Design of Torsion Flexures for MEMS Bearings

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***Abstract*-The design of microelectromechanical systems (MEMS) devices that assemble or move outside the wafer plane requires the development of selectively compliant load carrying mechanisms. We present an analysis and design of a torsion bearing fabricated in the Iolanthe process (combination of SOI with polysilicon surface micromachining). Modeling results indicate that a viable and robust design is possible within the process and application constraints. We present a test structure array to verify the modeling results and further characterize the device under parameter variations.**

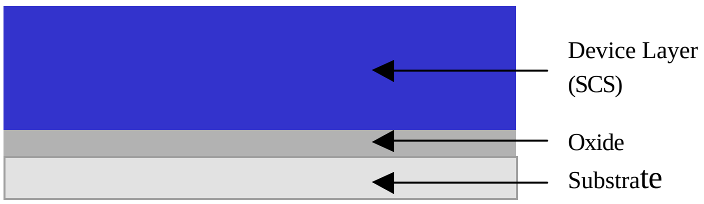
I. INTRODUCTION

The design of microelctromechanical systems (MEMS) devices that assemble or move outside the wafer plane is a relatively undeveloped field, the advancement of which requires reliable mechanisms capable of bearing and transmitting forces while maintaining a prescribed path of motion. That is, mechanisms that are selectively compliant are needed. Two bearings frequently encountered in this field are hinges with pins [5,8] of square cross section and torsion flexures [1,3,4]. These and other bearings have led to significant successes and advances in the field [3,8]. However, the hinge pins are too weak for some larger scale applications and exhibit considerable static friction [8], while a torsion flexure designed to operate under considerable transverse load is not yet in wide use. Thus, there is impetus to design a more robust bearing which is selectively compliant in the rotational direction. This project focuses on the design of a torsion bearing which is compatible with the requirements of the Iolanthe process and the Microrobot project, administered by the Berkeley Sensor and Actuator Sensor (BSAC), University of California, Berkeley. As such the bearing is designed for at least ±90° of rotation and rigidity under significant transverse force.

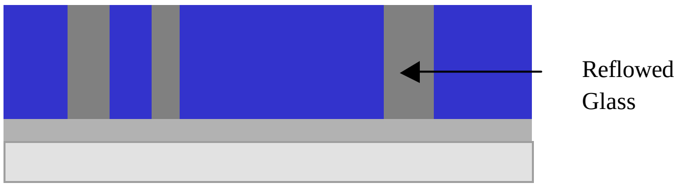
II. DESIGN OF THE FLEXURE

The fabrication process chosen for the device is the Iolanthe process, developed by BSAC. This process was originally developed for the Microrobot project [8], which requires a combination of large, strong structures built in single crystal silicon and small detailed structures built through polysilicon surface micromachining. The flexural bearing design described in this paper is specifically tailored for application in the Microrobot leg joints, and is thus fully compatible with the Iolanthe process. In brief, the Iolanthe process flow is as follows. 1) An SOI wafer is acquired, and the device layer patterned with DRIE. 2) Glass is deposited, baked out, and heated to flow into the spaces left from patterning. The surface is then polished flat with CMP. 3) A 2-poly process proceeds on top of the polished glass-device layer. There is an anchor etch between poly1 and poly2, which can also penetrate to the device later if desired. Nominally, the device layer is 50 µm thick, and each poly layer is 2 µm thick, with each oxide layer 1-2 µm. The minimum lines and spaces for the device patterning and polysilicon are, respectively: 3 and 1.5 µm, and 2 and 2 µm, with 2 µm mask alignment tolerance, and 5 and 5 µm line and space requirement for the anchor etch.

