Design and Characterization of a Fast Piezoelectric Micropositioning Stage

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Abstract

This paper details the design, simulation and characterization of a high resonance piezo actuated X-Y flexure stage. The goal of this project was to develop a low cost X-Y stage possessing both fast response, and large displacement micropositioning capabilities for use in fiber laser optics positioning. A flexure stage topology was proposed, and a discrete elastic rod model was developed in MATLAB. MATLAB simulation iteration was used in conjunction with FEM analysis to characterize and modify flexure geometry (thickness and spacing) to improve bandwidth, displacement and axis cross coupling characteristics. The geometry was tuned to yield a resonant frequency of 3.4 kHz, a displacement of 65μm (under 150 V), and a cross axis coupling error of less than 2% at max displacement.

Introduction

The DE-STAR project at UCSB Physics is researching directed energy as a propulsion method for interstellar flight. One significant aspect of this project is the design of a large phased fiber laser array [1]. Each laser in this larger array will need to be individually controlled at very high frequencies with small error to correct for atmospheric disturbances. For this application, a piezoelectrically actuated flexure stage with high bandwidth, high displacement and low cross axis coupling will be designed.

Piezoelectrically actuated flexure stages have a wide variety of applications ranging from metrology and scanning microscopy, to precision machining and nanofabrication systems. The capability of piezoelectrics for high bandwidth low error actuation combined with repeatability and low cost of flexure stages make them an attractive micropositioning option for this wide range of precision applications.

Flexure stages have been studied in detail for years by a number of research groups. Methods for motion amplification using levers, or skewed mechanisms [2] [3], have been analyzed and there are many commercially available flexure stages designed for scanning microscopy [4].

Most of the current flexure stages are designed to have either high bandwidth, or relatively large displacements, but not both. For example, the flexure stage in [2] has a high resonant frequency of 2.6 kHz, but a maximum displacement of only 25 um even with displacement amplification. Without some mechanism for displacement amplification, extremely high resonant frequencies are achievable. For example, the AFM stage developed in [5] has a first resonant mode of over 20kHz. In this case, multiple stack actuators are placed in series to increase deflection, but a scan range of only around 10 μm was obtained.

A simple low cost stage that possesses both high frequency properties, and relatively large displacement capabilities is needed for the DE-STAR laser array. Here a stage will be designed to meet target specifications of at least 2.5 kHz (resonant frequency), and a scan range of at least 50 μm (x and y). Additionally, the system will be designed to fit within the dimensional limitations an arrayed laser optics tube (7.5 cm outer diameter), and have a cross axial drift of less than ±1% over the entire scan range.

Simulation Methods and Design Process

The decoupled flexure stage design analyzed in [2] was used as a starting point for this project. The design topology was modified significantly to better suit the dimensional requirements. Figure 1 shows a CAD model of the final design with the main components labeled. The Displacement amplification mechanism has an amplification ratio of roughly 2.5 (under no load), and the anti-rotation flexures are necessary to prevent unwanted twisting motions. The selected stacked piezo actuator [6] has an overall length of 20 mm, and a displacement range of 31 μm (-30 to 150 V).

Two simulation methods (Discrete Elastic Rod and Finite element) were used during the design of this flexure stage. Discrete Elastic Rod (DER) modeling is a numerical modeling technique that breaks up mechanical systems into a series of nodes, and modeling each node as a mass spring oscillator with a stretching, bending, and twisting stiffness. Force balance equations are then written and used to calculate the jacobian so that the entire system can be solved iteratively using newton's method. This modeling technique is described in more detail in [7].

A DER model of a simplified geometry was developed in matlab. Figure 2A shows the nodal positions in a matlab plot, and Figure 2B shows the mechanical system that the MATLAB model is representing. This model was used to iterate through a range of geometries, varying both flexure thickness and spacing. Flexure thickness was varied from 0.3 mm to

0.9 mm, and flexure spacing was varied from 6 mm to 12mm. The out of plane thickness was assumed to be constant at 5mm for all cases.

Three outputs (resonant frequency, cross axis error, and displacement) were used to characterize each unique geometry using impulse and ramp force inputs applied in the Y direction. The time domain impulse response was simulated for 20 ms using a timestep of 0.002 ms. An FFT was then performed to find the frequency spectrum and extract the first resonance for each geometry. The steady state displacements in both the actuated and free directions were gathered from the time domain response under a ramped force input and used to find the cross axis error and deflection. It's important to note that the force inputs here are not representative of the stacked piezo, and were simply used for design characterization purposes.

Using the results from the design iteration study, a more complex CAD model (Figure 1) was created for FEM analysis. Several iterations were performed on the amplification mechanism to avoid excessive strain and improve force transfer. The Piezoelectric stack actuator was modeled using an equivalent thermal expansion coefficient found using equation 1 where deflection is taken to be the stack actuator nominal displacement (31 um [6]) and ΔT is assumed to be 180 C (the same range as the voltage input). This calculation yields an expansion coefficient of $8.6*10^{-6}$.

$$(\alpha\ \Delta T)\ L\ =\ deflection \quad (1)$$

The nominal actuator stiffness was used to back out the elastic modulus for modeling purposes. Based on the cross sectional area (A) of 5x5mm, length (L) of 20mm and an actuator spring constant (K) of 41N/um, the equivalent elastic modulus was found to be 32.8 GPa (equation 2).

$$K = \frac{E_{eq}*A}{L} \qquad (2)$$

These two equivalent material properties along with density were used in the Solidworks FEA simulations for the actuator. The stage was assumed to be made from aluminum 6061, and the thermal expansion coefficient was manually set to 0 so that the actuation could be modeled independently in a static temperature study.

