

DESIGN AND MODELING OF A PRECISION 6 DOF MEMS-BASED PARALLEL KINEMATIC TEM SAMPLE MANIPULATOR

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INTRODUCTION

The relatively large dimensions of 'conventional' transmission electron microscope (TEM) sample manipulators results in typical drawbacks such as thermal drift and compromised dynamics. Especially the requested stability of 0.1 nm/min requires a new manipulator concept. Miniaturizing creates the opportunity to fix the manipulator directly to the column guiding the electron beam, isolating external thermal and vibration noise. Secondly the manipulator can be made more stable by increasing the natural frequencies, decreases the thermal drift and decreases the thermal time constant of the manipulator. Potential solutions for miniaturizing can be found in Micro Electro Mechanical Systems (MEMS). Precision manipulation in MEMS seems sparse however. In this paper a design for a 6 Degree-of-freedom (DOF) precision MEMS-based manipulator will be discussed [1]. The elastic mechanism is modeled with the specific design considerations regarding kinematic constraint design and elastic energy storage. The fabrication will not be discussed in the paper.

In general the actuators used in MEMS exhibit low work density compared to the energy storage in elastic elements. Consequently the actuators in MEMS are relatively large and the elastic elements are generally long and slender. The typical relatively large deformations of elastic hinges in MEMS result in relatively large displacements and large rigid body rotations. Geometrically non-linear elasticity theory is a necessity for accurate analysis. A software package called Spacar considers elastic elements as multi-body-like finite elements, which considerably reduces the number of elements, which makes the analysis fast and effective [2].

HEXAPOD SYSTEM DESIGN

The MEMS-based stage is designed like a parallel manipulator with elastic mechanisms.

Elastic mechanisms are characterized by low hysteresis, zero backlash, no wear and high stiffness. Parallel manipulators in general have a large stiffness to mass ratio resulting in high natural frequencies and short settling times. In this case the parallel kinematics facilitates 6 actuators to be made in-plane of the wafer, using a mechanism to direct the motion out-of-plane to obtain 6 DOF (Figure 1). In this way one single technology can be used to manufacture six of the same type of electrostatic lateral comb-drive actuators as shown in Figure 2. Furthermore these can be combined with capacitive sensing by super positioning a high frequent signal on the actuation signal. The 6 comb-drives are arranged in 3 pairs, each pair controlling one of the 3 intermediate bodies in the two translational DOF of the wafer-plane. Within a pair, one comb-drive is connected to the other with a silicon leaf-spring, an intermediate body and a second leaf-spring. Three slanted leaf-springs connect the intermediate bodies to the end-effector. Because the total mechanism movement is achieved by purely elastic movements, and the mechanism is nearly exact kinematic constraint the positional repeatability will be high.

