MULTI-DEGREE OF FREEDOM DYNAMIC CHARACTERIZATION OF DEEP-ETCHED SILICON SUSPENSIONS

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ABSTRACT

Modern plasma etching techniques enable the fabrication of micromachined devices with extremely high height-to-width aspect ratio. This paper examines the mechanical performance of high aspect ratio silicon flexure suspensions. Flexures were etched with aspect ratios varying from 22:1 to 45:1 through the Bosch deep silicon etching process. Using non-contact measurement techniques and magnetic actuation, the ratio of the out-of-plane to in-plane suspension stiffness was measured as a function of in-plane displacement and flexure aspect ratio. This stiffness ratio was shown to be proportional to the square of the flexure aspect ratio, and diminished parabolically with in-plane displacement. A maximum out-of-plane to in-plane stiffness ratio of 700:1 was observed in a device with 45:1 flexures.

INTRODUCTION

The advent of semiconductor wafer bonding and deep reactive ion etching (DRIE) has enabled the fabrication of micromachined structures with extremely high height-to-width aspect-ratio. Such structures may be used to produce inertial sensors with large cross-axis mode separation or actuators with the ability to bear large out-of-plane loads[1],[2]. These properties come about because linear beam theory predicts that a rectangular cross-sectioned beam will have a ratio of out-of-plane to in-plane bending stiffness that is proportional to the aspect-ratio squared:

$$\frac{k_z}{k_x} = \frac{I_x}{I_z} = \left(\frac{h}{w}\right)^2 \quad (1)$$

where w and h are the beam width and height, respectively. Real-world devices may not match the predictions of Eq. (1), however. In reality, plasma etched beam profiles are never exactly rectangular. Additionally, linear beam theory does not capture such effects as the dependence of the in-plane and out-of-plane stiffness on in-plane deflections. Finally, because typical micromachined suspensions consist of folded or hairpin flexures connected by short linkages, losses in the folds and linkages may severely reduce the out-of-plane suspension stiffness.

To take a closer look at the mechanical performance of high aspect-ratio suspensions, 100 μm thick suspended test structures were etched into silicon-on-insulator (SOI) substrates using the Bosch DRIE process. A SEM image of a typical device is illustrated in Fig. 1. Each test structure consisted of a 1.4 mm $\times$ 1.4 mm $\times$ 0.1 mm mobile stage, flexurally suspended on four sides to allow two degree-of-freedom motion in the x-y plane while limiting out-of-plane (z-axis) motion. To allow the effect of varying aspect ratio to be investigated, flexures were designed with widths of 3 μm , 4 μm , and 5 μm . Mask undercut from the DRIE process varied from 0.5 μm to 1.0 μm depending on the device location on the wafer, resulting in fabricated aspect-ratios (AR) from 22:1 to 45:1.

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Figure 1. SEM images of completed test structure. Folded flexures provide x and y axis compliance, while limiting out-of-plane (z-axis) displacement. The center of the device is coated with NiFe to permit magnetic actuation.

THEORETICAL SUSPENSION PERFORMANCE

The x-y stages described in this paper utilize cascaded suspension elements to produce freedom of motion in two axes. The x and y axis stiffness of these suspensions were equal and will be denoted by k_x . The effective z-axis stiffness of the entire suspension, k_z , is the series combination of the z-axis stiffness of the x and y suspension elements. Thus, the maximum ratio of out-of-plane to in-plane stiffness is half that predicted by Eq. (1), or:

$$\frac{k_z}{k_x} = \frac{1}{2} \left(\frac{h}{w}\right)^2 \quad (2)$$

The difference between the stiffness ratio, k_z/k_x , achieved by a particular device and the value predicted by Eq. (2) can be considered to be a figure of merit. Additional losses in out-of-plane stiffness are contributed by bending in the linkages connecting the individual flexures, and by torsional bending of the flexures themselves. Careful suspension design is required to minimize these effects.