In addition to static temperature studies, frequency simulations were performed to find the first 2 (X and Y) resonances.

Results

Figure 3 shows the time domain responses for all geometric combinations under both the impulse and ramp force input. Due to the large number of data sets, the plot lines are not labeled (this will be covered later). While it is difficult to extract quantitative data from their plot, it is vital for ensuring that the time domain responses are in fact physical, that the model is appropriately constrained, and that the simulation parameters are correct. The ramp input response reaches a steady state value by the end of the simulation time, and the impulse responses show enough cycles for frequency extraction via FFT.

Figures 4A-B show a the DER simulated resonant frequency plotted against the flexure thickness and flexure spacing. Figure 4A is a 3D plot for qualitative evaluation of input cross coupling, and Figure 4B shows a 2D plot where the effects of the input variables can be observed individually. It is clear from this data that increasing the flexure thickness, and decreasing the flexure spacing both increase the resonant frequency. Both of these trends are expected, and can be explained by increasing stiffness. For a beam, the bending stiffness and stretching stiffness are both related to thickness (increased thickness means increased stiffness). In addition to this, shortening a flexure will also increase the stiffness. Assuming that the stage behaves like a simple harmonic oscillator, the resonant frequency proportional to the square root of the equivalent spring constant, and inversely proportional to the square root of the mass.

3 FEM frequency simulations were performed on the simplified design using different geometry to validate the DER model results. In all cases, the FEM predicted frequency matched the DER predicted frequency within 5%.

Figures 5A-B show the simulated cross axis error plotted against the flexure thickness and flexure spacing. Figure 5A presents a 3D plot for qualitative evaluation of input cross coupling, while Figure 5B shows a 2D plot where the effects of the input variables can be observed individually. It is clear from this plot that at large flexure spacings, the flexure thickness plays an important role in the cross axis error, but at small spacings, the effect is subdued. This is likely because if the flexures become too loose (low stiffness) then, then they can no longer guide the motion of the stage properly. Since flexural stiffness is related to both thickness and length, at small flexure spacing (and small outer flexure length) the thickness effect is reduced. Based purely on Figures 4A-B and 5A-B the clear choice would be large flexure thickness and small flexure spacing to maximize resonant frequency, and minimize cross axis coupling. The stacked piezo actuator however, is not connected directly to this stage, but rather connected through the amplification mechanism. This means that the input for this simplified stage cannot be treated as a displacement input and the stiffness needs to be taken into consideration.

Figures 6A-B show the steady state deflection plotted against the same geometric inputs. This result was simulated using the same ramp input for all geometries (ramp to 25 N at 1 ms, and hold for the duration of the simulation). Figure 6A is again a 3D plot where the cross coupling of inputs can be observed, and Figure 6B shows the same data on a 2D plot. These plots show clear trends of decreasing deflection with increasing flexure thickness and decreasing spacing. This trend can be described again by the above discussion on how these geometry inputs affect flexure stiffness. Since each iteration was performed using the same force input, and allowed to reach steady state, the deflection plot is directly proportional to effective stage stiffness. The stiffness of the maximum geometric combination was found to be 0.086 N/um, while the minimum was found to be 0.0189 N/um.

If this simplified geometry was driven by a stacked piezo directly, then the stage stiffness would play a very small role in the overall deflection. Equation 3 gives the relationship between the nominal actuator deflection (ΔL_o), and the actual actuator deflection (ΔL) as a function of the piezo actuator stiffness and the stage stiffness. Based on the MATLAB DER results, using the maximum stage equivalent stiffness (0.086 N/um) and the nominal actuator stiffness (41 N/um), the stiffness coefficient is 0.99. Since this coefficient is so close to 1, the stage stiffness will play a very small role in the actuator deflection.

$$\Delta L = \frac{k_{piezo}}{k_{stage}+k_{piezo}} \Delta L_o \quad (3)$$

Although the stage stiffness does not affect the actuators motion to any significant degree, the final design incorporates a motion amplification mechanism to improve the scan range. This mechanism is strongly affected by the stage stiffness, and will not properly transfer motion with the full amplification factor (2.5) if the stage is too stiff.

Several design iterations were modeled using FEM in SOLIDWORKS. The plots in Figures 4 - 6 served as design guidelines for further iteration resulting in a final selection of 0.6 mm for the flexure thickness, and 12mm for the flexure spacing.

Final static actuation simulation results on the are shown in Figure 7. This plot shows only the deflection in the actuated direction. From this simulation, the displacement (at 150V) was found to be 65 um, and the cross axis error was found to be around 1.5%. As predicted by equation 3, the actuator end displacement matches the datasheet no load deflection (25.8 um for 150V) well.

Figure 8 shows the first resonant mode (3.4 kHz) as predicted by a frequency study. It is important to note the asymmetry in this resonance. This is introduced by the inclusion of the actuator/amplification mechanism, and anti rotation flexures on only 2 sides of the stage. Future designs may add additional flexures on the 2 free sides to help balance the stiffness and produce more symmetric mode shapes. This would however, decrease the range slightly due to overall stiffening. The system was additionally simulated without the actuators in place, resulting in a more symmetric mode shape, and a reduced resonant frequency (2.5 kHz).

