Design & Simulation of a Mechanical Amplifier for Inertial Sensing Applications

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ABSTRACT

This paper describes the design and simulation of a mechanical amplifier used to improve the performance of a capacitive accelerometer. It comprises a long silicon beam attached to the proof mass and it amplifies the sensed motion based on the leverage system [1]. Finite element simulations have been used to validate the concept and facilitate the design. Although increasing the overall accelerometer dimensions, a mechanical amplifier provides amplification without adding significant noise to the system, unlike its electrical counterpart. Moreover, the increased amplification of the motion obtained with longer beam length results in higher sensitivity.

Keywords: capacitive accelerometer, mechanical leverage system, sensitivity.

1 INTRODUCTION

The magnitude of the signal in capacitive inertial sensing devices and particularly in differential sensors is often the limiting factor to sensor sensitivity and resolution. The signal is usually degraded by various noise sources. Due to the requirement of interface circuitry in capacitive sensing devices and therefore to the presence of inherent electronic noise sources in the circuit, noise can be minimised but never completely eliminated. In order to further improve the performance of a sensor, non-electrical circuit related alternatives need to be considered.

In this paper, a mechanical amplifier is presented which improves the sensitivity of a capacitive accelerometer by amplifying the sensed signal, as depicted in Fig.1. A schematic of the mechanical amplifier is shown in Fig.2. An advantage of using the mechanical amplifier is that it does not add significant noise to the system unlike its electrical counterpart. Modelling results for the amplifier are presented. Simulations were carried out using the MEMS dedicated finite element tool, Coventorware.

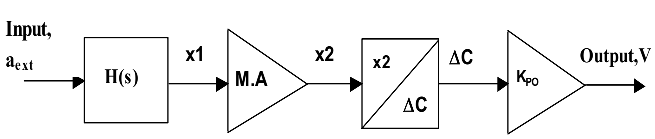

Fig. 1. An open loop, capacitive accelerometer including the mechanical amplifier (MA)

2 THEORY

The mechanical amplifier presented is based on the mechanical leverage system [1], which can be used as a way of increasing the scale factor of the sensor. An ideal pivot has the torsional stiffness, k_θ, which tends to zero and a vertical stiffness, k_v, which tends to infinity. Since true pivots cannot be realised using micromachining processes, they are approximated by flexural pivots. The model can be represented by Fig. 3 and described by Equation (1).

$$x1 = \frac{L2}{L1} x1 \qquad (1)$$

where L1, L2 are the beam lengths at either side of the pivot and x1, x2 are the deflections at the end of L1 and L2 respectively.

3 DESIGN

The design of the proof mass has been performed first and was followed by the addition of the mechanical amplifier. The two electrodes that are normally placed on either side of the proof mass in a conventional capacitive accelerometer, are placed at the end of the beam to detect the magnified deflection, x2, in this model.

Preliminary calculations have been performed to estimate the device characteristics and the values were verified using finite element analysis.

The proof mass measures $1x1mm^2$ with a thickness of 50µm. Fabrication of such a high thickness is enabled by the use of SOI-MEMS technology and deep reactive ion etching (RIE) process. $10x10\mu m^2$ etching holes are included in the proof mass to allow undercutting and release of the large structure during fabrication. In the simulation, the etch holes are lumped together and modelled as one large hole in the middle of the proof mass to increase simulation speed [2].

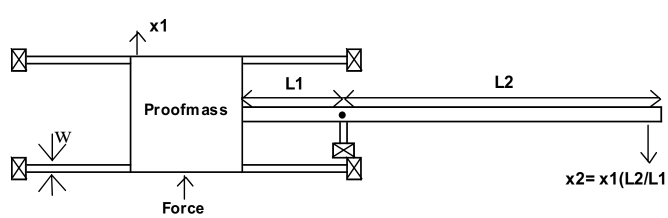

Fig. 2. Accelerometer with the mechanical amplifier

The volume of the proof mass is calculated to be 3.75×10^{-11} m^3 and the mass, about 100μg. The proof mass is supported by four suspension beams, each 1mm in length and 20μm in width, w. The depth of the beam is the same as the thickness of the proof mass.

