

Research Article

Design and Analysis of Silicon Resonant Accelerometer

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Abstract: In this study, we present an approach for micromechanical resonant accelerometer design, structural optimization and testing method of a kind of resonant accelerometer which is excited by the electrostatic force and detected by the change of capacitance. The structure parameters' influence on capability parameters are analyzed by MATLAB and the mechanical and model analysis of structure are carried out by ANSYS, also, we optimized the structure parameters. Simulating results validates the design thought.

Keywords: Model analysis, silicon resonant accelerometer, structure optimization

INTRODUCTION

With the help of micro-machined manufacturing process and technology, various types of MEMS (Micro-Electro-Mechanical System) inertial sensors such as accelerometer and gyroscope have been developed. Among these, a number of sensors take advantage of vibration as its operational principle. For instance, micromechanical accelerometers have been widely used both in military and civil applications and may lead to enormous benefit in society and economy. Acceleration of the MEMS resonant accelerometer is measured by detecting the change of resonant frequency. This kind of sensor has lots of advantages, such as direct frequency signal output, stable performance, excellent repeatability, high sensitivity and measuring precision.

During last few years, basing on resonant principle, several resonant accelerometers have been reported, including surface (Roessig, 1997; Seshia, 2002), bulk (Burrer, 1996) and mixed (bulk and surface) micro-machining process (Seok, 2002). In 1997, Roessig designed and fabricated a DETF (double-ended tuning fork) resonant accelerometer with a 45 Hz/g scale factor using a lever system to magnify the inertial force applied to the tuning fork (Roessig, 1997). In order to greatly increase the measurement sensitivity, a compound micro leverage mechanism is considered by stacking two single-stage micro levers together. In 2005 (Su, 2005), two-stage micro leverage mechanisms with an amplification factor of 80 are designed for force

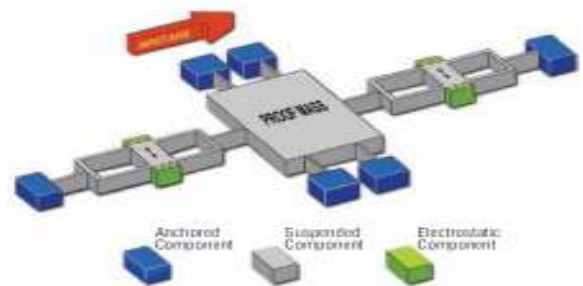


Fig. 1: SOA schematic by DRAPER

amplification to increase the overall sensitivity to 160 Hz/g. Although the micro leverage mechanism greatly increases the measurement sensitivity, it requires more on processing technology. In 2005 (Hopkins, 2005), a paper from DRAPER described the theory of operations, performance goals and the fabrication process for Silicon Oscillating Accelerometer (SOA) designs and achieve high sensitivity of 100 Hz/g without micro-leverage mechanism. Figure 1 is a schematic representation of the SOA sensor.

In this study, we design a structure with two double-ended tuning forks, drive/sense combs and a proof mass. Also, we give out the analysis results.

STRUCTURE AND PRINCIPLES

The structure of the resonant accelerometer designed by us is shown in Fig. 2. This accelerometer

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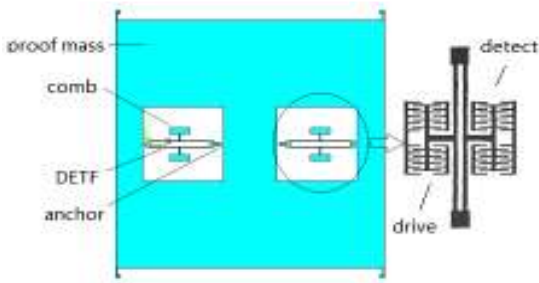


Fig. 2: Structure of the micromechanical resonant accelerometer

consists of two double-ended tuning fork, drive/sense combs and a proof mass surrounding two tuning forks and extending around. The proof mass is directly attached to the two double-ended tuning fork resonators. Each tuning fork is electro-statically actuated and sensed at resonance using comb electrodes. When an external acceleration is applied to the device, one tuning fork is subjected to a tensile force which raises its natural frequency and the other one experiences a compressive force, lowering its natural frequency. Two double-ended tuning forks provide for a differential output which is twice sensitivity of the one double-ended tuning fork and avoid or lower the ambient noise.