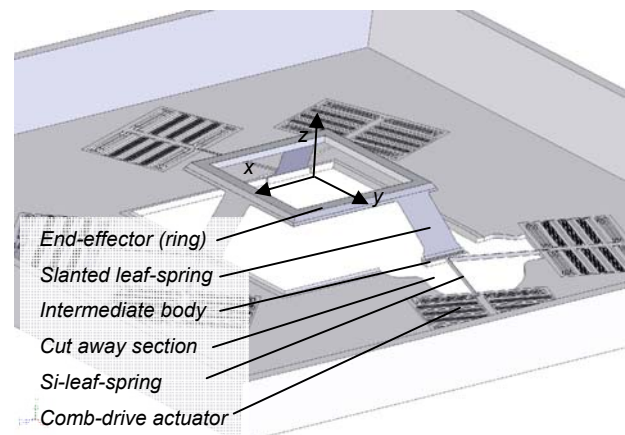


FIGURE 1: The MEMS-based hexapod design

INFLUENCE OF DRIVE STIFFNESS AND REINFORCEMENT ON LEAF-SPRING BEHAVIOR

The hexapod incorporates 6 comb-drives, each suspended by 4 folded flexures, which straight guide the actuator in actuation direction. Each folded flexure consists of 4 reinforced leaf-springs. The actuation direction of the suspension is compliant to minimize elastic energy storage. The other 5 Degrees-Of-Freedom are intended to be stiff in relation to the actuation direction. For a comb-drive a high longitudinal stiffness (Figure 2) of the leaf-springs of the folded flexures is important to minimize the possibility of side pull-in. The longitudinal stiffness however decreases with increasing deflection. Van Eijk [3] models the longitudinal stiffness of a leaf-spring while the leaf-spring is constrained at the actuated end in the actuation direction, $C_d = \infty$ in Figure 5. Legtenberg [4] leaves the translations free, $C_d = 0$. Van Eijk's model can be used for elastic guidances with a stiff drive train or a "stiff" control system at frequencies well below the control bandwidth. In MEMS however the drive train is usually in the order of the same compliance as the folded flexures, and currently, an ample control system lacks because of inadequate sensors. The longitudinal stiffness difference between the two models for relatively large deflection can be more than 2 orders of magnitude in favor of the constrained leaf-spring. The longitudinal stiffness as a function of the deflection in relation to the drive stiffness has been investigated for prismatic leaf-springs and for reinforced leaf-springs as shown in Figure 5.

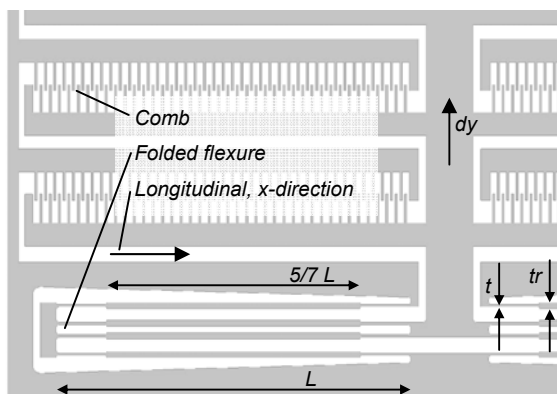


FIGURE 2. *Folded flexure in a comb-drive suspension.*

As folded flexures behave like individual leaf-springs, but then with half the displacement, leaf-springs are investigated. The leaf-spring dimensions (Figure 5) are taken L for length, w for width and t for thickness. A leaf-spring, prismatic or reinforced, is clamped at one end and constrained for rotation R_z at the other. An often implemented ratio of $5/7^{\text{th}}$ of the leaf-spring length is thickened. An extra one dimensional spring with spring constant C_d is attached to the free end of the leaf-spring. The displacement dy_2 is a constrained input. The corresponding y_2 and C_x are calculated by Spacar. C_x is related to the initial stiffness without deflection, C_{x0} .

C_{x0} of a reinforced leaf-spring is 3.5 times larger than C_{x0} of a prismatic leaf-spring with the reinforcement taken as a rigid body. This ratio increases during deflection in y -direction for $C_d/C_y > 100$. The reason is that by constraining the dy displacement the leaf-spring is forced in a "buckling like bending-mode" when loaded in x -direction. Reinforcement stiffens this "buckling like bending-mode" effectively, as the buckling length is shortened. Reinforcement for leaf-springs with $C_d/C_y < 100$ also increases the C_{x0} stiffness, but the initial increase of 3.5 becomes less and reaches down to 1.9 for $C_d = 0$ at $dy/t = 30$. Loading a leaf-spring in x -direction causes the leaf-spring to deflect mainly in y -direction. The movement in x -direction is a second order effect. Therefore bending stiffness C_y dominates C_x for large y/t . The C_y stiffness of a reinforced leaf-spring is 58% larger than the C_y of a prismatic leaf-spring.

In general exact kinematic constraint design is about creating exactly those degrees-of-freedom necessary in a design by, in the case of an elastic mechanism, introducing compliance in certain directions in relation to stiffness in others. For a straight guidance for a comb-drive actuator the ratio longitudinal to actuation direction is important. Reinforced leaf-springs increase this ratio over prismatic beams at zero deflection by a factor 2.2. However this ratio decreases fast when a leaf-spring is deflected for a $C_d = 0$ situation. Also the stiffness of the reinforcement has influence on the behavior of the leaf-spring and needs a more thorough investigation. While the longitudinal stiffness is of importance for the stability of a comb-drive, the out-of-plane stiffness is important for the stiffness of the end-effector of the hexapod. The longitudinal and out-of-plane stiffness as a function of the relative deflection in relation to

the reinforcement thickness has been investigated. For the out-of-plane stiffness the length of the leaf-spring in relation to the width is important [3]. In the case of the hexapod the L/w ratio is 11.7.