Conclusion

This document presents a high resonance (3.4 kHz) large displacement (65 um) flexure stage. Matlab DER modeling was used in conjunction with SOLIDWORKS FEA modeling to tune geometry to better tailor this design towards fast positioning of a laser fiber in the confined space of a 7.5 cm diameter optics tube. The off the shelf piezo stack actuators selected for this design require relatively low voltages (150 V), and the design allows for high volume low cost fabrication due to its 2D geometry, and common materials (aluminum 6061). The design approach presented here may easily be applied to other piezoelectric flexure stages, as the rapid DER simulations allow for fast design iteration to match application requirements.

References

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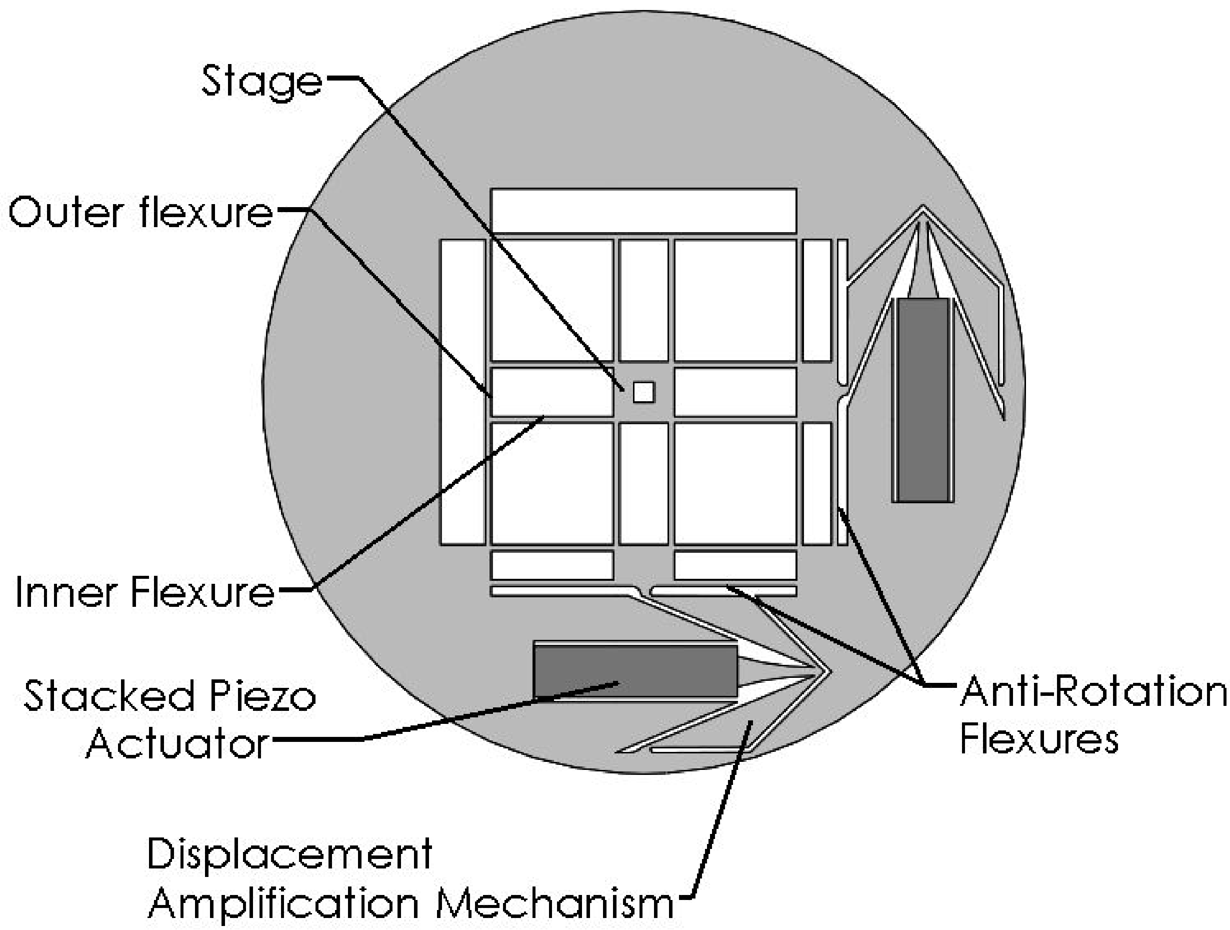