STRUCTURE OPTIMIZATION AND ANALYSIS

Structure optimization and simulation is very important during the design of MEMS device. The aim of optimization and simulation is to determine the best parameters of the structure to perform the best capability. The main job of optimization is to get the parameters of the resonator and mass. We use MATLAB to do some assistant analytic analysis and plot the relationship between the structure parameters and the performance index. We calibrate the design and optimize the results by ANSYS.

Theoretical analysis: For an initially flat beam, subjected to a axial force F , the equation of motion governing the transverse deflection φ as a function of the position x along the beam length is given by the following partial differential equation:

$$\frac{\partial^2}{\partial x^2} (EI \frac{\partial^2 \varphi}{\partial x^2}) - \frac{\partial}{\partial x} (F \frac{\partial \varphi}{\partial x}) - \rho A \frac{\partial^2 \varphi}{\partial t^2} = P(x) \quad (1)$$

where,

- E : The Young's modulus
- ρ : The material density
- A & I : The cross-sectional area and second moment of inertia, respectively
- $P(x)$: The driving force per unit length

When there is no force applied, Eq. (1) can be simplified as:

$$\frac{\partial^2}{\partial x^2} (EI \frac{\partial^2 \varphi}{\partial x^2}) - \rho A \frac{\partial^2 \varphi}{\partial t^2} = 0 \quad (2)$$

The key parameters of resonator are the length, height and width. The aim is to maximize the resonant frequency under 1 g acceleration.

By solving Eq. (2), we get the expression of the natural frequency of single beam:

$$f_0 = \frac{1}{2\pi} \sqrt{\frac{195.52}{L^3} \frac{EI}{0.3965m_t + m}} \quad (3)$$

By simplifying Eq. (3), the relationship of the first order natural frequency of single beam and the parameters of the structure is:

$$f_0 = \frac{6.9913}{\pi} \sqrt{\frac{1}{12} \left(\frac{h}{L}\right)^3 \frac{E}{\rho(0.3965hL + s)}} \quad (4)$$

Under an axial load, the natural frequency can be expressed as:

$$f = f_0 (1 \pm 0.2949 \times F \frac{L^2}{Eh^3b})^{1/2} \quad (5)$$

where,

- f_0 : The natural frequency without the axial load
- f : The natural frequency under the axial load
- m : The mass of the comb connected by the beam
- m_t : The mass of the single beam
- E : The Young's modulus
- ρ : The material density
- b : The width of the beam
- h : The thickness of the beam
- L : The effective length of the beam
- s : The total area of the driven and sense electrodes

When the tuning fork is subjected to a tensile force, the natural frequency raises, otherwise, decreases.

To substitute the structure parameters into Eq. (5), we can get the relationship between the structure parameters and the natural frequency.

From Fig. 3 and 4, we can see the length and height of the beam are of nonlinear relationship with the natural frequency. As the length of the beam increases, the frequency decreases, while as the height of the beam increases, the frequency also increases. We mainly consider about the requirement of driven and detect capacitance when choosing the length and height. The larger the length and height the greater the driven force and detect capacitance and the easier to detect the resonant statement.