To calculate the spring constant of the proof mass, the beam equation for a clamped-clamped beam with a point load at the centre (Fig. 4) is used [3]. For x less than or equal to L/2 away from a support, the deflection, y(x), corresponding to a point load, F, is given by:

$$y(x) = \frac{Fx}{48EI}(3Lx - 4x^2) \quad (2)$$

where E is Young's modulus, L is the length of the beam, t is the thickness of the beam and I is the bending moment of inertia. For a beam of a rectangular cross section, the bending moment of inertia is given by,

$$I = \frac{1}{12}wt^3 \quad (3)$$

where w is the width of the beam and t is the thickness.

Thus, in our model, the deflection of the beam, y(0.5L) is given by:

$$y(0.5L) = \frac{F(0.25L^3)}{48EI} \quad (4)$$

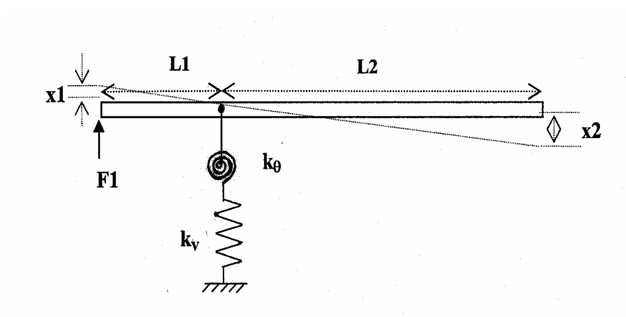

Fig. 3. Model used to predict leverage motion magnification

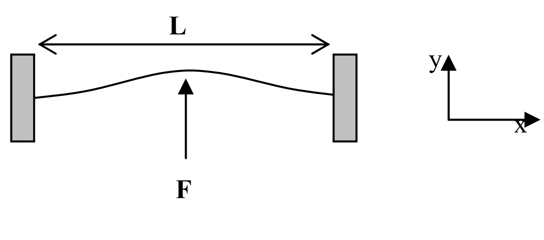

Fig. 4. Clamped-clamped beam [3]

The spring constant of a beam, k, can then be expressed as,

$$k = \frac{F}{y(0.5L)} = \frac{192EI}{L^3} \quad (5)$$

The effective spring constant of the proof mass was obtained by multiplying k by 2 as there are two suspension beam pairs on the proof mass. The resonant frequency can be calculated using the equation below:

$$\omega_r = \sqrt{\frac{k}{m}} \quad (6)$$

Several beam and pivot models have been evaluated in an effort to optimize the amplifier performance. To this end, finite element simulations were carried out using Coventorware. Optimized conditions for the accelerometer were determined based on the beam deflection, x2, which is sensed capacitively in the pick-off circuit.

During operation, the long beam, which acts as an amplifier, has to remain rigid while allowing maximum deflection. Parameters such as the beam width and the width of anchor (pivot) were evaluated. The beam thickness or depth is not a variable as it is decided by the fabrication process. In ref. [4], for the design of a silicon micromotion amplifier, silicon hinge-like systems were used to amplify motion. The same concept is applied here when various pivot structures, as shown in Fig. 5, were evaluated. The pivots are realised by etching the silicon material near the anchor in order to reduce torsional stiffness, k_θ, and allow larger beam deflection. Fabrication of these structures is limited by the specification of the minimum etching width.

Having optimised the beam and pivot parameters, the length, L2 is varied and the resulting structure simulated to explore the behaviour of the mechanical amplifier. A static input acceleration of 1g is assumed in the simulations throughout these experiments.

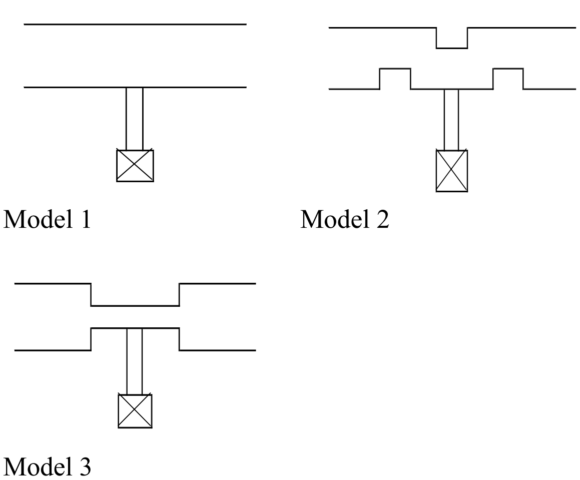

Fig. 5. Different pivot models

4 SIMULATION RESULTS AND ANALYSIS

Preliminary calculation shows that the proof mass has an effective spring constant of 34N/m and a fundamental resonant frequency of 2.93 kHz. These are comparable to the finite element simulation results, as shown in Fig. 6, which show the fundamental resonant mode occurring at 3.14 kHz (y-direction). The second and third resonant modes occur at 14.5 kHz (z-direction) and 22.6 kHz (torsional) respectively.