1. SOI Wafer



2. DRIE, Glass Deposition, CMP



3. 2-Poly Process on Top of Prepared Wafer

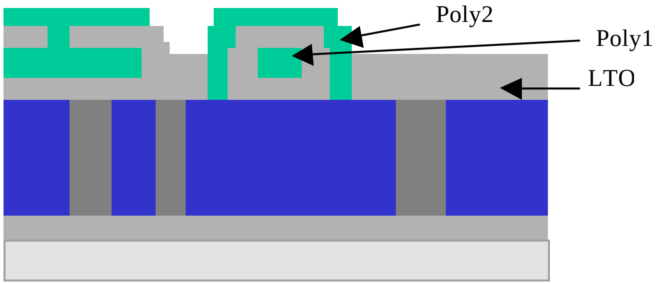


Fig. 1. Process flow cross sections [8].

As stated above, currently existing bearings do not meet the requirements of a heavily stressed joint. Specifically, the design presented must survive an angular deflection of ±90° and a transverse force of 0.022 N with little deflection from bending. The angular direction must be compliant enough that the bearing easily rotates under load.

Flexural bearings are attractive because they are integrated into one part, and therefore lack some of the inherent problems encountered with pin joints. These include the square pin getting lodged in the joint, excess frictional contact, and susceptibility to occlusion of the joint by foreign particles.

Most flexural members consist of long, thin ribbons of polysilicon [3]. This configuration is chosen to minimize the stress experienced by the flexure during the considerable deflections required of it, and to minimize process complexity. In fact, all open beam cross sections provide very similar resistance to deflection in torsion. Further, and in a similar geometric domain, resistance to bending is dominated by the thickness (in the direction of bending force) of the material, as opposed to it width. Thus, a ribbon is only suitably stiff in one direction. A segmented cross section is required to resist bending in two directions, while remaining relatively compliant in rotation. Therefore, if a ribbon type torsion flexure is used in a high load application the bearing will rotate outside the desired plane of motion, frustrating the design intention. No matter how a ribbon is oriented, if torqued through 90° of deflection, it results in a stiff spring in series with a compliant spring. This configuration results in a joint that is compliant when actuated, regardless of the way in which force is applied.

A "U" type channel is chosen for ease of fabrication, with each of the three walls as thin as possible to allow rotation. The geometry is clearly shown in Figure 5 below. Given the design requirement for rotation of ±90°, the strain developed in the flexure is necessarily quite significant. In fact, if the device is to be of any useful size (less than 1 mm in length) the strains must be roughly 1%. Due to the particular properties of silicon this is acceptable, while almost any other material would either shatter or plastically deform [2].

There are some manufacturing concerns that are linked to the design constraints as well. The flexure itself is composed of walls (3-6 µm) that are etched out of the 50 µm device layer, and the channel is capped with a 2 µm thick sheet of polysilicon. In order for the cap to contact the sidewalls, the walls must be wide enough to accommodate mask misalignment the line width requirements of the anchor etch. If one considers this at its worst, the minimum width of each wall must be at least 9 µm (5 µm for the anchor + 4 µm for the alignment tolerance). This is three times the minimum line width in the device layer etch, and is undesirable because of its unnecessary stiffness. Thus using a Poly1 layer as a mask over the substrate, the wall thickness can be reduced to 6 µm, which is much closer to the desired value. A layer schematic is shown as Figure 1. Another approach is to use the minimum line width for the walls, and have wide pylons periodically to ensure contact between the cap and the walls. This means that there will be parts of the cap not connected to the walls, and as such localized bending of the sidewalls is allowed, which may become detrimental at large strains. Nonetheless, this is presented as an alternative design to the one above, and is to be tested and compared with the continuous beam within the test structure arrays.

III. TEST STRUCTURES

An array of 96 unique test structures achieves the following goals: to demonstrate at least 90° rotation of the flexure; to demonstrate compatibility of the flexure design with the process constraints; to facilitate measurement of the flexure's spring constant under torsion and transverse loading, while varying the length, width, and wall design of the flexure. Two fundamental conformations of the test structure (Figure 2-a) were required to achieve these goals. In both designs, the load carrying components are patterned in the single crystal silicon device layer (50 µm), the first polysilicon layer (2 µm) and second polysilicon layer (2 µm) are used to "cap" the channel and for fine details, and the single crystal silicon substrate (350 µm) is kept or etched away as necessary.