Structure of the resonator: We consider about some novel resonator designs which are shown in Fig. 5. We choose a set of typical parameters and make simulation on the structure. The parameters are:

$$L = 550 \mu\text{m}, h = 4 \mu\text{m}, b = 50 \mu\text{m}, \text{gap} = 8 \mu\text{m}, \\ \text{fsl} = 45 \mu\text{m}, s = 2 \mu\text{m}, \text{sl} = 55 \mu\text{m}, \text{sw} = 12 \mu\text{m}$$

The results are shown in Table 1. One mode of the resonator comprises an asymmetric mode and a symmetric mode. The asymmetric mode is the working state and the symmetric mode is the disturbing state. The difference of these two modes' natural frequency is one of the most important criterion for resonator design. The larger the difference, the better the performance. From Table 1 we can see that the last two designs have much larger difference, which means that the last two have better performance. And the stubs can reduce the influence of the symmetric mode. Finally, we choose number 4 as the structure of the resonator. The first four mode of the resonator are shown in Fig. 6.

Also, we analyze the difference of the asymmetric mode and symmetric mode natural frequency as the beam length changes, which is shown in Fig. 7. We can conclude that as the length of the beam increases the difference of the two modes' natural frequency decreases. It means that the beam length should be as short as possible. However, when it comes to circuit detection, we want a longer beam. Therefore, when we

Table 1: Frequency analysis

Resonator	Mode	Asymmetric mode natural frequency	Symmetric mode natural frequency
1	2	118.665	118.366
2	2	118.441	118.295
3	2	118.653	109.283
4	2	119.455	109.730