The simulation results showing optimization of the beam and pivot are presented in Figures 7 and 8. The figures illustrate the beam deflection, x2, for a range of beam/pivot parameters. Fig. 7(a) shows an optimum beam width of 30μm. For beams narrower than 30μm, the torsional vibration mode becomes the fundamental mode.

The length L1 was then varied for an assumed beam width of 30μm and shown in Fig. 7(b). Although increase in L1 results in greater beam deflection and therefore signal amplification, it increases the size of the device. Moreover, the dependence of x2 on L1 is less than on L2. For this reason, a value of 1mm was chosen for L1. Fig. 7(c) shows the variation in x2 with the pivot width. A pivot width of 5μm was chosen as it gives the highest beam deflection and is well above the specification for minimum etching width.

The beam length parameter, L2, was varied between 3 and 7mm and using the optimized conditions mentioned above and pivot model 1, the result is depicted 'model 1' in Fig. 8. It is found that the theoretical condition stated in Equation (1) is met when length of L2 is 6mm. Similarly, evaluations were carried out for pivot models 2 and 3. The figure shows that the improved pivot structures effectively reduce the length L2 to 5mm in model 2 and to 4mm in model 3.

In the simulation of these models, the 10x10μm^2 etching holes were not included in the long beam. As they are required for fabrication, they are incorporated both in the beam and in the proof mass. The modified structure yields a greater beam deflection denoted 'modified' in Fig. 8. The Coventorware model is shown in Fig. 9. The modal simulation for an input acceleration of 1g is shown in Fig. 10.

The simulation results can be summarized that L2 has to have a minimum length to act as a mechanical amplifier. Furthermore, the beam deflection is rapidly increased when length L2 is further increased. This property results in the rapid increase of the pick-off voltage signal and thus a higher resolution of the device is achieved.

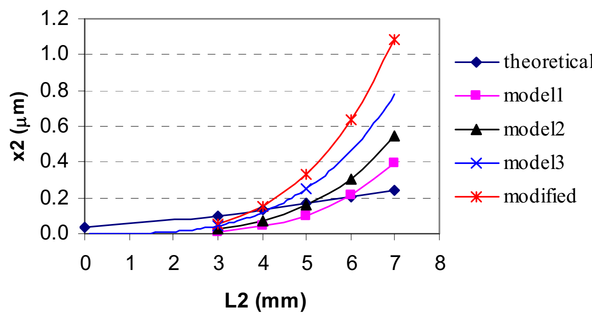

Fig. 8. Deflection of beam at various L2 lengths and different pivot models at input acceleration of 1g