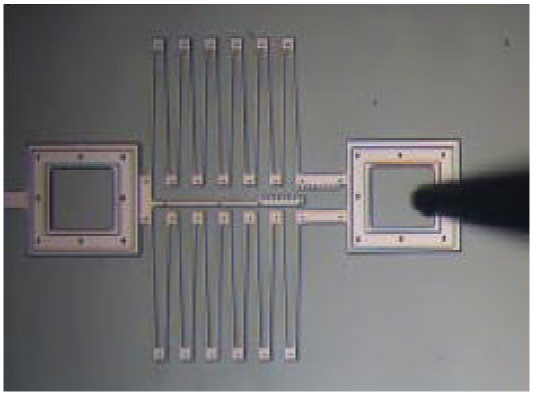


Fig. 2. (a) Spring calibrated to indicate applied force [6].

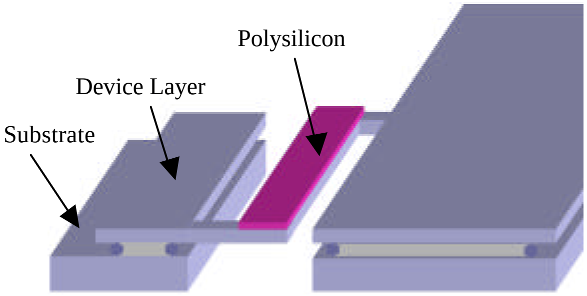


Fig. 2. (b) Concept sketch of test structure to determine torsional spring constant.

The first test structure design (Figure 2-a) consists of the flexure anchored to the substrate at one end and attached to a large mass (2 mm x 2 mm, device layer and substrate). Results form modeling indicate that this mass will deflect a nominal flexure (500um long, 200 µm wide) approximately 10° under the force of gravity. After fabrication this deflection is measured under a calibrated microscope, as a means of deriving the torsional spring constant of the each flexure. A probe lifts the mass upward, out of plane, demonstrating the range of motion of the flexure.

The second test structure design (Figure 2-b) consists of the flexure anchored to the substrate at one end and attached to a structure of known spring constant (58.6 N/m) at the other end [6]. The spring structure facilitates the application force by a probe tip, and the measurement of the resulting deflection along a "ruler-like" gage. As the probe tip applies force to the structure, it deflects, moving a finger of polysilicon along the ruler. This deflection and the known spring constant of the spring structure will yield the lateral spring constant of the flexure. These two types of test structures were each coupled to 48 unique flexure designs. These design consists of all possible combinations of four flexure lengths (200 µm, 500 µm, 1000 µm, 2000 µm), three flexure widths (50 µm, 200um, 500um), and four side wall designs (6um wide, 3 µm wide with pylons every 25 µm, 50 µm, 100 µm). These parameter variations were derived from the modeling results reported in the next section.

IV. RESULTS AND DISCUSSION

Two models of the proposed device are formulated. A parametric study of the flexure is run using Matlab, in which various dimensions of the flexure are varied and the results tabulated. A model of nominal dimensions is tested in Ansys to compare and verify the results obtained from the Matlab model, which makes extensive simplifications in modeling the device.

The purpose of the Matlab simulation is first to get an approximation of the stresses in the flexure, and how parameter variations affect them. The model employed uses equations taken from the standard solid mechanics literature for non-circular cross-sections in torsion [7]. Below is a synopsis of the analysis.