Even the best suspension design is subject to limitations from the cross-sectional geometry of the fabricated flexures. Although the side-wall profile obtained by DRIE is very close to vertical, an angular deviation of 0.1° results in nearly 0.2 μm of undercut at a depth of 100 μm . Beam width variations have a more pronounced effect on in-plane stiffness than on out-of-plane stiffness. For small $\Delta w/w_0$, the change in the stiffness ratio is approximately:

$$\frac{k_z}{k_x} = \frac{k_{z0}}{k_{x0}} \left(1 - 2 \frac{\Delta w}{w_0} \right) \quad (3)$$

where k_{x0} and k_{z0} are the in-plane and out-of-plane stiffnesses of a rectangular beam of width w_0 . Assuming that the beam profile is independent of width, the stiffness loss predicted by Eq. (3) will be more pronounced at narrower beam widths, since Δw will be a larger fraction of w_0 . Although variations in beam height can also result in stiffness losses, the use of SOI substrates guarantees less than 0.5% variation in the flexure height from wafer to wafer.

In addition to the stiffness ratio at zero displacement, it is important to examine the effects of in-plane displacements on flexure stiffnesses. In general, both k_x and k_z can be effected by in-plane displacements. Increased torsional stress from in-plane displacement has the effect of reducing k_z . As a consequence, the stiffness ratio is in turn reduced. For a rectangular cross-sectioned beam, this stiffness loss can be calculated from the inverse of the stiffness ratio:

$$\frac{k_x}{k_z} = \left(\frac{w}{h} \right)^2 + \alpha \left(\frac{x}{L} \right)^2 \quad (4)$$

where x is the in-plane displacement, L is the beam length, and α is a constant that is approximately equal to 0.14 for aspect ratios greater than 5:1. At displacements greater than 10% of the flexure length, tensile stiffening effects begin to increase the in-plane stiffness. However, the displacements in this study were sufficiently small to prevent tensile stiffening effects from occurring.

FABRICATION

Deposit 2 μm NiFe on 100 nm Ta adhesion layer.

DRIE flexures, stop on buried oxide.

DRIE backside cavities, stop on buried oxide. HF etch to release suspended test structure.

Figure 2. Simplified fabrication process flow.

A simplified fabrication process flow is shown in Fig. 2. Processing began on SOI wafers with a 100 μm thick device layer bonded to a 450 μm thick handle wafer via a 1 μm thick buried oxide layer. A 2 μm NiFe layer was sputtered onto the front surface and patterned with a lift-off technique. Next, 100 μm tall flexures were deep etched into the device layer using the Bosch silicon etching process. The silicon etch rate was observed to vary by approximately 3% from the edge to the center of each wafer. This etch-rate non-uniformity necessitated a slight overetch to clear the silicon at the wafer center. A slower etching recipe was used for this overetch in order to minimize undercutting of the silicon layer at the Si/SiO₂ interface. Although this approach improved device yield, it did

result in significant variations in the across-wafer sidewall profile of finished beams. Following the flexure etch, 450 μm deep cavities were etched into the back-side of the wafer. The buried oxide was used as an etch stop layer for the back-side cavity etch, leaving the stage supported by a thin oxide membrane. The final processing step is to remove this oxide with hydrofluoric acid, releasing the finished stage from the wafer.

A cross-sectional view of the deep etched flexures before removal of the oxide membrane is illustrated in Fig. 3. The cross-section was taken near the end of the first, third and fifth trench, and some silicon residue is visible at the base of these flexures. Although this residue was typically observed only near the flexure ends, the base of each fabricated flexure was either slightly narrower or slightly wider than the as-drawn linewidth, depending on the location of the beam on the wafer.

Figure 3. Cross-sectional view of deep-etched flexures. This cross-section taken at A-A reveals some silicon residue at the base of the 1st, 3rd, and 5th trenches.