Figure 1. Final CAD design schematic including all major components

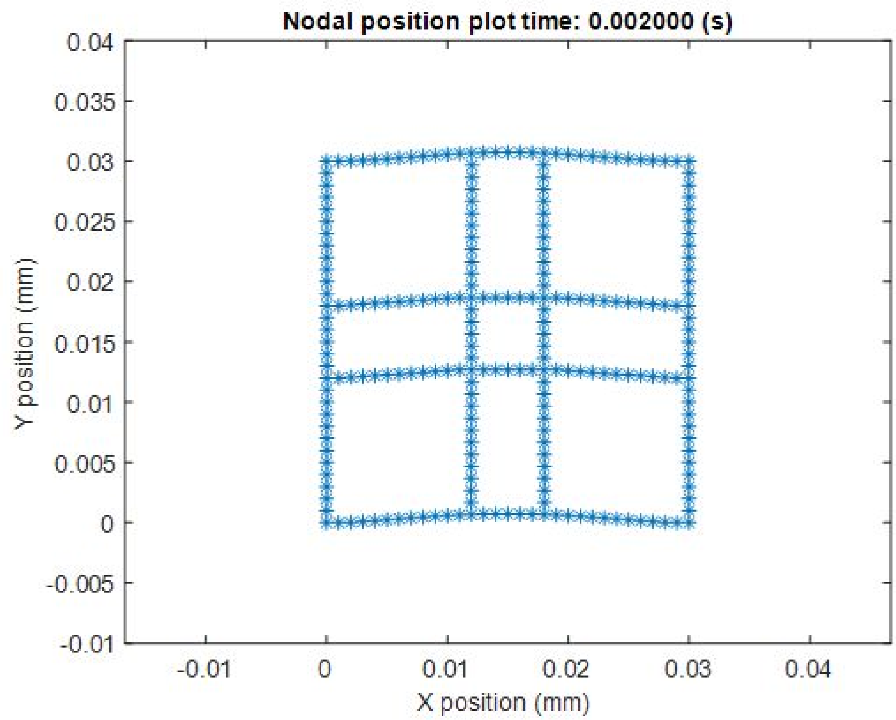

Figure 2A. Nodal position plot for DER simulation (deformed state)

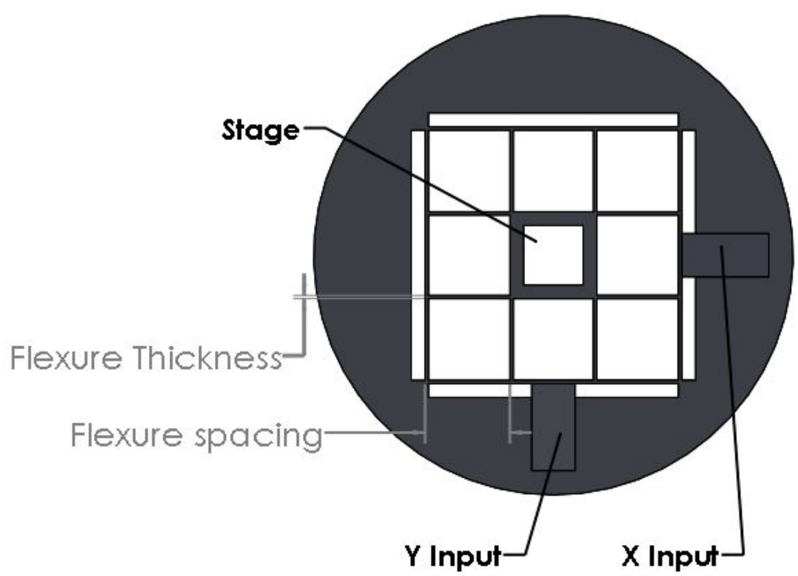

Figure 2B. Simplified topology for DER simulation

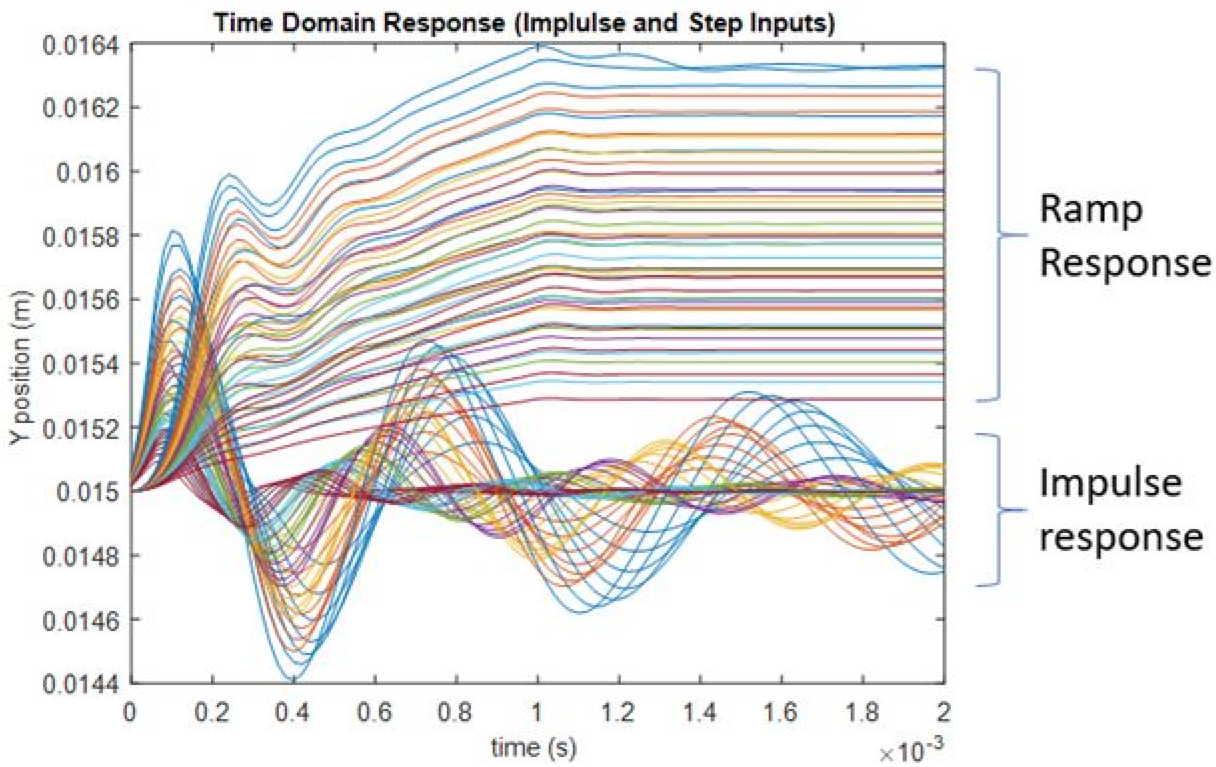

Figure 3. Time domain responses for iterated geometries (Y actuation only)