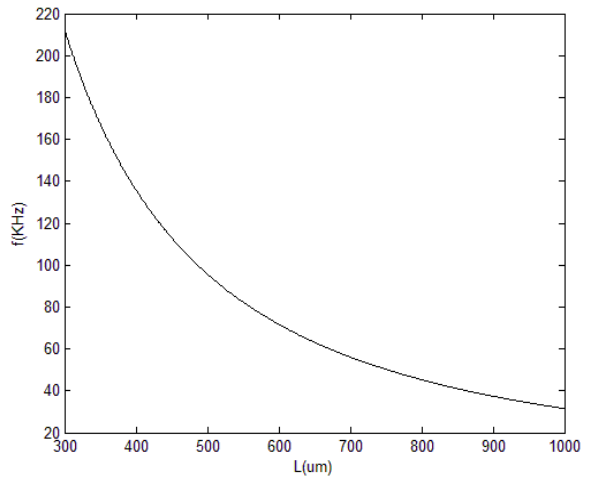


Fig. 3: Relationship between length and frequency

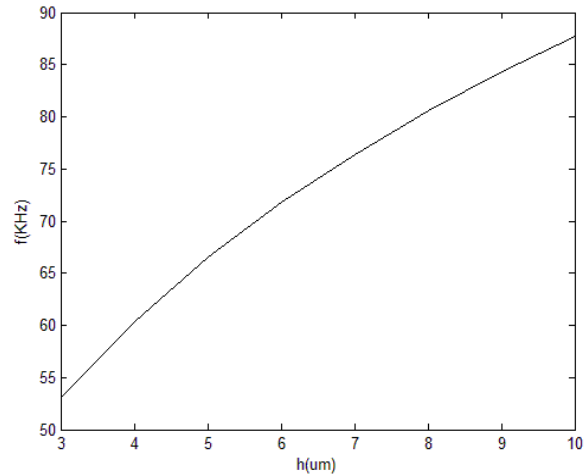


Fig. 4: Relationship between height and frequency

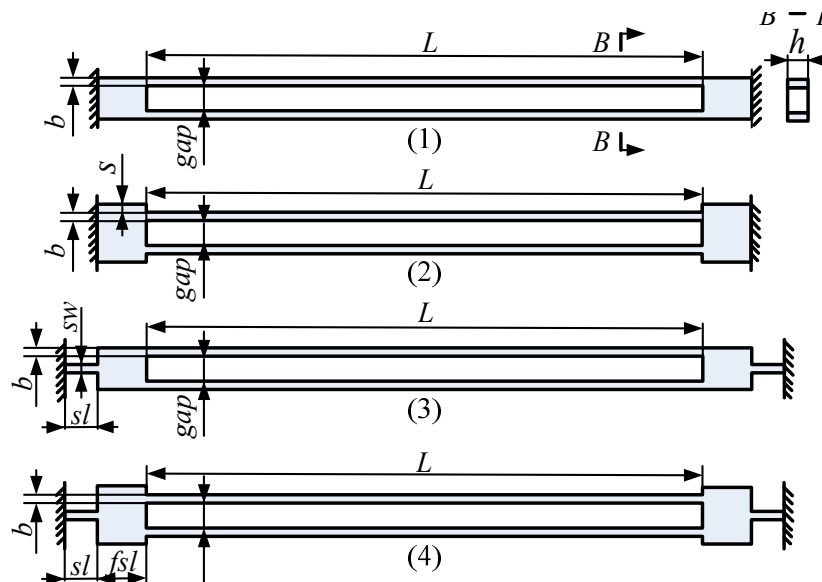


Fig. 5: Different kinds of resonators

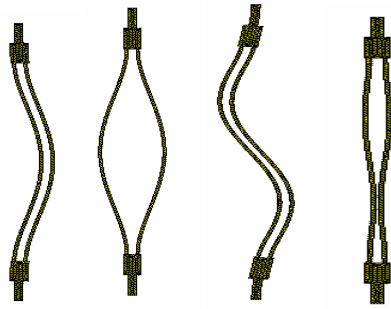


Fig. 6: The first four mode of the resonator

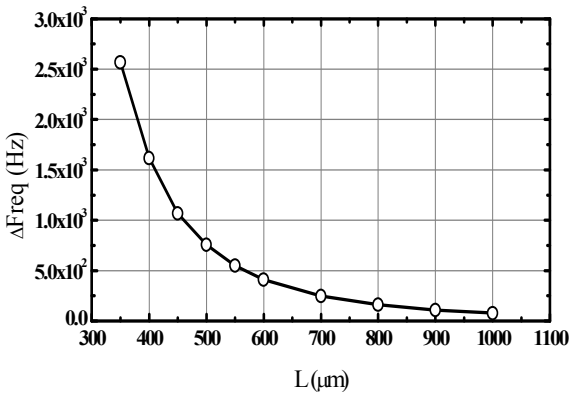


Fig. 7: The difference of asymmetric mode and symmetric mode natural frequency as the beam length changes

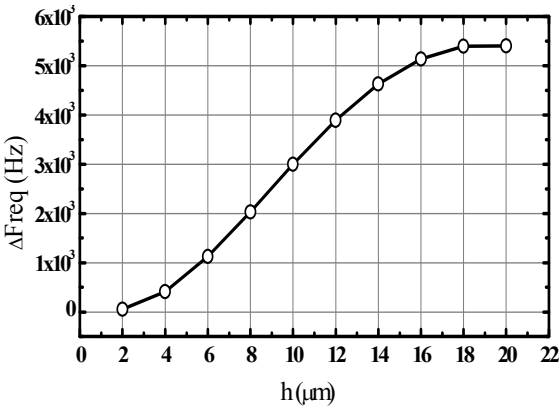


Fig. 8: The difference of asymmetric mode and symmetric mode natural frequency as the beam length changes

design the length of the beam, we should consider about these two conditions.

Similarly, we analyze the difference of the asymmetric mode and symmetric mode natural frequency as the beam height changes, which is shown in Fig. 8. We can conclude that as the height of the beam increases the difference of the two modes' natural frequency increases. It means that the beam height should be as large as possible. However, when it comes

Table 2: Structure parameters

Notation	Parameter	Value
b[μm]	Beam width	6
l[μm]	Accelerometer thickness	80
L[μm]	Beam length	450
E[N/m ²]	Young's modulus	1.69×10 ¹¹
ρ[kg/m ³]	Density	2.5×10 ³
A _m [μm ²]	Proof mass surface area	2000×2000