Figure 3 shows that the longitudinal / actuation stiffness ratio gain of a reinforced leaf-spring folded flexure with respect to a prismatic leaf-spring folded flexure decreases from a factor 2.2 to a factor 1.22 at $dy/t = 10$. It must be taken into account that reinforcement given a certain leaf-spring length will need a 26% larger drive-voltage or a 58% larger comb-drive to compensate the increased actuation stiffness, if the leaf-springs can not be made thinner, as is often the case in MEMS. A comb-drive can benefit from a reinforced leaf-spring folded flexure if this leaf-spring is pre-curved and is straightened during actuation. At the point of maximum pull-in force the reinforced leaf-spring is then straight with maximum longitudinal stiffness [5].

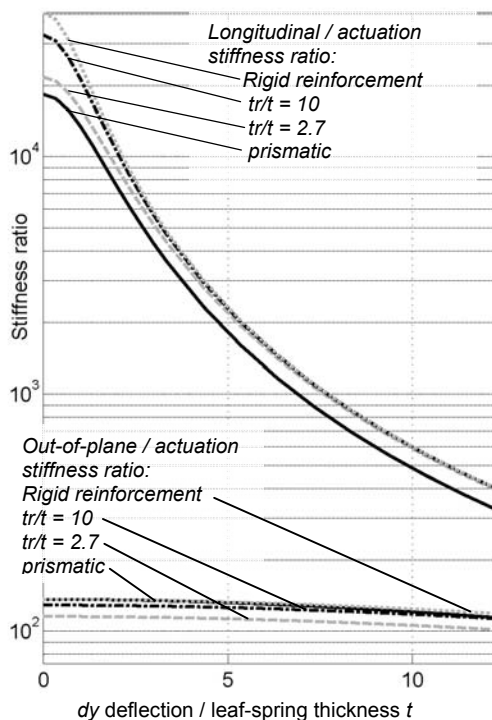


FIGURE 3. Stiffness ratios for a 4x folded flexure suspension as a function of the relative displacement. $L/w = 11.7$, $L = 406\mu\text{m}$, $w = 35\mu\text{m}$, $t = 3\mu\text{m}$.

The out-of-plane / actuation stiffness ratio does not benefit from reinforcement at zero

displacement. The bending stiffness for out-of-plane bending is increased the same amount as the bending stiffness increase in actuation direction. In fact reinforcing $tr/t = 2.7$ shows a decrease in the out-of-plane / actuation stiffness when compared to a prismatic leaf-spring. The actuation stiffness increases faster than the out-of-plane stiffness for increasing tr . The out-of-plane / actuation stiffness doesn't change much due to deflection, because L/w is relatively large [3]. A small L/w results in a relatively high out-of-plane bending stiffness to torsion stiffness ratio, which causes decrease of out-of-plane stiffness at deflection. Cd has no influence on the out-of-plane stiffness.

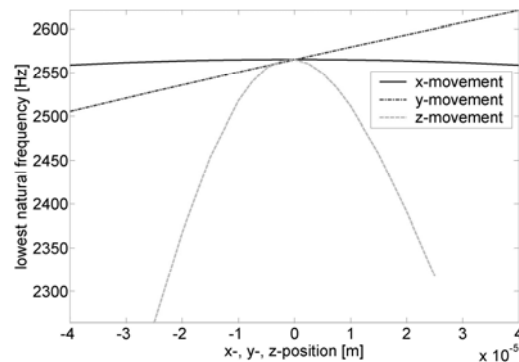


FIGURE 4. The lowest natural frequency as a function of the hexapod end-effector position in the individual x-, y- and z-direction.