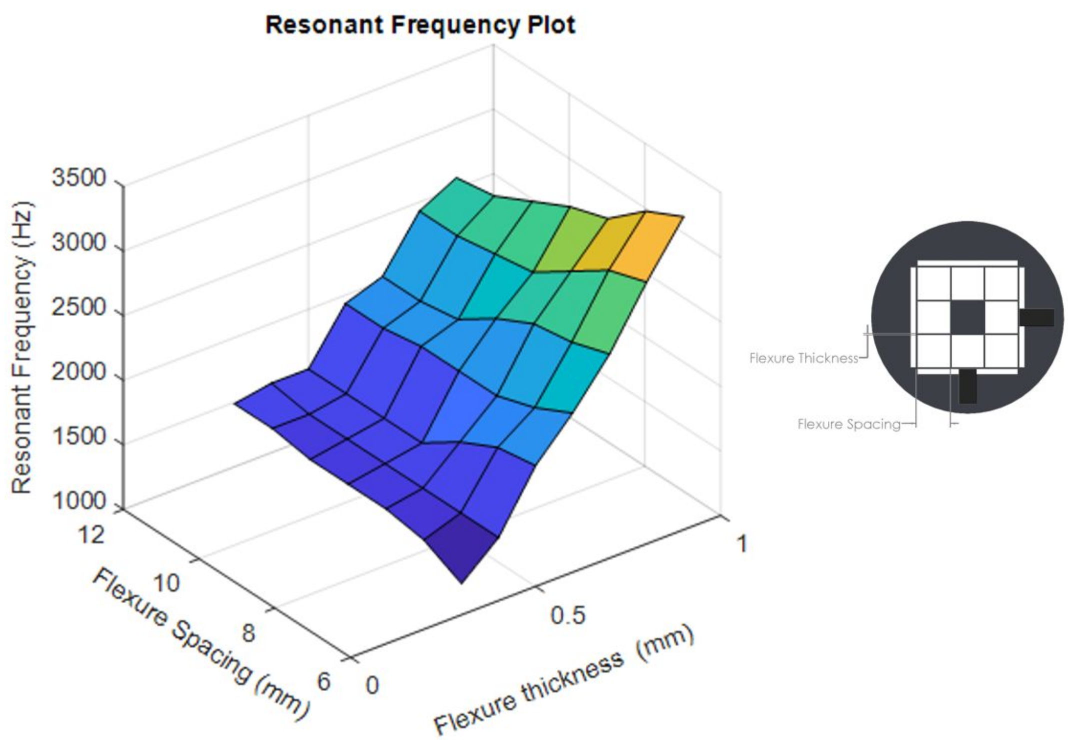

Figure 4A. 3D DER simulated resonant frequency plot

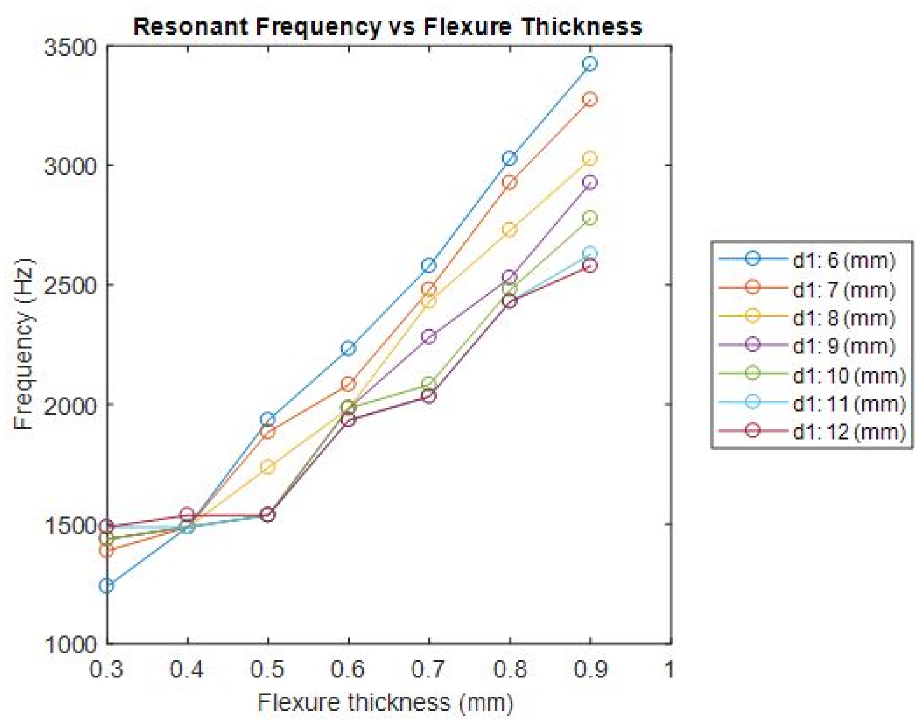

Figure 4B. 2D DER simulated resonant frequency plot