U= total section perimeter
t = section thickness
l = Flexure length
f =Angular rotation of flexure
G = Shear modulus
K = Section stiffness
T= Torque on flexure
t_{max} = Maximum Shear Stress

$$K = \frac{1}{3}Ut^3 ; \qquad Q = \frac{U^2 t^2}{3U + 1.8t}$$

$$f = \frac{Tl}{KG} ; \qquad t_{max} = \frac{T}{Q}$$

The assumption for this model is that there are no sharp corners in the section, and that the walls are relatively thin. Of course, the wall-cap interface produces a sharp corner, so the stresses produced are almost certainly higher than predicted by the model. Further, the model assumes that linear beam theory holds, and under large angular deflections this is lost as well. However, this model is deemed useful in judging feasibility. The performance of the flexure will ultimately be compared to the expected results after the test structures are fabricated and tested.

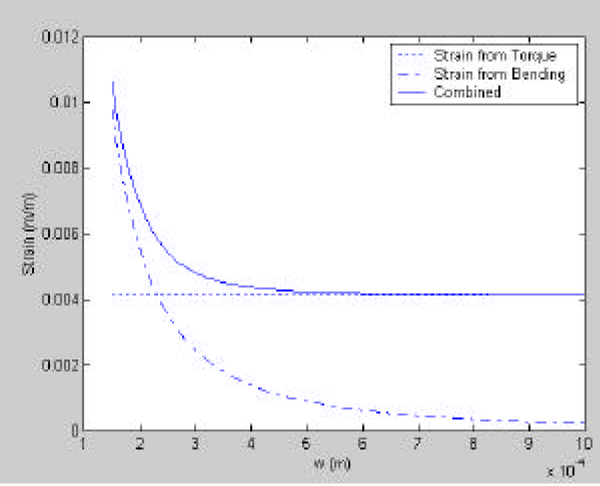


Fig 3. Strain with varying flexure widths.

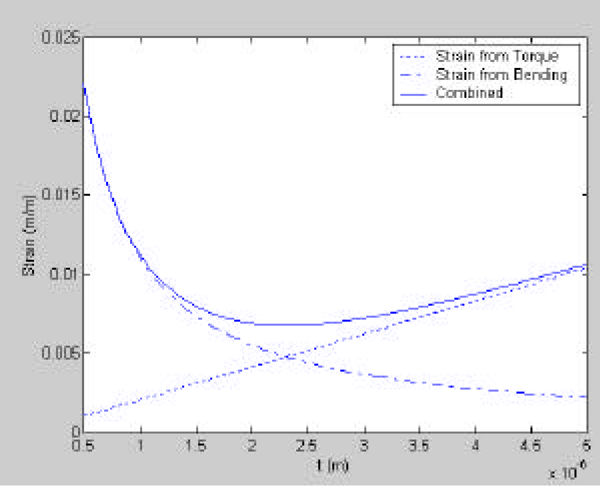


Fig. 4. Strains with varying flexure wall thickness.

Figures 3 and 4 above present the variation in strain experienced by the flexure with flexure width and wall thickness. It is important to note that the figures are for a bearing that is rotated 90° from equilibrium, using a 0.022 N force on some moment arm to actuate the joint. All simulations have a length of 500 µm, and flexure length variations are not modeled because the stress varies linearly with the length. A length of 500 µm is chosen for packaging reasons. However, as noted above, the length is varied in the test structures in case the stresses are higher than expected.

From the figures above, it is apparent that there is a preferred set of dimensions for the flexure. The width of the flexure need only be greater than 200 µm according to Figure 3 (width variation is conducted with cap thickness of 2 µm, and neglecting the effect of thick walls, since the cap does the most flexing). The next figure shows an elbow in the graph between 2 µm and 3 µm, the layer thickness of the Poly2 layer that makes up the flexible part of the bearing. Thus 200 µm width and 2 µm thickness are chosen as the nominal dimensions for the flexure, with the length 500 µm.