TESTING APPARATUS

It is possible to directly measure flexure stiffness using load cells and displacement transducers[4]. However, this technique requires careful positioning of the applied load and suffers from low precision when applied to devices with a limited range of travel or very high compliance. For this reason, resonant frequency measurements were used to produce higher precision measurements of suspension stiffness. The relationship between the resonant frequencies and the stiffness ratio is simply:

$$\frac{k_z}{k_x} = \left(\frac{f_z}{f_x} \right)^2 \quad (5)$$

where f_x and f_z are the in-plane and out-of plane resonant frequencies, respectively. Natural frequency can be easily measured with an accuracy of better than 1%, resulting in a 2% accuracy in the calculated stiffness ratio. In addition, the dependence of the stiffness ratio on in-plane displacements can be studied by measuring the local natural frequencies at a fixed displacement. Calculation of the spring constants themselves, k_x and k_z , requires knowledge of the effective mass of the stage, but this value can be accurately esti-

mated based on the SOI wafer thickness and the stage geometry.

Each device was tested by exciting and measuring the in-plane and out-of-plane vibration modes at selected points over a $\pm 30 \mu\text{m}$ range of in-plane displacements. As illustrated in Fig. 4, out-of-plane vibration was induced using a vibration exciter (shaker), and external magnets were used for in-plane displacement. These external magnets consisted of a wire coil for ac stimulation and a permanent magnet assembly for dc inputs. The magnets generated a maximum force of approximately $800 \mu\text{N}$. Static in-plane displacements were accomplished by moving the permanent magnet assembly on a micrometer driven stage. A frame-grabber and CCD camera were used to measure the resulting in-plane displacement of the stage with $0.25 \mu\text{m}$ accuracy. A microscope mounted laser-doppler vibrometer was used to measure the out-of-plane natural frequency, while the in-plane natural frequency was measured from the blur envelope of the video image[5].

Figure 4. Excitation and measurement setup. The sample is mounted on a z-axis vibration exciter. NdFeB magnets may be moved relative to sample to vary the dc magnetic field, while ac fields are applied with a coil.

The measurement system allowed accurate measurement of both f_x and f_z at a given displacement. However, nonlinearities in the magnetic force were the dominant contributor to the measured variation in f_x as a function of displacement. Denoting the NiFe saturation magnetization as M_s , the magnetic field, $H(x)$, produces an in-plane force on the NiFe layer that is proportional to the magnetic field gradient:

$$F(x_0) = M_s S \left[H\left(x_0 - \frac{L}{2}\right) - H\left(x_0 + \frac{L}{2}\right) \right] \approx M_s S L \frac{dH}{dx} \bigg|_{x_0} \quad (6)$$

where x_0 is the distance from the edge of the permanent magnet to the center of the test sample, and L and S are the length and cross-sectional area of the NiFe film. However, a magnetic spring effect

is produced by the derivative of Eq. (6):

$$\frac{dF}{dx}(x_0) = M_s S \left[\frac{dH}{dx} \bigg|_{x_0 - \frac{L}{2}} - \frac{dH}{dx} \bigg|_{x_0 + \frac{L}{2}} \right] \approx M_s S L \frac{d^2 H}{dx^2} \bigg|_{x_0} \quad (7)$$

The permanent magnet assembly was designed to produce a large x-axis magnetic field gradient, maximizing the difference in the field at the two edges of the NiFe sheet. An unfortunate consequence of the design is that it also resulted in a large second derivative of the magnetic field. Although it was possible to remove the permanent magnets to measure f_x and f_z at zero displacement, introducing the magnets resulted in a measurable shift in f_x . Since in-plane displacements were induced by moving the magnet assembly, the magnetic shift in f_x was displacement dependent. The measured shift was roughly 5% over the full $30 \mu\text{m}$ range of travel, in agreement with the value predicted using an electromagnetic finite element analysis (FEA). In contrast, mechanical FEA predicted a frequency shift of less than 1%, suggesting that the magnetic effect dominated over any mechanical stiffening.

This measurement limitation was not observed in the z-axis. Because the NiFe is a thin sheet, the magnets did not produce appreciable out-of-plane force or spring stiffness. In addition, the out-of-plane mechanical suspension stiffness is more than two orders of magnitude greater than the in-plane stiffness, rendering the out-of-plane measurements relatively insensitive to magnetic field effects.