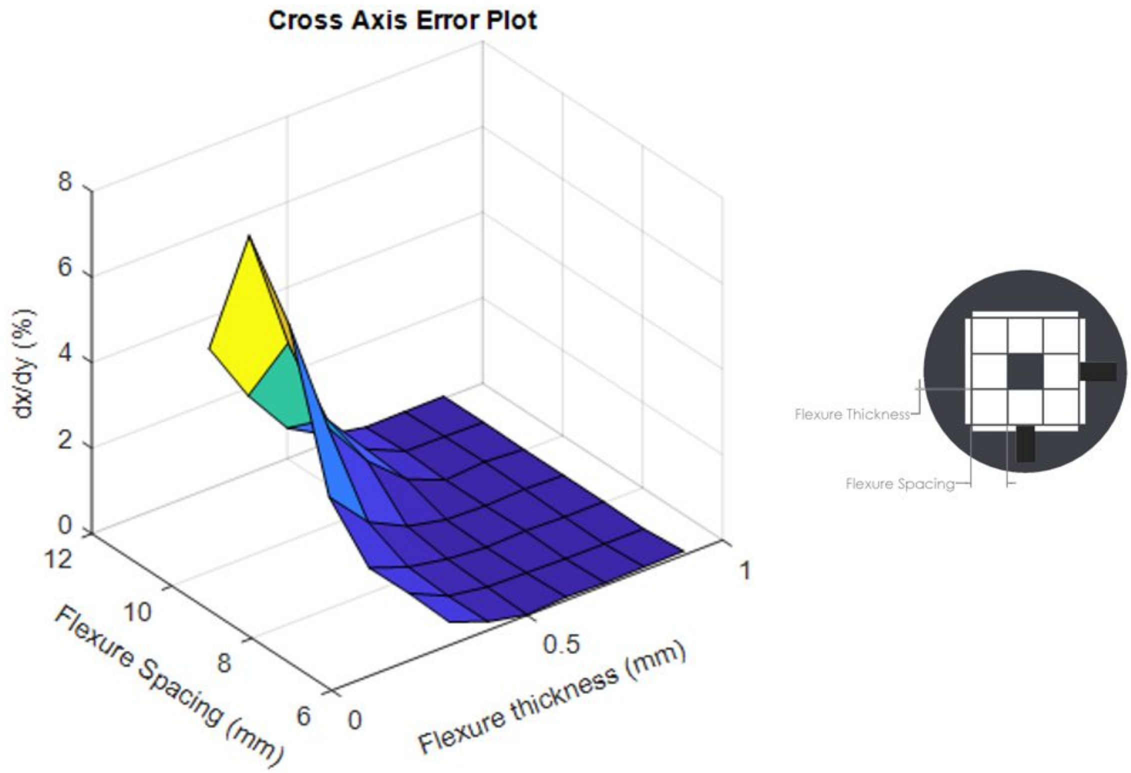

Figure 5A. 3D simulated cross axis error plot

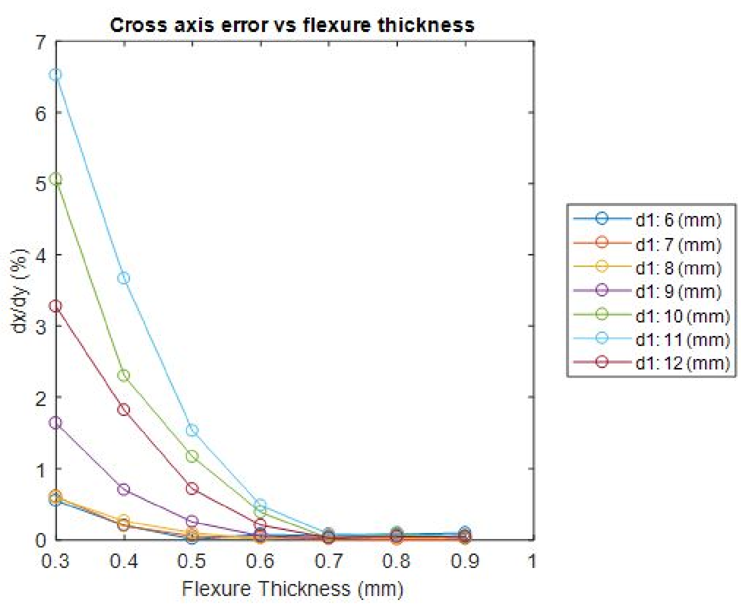

Figure 5B. 2D DER simulated cross axis error plot