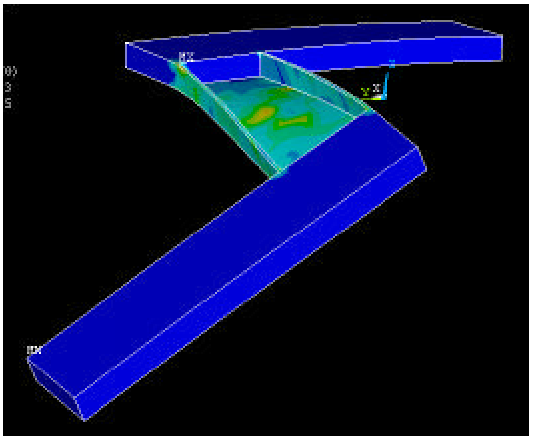


Fig. 5. Ansys deflected geometry.

An Ansys model of the nominal configuration under force is shown in Figure 5. As stated before, the purpose of this is to verify the validity of the Matlab model, and to identify potential failure modes and stress concentrations. The figure shows the approximate deflected shape, and also illustrates the stress distribution in the flexible part of the device, which lies in the range expected from the parametric model.

V. CONCLUSIONS AND IMPLICATIONS

The device presented above provides a viable design alternative to conventional MEMS bearing designs for applications that require rotational motion under transverse forces. The integrated "single-piece" design is robust- if the flexure survives several cycles it is likely that it will last indefinitely [2]. Since the design is stiff, it lends itself to use in foldable structures as well, with the same advantages over hinge pins.

The limitations of the analysis lie in the non-linear nature of the flexure, and some generalizations in the analysis, such as neglecting the effect of the width of the sidewalls as important- given the springs in series argument. Some improvements might include mixing the thick wall design with the pylon design, with pylons placed in areas of high stress- such as at the corners and in the center of the flexure. Further, the process might be refined to allow for a more fine line and space for the anchor step, which would allow for the theoretical 3um wall thickness for the sides allowable in the SOI processing steps.

Ultimately, the performance of the flexural bearing must be tested in a range of appropriate application. One might envision a direct comparison between one of these bearings and another joint by simultaneously loading each with the same actuator, until one fails. After a series of applications such as these, the relative performance of the torsion flexure and existing flexure designs may be assessed.

ACKNOWLEDGMENT

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REFERENCES

[1]O. Dagani, and Y. Nemirovsky, "Design Considerations of Rectangular Electrostatic Torsion Actuators Based on New Analytical Pull-In Expressions," Journal of Microelectromechanical Systems, vol. 11, pp. 20-26, February 2002.

[2]K. Petersen, "Silicon as a Mechanical Material," Proceedings of the IEEE, vol. 70, pp.420-457, May 1989

[3]H. Toshiyoshi and H. Fujita, "Electrostatic Micro Torsion Mirrors for an Optical Switch Matrix," Journal of Microelectromechanical Systmes, vol. 5, pp 231-237, December 1996.

[4]J. Yeh, H. Jiang, and N. Tien, "Integrated Polysilicon and DRIE Bulk Silicon Micromachining for an Electrostatic Torsional Actuator," Journal of Microelectromechanical Systems, vol. 8, pp 456-465, December 1999.

[5]R. Yeh, S. Hollar, and K.S.J. Pister, "A Single Mask, Large Force, and Large Displacement Electrostatic Inchworm Motor," Proceedings of the Fourteenth Annual International Conference on Microelectromechanical Systems (MEMS 2001), Interlaken, Switzerland, pp. 260-264, January 21-25, 2001.

[6]Yeh, R.; Pister, K.S.J. "Measurement of Static Friction in Mechanical Couplings of Articulated Microrobots," Proc. SPIE Conf. Micromachined Devices and Components, Austin, TX, October 1995, pp. 40-40.

[7] Norton, R. L., Machine Design, an Integrated Approach, Prentice Hall, Upper Saddle River, NJ 2000.

[8] Hollar, S; Flynn, A; Bergbreiter, S.; Pister, K.S.J. "Robot Leg Motion in a Planarized-SOI, 2 Poly Process," Hilton Head 2002 Workshop, Hilton Head Island, S.C., June 2-6, 2002.