EXPERIMENTAL RESULTS

The measured stiffness ratio (k_z/k_x) of seven devices is displayed in Fig. 5 along with the ratio calculated by FEA. Test samples with both $3 \mu\text{m}$ and $4 \mu\text{m}$ as-drawn flexure widths were tested, with the $3 \mu\text{m}$ beams having a higher stiffness ratio than the $4 \mu\text{m}$ beams. However, varying beam profile caused the stiffness ratio to vary from device to device. Devices with $5 \mu\text{m}$ flexures were too stiff to be displaced by more than $10 \mu\text{m}$ and do not appear on the plot. The FEA model had the same geometry as the test samples and a $3 \mu\text{m}$ flexure width. As described in the preceding section, it was not possible to measure the variations in the in-plane stiffness versus displacement, so the nominal stiffness at zero displacement, k_{x0} , was used in preparing this plot.

In general, the out-of-plane stiffness diminished parabolically with in-plane displacement, as described by Eq. (4). The data in Fig. 5 correspond to the inverse of Eq. (4). When plotted in this form, the stiffness loss increases with aspect ratio, however, the parabolic loss term, α , is largely independent of aspect ratio. The constant is theoretically equal to 0.14 for a simple beam, but tested suspensions consisted of cascaded beams of two lengths, each of which was subjected to a different deflection to achieve a particular stage displacement. Correcting for these factors, the predicted parabolic loss term for these suspension designs was 0.09. The $3 \mu\text{m}$ flexures had an average measured parabolic loss constant of $\alpha = 0.09$, while the $4 \mu\text{m}$ flexures had a slightly lower measured loss constant of $\alpha = 0.07$.

While the devices with $3 \mu\text{m}$ flexures have a higher stiffness ratio than those with $4 \mu\text{m}$ flexures, there is a great deal of variation from device to device. This variation is due to the edge-to-center beam profile variations. A convenient way of comparing the vari-

Figure 5. Measured stiffness ratio versus in-plane displacement. The as-drawn beam width of each sample is indicated on the legend.

ous devices is to plot the nominal stiffness ratio of each as a function of the flexure aspect-ratio. Direct measurement of the beam profile would require destructive cross-sectional analysis, so a simpler approach was adopted. Assuming a rectangular beam profile, the nominal beam width was estimated from the ratio of the measured natural frequency to that predicted by the FEA model:

$$w = w_{\text{FEA}} \left(\frac{f_x}{f_{x, \text{FEA}}} \right)^{2/3} \quad (8)$$

Despite the fact that the fabricated beam profiles deviated slightly from rectangular, the estimates calculated from Eq. (8) proved reasonably accurate. Two cross-sections measured using a SEM are shown along with estimated widths in Fig. 6

The nominal stiffness ratio at zero displacement is plotted in Fig. 7 as a function of the aspect ratio computed from Eq. (8). The theoretical relationship between the stiffness ratio and aspect ratio from Eq. (2) is plotted as a solid line, while a linear fit to the data is plotted with a dashed line. The slope of this fit is 0.32, or 36% lower than the theoretical maximum stiffness ratio. The fact that the FEA data also fits close to this curve suggests that the suspension design is the main source of this loss, rather than any nonideality in the flexures themselves.

Figure 6. Measured beam profiles and widths estimated by Eq. (8)

Figure 7. Nominal stiffness ratio (k_z/k_0) vs. aspect ratio squared. A linear fit to the data is indicated with a dashed line, while the solid line indicates the theoretical limit from Eq. (2)

CONCLUSIONS

This paper presented a study of the mechanical performance of deep etched silicon flexure suspensions. While the ratio of the in-plane to out-of-plane flexure stiffness was shown to be proportional to the square of the aspect ratio, this ratio was 36% lower than predicted for a single, rectangular cross-sectioned beam. The source of this additional stiffness loss is bending in the linkages connecting the individual flexure elements, as well as torsional bending introduced at these linkage points. Variations in k_x and k_z as a function of in-plane displacements were also measured. Variations in k_x were measured to be less than 5% at a 30 μm displacement, and the measured variation was magnetically, rather than mechanically, induced. Variations in k_z were measured with accuracy of less than 1%, and were shown to have a parabolic dependence on in-plane displacement.

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