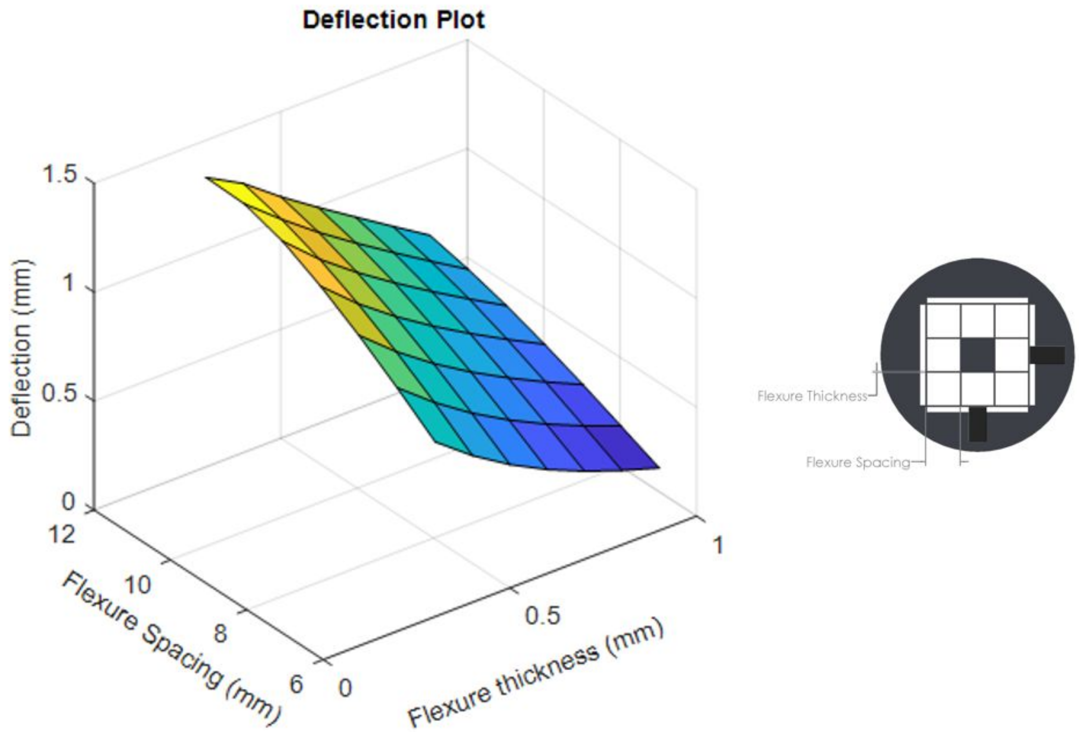

Figure 6A. 3D DER simulated deflection plot

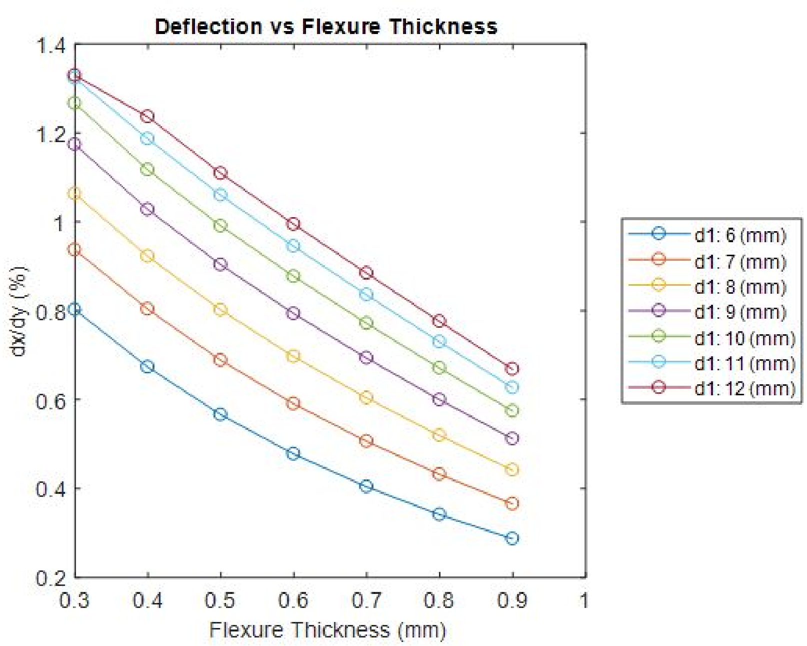

Figure 6B. 2D DER simulated deflection plot