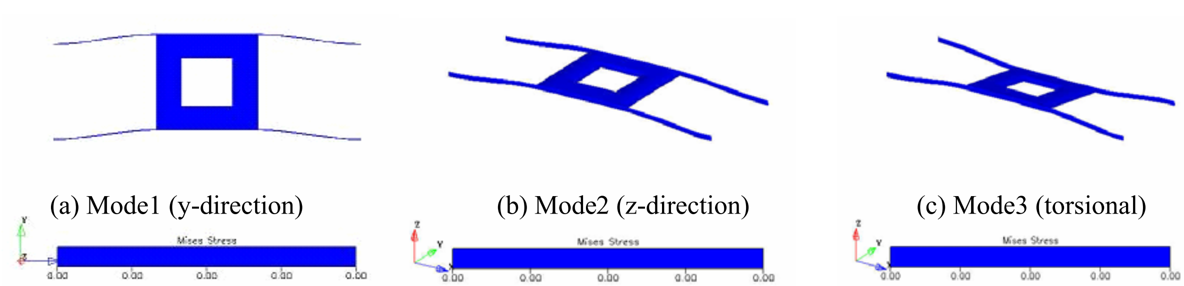

Fig. 6 (a-c). The resonant modes of the proof mass

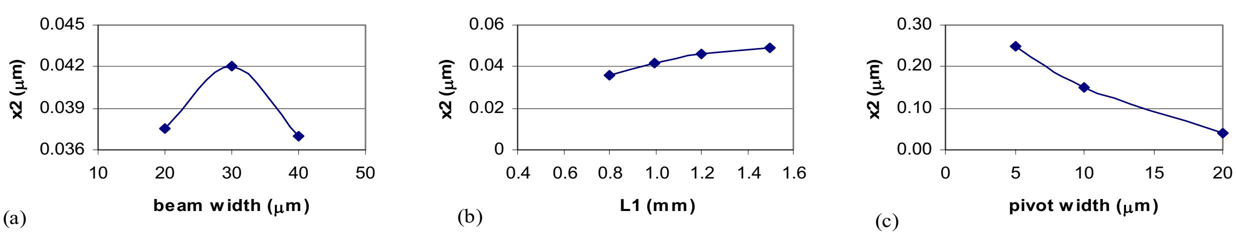

Fig. 7(a-c). Simulation results of optimization of beam and the pivot parameters

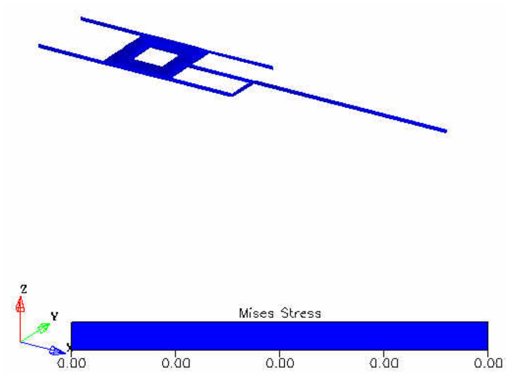

Fig. 9. Final simulation model

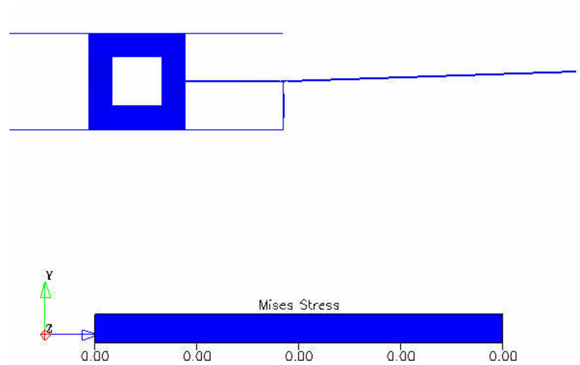

Fig. 10. Model under static input acceleration 1g

Device Parameter	**Without MA**	**With MA (L2=4mm, minimum length)**	**With MA (L2= 7mm)**
Mass	~100 μg	~100 μg	~100 μg
Fundamental res. Freq.	3.14 kHz	1.2 kHz	0.7 kHz
Area (Dimensions)	3 μm^2 (3mm x 1mm) **(x 1)**	7 μm^2 (3mm x 1mm) **(x 2.5)**	10 μm^2 (10mm x 1mm) **(x 3.3)**
Deflection	x1 = 0.034 μm	x2 = 0.15 μm	x2 = 0.632 μm
Sensitivity = $\Delta C/G$ where $\Delta C = 2\varepsilon A x/(d_o^2 - x^2)$	7.5 fF/G **(x 1)**	30 fF/G **(x 4)**	233 fF/G **(x 30)**

Table 1: Comparison of the performance of accelerometer with and without the mechanical amplifier

Finally, the parameters of the capacitive accelerometer with and without the mechanical amplifier are summarized in Table 1. The effect of exponential beam deflection when L2=7mm is evident. Although the total area occupied by the accelerometer is tripled, a 30-fold increase in sensitivity can be achieved.

5 CONCLUSION

Simulation results and calculation show that the concept of mechanical amplification based on a leverage system is feasible and it does not add significant noise to the system. Thus, incorporation of the amplifier into an inertial sensing system potentially can increase sensitivity. Currently we are extending the analysis to time varying input accelerations.

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