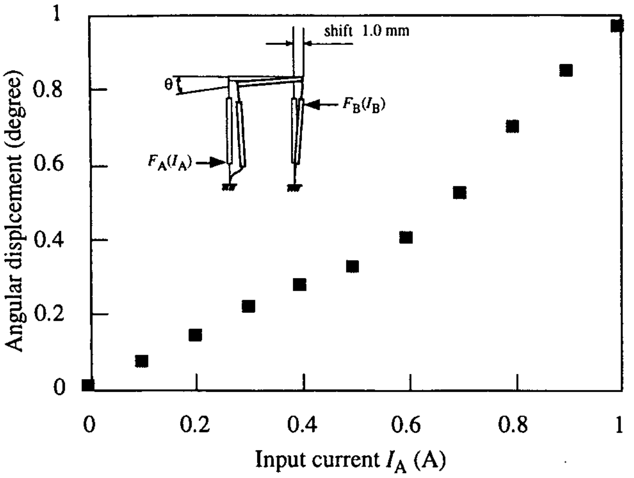


Fig.12 Angular displacement of 1 *mm* shifted mechanism to X direction

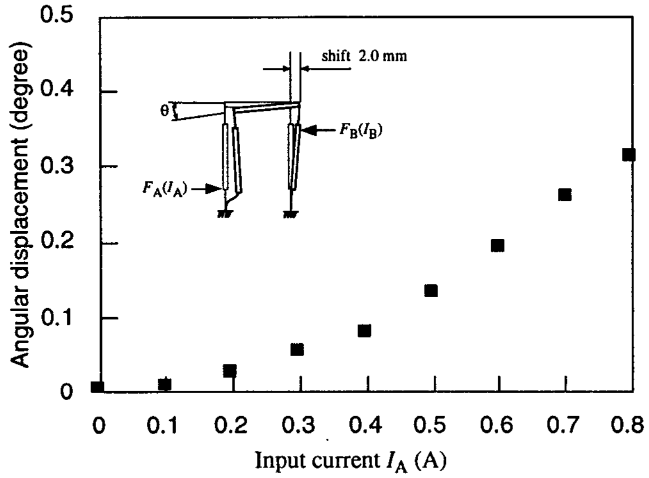


Fig.13 Angular displacement of 1 *mm* shifted mechanism to X direction

5. Acknowledgment

This work is supported by the Scientific Research Grant-in-Aid No.10555049 from the Japanese Ministry of Education, Science, Sports and Culture.

6. References

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