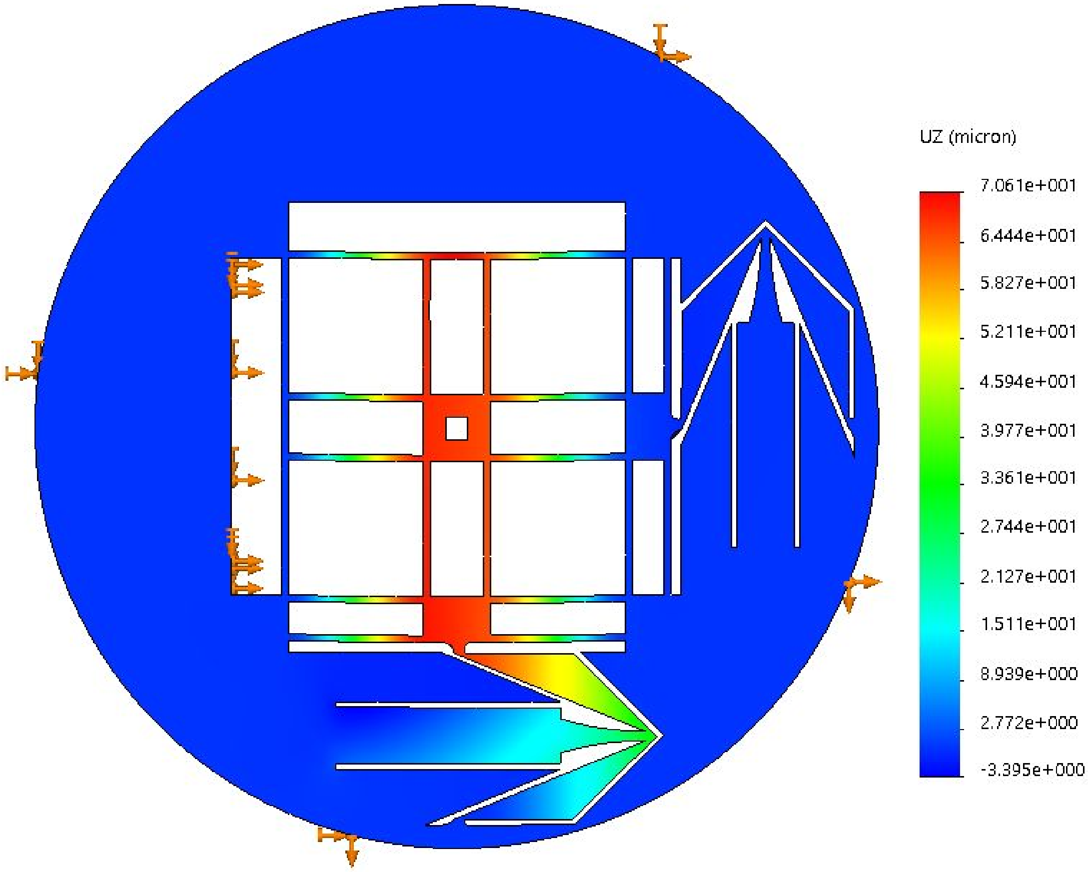

Figure 7. FEM simulated static actuation (0-150V)

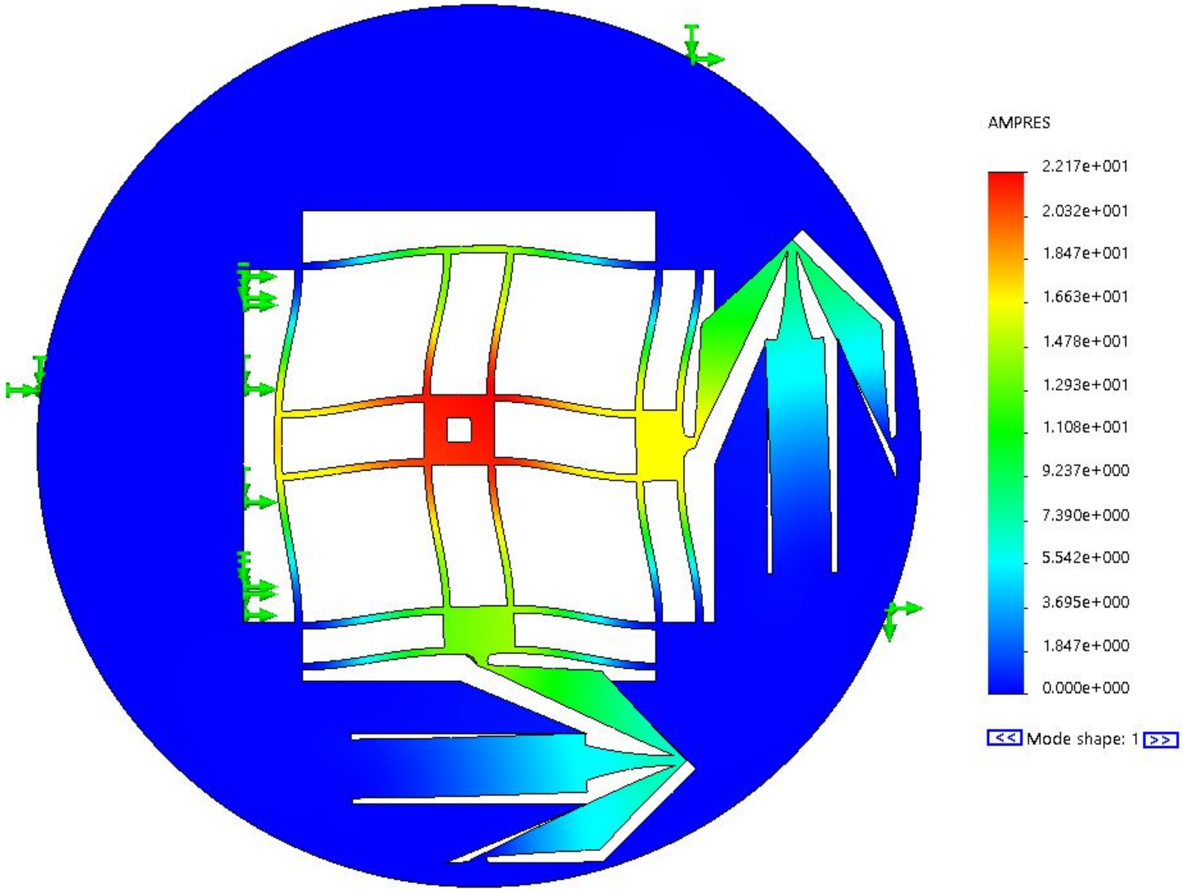

Figure 8. FEM simulated first resonant mode (3.4 kHz)