THE HEXAPOD

The hexapod has been modeled by regarding the elastic energy storage in all the leaf-springs. Drive stiffness Cd for the six actuator suspensions is low and varies between 0.051 and 0.21. Most actuator energy is stored in the folded flexures. The lowest natural frequency in relation to a single movement in x, y or z-direction has been calculated as shown in Figure 4. The lowest natural frequency has a mode mainly in the out-of-plane direction. In the x- and y-directions the stiffness change is small compared to the stiffness change due to movement in z-direction. During an x- or y-displacement only the folded flexures and the silicon leaf-springs, as shown in Figure 1, are deflected. The z-stiffness of these flexures changes only slightly by deflection. The lowest natural frequency increases slightly when the end-effector is moved in the positive y-direction. The mass of the end-effector, which is the most important, is distributed more equal over the actuator folded flexure suspensions. The longitudinal stiffness of the slanted leaf-springs

does change significantly due to deflection when the end-effector is displaced in z-direction. Because the slanted leaf-springs are relatively thin ($0.7\mu\text{m}$) the decrease in natural frequency is relatively fast. Although an end-effector displacement of tens of micrometers shows a change in lowest natural frequency, the change in actuation force is neglectable.

CONCLUSION

The drive stiffness has great influence on the longitudinal stiffness of leaf-springs. For leaf-springs used for a relatively large drive stiffness, reinforcement is beneficial. Leaf-springs used at a relatively low drive stiffness, as is usually the case in MEMS, can also benefit from reinforcement, but only in certain cases. If the relative deflection is small there is a substantial gain in the ratio longitudinal to actuation stiffness. If the length to width ratio is small, the out-of-plane stiffness during deflection can be increased. The stiffness and so the natural frequency decrease due to deflection of elastic elements in a system can be drastic. In the proposed hexapod design the maximum lowest natural frequency shift due to a displacement of $25\mu\text{m}$ is only 10%.

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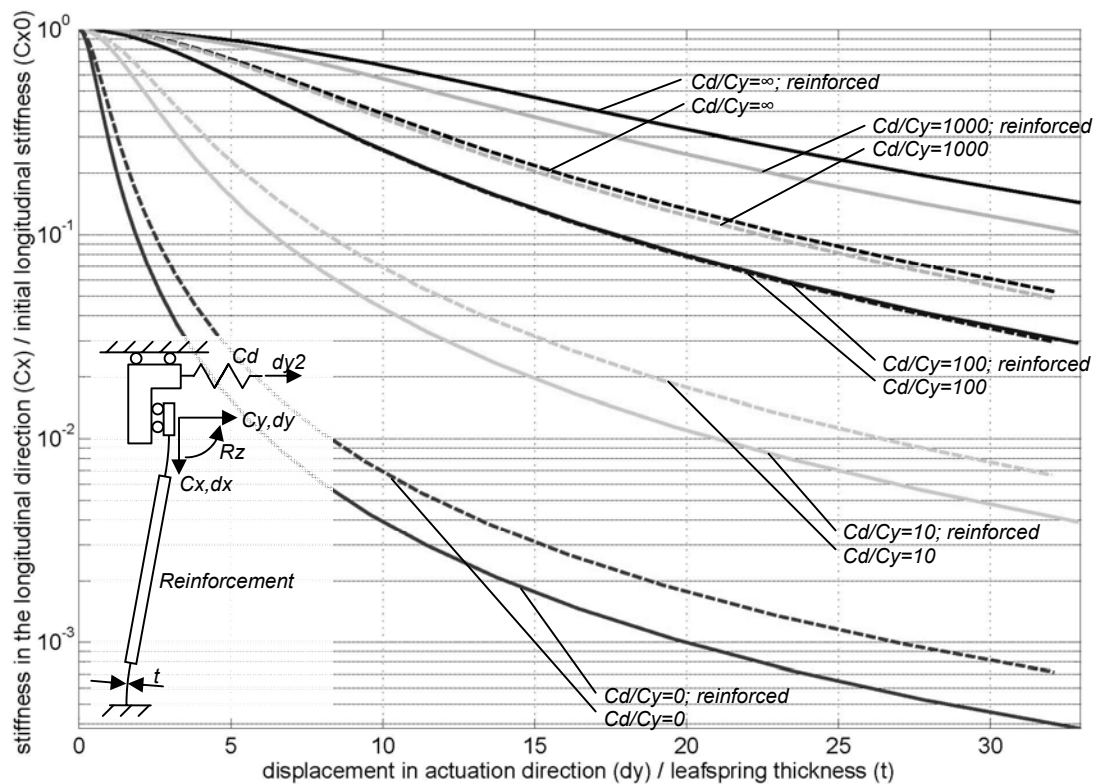


FIGURE 5. The longitudinal stiffness as a function of the deflection in relation to the drive stiffness for prismatic and for reinforced leaf-springs.