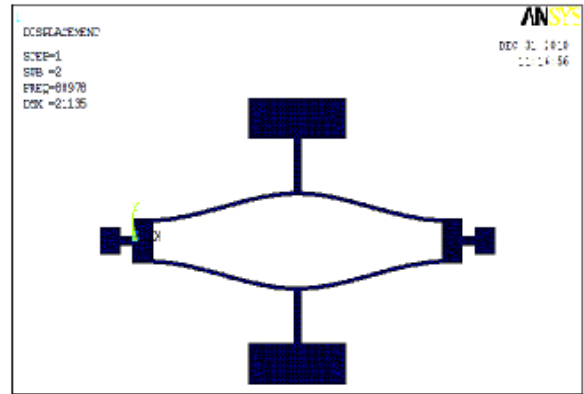


Fig. 9: Vibration mode with no acceleration

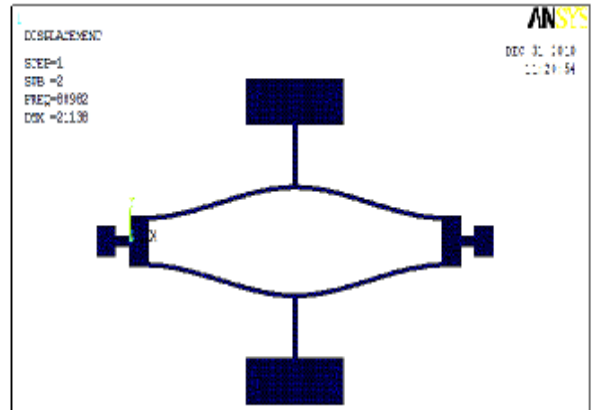


Fig. 10: Vibration mode with 1 g acceleration

to drive the structure, we want a thinner beam. Therefore, when we design the height of the beam, we should consider about these two conditions.

The whole structure: By the analysis above, we optimize all of the parameters of the accelerometer. The structure parameters of the micro resonant accelerometer are shown in Table 2. We use Finite Element Method (FEM) to validate the results and get the vibration mode shown in Fig. 9 and 10. We apply 0 and 1 g to the DETF respectively and the resonant frequencies are 81978 and 81982 Hz, changed 4 Hz. In Fig. 11, we show the relationship between the acceleration and output frequency and with 1 g input acceleration changes, the output frequency changes about 9 Hz. Figure 12 shows the sample we processed.

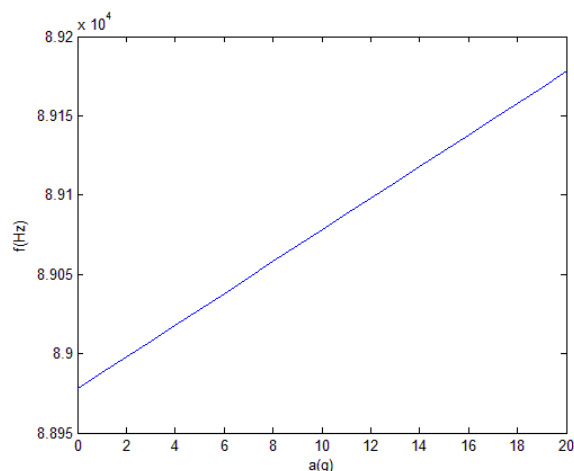


Fig. 11: Relationship between acceleration and frequency



Fig. 12: The sample

CONCLUSION

In this study, a structure of silicon resonant accelerometer is presented, which provide differential output. This kind of structure can improve the sensitivity and avoid or lower the ambient noise. Besides, we show the simulation analysis of a resonant accelerometer with ANSYS and MATLAB, also optimize the structure parameters. From the vibration mode analysis, we can see that when the tuning fork under a tensile force, its natural frequency raises by 4 Hz. Meanwhile, the other

one experiences a compressive force and its natural frequency decreases. As discussed in section 2, the total sensitivity should be around 8 Hz. In Fig. 7, our theoretical result shows that the total sensitivity is about 9 Hz, which conforms with the simulation results above. Future work includes the test of the sample and further optimization of the structure.

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