The effects of beam geometry on (TED) on Q-factor in vibratory Dual-Mass Microgyroscope

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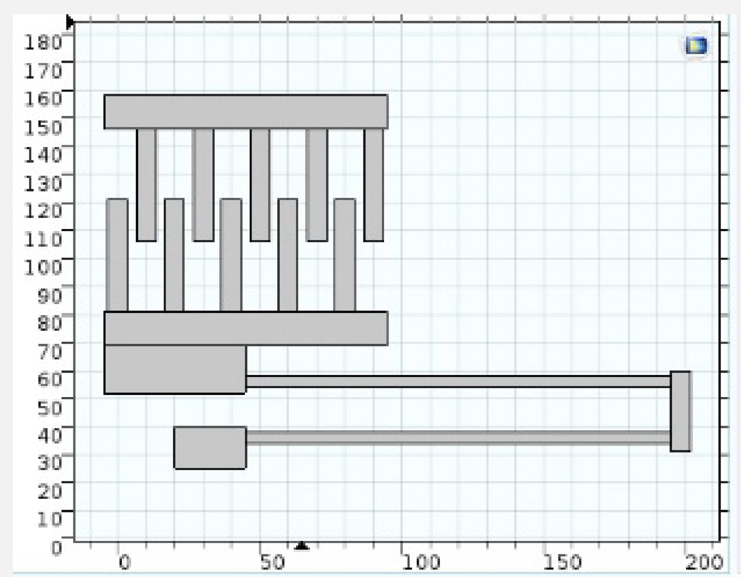
INTRODUCTION

The effects of beam geometry , length-to-thickness aspect ratio , natural frequency , flexural mode shape and structural boundary conditions on TED are investigated for the representative case of single Silicon micro beam resonators. The effects of beam geometry Nonsymmetric tuning of stiffness in a coupled 2-DOF gyroscope is completed through the use of the negative electrostatic spring effect. This variable stiffness is shown to be able to adjust the reaction forces of the structure at the anchors, effectively balancing any spring imperfections caused by fabrication imperfections.

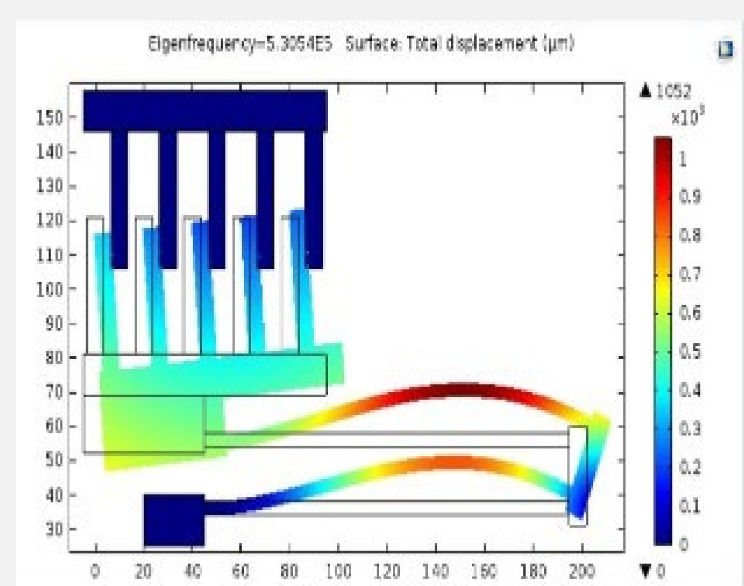
For many resonant modes the limit to the achievable quality factor is determined by thermoelastic damping. thermoelastic friction takes place when you subject any material l to cyclic stress. The stress results in deformation, and the required energy is mostly stored as internal potential energy.

PROBLEM FORMULATION

A typical Comb-Drive consists of movable and fixed combs. The fixed comb is anchored to the base and movable plate is supported by a spring through a shuttle mass plate.



An increased displacement of lateral comb drive actuator will subsequently be accomplished with the same actuation voltage.



Stress distribution over different spring designs are simulated by COMSOL 5.0 a using a standard comb drive with increase in flexure length from 220µm to 280µm, displacement and capacitance increases from 1.063µm to 2.85µm and 327pf to 352pf respectively at 130V.

RESULTS

The lateral electrostatic force applied on the movable plate of the capacitor is

$$F_{el} = -\frac{\partial E(x)}{\partial x} = \frac{\varepsilon_0 W}{2d} V_d^2$$

And keeps constant when width, W , of the plates is scaled down with the distance d .

A displacement of the mass from its balance position, x .where k is the elastic constant of the spring.

$$F = -kx$$

The differential equation for the mass movement is

$$M\dot{x} = -kx$$

$$\omega^2 = {}^{k}/_{M} \ , \quad \dot{x} + \omega^2 x = 0$$

$$x = Asin(\omega t + \alpha)$$

$$\omega = \sqrt{\frac{k}{M}}$$

The amplitude A and the phase lag α can be decide by the initial conditions of the system.

$$A = \sqrt{x_0^2 + (\frac{v_0}{\omega})^2}$$

$$Q = \frac{2\pi W_0}{\Delta W} = \frac{\omega_0}{2\delta} = \frac{\omega_0}{\Delta\omega}$$

W_0 is the total stored vibrational energy, ΔW is the energy lost per cycle

SIMULATION RESULTS

The resonant frequency of the beam can also be computed

Name	Expression	Value
Young's modulus	157 [Gpa]	1.5700e11 Pa
Density	2330 [kg/m^3]	2330.0 kg/m^3
Poisson's ratio	0.3	0.30000
Coefficient of thermal expansion	2.6e-6 [1/k]	2.6000e-6 1/k
Heat capacity at constant pressure	700 [J/(kg*k)]	700.00 J/(kg.K)
Thermal conductivity	90 [W/ (m*k)]	90.000 W/(m.K)
T0	300 [k]	300.00 K

Resonator	Length (µm)	Width (µm)	Thickness (µm)	F0 (KHZ)
SC1	390	52	9	1445
SC2	191	52	9	1445
SC3	189	26	9	1445
SC4	190	26	9	1445
SC5	188	26	14	641
SC6	182	26	14	641
SC7	189	52	16	471
SC8	202	26	16	471
SC9	179	26	16	471
SC10	438	52	12	632

Resonator	Frequency F (KHZ)	Quality Q
SC1	81.40	65.200
SC2	318.50	15.500
SC3	343.40	12.200
SC4	310.20	9.600
SC5	556.00	9.400
SC6	584.50	8.060
SC7	607.00	7.800
SC8	478.40	8.200
SC9	640.00	8.900
SC10	600.00	7.500

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